# NEW CHAPTER

## Gears [spur Gear]

Addendum 'a' = 1m

Dedendum 'd' = 1.157 m

clearance 'c' = .157 m

Face width 'b' = 10 m

The aim of this Topic is to determine Module of gear.

$\boxed{P_c = \pi m}$

$P_d = \frac{1}{m}$

$P_c \cdot P_d = \pi$

$m = \frac{D}{z}$ ← no. of Teeth

T = Torque

## CIRCULAR PITCH

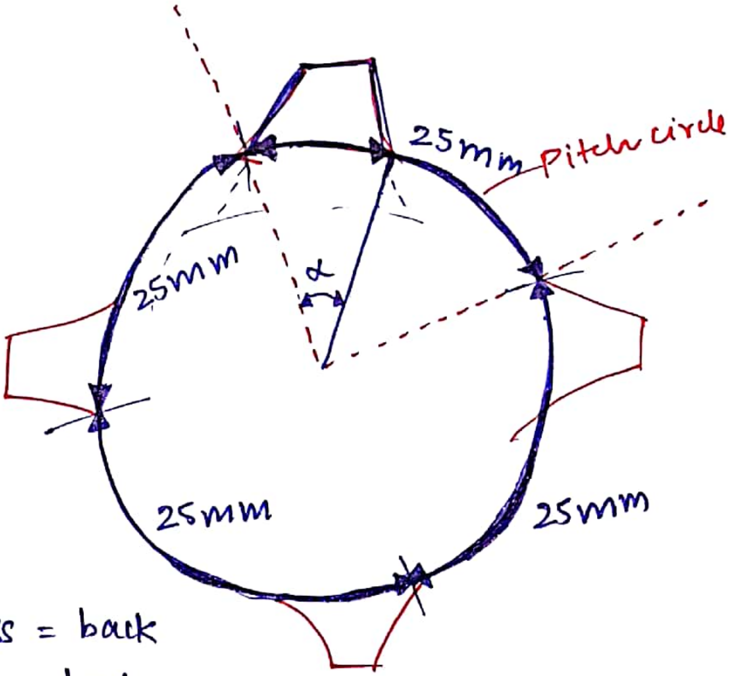
$P_c = \pi m = \frac{\pi D}{z}$

eg :-

$\pi D = 100\,mm$

$z = 4$

$P_c = 25\,mm$



Tooth thickness + Tooth space = $P_c$

Tooth space - Tooth thickness = back lash

Tooth space ≈ Tooth thickness

$$\boxed{\text{Tooth thickness} = \text{Tooth Space} = \frac{P_c}{2}}$$

when two gears are meshing together, their circular pitch must be equal.

$$\boxed{\alpha = \frac{360}{2 \times Z}}$$

$\Downarrow$

Angle cover by Thickness /space on centre.

