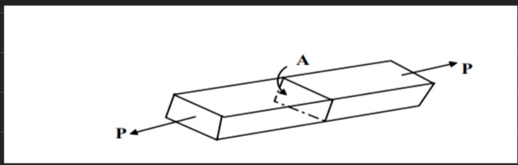
MOS Revision

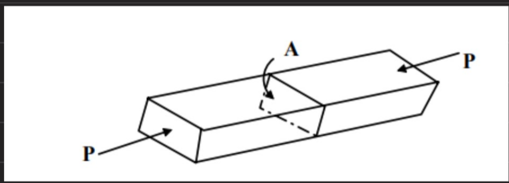
i) Stress Analysis :

a) Tensile

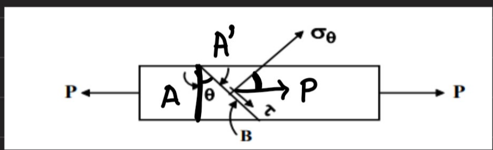


$$\sigma_t = \frac{P}{A}$$

b) Compressive



$$\sigma_c = \frac{P}{A}$$

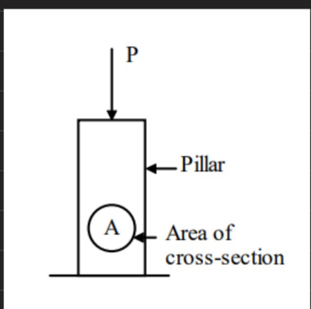


$$\sigma_\theta = \frac{P\cos\theta}{A/\cos\theta} \quad , \quad \tau = \frac{P\sin\theta}{A/\cos\theta}$$

$\uparrow$ Normal stress $\qquad$ $\uparrow$ Sheer stress

ii) Bearing Stress : When a body is pressured against another, the compressive stress developed is termed as bearing stress

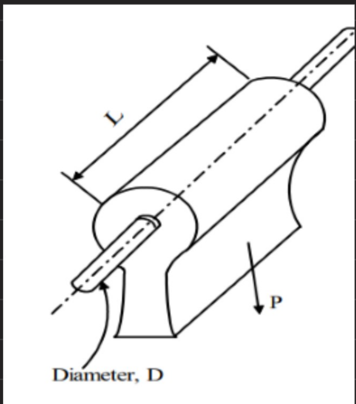
a)

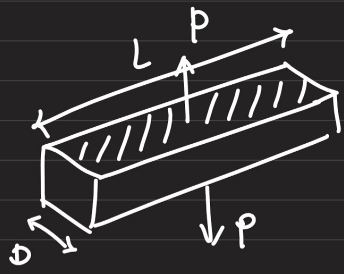


$\equiv$

$$f_{br} = \frac{P}{A}$$

b)

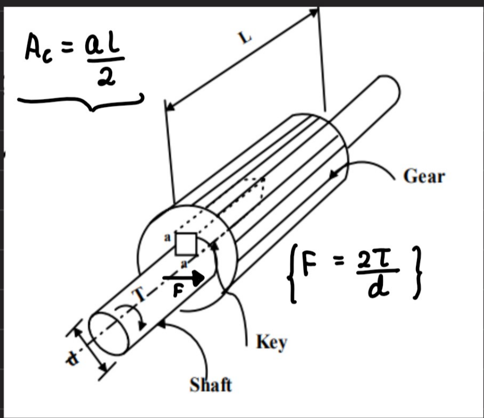




$$\sigma_{br} = \frac{P}{LD}$$

projected area

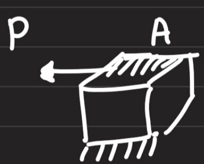
c)

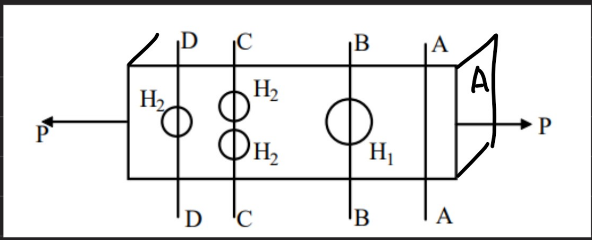


$$\sigma_{br} = \frac{4T}{aLd}$$

iii) <u>Shear stress</u>:

$$\tau = \frac{P}{A}$$





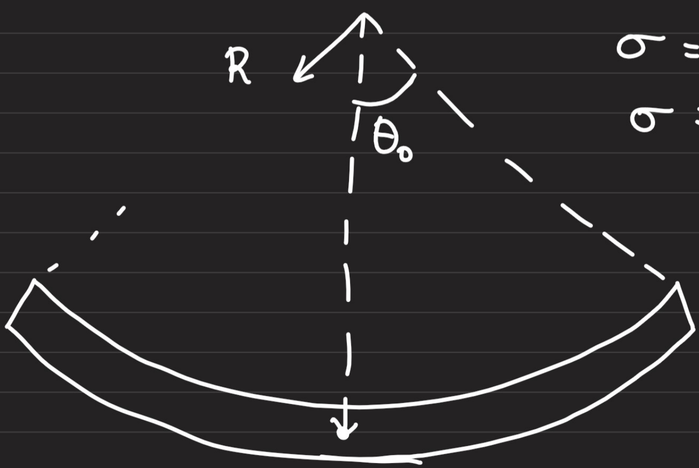
Let the cross-sectional area of the plate, the larger hole H_1 and the smaller holes H_2 be A, a_1, a_2 respectively. If $2a_2 > a_1$ the critical section in the above example is CC and the average normal stress at the critical section is

$$\sigma = \frac{P}{A - 2a_2}$$

If $2a_2 < a_1$, $\sigma_{critical} = \dfrac{P}{A - a_1}$ (BB → critical secⁿ)

Bending of Beams:

$$\left\{ \sigma = \frac{My}{I} \right\} \leftarrow$$

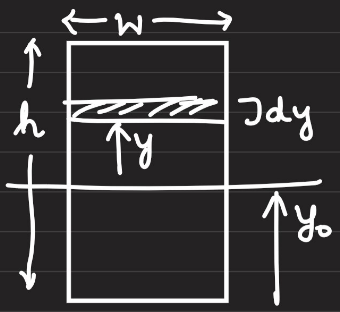


$$\sigma = E\varepsilon$$

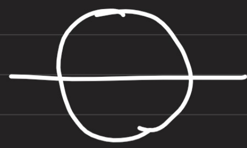
$$\sigma = \frac{Ey}{R}$$

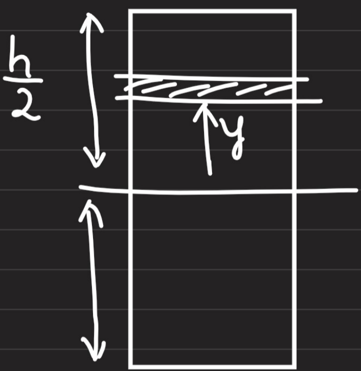
$l_0 = 2R\theta_0$, $l_f = 2(R-y)\theta_0$

