MACHINE DESIGN

DESIGN OF MACHINE ELEMENT :

Topics : * 1) Review of stress analysis

2) Theory of failures

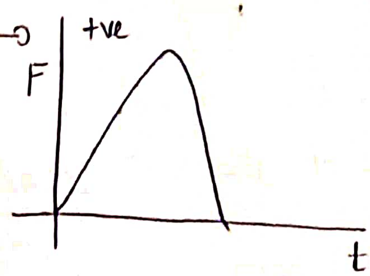
3) MD in continuation of strength of material.

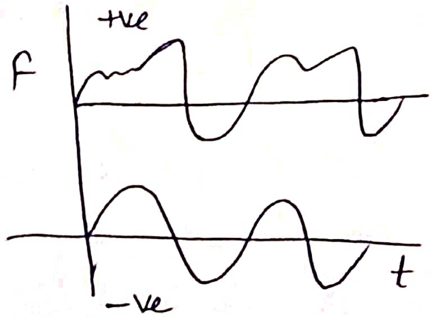
4) * General prin. & proc. of MD.

* FOS and service factor.

* Design of shaft under torsion, bending, axial & Combined Load.

* Impact load →





← Fatigue Load *

→ MoI ↑ deflection ↓

→ Intersec. of Neutral layer & Cross sec - sec. → Neutral Axis.

→ Transverse moment will Create → BM

→

Types of M/c Design :

1)* Adaptive M/c designer → M/c which is already been designed

(Bigger m/c cannot be used for doing small task due to sensitivity issue)

2) Modified M/c design :- When some parts are introduced into the M/c.

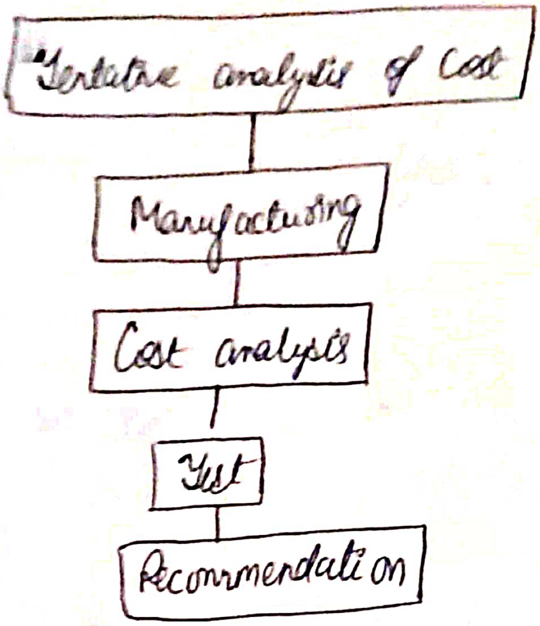
3) New M/c design →

NEED

1. Qualitative statement
2. Quantitative statement
3. Mathematical statement

1. Kinematic Synthesis
2. Kinematic Analysis
3. Force Analysis
4. Material selection

1. Set suitable dimension
2. Selection of failure modes & failures.
3. Suitable prevention of failures.
4. Manufacturing Consideration.
5. Production drawing



(A) For the bulk of the body :

1). Strength
2) Deflection or deformation
3) Weight
4). Size & shape

(B) For the surface of Component :

1). Wear
2) Lubrication
3). Friction
4) Frictional heat generation

(C) Cost

<u>Modern consideration</u> :

1) Safety
2) Ecology
3). Quality of life

<u>Miscellaneous Consideration</u> :

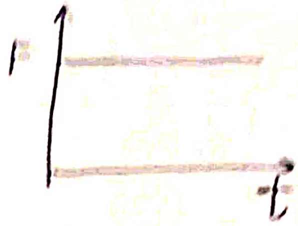
1) Reliability & maintainability

2) Athletics

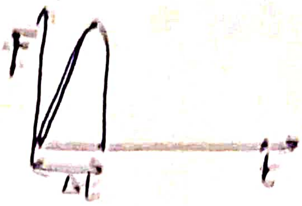
3) ~~Economic~~ Ergonomic Consideration

*(Isotropic , anisotropic ; Unisotropic)

→ Static force : Mag. , dirn, point of app. & time invariant.



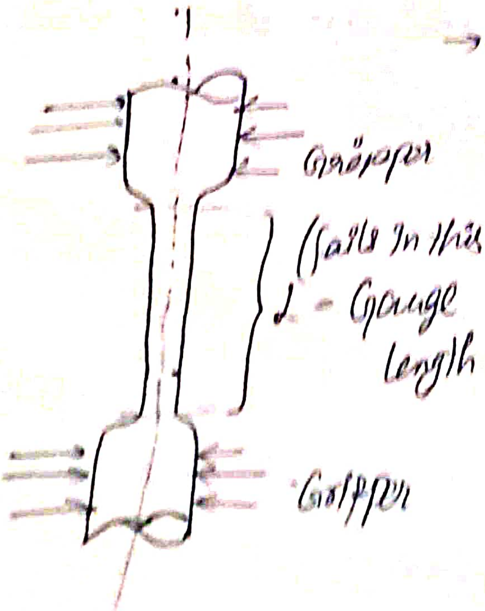
Dynamic force → If time of application of force is very low.