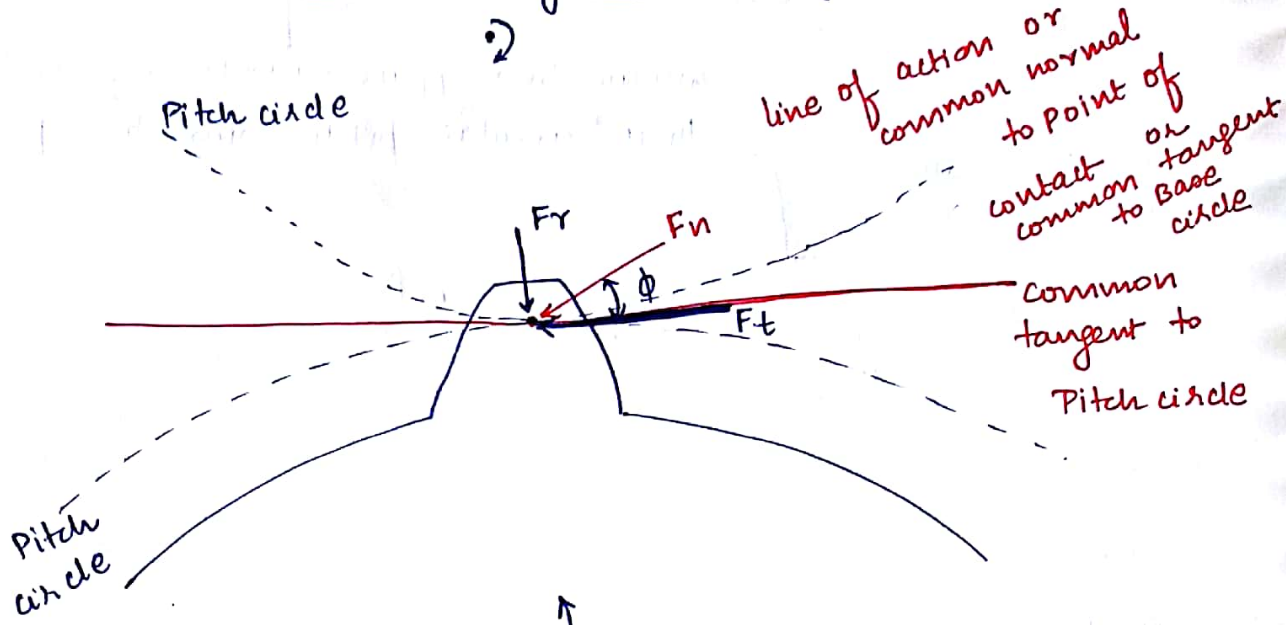
$P_{c_1} = P_{c_2}$

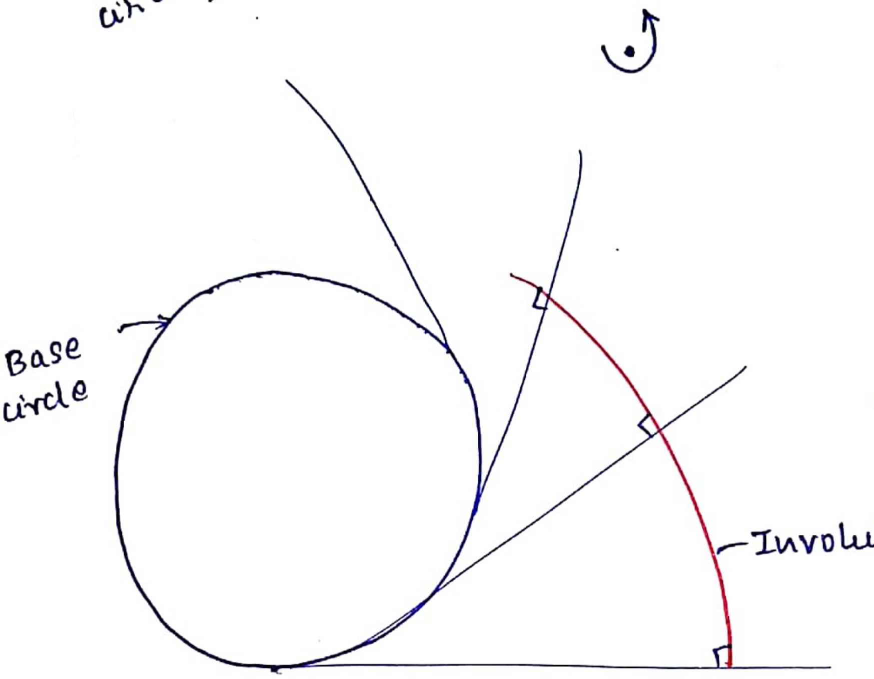
$\pi m_1 = \pi m_2$

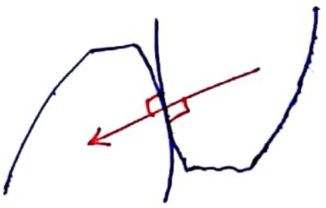
$$\boxed{m_1 = m_2}$$

15/12/2016

# Force analysis used for gears :-







$$\boxed{\phi \leq 25^\circ}$$

$\phi$ = Pressure angle

$$F_t = F_n \cos\phi$$

$$F_n = \frac{F_t}{\cos\phi}, \quad F_r = F_n \sin\phi$$

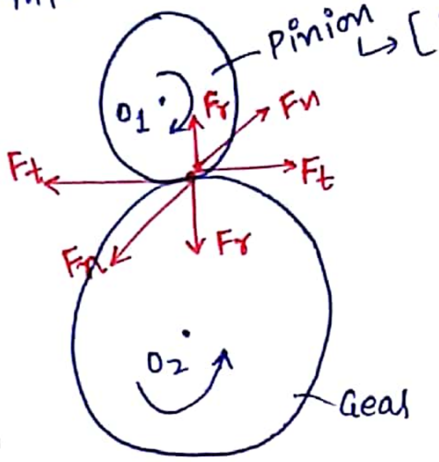
$$F_r = F_t \tan\phi$$

$$T = F_t \times R$$

$$\boxed{F_t = \frac{2T}{D}}$$

$\phi$

$m_1 = m_2$



↳ [Prime mover] → ∵ Inertia ↓

$P_1 ; N_1$

Starting Torque less for Power transmission

**Pinion**

$$P_1 = \frac{2\pi N_1 T_1}{60}$$

$T_1$ = known

$$\boxed{m, Z_1}$$

$$F_t = \frac{2T_1}{D_1} , \quad D_1 = mZ_1$$

$$F_t = \frac{2T_1}{mZ_1}$$

$$\boxed{F_t = \text{known}}$$

$$F_n = \frac{F_t}{\cos\phi} , \quad F_r = F_t \tan\phi$$

$F_n, F_r$ are known

**Gear**

$$T_2 = F_t \times \frac{D_2}{2}$$

$$T_2 = F_t \times \frac{mZ_2}{2}$$

$T_2$ = known

law of gearing

$$\boxed{G.R. = G \geqslant 1 = \frac{D_2}{D_1} = \frac{Z_2}{Z_1} = \frac{N_1}{N_2} = \text{Fixed}}$$

**case(1)** if $\eta_m = 100\%$

$$P_2 = P_1$$

$$\frac{2\pi N_2 T_2}{60} = \frac{2\pi N_1 T_1}{60}$$

$\frac{N_1}{N_2} = \frac{T_2}{T_1}$ only when $\eta_m = 100\%$

$$\boxed{G = \frac{D_2}{D_1} = \frac{Z_2}{Z_1} = \frac{N_1}{N_2} = \frac{T_2}{T_1}}$$ when $\eta_m = 100\%$

**Case II**

when $\eta_m \neq 100\%$

$P_2 = \eta_m \cdot P_1$

Real Torque

$$\frac{2\pi T_2 N_2}{60} = \eta_m \frac{2\pi N_1 T_1}{60}$$

$$\frac{N_1}{N_2} = \frac{T_2}{\eta_m \cdot T_1}$$

$$\boxed{G.R. = \frac{D_2}{D_1} = \frac{Z_2}{Z_1} = \frac{N_1}{N_2} = \frac{T_2}{\eta_m T_1}}$$

$T_2 \neq F_t \cdot \frac{D_2}{2}$

$\downarrow P = T \downarrow \cdot \omega$

Torque loss = $\frac{F_t \cdot D_2}{2} - T_2$

Resultant ~~Torq~~ force on Pinion = $F_n$

'' '' '' Gear = $F_n$

WORK

Gate book

Design-Gear

5.3 → 1 KW = P    N = 1440 rpm

T = 56.36 NM

10 : 1 ✓   $\frac{N_2}{N_1}$ ✗

$T_1$ ✓   $T_2$ ✓

$\frac{2\pi NT}{60} = P$

$\eta_m^2$.   $N_1$ $N_2$ ✓

$$\frac{N_1}{N_2} = \frac{T_2}{\eta_m \cdot T_1}$$

$$10 : 1 = \frac{56.36}{\eta_m T_1}$$

$P_{ow} = \frac{2\pi N_1 T_1}{60}$

$T_1 = 6.63$ ✓

$\eta_m =$

5.19 $T_1 = 20$ $N_1 = 30$ rev/s $\phi = 20°$

$T_2 = 40$ $P = 20 \times 10^3 W$ Full-depth system

17 21
18
19
20
21

$m = 5mm$

SIR $l = 19 mm$

$$\text{Power} = 2\pi n_1 T_1 = \frac{2\pi N_1 T_1}{60} = T_1 \omega_1$$

$$20 \times 10^3 = 2\pi(30) T_1$$

$$T_1 = 106.1 N\text{-}m$$

$$F_t = \frac{2T_1}{D_1} = \frac{2T_1}{mz_1} = \frac{2 \times 106.1}{5 \times 20 \times 10^{-3}}$$

$$F_t = 2122 N$$

$$F_n = \frac{F_t}{\cos\phi} = 2258 N$$

5.23 20° $m = 4mm$ $P = 15 kW$ $b = 25 mm$

21 teeth $N = 960$ rpm

SIR $$P = \frac{2\pi NT}{60}$$

$$15 \times 10^3 = \frac{2\pi(960)T}{60}$$

$$T = 149.2 N$$

$$F_t = \frac{2T}{mz} = \frac{2 \times 149.2}{4 \times 21 \times 10^{-3}} = 3552 N$$