$$\left\{ \varepsilon = \frac{l_f - l_0}{l_0} = \frac{y}{R} \right\}$$



$$\underbrace{\int dA\, y = 0}_{\text{1st moment of area}}$$





$$\int_{-y_0}^{(h-y_0)} y\,(dy\,w) = 0$$

$$\Rightarrow \frac{w}{2}\left[(h-y_0)^2 - y_0^2\right] = 0$$

$$\Rightarrow -2hy_0 + h^2 = 0$$

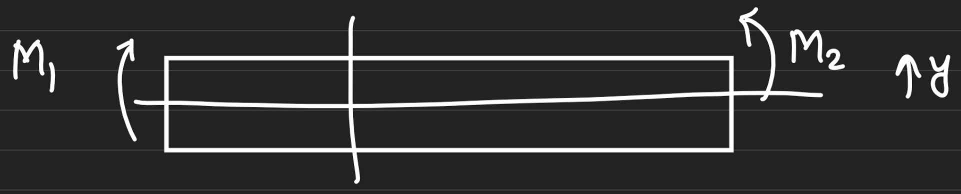
$$\Rightarrow \left\{ y_0 = \frac{h}{2} \right\} \checkmark$$

2nd moment of area, $I = \int_{\text{about N.A}} y^2\, dA$

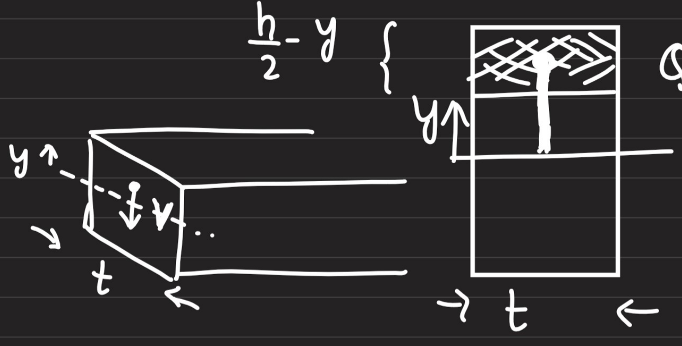
$$I = \int_{-h/2}^{h/2} y^2 (w\,dy) = \frac{w}{3}\left[\left(\frac{h}{2}\right)^3 + \left(\frac{h}{2}\right)^3\right]$$

$$= \frac{wh^3}{12}$$

Now,



$$\boxed{\tau = \frac{VQ}{It}}$$



$$Q = \frac{t}{2}\left(\frac{h}{2} - y\right) \times \left(\frac{h}{2} + y\right)$$

$$\left\{ Q = \frac{t}{2}\left(\frac{h^2}{4} - y^2\right) \right\}, \quad I = \frac{th^3}{12}$$

Torsion :

$(r\theta = L\gamma)$, Now, $\tau = G\gamma$

$$\Rightarrow \frac{\tau}{r} = \frac{G\theta}{l}$$

$$df = \tau\,(2\pi r\,dr)$$

$$dT = \tau\,(2\pi r\,dr)\,r$$

$$\Rightarrow T = \left(\frac{G\theta}{l}\right) 2\pi \int_0^{r_0} r^3\,dr$$

$$\Rightarrow T = \frac{G\theta}{l}\left(\frac{\pi r_0^4}{2}\right) \quad , \quad J = \frac{\pi r_0^4}{2}$$

$$\Rightarrow \boxed{\frac{T}{J} = \frac{G\theta}{l} = \frac{\tau}{r}}$$

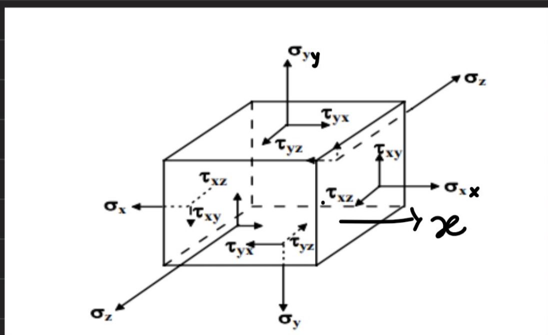
$$\therefore \tau = \frac{Tr}{J} \quad \Rightarrow \left.\tau_{max} = \frac{T r_0}{J}\right\} \checkmark$$

Buckling :-

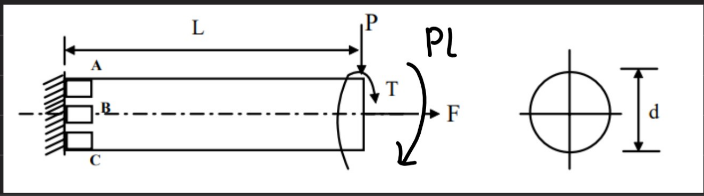
$$\left\{ EI \frac{d^2 y}{dx^2} = M \right\} \qquad F_{cr} = \frac{\pi^2 EI}{l^2}$$

Stress

$$\begin{bmatrix} \sigma_x & \tau_{xy} & \tau_{xz} \\ \tau_{yx} & \sigma_y & \tau_{yz} \\ \tau_{zx} & \tau_{zy} & \sigma_z \end{bmatrix}$$



$$\tau_{xy} = \tau_{yx} ,$$



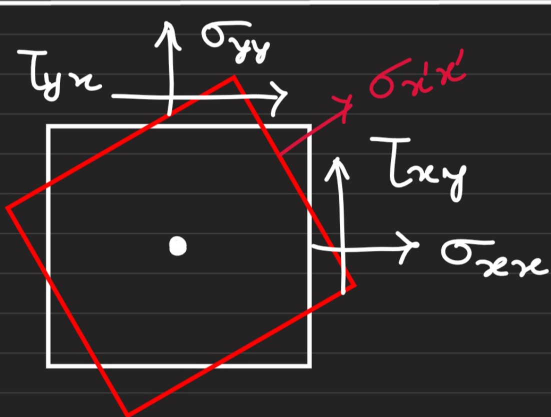
$$\sigma_{axial} = \frac{4F}{\pi d^2}, \quad \tau_{torsion} = \frac{2T\cancel{(d/2)}}{\pi (d/2)^{\cancel{4}\,3}} = \frac{16T}{\pi d^3}$$

$$\sigma_{bending} = \frac{My}{I} = \frac{4P}{\pi\left(\frac{d}{2}\right)^3}$$

$$\left\{ \sigma_{bending} = \frac{32P}{\pi d^3} \right\}$$

$$\frac{\pi r_o^4}{4}$$

Stress Strain Transformation :



$$\tau_{x'y'} = \frac{(\sigma_y - \sigma_x)}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$$