Impact load → 

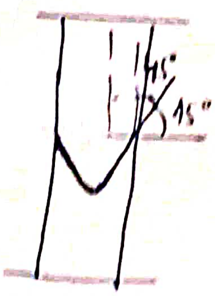
(Δt is very small)

→ forces are static.

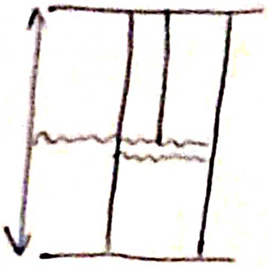
→ failure will take place where dia. is min.



d → gauge dia.



Cup-Cone type failure.

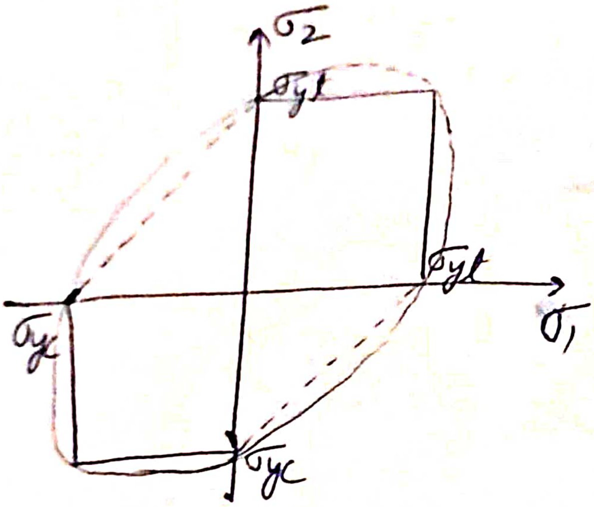


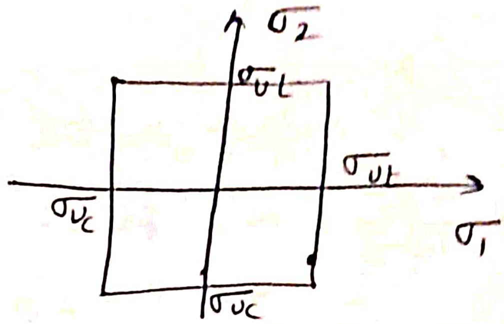
Fragmentation type

For ductile material,

1) Max normal stress th.

2) Distortion th.





Uncertanities :

→ Due to as molecular str. or molecular arrangement are diff. of two same type body.

1). Degree of Uncertanity in load.

2) Degree of ~~uncertainity~~ uncertainity in material prop. (F.S. material)

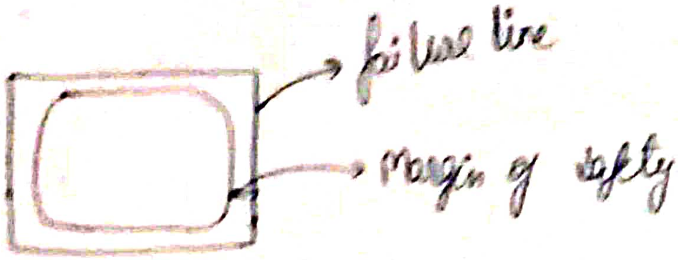
3) Degree of uncerteinty in stress calculation method. ($F.S._{stress}$)

4). Degree of uncertainty in theories of failure. ($F.S._{failure\ th}$)

5). Releibility requirement. ($F.S._{reliability}$)

6). Manufacturing tolerance in geometry. ($F.S._{manufacturing}$)

→ For diff. type of users margin of safety is diff.



→ How much margin of safety is given is totally depends upon "Engineering understanding".

<u>Margin of Safety</u> = Failure stress − Allowable stress

→ Ex- steel load = 200 MPa
we are taking stress upto 150 MPa } ⇒ MOS = 50 MPa

→ $MOS = \cancel{\text{Allowable}}\ \text{Failure stress}\left(1 - \dfrac{\text{Allowable stress}}{\text{Failure stress}}\right)$

→ $MOS = \text{Failure stress}\left(1 - \dfrac{1}{\dfrac{\text{Failure stress}}{\text{Allowable stress}}}\right)$

→ $MOS = \text{Failure stress}\left(1 - \dfrac{1}{\text{Factor of safety}}\right)$

→ $\boxed{\text{Factor of safety} = \dfrac{\text{Failure stress}}{\text{Allowable stress}}}$

→ FOS ↑ allowable stress ↓

→ $\boxed{\text{Fe E 200}}$ → yield strength → stress

→ Safe Design → High FOS

→ Near Optimum design → Less FOS

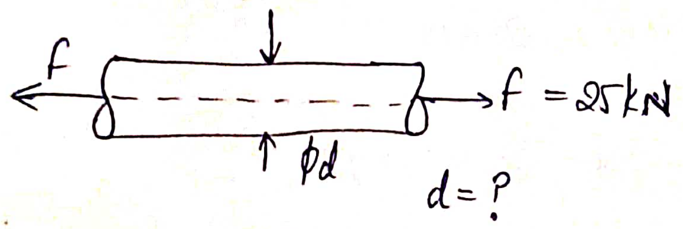
⇒ FOS = ~~4~~ 1 − 3

~~Near optimum design~~
Safe design

Near optimum design

→ $F.S. = F.S._{material} + F.S._{stress} + F.S._{failure\ th.} + F.S._{Reliability} + F.S._{Manufacture}$

→ Margin of Safety = Failure stress − Allowable stress

Q. 