5.29 $D = 50 mm$ $N = 200$ rad/s $P = 3 kW$ $\phi = 20°$

$F_n$ ✓

$$\frac{2\pi NT}{60} = P$$ ✓

$$F_t = \frac{2T}{D} = \frac{2 \times T}{D}$$ ✓

$$F_r = F_t \times \tan\phi$$ ✓

$$F_r = F_n \sin\phi$$ ✓ ?✓

$P = T\omega$

$3 \times 10^3 = T \times 200$

$T = 15 N\text{-}m$

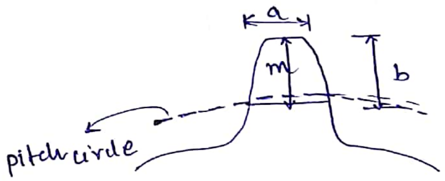
$F_t = \frac{2T}{D} = \frac{2 \times 15}{.05} = 600 N$

$F_n = \frac{F_t}{\cos\phi} = 638 N$

*centre distance वाले →

5.12
5.13
5.14
5.16
5.17
→ Do it.
5.18

5.22  $m = 4$

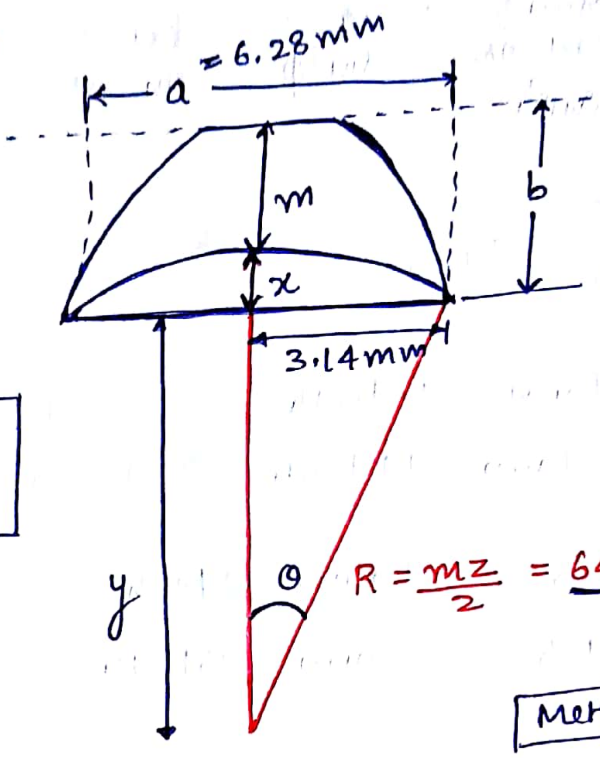


N.P.

5.22

$a \approx \frac{P_c}{2}$

$a = 6.28 mm$



$R = \frac{mz}{2} = 64 mm$

$b = m + x$

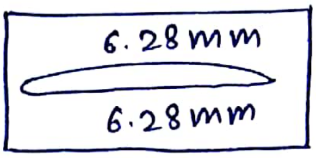
$x = R - y$

$y = \sqrt{64^2 - 3.14^2}$

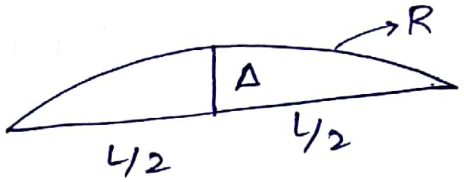
$y = 63.92 mm$

$x = .08$

$b = 4.08$



Method-3



$$\left(\frac{L}{2}\right)^2 = \Delta(2R - \Delta).$$

$$(3.14)^2 = x(2 \times 64 - x)$$

$x = .08 mm$

Method-2

$$\theta = \frac{360}{2 \times 2 \times 32} = \frac{\alpha}{2}$$

$\theta = 2.8°$

$y = R\cos\theta$

5.11 $\phi = 20°$

$$T_{min} = \xcancel{\frac{2A_p}{\sqrt{1 + a(a+2)\sin^2\phi} - 1}}$$

$G = ?$

Conclusion → ① More the addendum, more the chance of interference.

② hence ~~as~~ there is maxm. chance of interference in case of Rack and pinion arrangement. so always design any gear by assuming Rack & pinion to avoid interference.

$$T_{min} \text{ Rack or Pinion} = \frac{2a_r}{\sin^2\phi} = \frac{2\times1}{\sin^2 20^\circ} = 17.09^\circ \approx 18 \text{ Teeth}$$

$a_r = 1$ Full depth

$a_r = \cdot 8$ Stub depth

Full depth, $\phi = 20^\circ$ $T_{min}$ = 18 teeth

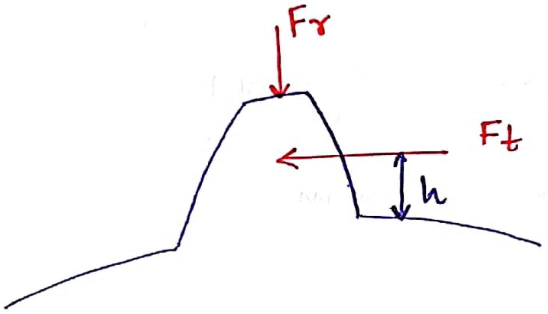
Best

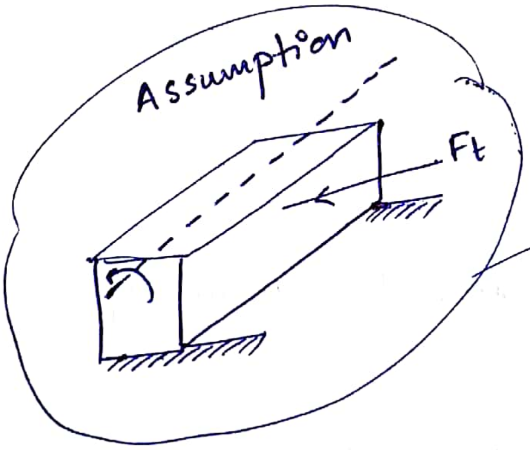
Stub tooth, $\phi = 20^\circ$ $T_{min}$ = 14 teeth