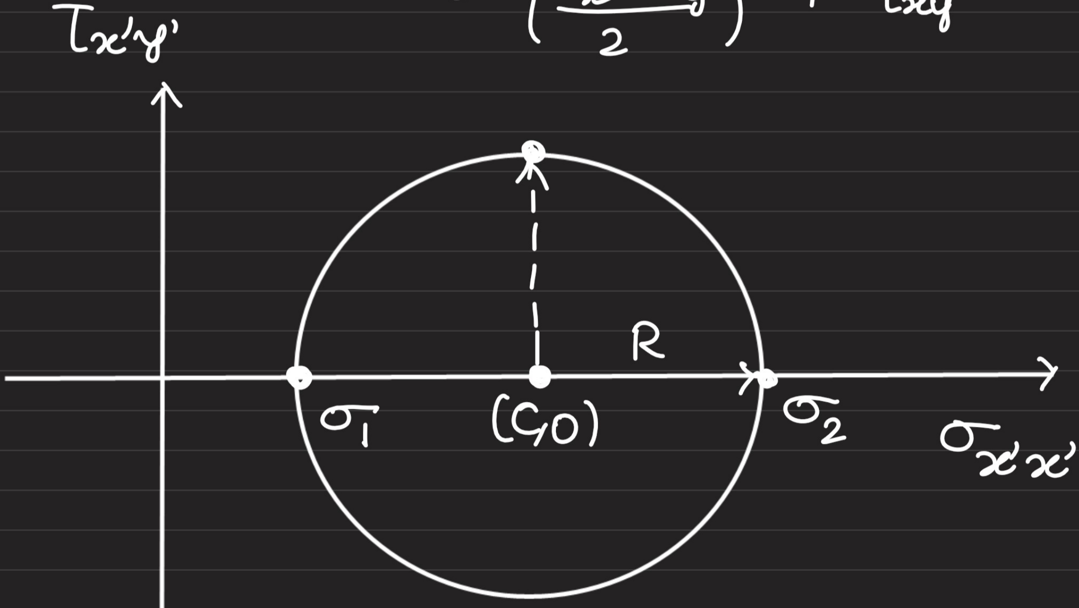
$$\underbrace{\sigma_{x'x'}} = \frac{(\sigma_x + \sigma_y)}{2} + \frac{(\sigma_x - \sigma_y)}{2}\cos 2\theta + \tau_{xy}\sin 2\theta$$

$$\tau_{xy'} = \frac{(\sigma_y - \sigma_x)}{2}\sin 2\theta + \tau_{xy}\cos 2\theta$$

$$\underbrace{\sigma_{x'x'}} = \frac{(\sigma_x + \sigma_y)}{2} + \frac{(\sigma_x - \sigma_y)\cos 2\theta}{2} + \tau_{xy}\sin 2\theta$$

$$\left[\sigma_{x'x'} - \left(\frac{\sigma_x + \sigma_y}{2}\right)\right]^2 + \tau_{x'y'}^2$$

$$= \left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2$$



$$C = \frac{\sigma_x + \sigma_y}{2}, \quad R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\left.\sigma_1 = C - R, \quad \sigma_2 = C + R\right\} \text{principal stress}$$

eg: $\sigma_x = -10$ MPa , $\tau = -20$ MPa

$\sigma_y = 20$ MPa

sol^n:

$$\sigma_{x'x'} - C = \frac{(\sigma_{xx} - \sigma_{yy})}{2} \cos 2\theta + \tau_{xy} \sin 2\theta$$

$P = C-R, \quad C+R$

$$C = \frac{\sigma_x + \sigma_y}{2}, \quad R = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$\{ C = 5, \; R = 25 \}$

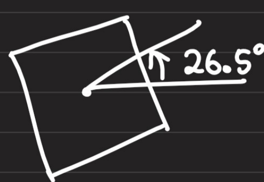
$\underbrace{\sigma_1 = -20 \text{ MPa}}, \quad \underbrace{\sigma_2 = 30 \text{ MPa}}$

NOW

$$-25 = -15 \cos 2\theta - 20 \sin 2\theta$$

$$\cos 2\theta = \frac{3}{5} \quad \text{or} \quad \sin 2\theta = \frac{4}{5}$$

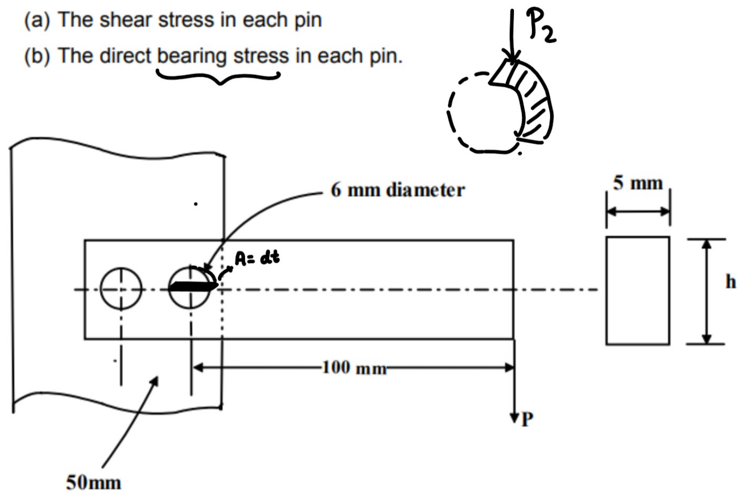
$\Rightarrow 2\theta = 53^\circ \Rightarrow \theta = 26.5^\circ$



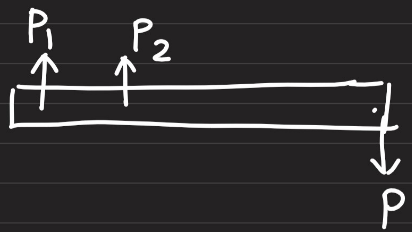
A 5mm thick steel bar is fastened to a ground plate by two 6 mm diameter pins as shown in figure-2.2.7.1. If the load P at the free end of the steel bar is 5 KN, find

(a) The shear stress in each pin

(b) The direct bearing stress in each pin.



Sol^n:

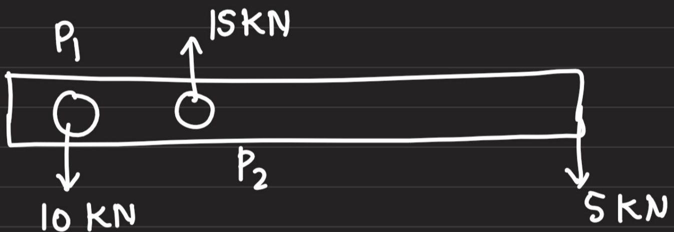


$$P_1 + P_2 = P$$

And

$$P_2 (50\ mm) = P (150\ mm)$$

$$P_2 = 3(P) = 15\ KN$$



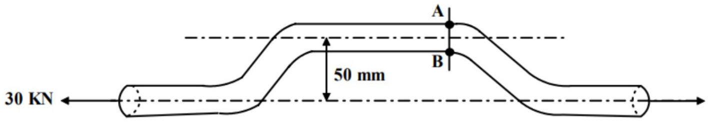
a)

$$\left.\begin{aligned} \tau_1 &= \frac{10 \times 1000}{\pi (3 \times 10^{-3})^2} \\ \tau_2 &= \frac{15 \times 1000}{\pi (3 \times 10^{-3})^2} \end{aligned}\right\} \text{Ans}$$