Calculate design dim. of load.

FeE 200

⇒ $\sigma_{yt} = \sigma_{yc} = \sigma_{yield} = 200\ MPa$

⇒ F.S. = 3−4

⇒ σ Allowable stress = 66.67 MPa

" " = 50 MPa

⇒ Let us take working stress as 65 MPa.

⇒ $65 \times 10^6 = \dfrac{25 \times 10^3 \times 4}{\pi \times d^2} \Rightarrow d^2 = \dfrac{1}{650\pi} = 22.129\ mm$

$d = 24\ mm$

$\Rightarrow \sigma = \frac{25 \times 10^3 \times 4}{\pi \times (24 \times 10^{-3})^2} = \frac{10^5}{\pi \times 10^{-6} \times (24)^2} = 55.26\ MPa$

$\Rightarrow \sigma = \frac{\sigma_Y}{FoS} \Rightarrow FoS = \frac{200}{55.26} = 3.619$ $\left\{ \begin{array}{l} \text{for } d = 22\ mm \\ F.S. = 3.04 \end{array} \right\}$

$\rightarrow F.S_{material} = 1.2$, $F.S_{manufacturing} = 1.1$

$\Rightarrow F.S_{stress\ load} = 1.2$, $F.S._{reliability} = 1.4$

$\Rightarrow F.S_{failure\ th.} = 1.2$

$\Rightarrow F.S. = 1.2 \times 1.2 \times 1.2 \times 1.1 \times 1.4 = 2.66$

$\rightarrow$ The above F.S. range is for safe design.

$\Rightarrow \tau_{max} = max. \left\{ \frac{|\sigma_1 - \sigma_2|}{2}, \frac{|\sigma_2 - \sigma_3|}{2}, \frac{|\sigma_3 - \sigma_1|}{2} \right\}$ (Von - Mises)

$= \frac{\sigma_{max.} - \sigma_{min.}}{2}$

$\Rightarrow \sigma_{max.} = max (\sigma_1, \sigma_2, \sigma_3)$

$\Rightarrow \sigma_{min.} = min. (\sigma_1, \sigma_2, \sigma_3)$

$\rightarrow$

$\Rightarrow$ If $\sigma_1 > \sigma_2 > \sigma_3$ & $\sigma_3 = 0$

then max. $= \sigma_1$ & min. σ_3

$\Rightarrow$ If $\sigma_1 > \sigma_2 > \sigma_3$ & $\sigma_2 = 0$

then max. $= \sigma_1$ & min. $= \sigma_3$

Expression for Von-Misses th. $\rightarrow$

$$\rightarrow \sigma_e = \sqrt{\frac{(\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_3-\sigma_1)^2}{2}}$$

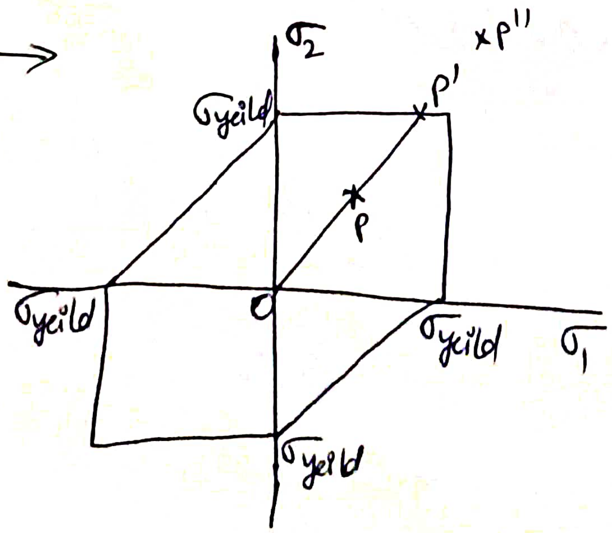
$\rightarrow$ $\sigma_{yeild} \geq \sigma_e$ [when $\sigma_y = \sigma_e \Rightarrow$ it is on the virtue of failure]

$\rightarrow$ $\tau_{yeild} \geq \tau_{max.}$

$\Rightarrow$ $\tau_{yeild} = \frac{\sigma_{yeild}}{2}$

$\Rightarrow$ $\tau_{yeild} \geq 0.6\, \sigma_{yeild} = \frac{\sigma_{yeild}}{\sqrt{3}} = 0.577\, \sigma_{yeild}$

$\rightarrow$



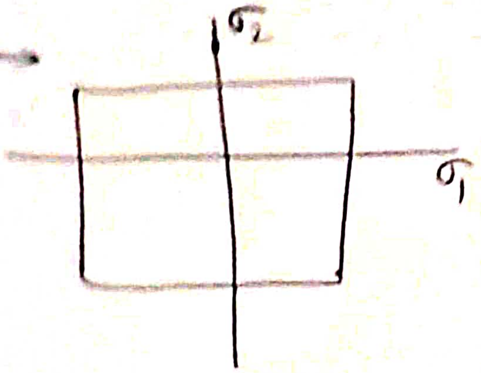
$\Rightarrow$ $\boxed{F.S. = \frac{\sigma_{yeild}}{\sigma_e}}$