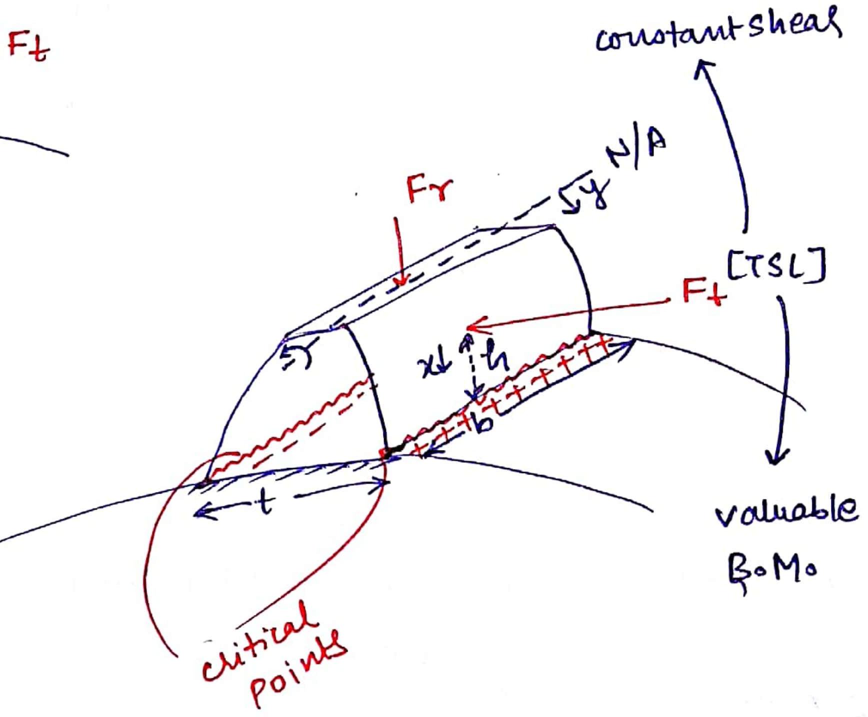
Full depth, $\phi = 14\frac{1}{2}^\circ$. $T_{min}$ = 32 teeth.

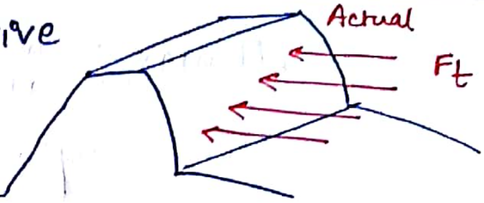
Stub tooth, $\phi = 14\frac{1}{2}^\circ$ $T_{min}$ = 26 teeth.

## DESIGN OF SPUR GEAR









**Conclusion :-** ① Due to axial compressive force $F_r$, gear tooth is subjected to compressive stresses.

② Due to shear force $F_t$, gear tooth is subjected to direct shear stresses. (TSL effect).

③ Due to variable Bending stress ( $F_t \times x$ ) gear tooth is subjected to Bending stresses.

④ For the safe design of the gear tooth, the effect of direct shear and compressive stresses are neglected, only Bending stresses willbe taken into consideration.

$$(\sigma_b)_{max} = \frac{M_{max} \cdot y_{max}}{I_{NA}}$$

$$M_{max} = F_t \cdot h$$

$$I_{NA} = \frac{bt^3}{12}$$

$$y_{max} = \frac{t}{2}$$

$$(\sigma_b)_{max} = \frac{6 F_t \cdot h}{bt^2}$$

Safe condn.

$$(\sigma_b)_{max} \le \sigma_{pel}.$$

$$\frac{6 F_t h}{bt^2} \le \sigma_{pel}.$$

$$(F_t)_{max} = \frac{bt^2 m \cdot \sigma_{pel}}{6h\, m}$$

$$\frac{t^2}{6hm} = Y \text{ Lewis's form factor}$$

$$\boxed{(F_t)_{max} = b \cdot m \cdot Y [\sigma_b]_{per}} \Rightarrow \text{Lewis equation.}$$

$\Downarrow$

Beam strength of gear tooth

Safe cond.

$$\boxed{F_{act} \leq (F_t)_{max}}$$

* BEAM STRENGTH :- It is defined as the maxm. value of the tangential load that a gear tooth can bear without any Bending.

→ Lewis Form factor :-

or

Form factor

or

Tooth Geometry factor

$Y = \pi y$

$y$ = Tooth form factor

$y = \left(.154 - \frac{.912}{z}\right)$ for full depth, $\phi = 20^\circ$.

$Y = \pi\left(.154 - \frac{.912}{z}\right)$ for full depth, $\phi = 20^0$

$Y_{(L.F.F.)}$ depends upon no. of teeth, Geometry of the tooth profile, pressure angle.

$Z_P < Z_G$

$Y_P < Y_G$.

$$(F_t)_{max} = bmY[\sigma_b]_{perm}.$$

Conclusion :-

① → weaker Gear is a Gear which has minimum value of the Beam strength and always design for weaker gear

② → when pinion and gear, both are made of same material

then $[\sigma_b]_P = [\sigma_b]_G$

$$Y_P < Y_G$$

$$(F_t)_{max_P} < (F_t)_{max_G}$$

hence, pinion is weaker so design for pinion in this case.

③ → when pinion and gear both are made of different material then design for the gear which has minm. value of the product.

$$\boxed{Y[\sigma_b]_{per}}$$

Actual load :-

$$\text{Power} = \frac{2\pi NT}{60}$$

$$T = \text{Known}$$

$$F_t = \frac{2T}{D}$$

$$F_t = \frac{2T}{mZ}$$

[Static Load] ← $F_t$ = Known

## safe condno

$$F_{act} \le (F_t)_{max}$$

$$F_{act} = F_{dynamic}$$

$$F_{dynamic} = F_{static} \times C_v \times \delta$$

$$F_d = F_t \times C_v \times \delta$$

$C_v$ = velocity factor

$\delta$ = service factor

$$F_d = F_t \times C_v \times \delta$$

$F_{dynamic} > F_{static}$

$$F_t C_v \delta \le (F_t)_{max}$$

***

$$\boxed{(F_t)\, C_v\, \delta \le b m Y [\sigma_b]_{per}}$$

$C_v = \frac{3+V}{3}$ when $V \le 10 m/s$

$C_v = \frac{6+V}{6}$ when $V > 10 m/s$

Gate book

⑳ $m = 3$

$T_1 = 16$

$b = 36$ mm

$\phi = 20°$

3kw $N_2 =$

$$F_t \cdot C_v \cdot \delta \le b m Y [\sigma_b]_{per}$$

(96M) ✓

$$994.7 \times 1.5 \le 36 \times 3 \times .3 [\sigma_b]_{per}$$

$$\boxed{[\sigma_b]_{per} = 46 MPa}$$

Power $= 2\pi N T$

$3 \times 10^3 = 2\pi (20) T$

$T = 23.87$ N-m

$$F_t = \frac{2T}{m z} = 994.7 N$$

㉔ Gate Book

$\phi = 20^\circ$
$m = 4$
$z_1 = 21$
$P = 15$
$N = 9.0$

TOM
FM
SOM
T/M.
PRo.