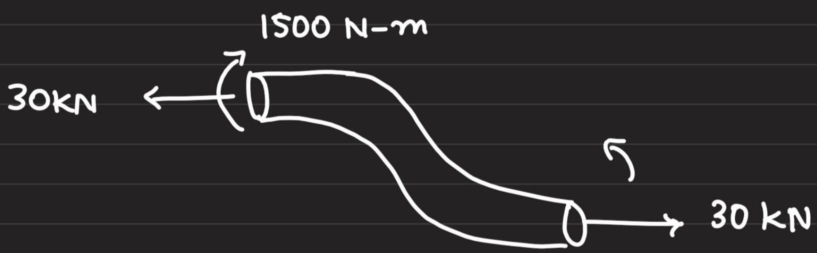
b)

$$\left.\tau_{b1} = \frac{10 \times 10^3}{(0.006)\ (0.005)}\right\} \text{Ans}$$

Q.2: A 100 mm diameter off-set link is transmitting an axial pull of 30 KN as shown in the figure- 2.2.7.3. Find the stresses at points A and B.



$$30 \times 10^{3} \times 50 \times 10^{-3}$$



$$\sigma_{y_A} = -\frac{My}{I} = \frac{-1500\,(\cancel{r})\,(4)}{\pi\,\cancel{r^4}\,r^3}$$

$$= -\frac{6000}{\pi \times (0.05)^3} = -15278.87454 \ \text{KN}/m^2$$

$$\sigma_{A_S} = \frac{30000}{\pi\,(0.05)^2} = 3819.718634 \ \text{KN}/m^2$$

$$\left\{ \sigma_{T_A} = -11459.156 \ \text{KN}/m^2 \right\}$$

$$\left\{ \sigma_{T_B} = 19098.593 \ \text{KN}/m^2 \right\}$$

Generalized Hooke's law

$$\varepsilon_1 = \frac{1}{E}\left(\sigma_1 - \nu(\sigma_2 + \sigma_3)\right)$$

$$\varepsilon_2 = \frac{1}{E}\left(\sigma_2 - \nu(\sigma_1 + \sigma_3)\right)$$

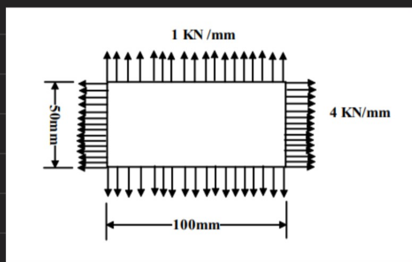
$$\varepsilon_3 = \frac{1}{E}\left(\sigma_3 - \nu(\sigma_1 + \sigma_2)\right)$$

*

$$\boxed{E = 2G(1+\nu)}$$

$$\boxed{E = 3K(1-2\nu)}$$

eg: A rectangular plate of 10mm thickness is subjected to uniformly distributed load along its edges as shown in figure-2.3.7.1. Find the change in thickness due to the loading. E=200 GPa, ν = 0.3



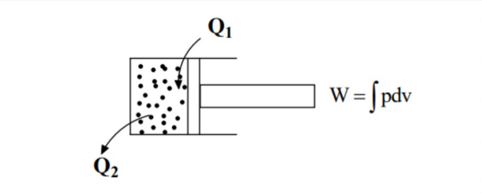
sol^n: $\sigma_x = \dfrac{2\cancel{00}\ \text{KN}}{5\cancel{00}\ mm^2}$ $\quad (\sigma_z = 0)$

$\sigma_y = \dfrac{100\ \text{KN}}{1000\ mm^2}$ $\quad\Big|\quad \sigma_x = 400\ \text{MPa}, \ \sigma_y = 100\ \text{MPa}$

$$\varepsilon_z = -\frac{(0.3)(5\cancel{00}) \times 10^6}{2\cancel{00} \times 10^9}$$

$$\left.\frac{\Delta t}{10} = -\frac{1.5}{2} \times 10^{-3} = -7.5\ \mu m \right\} \text{Ans}$$

Module - 1 :

- 

Heat → Mechanical energy
$\underbrace{\text{Heat} \rightarrow \text{Mechanical energy}}_{\text{Heat engine}}$

Types of design :-

Adaptive design

This is based on existing design, for example, standard products or systems adopted for a new application. Conveyor belts, control system of machines and mechanisms or haulage systems are some of the examples where existing design systems are adapted for a particular use.

Developmental design

Here we start with an existing design but finally a modified design is obtained. A new model of a car is a typical example of a developmental design .

New design

This type of design is an entirely new one but based on existing scientific principles. No scientific invention is involved but requires creative thinking to solve a problem. Examples of this type of design may include designing a small vehicle for transportation of men and material on board a ship or in a desert.

Based on methods

Rational design

This is based on determining the stresses and strains of components and thereby deciding their dimensions.

Empirical design

This is based on empirical formulae which in turn is based on experience and experiments. For example, when we tighten a nut on a bolt the force exerted or the stresses induced cannot be determined exactly but experience shows that the tightening force may be given by P=284d where, d is the bolt diameter in mm and P is the applied force in kg. There is no mathematical backing of this equation but it is based on observations and experience.

Industrial design

These are based on industrial considerations and norms viz. market survey, external look, production facilities, low cost, use of existing standard products.

Mechanism used

Materials

Shear / Tensile

Inspectibility ← Factors → Shape, size Space & Weight

Manufacturing method

Reliability

Maintenance, cost

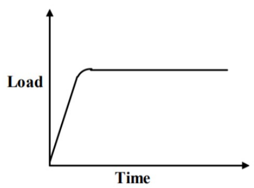
Material- This is a very important aspect of any design. A wrong choice of material may lead to failure, over or undersized product or expensive items. The choice of materials is thus dependent on suitable properties of the material for each component, their suitability of fabrication or manufacture and the cost.

Load -

Load- The external loads cause internal stresses in the elements and these stresses must be determined accurately since these will be used in determining the component size. Loading may be due to:

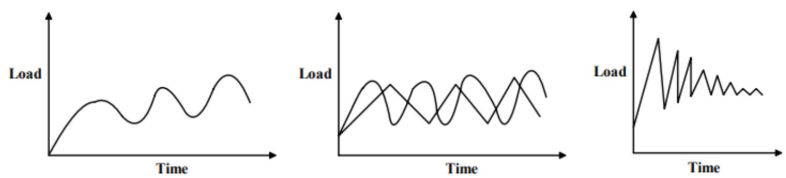
i) Energy transmission by a machine member.
ii) Dead weight.
iii) Inertial forces.
iv) Thermal effects.
v) Frictional forces.