$\Rightarrow$ $F.S. = \frac{OP'}{OP}$

$\Rightarrow$ $F.S. = \frac{OP''}{OP}$

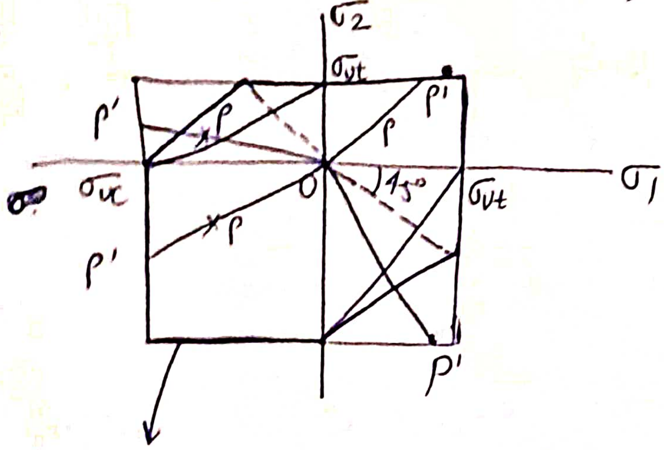
→ In Von-Mises,

→ All lines are st. lines & eqn's are linear

→ 

$\sigma_{max.} \rightarrow \sigma_{ut}$

$\sigma_{min.} \rightarrow \sigma_{uc}$

→ 

(This th. is less Conservative)

→ F.S. is fr. ⇒ Allow. stress > fail. stress (fraction)

$\Rightarrow F.S. = \frac{OP'}{OP}$

⇒ For brittle mat. → F.S. ↑ (is high)

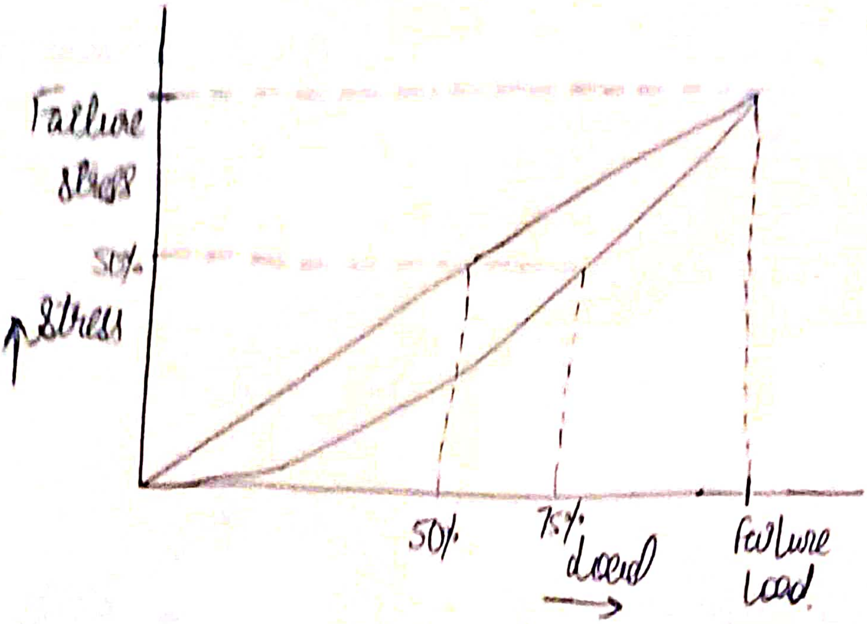
⇒ Compress. load is pre-dominent → use of brittle mat. (Preffarable)

→ $F.S. = \frac{\text{failure stress}}{\text{Allowable stress}} = \frac{\text{failure stress}}{\text{working stress}}$

→ $F.S. = \frac{A \times \text{failure stress}}{A \times \text{Allowable stress}} = \frac{A \times \text{failure stress}}{A \times \text{working stress}}$

Design over load

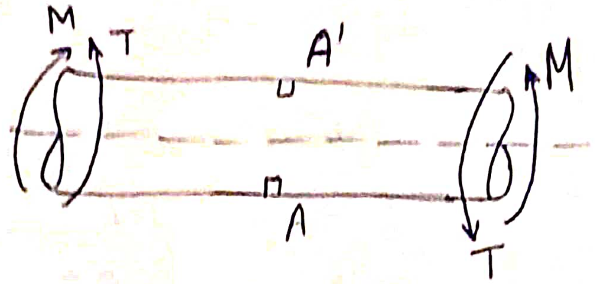
Normal load

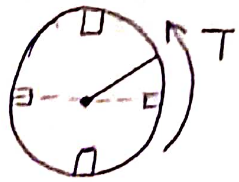


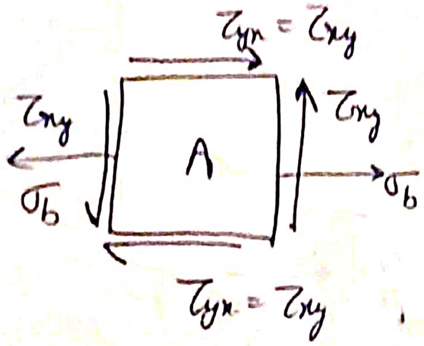
→ For designer point of view, stress based F.S. is more imp.

(Conservative th. → Tresca's th.)