M
EM
TOM1
TOM 2
PP

SIR

$$3552 \times 1.5 \leq 25 \times 4 \times .32 [\sigma_b] = \underline{166.5 \text{ MPa}}$$

$$F_t \cdot C_v \cdot \delta \leq b\, m \cdot Y [\sigma_b]_{per}$$

$$994.7 \times 1.5 \leq 36 \times 3 \times \cdot 3 [\sigma_b]_{per}$$

$$\boxed{(\sigma_b)_{per} = 46 \text{ MPa}}$$

## * Wear strength of Gear Tooth.

It is defined as the maxm. value of the load that a gear tooth can wear without any wear failure.

wear strength is always check for pinion only because pinion is subjected to more wear.

$$F_w = D_p \cdot b \cdot Q \cdot K$$

wear strength

$D_p$ = pitch circle dia. of the pinion

$b$ = face width

$Q$ = Ratio factor

$$Q = \frac{2G}{G \pm 1}$$

(+) — External gear.

(−) — Internal gear.

K = material combination factor

$$\boxed{K = \frac{\sigma_{es}^2 \sin\phi \left[\frac{1}{E_P} + \frac{1}{E_G}\right]}{1.4}}$$

$$\left[\begin{array}{l} K \propto (BHN)^2 \\ \\ K \propto (S_{ut})^2 \end{array}\right]$$

$\sigma_{es}$ = surface endurance limit of the gear tooth.

$E_P$ and $E_G$ = Young's Modulus of pinion and gear.

$\phi$ = Pressure angle.

Safe condition

$F_{act} \leq F_W$

$$\boxed{F_t \cdot C_v \cdot \delta \leq D_P \cdot b \cdot Q \cdot K}$$

Hence safe from wear

Practical case

$F_W \geq (F_t)_{max} \geq F_{act}$.

wear strength ≥ Beam strength ≥ Actual load.

eg:

$F_W = 15\,kN$

$(F_t)_{max} = 20\,kN$

Power = ?

$F_{act} \leq 15$

$F_t \cdot C_v \cdot 8 \leq 15$

$F_t$ = known

T = known

Power = known

## * Assumptions made in Lewi's Equation :-

① Gear Tooth assumed as a cantilever Beam fixed at the root portion.

② The effect of direct shear and compressive stresses are neglected.

③ Gear Tooth assumed as a prismatic throughout.

④ The effect of stress concentration factors are neglected.

⑤ Inertia of the Rotating part neglected.

⑥ Deflection of the Tooth under load neglected.

⑦ Errors in Tooth manufacturing and spacing are neglected.

⑧ Contact Ratio assumed as 1 (one).

all these assumptions are the reason for the dynamic loading.

# Type of Wear

*Abrasive wear → ① These type of wear occurs b/w meshing gear surface due to presence of foreign material by the dust deposit or something by the lubricant.

② occurs more in open gears.

*Scoring and Scuffing ~~Gear~~ Wear :- This type of wear occurs between meshing gear surface due to failure of lubrication.

Scoring → Scratches in sliding dirn

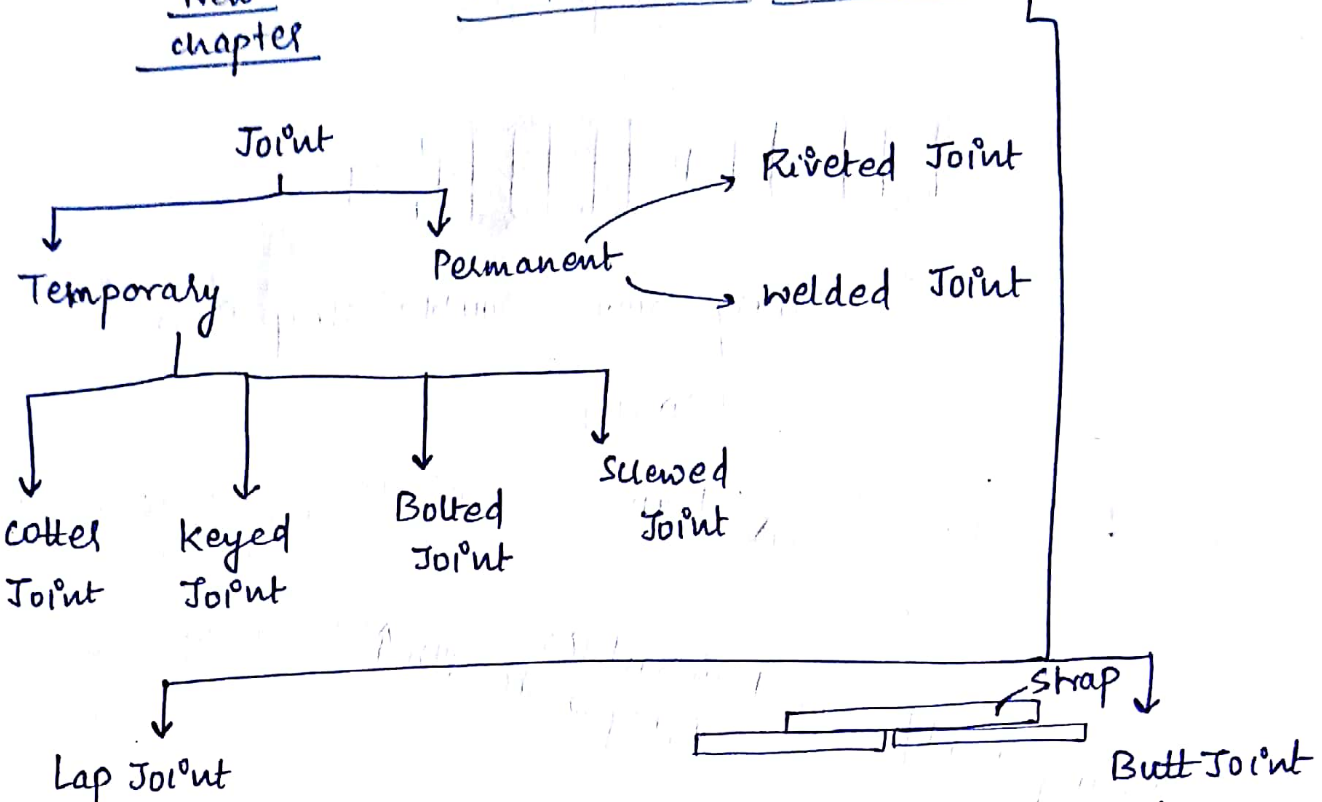
Scuffing → welding due to heating.