Static load:



Static load

Dynamic load:



Size, shape, space requirements and weight- Preliminary analysis would give an approximate size but if a standard element is to be chosen, the next larger size must be taken. Shapes of standard elements are known but for non-standard element, shapes and space requirements must depend on available space in a particular machine assembly. A scale layout drawing is often useful to arrive at an initial shape and size.
Weight is important depending on application. For example, an aircraft must always be made light. This means that the material chosen must have the required strength yet it must be light. Similar arguments apply to choice of material for ships and there too light materials are to be chosen. Portable equipment must be made light.

Reliability and safety

Reliability is an important factor in any design. A designed machine should work effectively and reliably. The probability that an element or a machine will not fail in use is called reliability. Reliability lies between $0 \le R < 1$. To ensure this, every detail should be examined. Possible overloading, wear of elements, excessive heat generation and other such detrimental factors must be avoided. There is no single answer for this but an overall safe design approach and care at every stage of design would result in a reliable machine.

Safety has become a matter of paramount importance these days in design. Machines must be designed to serve mankind, not to harm it. Industrial regulations ensure that the manufacturer is liable for any damage or harm arising out of a defective product. Use of a factor of safety only in design does not ensure its overall reliability.

Engineering Materials :

Metal → Ferrous, Non-Ferrous

Non-metal

Ferrous
- → Cast iron (1.7 - 3 %) { Fe_3C }
- → wrought iron (99.5 % Fe) → pure iron
- → steel (< 1.7 % C)

Steel

Plain

→ C & other alloying materials are usually present in not more than 0.5 - 1 %.

C01, C14, C45, C70

Alloy

Alloy steel- these are steels in which elements other than carbon are added in sufficient quantities to impart desired properties, such as wear resistance, corrosion resistance, electric or magnetic properties. Chief alloying elements added are usually nickel for strength and toughness, chromium for hardness and strength, tungsten for hardness at elevated temperature, vanadium for tensile strength, manganese for high strength in hot rolled and heat treated condition, silicon for high elastic limit, cobalt for hardness and molybdenum for extra tensile strength. Some examples of alloy steels are 35Ni1Cr60, 30Ni4Cr1, 40Cr1Mo28, 37Mn2. Stainless steel is one such alloy steel that gives good corrosion resistance. One important type of stainless steel is often described as 18/8 steel where chromium and nickel percentages are 18 and 8 respectively. A typical designation of a stainless steel is 15Si2Mn2Cr18Ni8 where carbon percentage is 0.15.

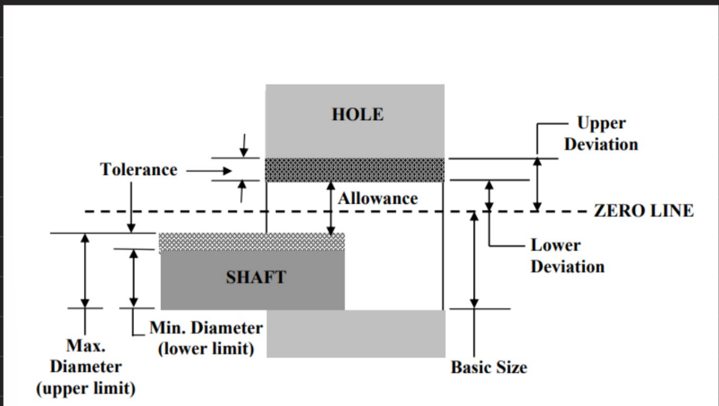


Fig. 1.3.1 Interrelationship between tolerances and limits

Tolerance :

↳ difference b/w maximum and minimum dimensions of a component.

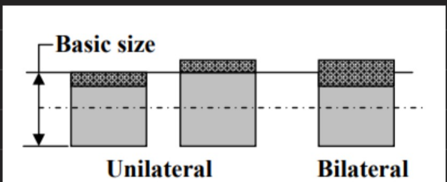


Fig. 1.3.2 Types of tolerance

50^{0}_{-y} , 50^{+x}_{0} , 50^{+x}_{-y}

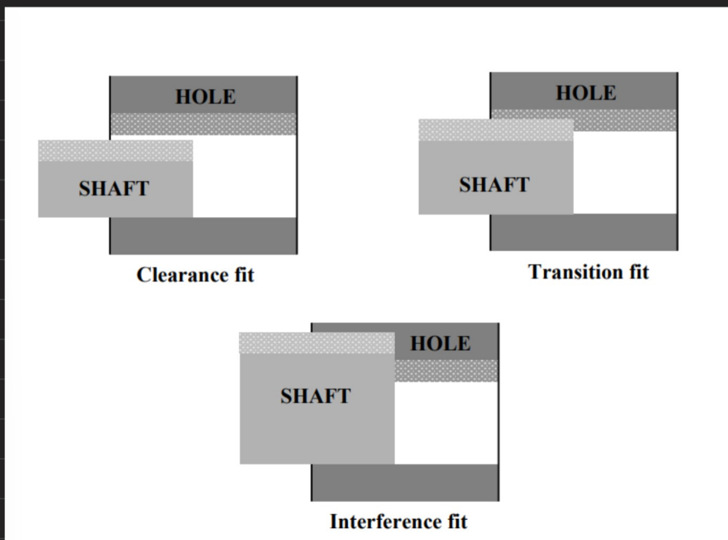
Allowance : Difference of dimensions of 2 mating parts.

Upper deviation : Max psbl dimen − nominal size dimen.

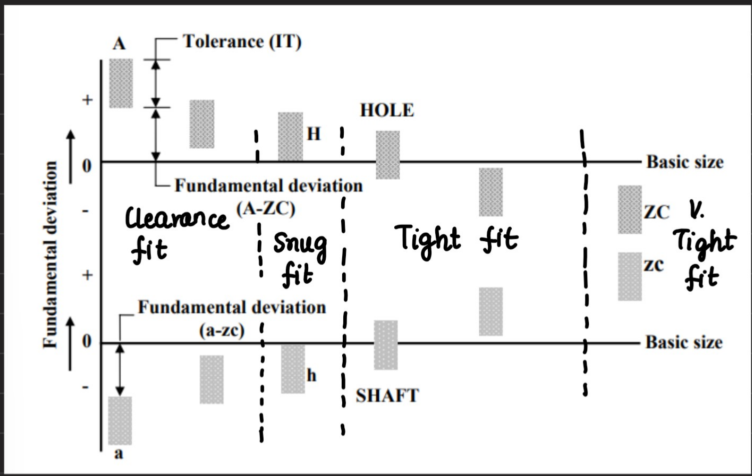
lower deviation : Min psbl dimension − nominal size dimen.

Fundamental deviation : It defines the location of the tolerance zone with respect to the nominal size. For that matter, either of the deviations may be considered.