→







$$\tau_{yx} = \tau_{xy} = \frac{16T}{\pi d^3}$$

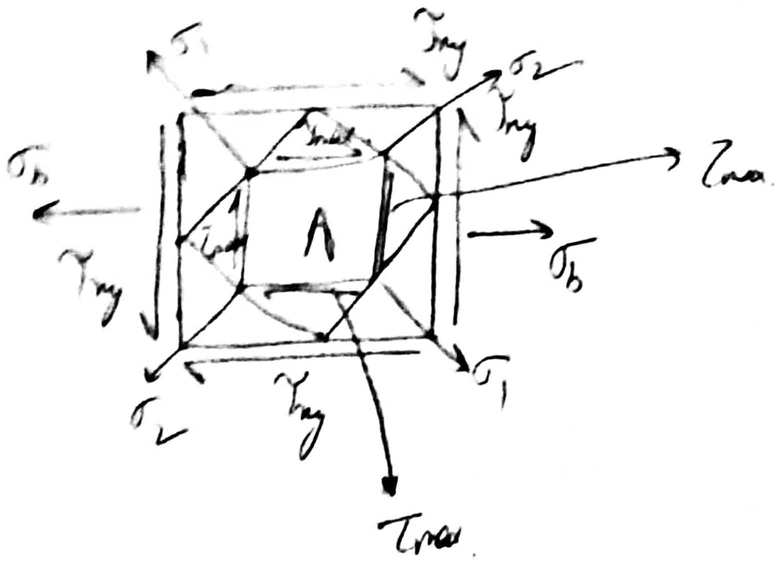
$$\sigma_b = \frac{32M}{\pi d^3}$$

$$\tau_{max.} = \sqrt{\left(\frac{16M}{\pi d^3}\right)^2 + \left(\frac{16T}{\pi d^3}\right)^2}$$

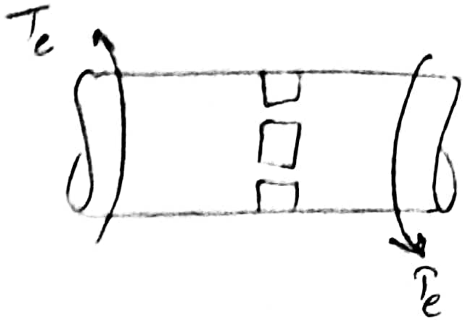
$$\Rightarrow \quad \tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2+T^2} = \frac{16T_e}{\pi d^3}$$

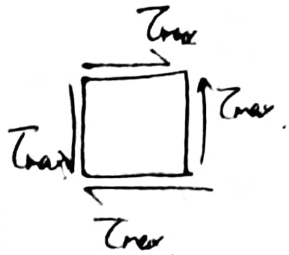
$$\Rightarrow \quad T_e = \sqrt{M^2+T^2}$$

↳ equivalent torque



$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{M^2+T^2}$$
$$= \frac{16T_e}{\pi d^3}$$

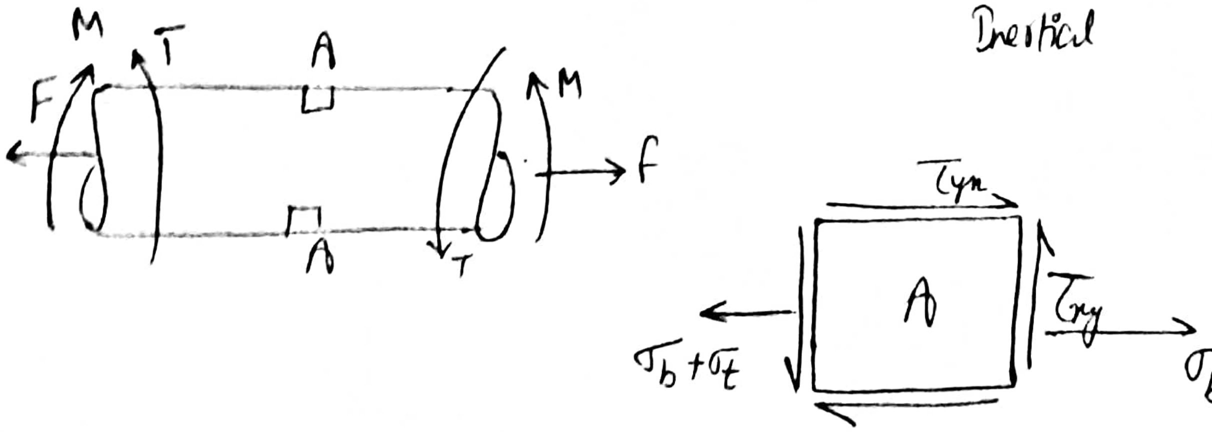




Mass

Inertial

grav.



$$\sigma_b = \frac{32M}{\pi d^3}$$

$$\sigma_b + \sigma_t$$

$$\Rightarrow \sigma_t = \frac{f}{\frac{\pi}{4}d^2} = \frac{4f}{\pi d^2} = \frac{4\times 8\times fd}{\pi d^3\times 8} = \frac{32\,fd}{\pi d^3\times 8}$$

$$\Rightarrow \sigma_x = \sigma_b + \sigma_t = \frac{32M}{\pi d^3} + \frac{32fd}{\pi d^3\times 8} = \frac{32}{\pi d^3}\underbrace{\left(M+\frac{fd}{8}\right)}_{M' \text{ — Modified Bending Moment}}$$