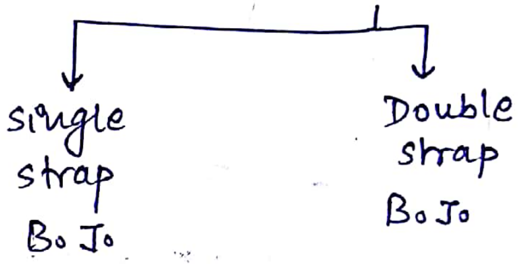
* Corrosive wear :→ These type of wear occurs Due to chemical $Rx^n$ between lubricant and mating/meshing surfaces.

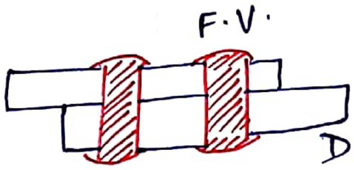
* pitting :→ This type of wear occurs between meshing gear surface due to repeated stress occur under cyclic loading.

New chapter

# Riveted Joint [Permanent Joint]

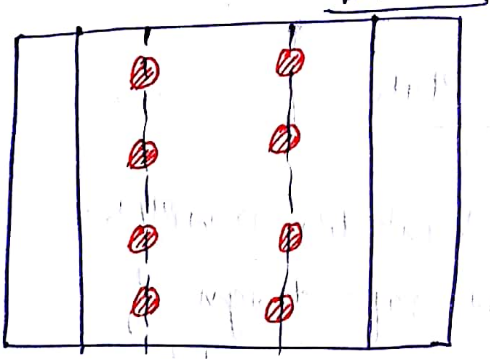




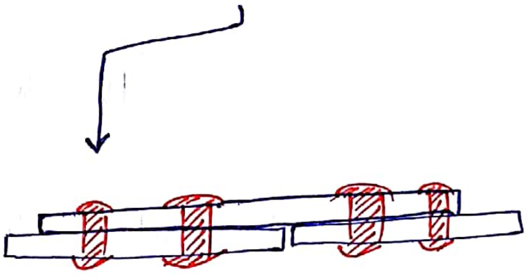


Double Row-lap Joint

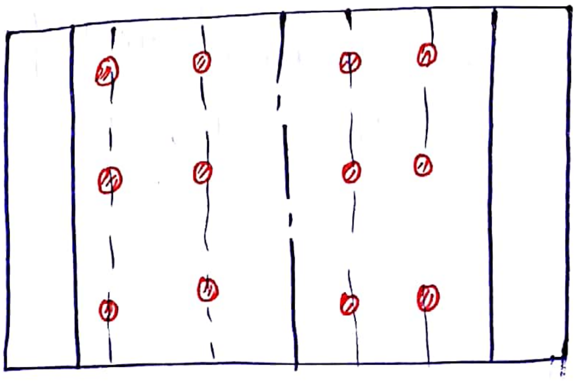
D.R.L.J.



Top view

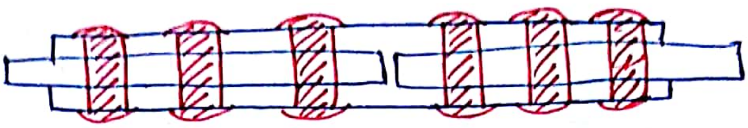


Double row - single strap Bo Jo



DRSSBJ

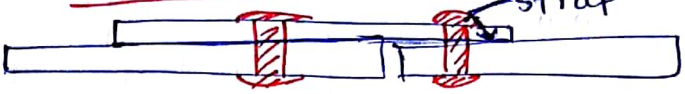
# Double strap B.J.



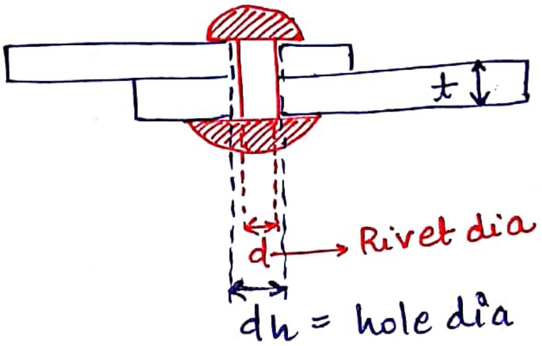
Tertiary Row - Double strap B.J.

TRDSBJ

SRSSBJ



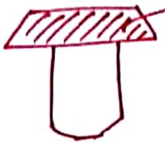
[Snap Head] ⇒ used in boilers
fatigue strength ↑



d = shank dia. / Rivet dia.

**Riveted Joint = Rivet + Plate**

For the safe design of the Rivet, shank dia D will be taken into consideration and for the safe design of the plate, hole dia. ~~dh~~ Dh will be taken under consideration.



Pan head / cone head

① ship hull's / boiles

② Corrosive strength ↑

counter sunk head

[underwater working]

Flat head

① Thin Plates.

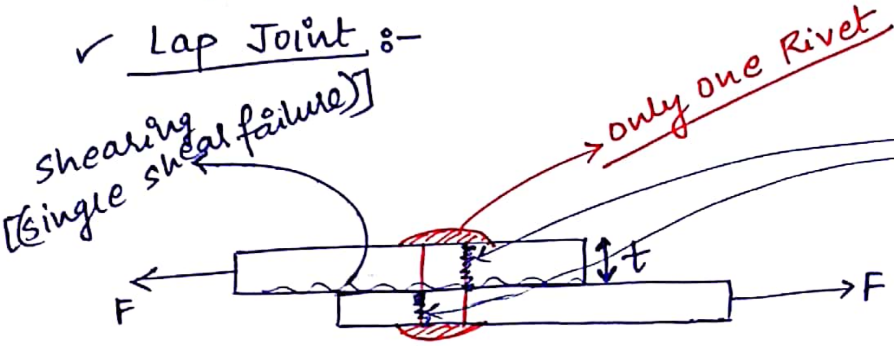
② structural work.

Unwin's formula ⇒ $\boxed{d = 6\sqrt{t}}$

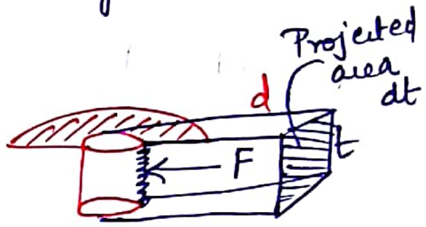
$d$ → m.m. $t$ → m.m.