Fit System :



Standard limit & Fit System :



Fundamental deviation

Hole → A - zc, + H -

Shaft → a - zc, - H +

Standard tolerance : 18 grades

- → IT01
- → IT0
- → IT1 - IT16

eg: 50 H6/g5

50 → nominal size = 50 mm

H → hole deviation (fun) = 0
6 → IT6 (tolerance)

g → shaft deviation (fun.) = -ve
5 → IT5 (tolerance)

Preffered Numbers :

- → standard sizes
- → manufacturer follows a pattern
- → sizes are in GP. with $a = 1$

r (basic)

R5 R10 R20 R40

$r = (10)^{\frac{1}{5}}$ $r = (10)^{\frac{1}{10}}$ $r = (10)^{\frac{1}{20}}$ $r = (10)^{\frac{1}{40}}$

R_{10} , R_{20} and R_{40} : Thickness of sheet metals, wire diameter
R_5 , R_{10} , R_{20} : Speed layout in a machine tool (R_{10} : 1000, 1250,1600, 2000)
R_{20} or R_{40} : Machine tool feed
R_5 : Capacities of hydraulic cylinder

Module - 3 :

Factor of Safety :

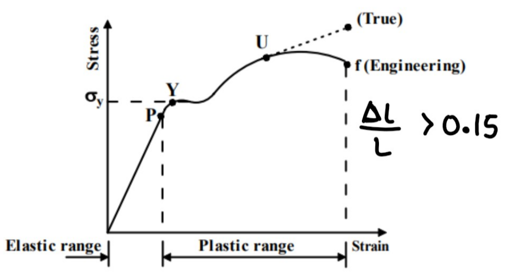
- The load at which the specimen finally ruptures is known as ultimate load and the ratio of load to original cross-sectional area is the ultimate stress.

An allowable stress is set considerably lower than the ultimate strength. The ratio of ultimate to allowable load or stress is known as factor of safety i.e.

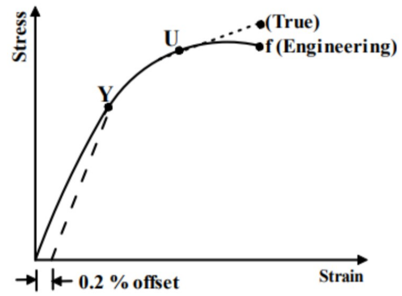
$$\frac{\text{Ultimate Stress}}{\text{Allowable Stress}} = \text{F.S.}$$

The ratio must always be greater than unity. It is easier to refer to the ratio of stresses since this applies to material properties.

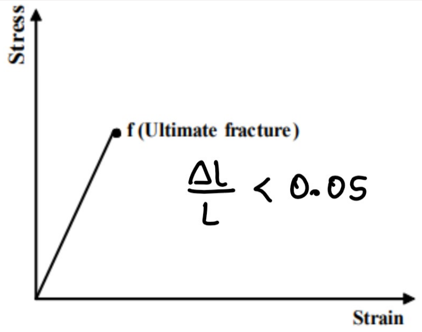
- Yielding - Excessive inelastic deformation
- Fracture - Component tears apart



3.1.3.1F- (a) Stress-strain diagram for a ductile material e.g. low carbon steel.

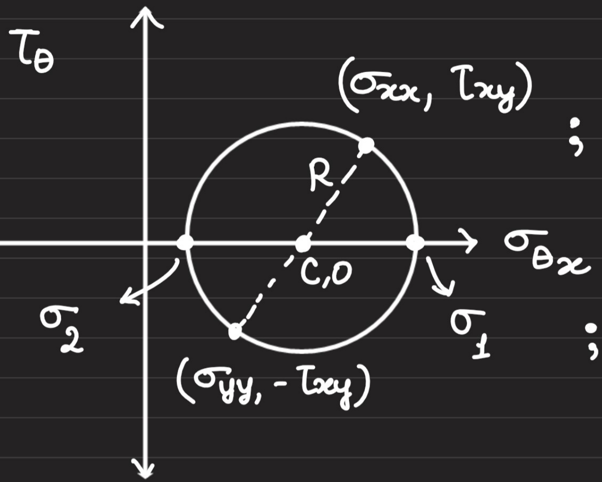


3.1.3.1F- (b) Stress-strain diagram for low ductility.

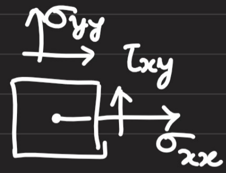


MSS (Tresca Theory) :

at tensile yeild point $\sigma_2 = \sigma_3 = 0$, $\sigma_1 = \sigma_Y$



$$; \underbrace{C = \frac{\sigma_{xx} + \sigma_{yy}}{2}}$$



$$; R = \sqrt{\left(\frac{\sigma_{xx} - \sigma_{yy}}{2}\right)^2 + \tau_{xy}^2}$$

$\sigma_2 = C - R$ & $\sigma_1 = C + R$,

$$\boxed{\tau_{max} = R = \frac{\sigma_1 - \sigma_2}{2} = \frac{\sigma_Y}{2}}$$ at yeild point

Distortion energy theory (Von mises yield)

Strain energy (E_T) $= \frac{1}{2}(\sigma_1 \varepsilon_1 + \sigma_2 \varepsilon_2 + \sigma_3 \varepsilon_3)$

E_v (volumetric S.E) $= \frac{3}{2} \sigma_{av} \varepsilon_{av}$

Distortion energy $= E_d = E_T - E_v$

$$= \frac{(1+\nu)}{3E}(\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1)$$

at yeild point , $E_d = \frac{(1+\nu)}{3E} \sigma_Y^2$

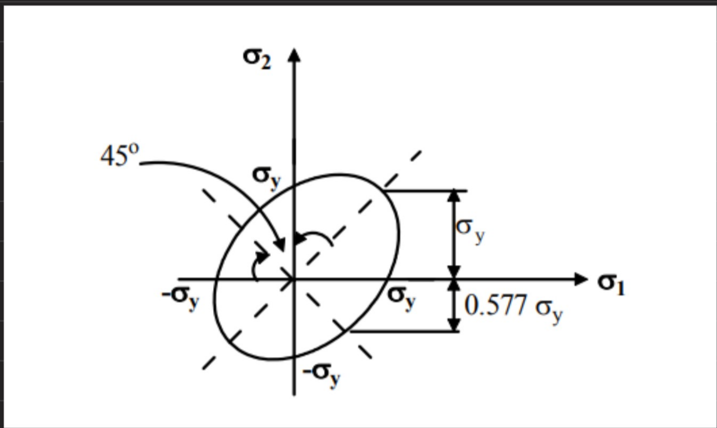
Failure - criteria (Boundary condition)

$$\frac{\cancel{(1+\nu)}}{\cancel{3E}}\left(\sigma_1^2+\sigma_2^2+\sigma_3^2-\sigma_1\sigma_2-\sigma_2\sigma_3-\sigma_3\sigma_1\right) = \frac{\cancel{(1+\nu)}}{\cancel{3E}}\sigma_Y^2$$

$$\Rightarrow (\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2 = 2\sigma_y^2$$