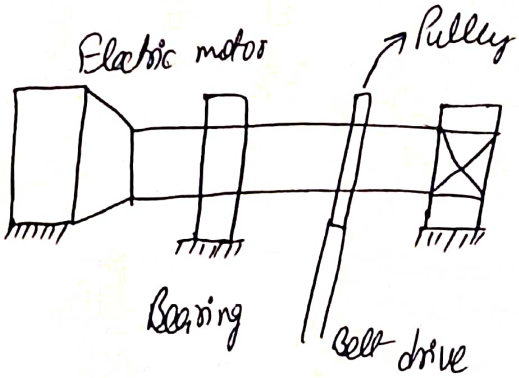
RHS $\Rightarrow$ put $d=0$ $\Rightarrow$ $f = \checkmark$

$$\Rightarrow [\tau] = \frac{\tau_{yield}}{F.S.}$$

$$\tau_{max.} \leq [\tau]$$

$$\frac{16 T_e}{\pi d^3} \leq [\tau]$$

$$d^3 \geq \frac{16 T_e}{\pi [\tau]} \Rightarrow d \geq \sqrt[3]{\frac{16 T_e}{\pi [\tau]}}$$



$$\rightarrow \quad d \geq \sqrt[3]{\frac{16 T_e}{\pi [\tau]}} \quad , \quad d_o \geq \sqrt[3]{\frac{16 T_e}{\pi [\tau] f}}$$

$$\Rightarrow T_e = \sqrt{M'^2 + T^2} \qquad f = 1 - k^4 \quad \text{where } k = \frac{d_i}{d_o}$$

$$M' = M + \frac{Fd}{8} \qquad T_e = \sqrt{M' + T^2}$$

$$M' = M + \frac{F d_o (1 + k^2)}{8}$$

 → By going inward Bending stress ↓

k_m → shock & fatigue factor

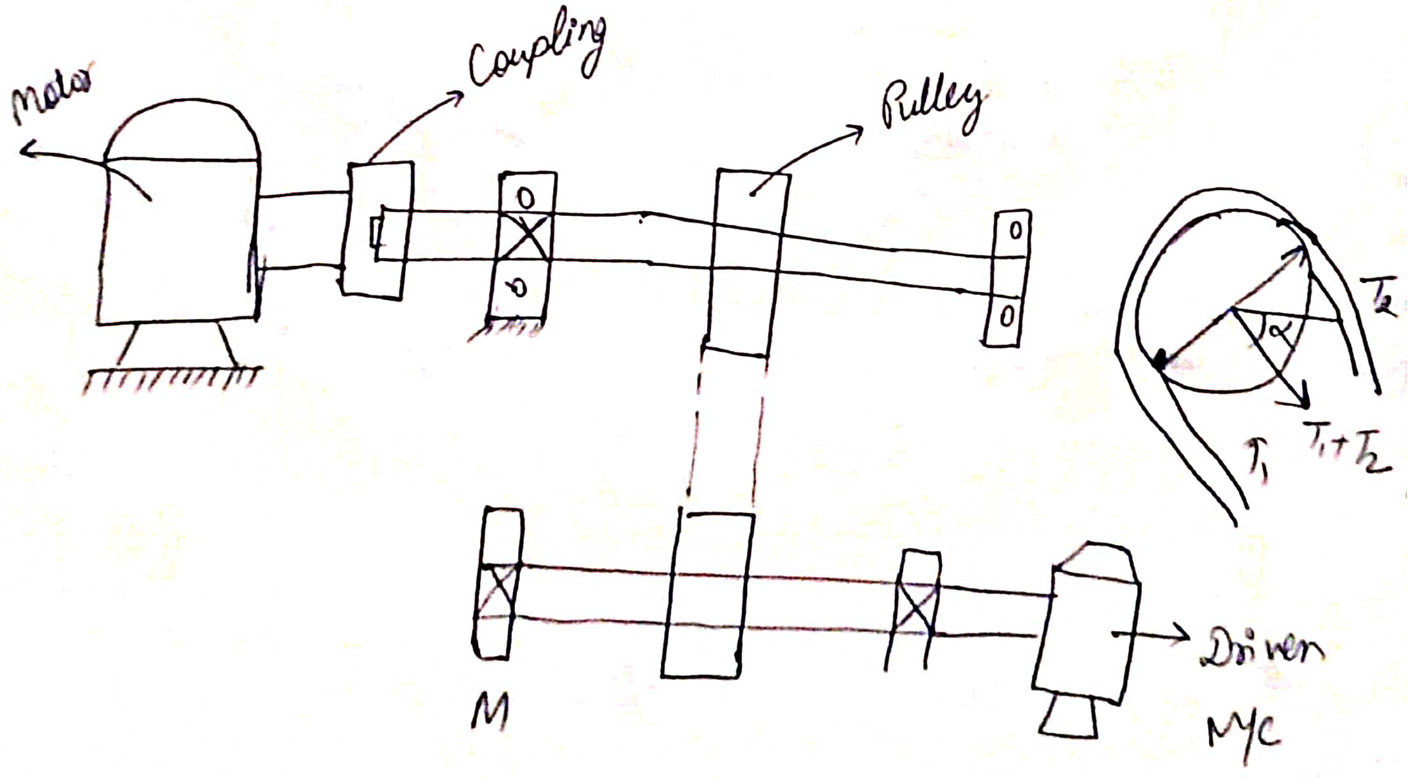
⇒ K_t & $\alpha.F$ (For no axial force → α or $f = 0$)