## * Type of failure :-

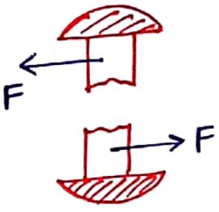
✓ **Lap Joint :-**



Crushing

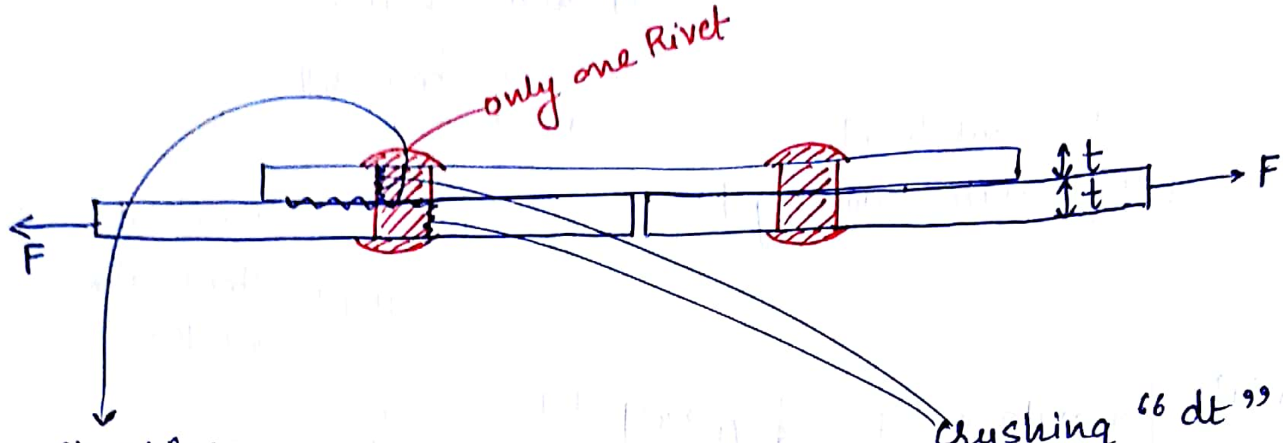


$$(\sigma_{ind})_{crush} = \frac{F}{dt}$$



$$\tau_{ind} = \frac{F}{\frac{\pi}{4}d^2}$$

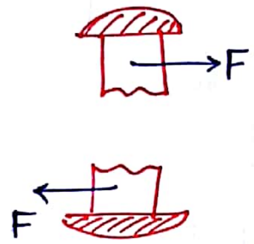
## Single strap B. J.



Shearing
[single shear failure]

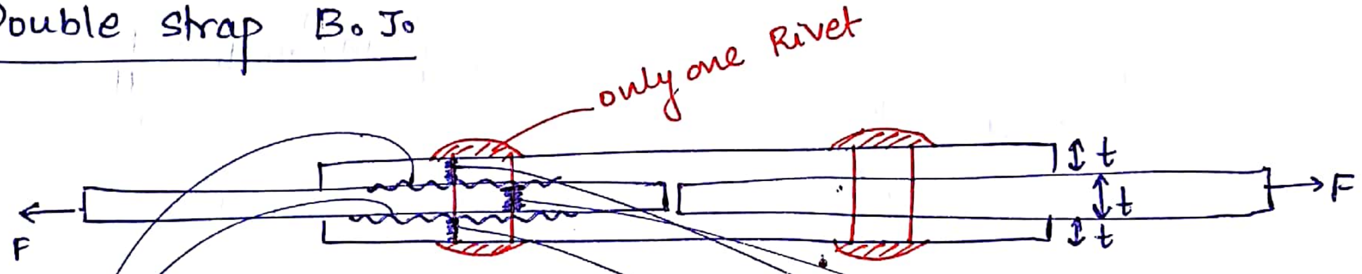
Crushing "dt"

$(\sigma_{ind})_{crush.} = \frac{F}{dt}$



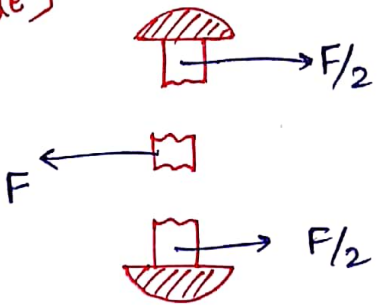
$\tau_{ind} = \frac{F}{\frac{\pi}{4}d^2}$

## Double strap B. J.



Shearing
[Double shear failure]

$\tau_{ind} = \frac{F}{2 \cdot \frac{\pi}{4}d^2}$



Crushing

$\sigma_{ind\ crush\ max} = \frac{F}{dt}$

$$\tau_{ind} = \frac{F}{k \cdot \frac{\pi}{4} d^2}$$

$k = 1 \rightarrow$ single shear

$k = 2 \rightarrow$ Double shear

$$(\sigma_{ind})_{crush} = \frac{F}{dt}$$

For all riveted Joint

---

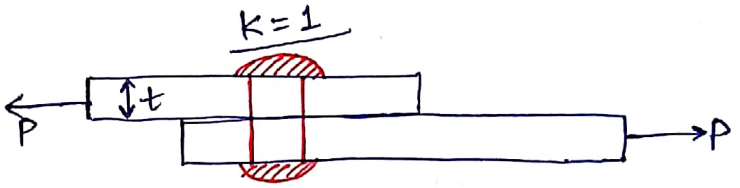
## Case No.1 finite Riveting

$$\tau_{ind} = \frac{P}{4 \cdot K \frac{\pi}{4} d^2}$$

safe condn.

$$\tau_{ind} \leq \tau_{pr}$$

$$\frac{P}{4 \cdot K \frac{\pi}{4} d^2} \leq \tau_{pr}$$

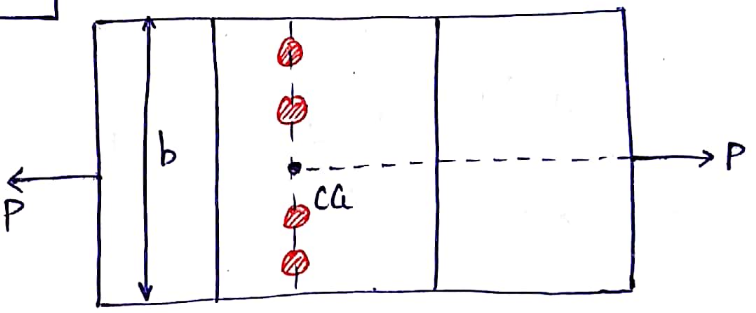


$$\boxed{P_{max} = 4 \cdot K \frac{\pi}{4} d^2 \tau_{pr}}$$

$\Downarrow$

shear strength of RJ/ Rivet

$F_{rivet} = P/4$.