2D , $\sigma_3 = 0$

$$\left\{ \sigma_1^2+\sigma_2^2-\sigma_1\sigma_2 = \sigma_Y^2 \right\}$$



proof:

$$E_d = E_T - E_V$$

$$= \frac{1}{2}(\sigma_1\varepsilon_1+\sigma_2\varepsilon_2+\sigma_3\varepsilon_3) - \frac{3}{2}\sigma_{av}\varepsilon_{av}$$

$$= \frac{(1+\nu)}{3E}\left(\sigma_1^2+\sigma_2^2+\sigma_3^2-\sigma_1\sigma_2-\sigma_2\sigma_3-\sigma_3\sigma_1\right)$$

$$\varepsilon_1 = \frac{1}{E}\left(\sigma_1 - \nu(\sigma_2+\sigma_3)\right)$$

$$\varepsilon_2 = \frac{1}{E}\left(\sigma_2 - \nu(\sigma_1+\sigma_3)\right)$$

$$\varepsilon_3 = \frac{1}{E}\left(\sigma_3 - \nu(\sigma_1+\sigma_2)\right)$$

BEM

Q.1: A shaft is loaded by a torque of 5 KN-m. The material has a yield point of 350 MPa. Find the required diameter using

(a) Maximum shear stress theory

(b) Maximum distortion energy theory

Take a factor of safety of 2.5.

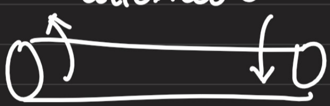
Sol^n:

$T = 50$ KN-m

$\sigma_Y = 350$ MPa

(a) $\tau_{max} = \dfrac{\sigma_Y}{2} = 175$ MPa

allowable



$\tau = \dfrac{Tr}{J}$, $J = \dfrac{\pi r^4}{2}$

$$\tau_{allo} = \frac{175 \times \cancel{10^6}\,10^3}{2.5} = \frac{5\times \cancel{10^3} \times d}{\cancel{2}\,\dfrac{\pi d^4}{\cancel{32}\,16}}$$

$$\Rightarrow d^3 = \frac{80 \times 2.5}{175 \times 10^3 \times \pi}$$

$(d = 0.071 \text{ m})$ Ans

Q2 (a) , $\sigma_Y = 400$ MPa

$T = 200 \times 10^3$

$(C = 82.5)$, $(R = 46.97$ MPa$)$

$\{ \sigma_1 = 129.47073 \}$

$$\tau_{max\ allow} = \left| \frac{\sigma_1 - \sigma_2}{2} \right| = R$$

$$\tau_{max,Y} = 200 \text{ MPa}$$

$$FS = \frac{200}{46.97} = 4.258 \}$$ Ans

$$* \quad \boxed{\frac{\sigma_1 - \sigma_2}{2} = \frac{\sigma_Y}{2\text{xF.S}}}$$

$$* \quad \boxed{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2 = 2(\sigma_Y / \text{F.S})^2}$$

Bearing :

Rolling contact bearing

Ball bearing

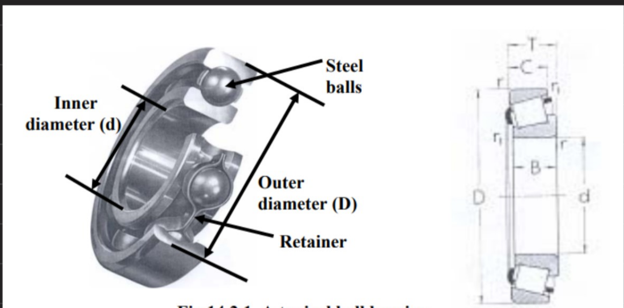
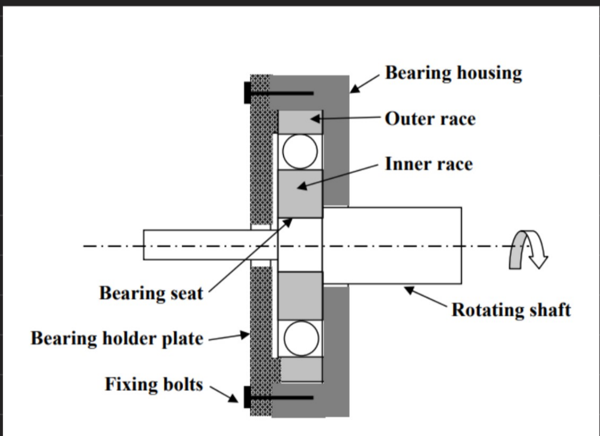


Fig.14.2.1 A typical ball bearing



single row, deep groove ball bearing.

Rolling contact bearing selection :-

i) Rating life – Rating life is defined as the life of a group of apparently identical ball or roller bearings, in number of revolutions or hours, rotating at a given speed, so that 90% of the bearings will complete or exceed before any indication of failure occur.

L_{10} : time after which 10% bearing fail

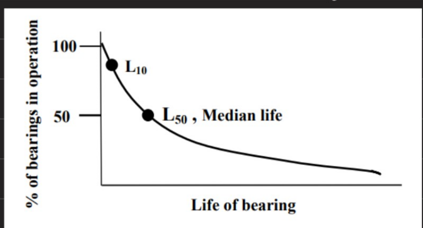


Fig. 14.2.9 Definition of life of rolling contact bearing

$$C = P(L)^{1/a}, \quad a = \text{const}$$

$\underbrace{C}$ ↑ Basic/Dynamic load rating

a = const: ball bear. 3; roller bear. 10/3

for 2 identical bearings

$$\Rightarrow P_1 (L_1)^{\frac{1}{a}} = P_2 (L_2)^{\frac{1}{a}}$$

C → represents load carrying capacity for 1M revolutions.

Equivalent radial load

The load rating of a bearing is given for radial loads only. Therefore, if a bearing is subjected to both axial and radial load, then an equivalent radial load is estimated as,

$$P_e = VP_r \quad \text{or}$$
$$P_e = XVP_r + YP_a$$

(14.2.3)

Where,

P_e : Equivalent radial load
P_r : Given radial load
P_a : Given axial load
V : Rotation factor (1.0, inner race rotating; 1.2, outer race rotating)
X : A radial factor
Y : An axial factor

V = rotation factor: inner (1); outer (1.2)

Q.

Select a deep groove ball bearing and provide the basic design number for the following parameters.

Design (desired) load	= 1.85 kN
Service factor	= 1.2
Speed of the shaft	= 300 RPM
Desired life	= 30 kH
Desired reliability	= 0.99
Diameter of the shaft	= 55 mm

[Consider the constants, a = 3 and b = 1.34]

Solⁿ:

$$L_{10} = \text{Speed of shaft} \times \text{life}$$
$$= \frac{9000}{10^6} \times 1000$$
$$= 9$$

$$C = (1.85) \times 1000 \times 9^{\frac{1}{3}}$$
$$= 3848.155 \text{ N}$$
$$= 392.668 \text{ kg}$$

The dimensions of a machine component are given in the Figure 4. All the dimensions are mentioned on the **x-y** plane and a force F is acting (see figure) out of the plane (along the +ve **z** axis). The component is made of steel 30C8 (yield strength 400 N/mm²) and the factor of safety is 2.5. Using the Distortion Energy (Von Mises) theory and Maximum Shear Stress (Tresca) theory of failure, determine the diameter **d** at the section **PP'**.