$\alpha.F$ → 1/0 → Tensile load

Service factor :

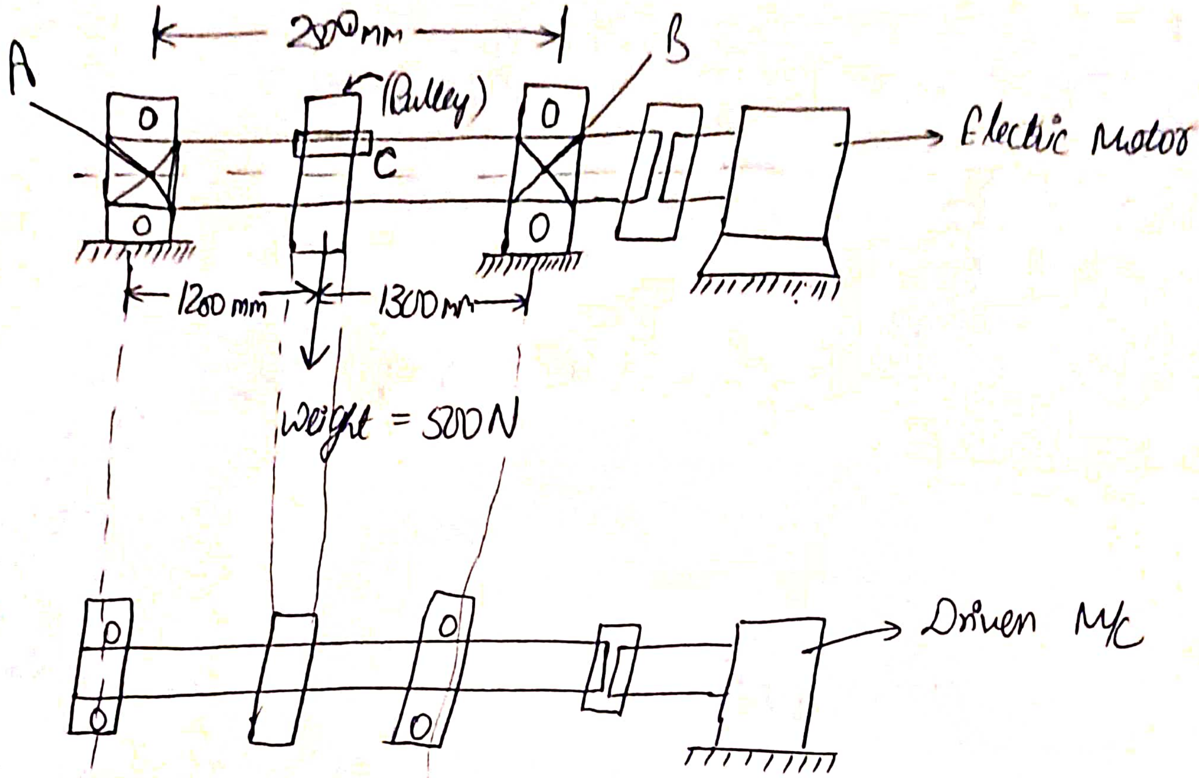
→ $T_{avg.} = \frac{9550 \times [kW]}{rpm} \times 10^3$ N-mm

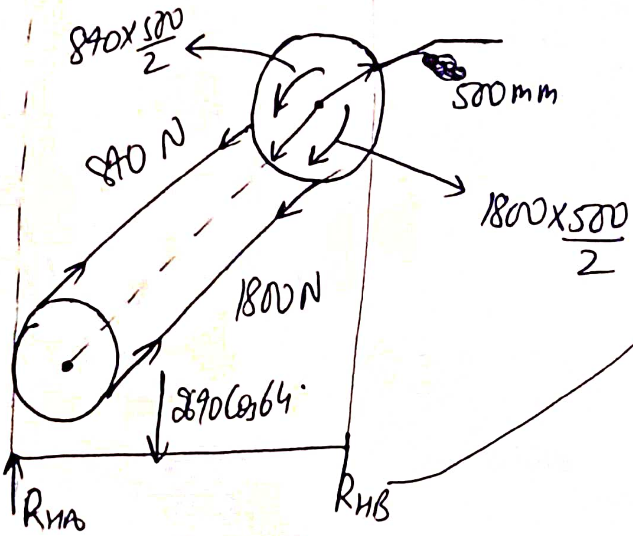
→ $C_s = \frac{T_{max}}{T_{avg.}}$

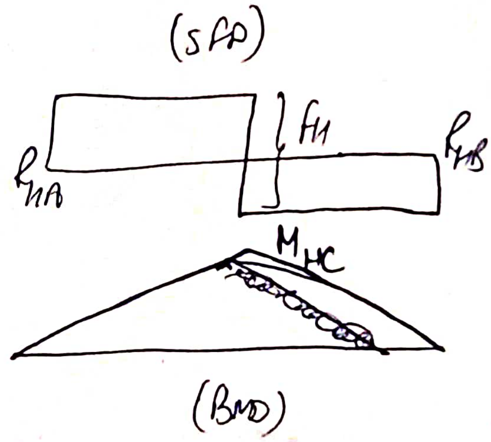


SHAFT DRIVE

Draw the Sketch i.e Space dia.