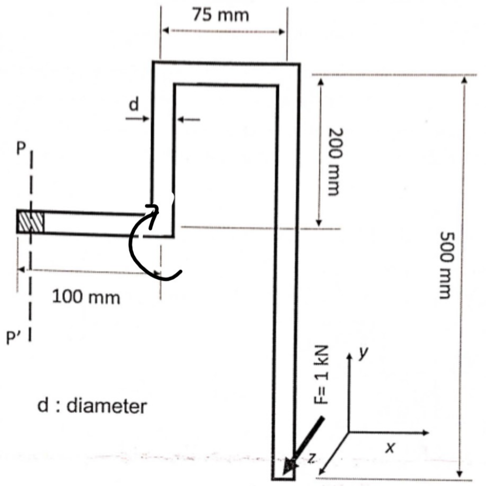


Figure 4

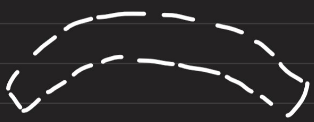
$\sigma_Y = 400 \ N/mm^2$

$T = 175 \times 1000$

$= 175000$ N-mm

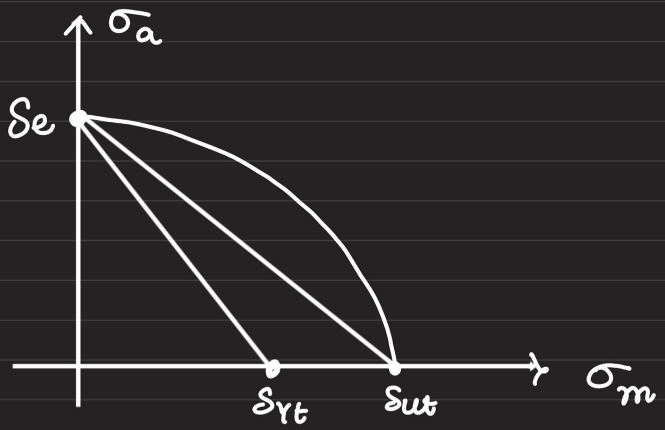
1 KN $\qquad M = 1000 \times 100$

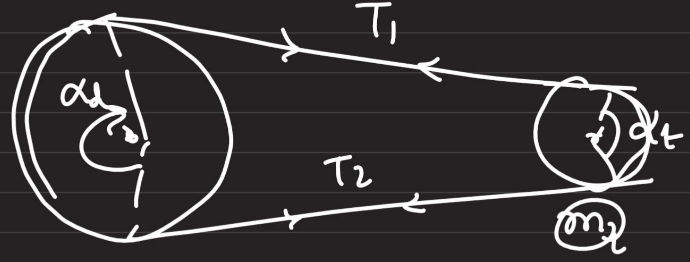
$\sigma = \frac{My}{I}, \quad \tau = \frac{VQ}{It}$



$$\tau_{max} = \frac{Tr}{J} = \frac{2T\cancel{d/2}}{\pi (d/2)^{\cancel{4}3}} = \frac{16T}{\pi d^3}$$

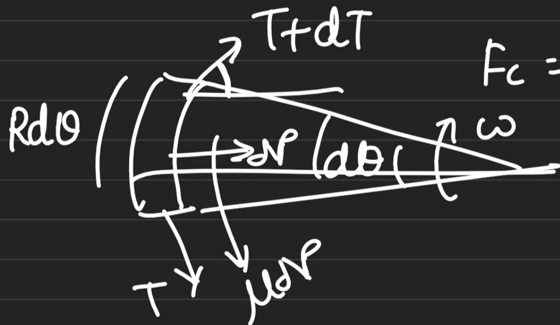
$$\underbrace{f_{critical}}_{\text{critical speed.}} = \frac{1}{2\pi}\sqrt{\frac{g(W_1\delta_1 + W_2\delta_2 + \cdots)}{(W_1\delta_1^2 + W_2\delta_2^2 + \cdots)}}$$



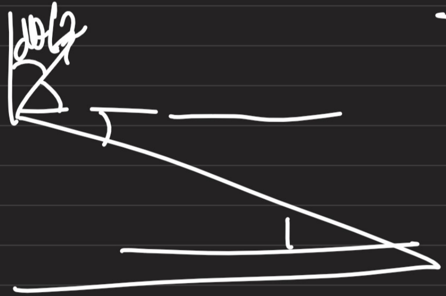


$\pi + 2\theta$ $\qquad\qquad$ $\pi - 2\theta$

$\boxed{\alpha_d = \pi + 2\theta}$ $\qquad$ $\alpha_t = \boxed{\pi - 2\theta}$



$$F_c = (m_l)\, R\,d\theta\, \omega^2 R$$
$$= m_l \omega^2 R^2 d\theta$$
$$= m_l v^2 d\theta$$



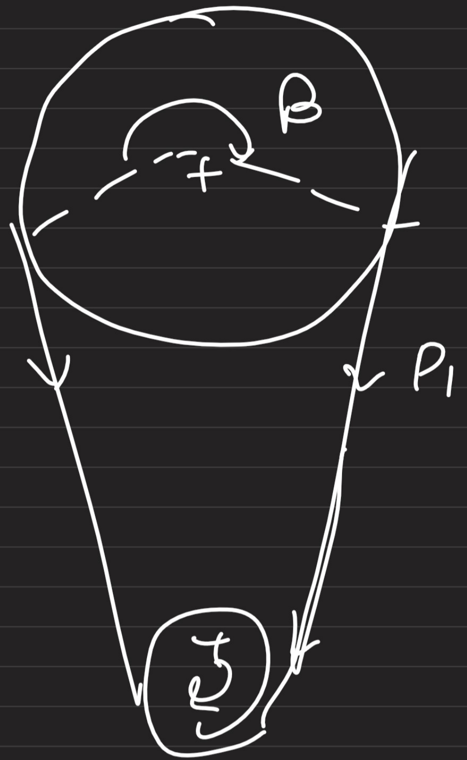
$$\Longrightarrow \cancel{2}T\frac{d\theta}{\cancel{2}} + dN \qquad \frac{-dT}{\mu}$$
$$= m_l v^2 d\theta$$

$$(\cancel{T}+dT)\cos\frac{d\theta}{2} - \cancel{T\cos\frac{d\theta}{2}} - \mu dN = 0$$

$$\boxed{dT = -\mu dN}$$

$$\boxed{\frac{dT}{T - m_l v^2} = -\mu\, d\theta}$$

$$\boxed{\frac{T_1 - m_l v^2}{T_2 - m_l v^2} = e^{\mu\alpha}}$$



$$\boxed{\frac{P_1}{P_2} = e^{\mu\beta}}$$