$\Rightarrow \quad T = (1800 - 840) \times \frac{500}{2}$ ← (for symm. dia if there is diff. in dia. then Cos( ) term will come)

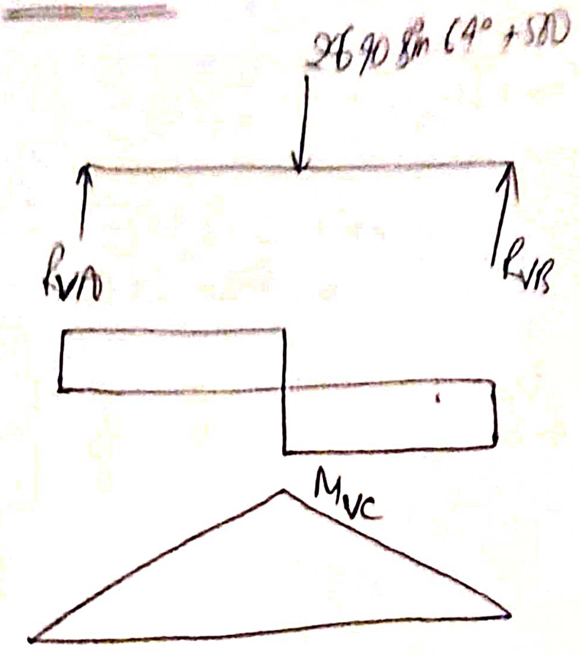
$\Rightarrow \quad F = 2640 \text{ N}$

$\Rightarrow \quad F_H = 2640 \cos 64^\circ \text{ N}$

$F_V = 2640 \sin 64^\circ \text{ N}$

$+ 500$ (weight of pulley)

for ver →



$M = \sqrt{M_{HC}^2 + M_{VC}^2}$

$T_e = \sqrt{(K_m M)^2 + (K_t T)^2}$

$T = 240\ 000$

$M_{HC} = F_{HA} \times 1200$

$= \frac{}{2500}$

$d \geq \sqrt[3]{\frac{16\ T_e}{\pi[\tau]}}$

$F_{HA} + F_{HB} = 1157.3$

$F_{HA} \times 2500 = 1157.3 \times 1300$

~~3105589 N-mm~~

$T_e = 305589.977$ N-mm

$F_{HA} = 601.79$ N

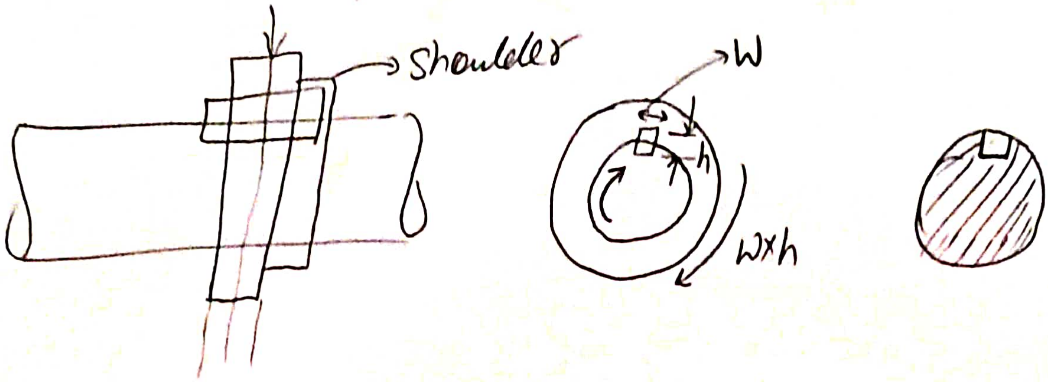
$F_{HB} =$

$[\tau] = \frac{\tau_y \rightarrow (\sigma_y/2)}{FS \rightarrow (2.5)}$

FeE200 ⇒ $\sigma_y = 200$

$d = d \times 1.1$

Splined shaft → M/C shaft



$$d \geq \sqrt[3]{\frac{16T_e}{\pi[\tau]\times 0.75}} = \sqrt[3]{\frac{16T_e}{\pi[\tau]}} \times \underbrace{\sqrt[3]{\frac{1}{0.75}}}_{\approx 1.1}$$

(i.e. 10% rise)

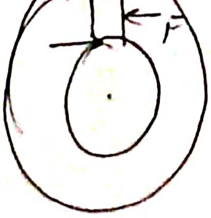
$$\Rightarrow d_o \geq \sqrt[3]{\frac{16T_e}{\pi[\tau]f}} \quad , \quad T_e = \sqrt{(K_m M')^2 + (K_t T)^2}$$

$$M' = M \quad (F = 0)$$

→ when force (F) is there then it is modified,

→ $M \to M + \frac{F\alpha d_o(1+k^2)}{8}$ (for hollow → d_o; for solid → d)

→ α = Column action factor
It signifies column effect.



$F \times d/2 = T_{key}$

$$\boxed{\frac{T_d \times 2}{d_s} = F_t}$$