## Shear Design of Riveted Joint ~~oblique~~/ Rivet.

$$\sigma_{\substack{Ind.\\Crush}} = \frac{P}{4dt}$$

safe condn.

$$\sigma_{\substack{ind.\\Crush}} \le \sigma_{pr}$$

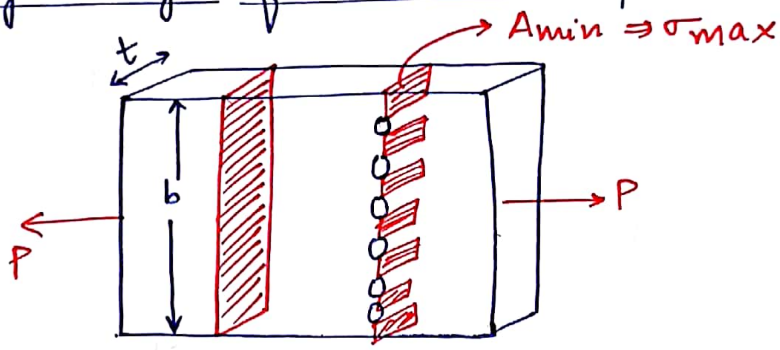
$$\frac{P}{4dt} \le \sigma_{pr}$$

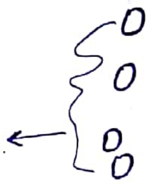
$$\boxed{P_{max} = 4dt \cdot \sigma_{pr}}$$

⇓

crushing strength of R.J./Rivet.

## Tearing design of Riveted Joint/plate





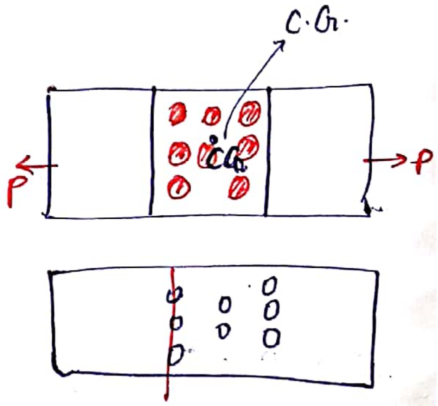
$$\sigma_{\substack{max\\Tearing}} = \frac{P}{(b-4dh)t}$$

$$\boxed{P_{max} = (b-4dh)t \cdot \sigma_{pr}}$$

⇓

Tearing strength of R.J./Plate

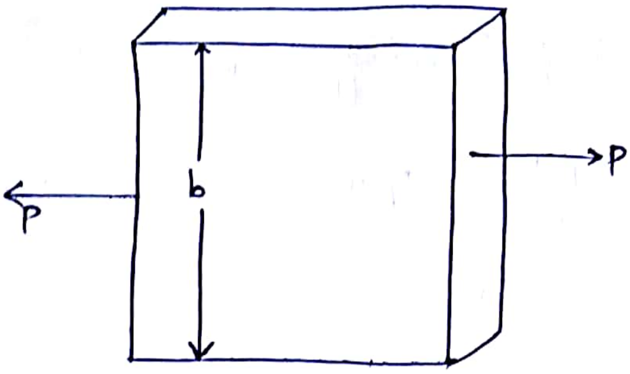
Ex:-

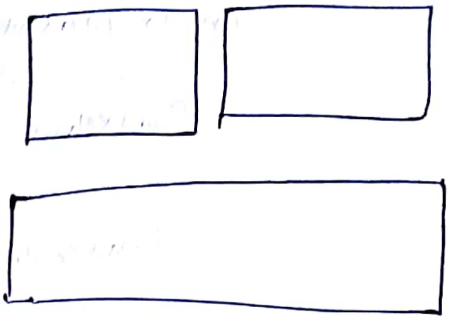


Actual strength of R.J. = min of$\left[(P_{max})_{shear}, (P_{max})_{crush}, (P_{max})_{Tearing}\right]$

——"—— Rivet = min of $\left[(P_{max})_{shear}, (P_{max})_{crush}\right]$.

## * Strength of the solid plate :-





$$\sigma_{ind} = \frac{P}{bt}$$

$$\boxed{P_{max} = bt \cdot \sigma_{pr}}$$

Solid

→ Best possible joint forever. for the comparison purpose only

$$\eta_{shearing} = \frac{(P_{max})_{shear}}{(P_{max})_{solid}} = \frac{4 \cdot \frac{\pi}{4} d^2 k \, \tau_{ind}}{bt \cdot \sigma_{pl}}$$

$$\eta_{crushing} = \frac{4 \cdot d \cdot \cancel{t} \cdot [\sigma_{pl}]_{Rivet}}{b\cancel{t} \, [\sigma_{pl}]_{plate}}$$

$$\eta_{Tearing} = \frac{(b - 4d_h)\cancel{t} \cdot \cancel{\sigma_{pr}}}{\cancel{bt} \cdot \cancel{\sigma_{pr}}} = 1 - \frac{4d_h}{b}$$

$$\eta_{actual} = \text{min. of } [\eta_{shearing}, \eta_{crushing}, \eta_{Tearing}]$$

[redacted]

Gate

[redacted]

3.12 $P_{max}$ crushing $= 3 \cdot d \cdot t \cdot \sigma_{pr}$

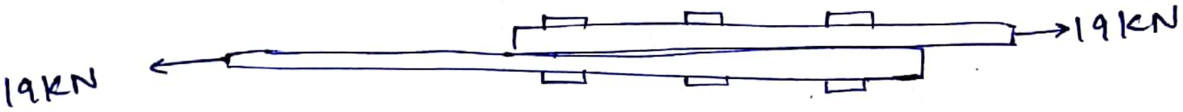
$P_{max}$ Tearing $= (w - 3d_h) + \sigma_{ps}$

$P_{max}$shear $= 3 \cdot \frac{\pi}{4} d^2 \cdot \tau_{per}$ ✓

$P_{max_{solid}} = 4\, l \cdot t \cdot \sigma_{per}$.

3.14 $\sigma_{ys} = 200$ MPa

FOS = 2



Bolt will fail (shear) → single ∵ load sey paas hore hue.

M-10 ✓ 8.97 = D ✓

SIR

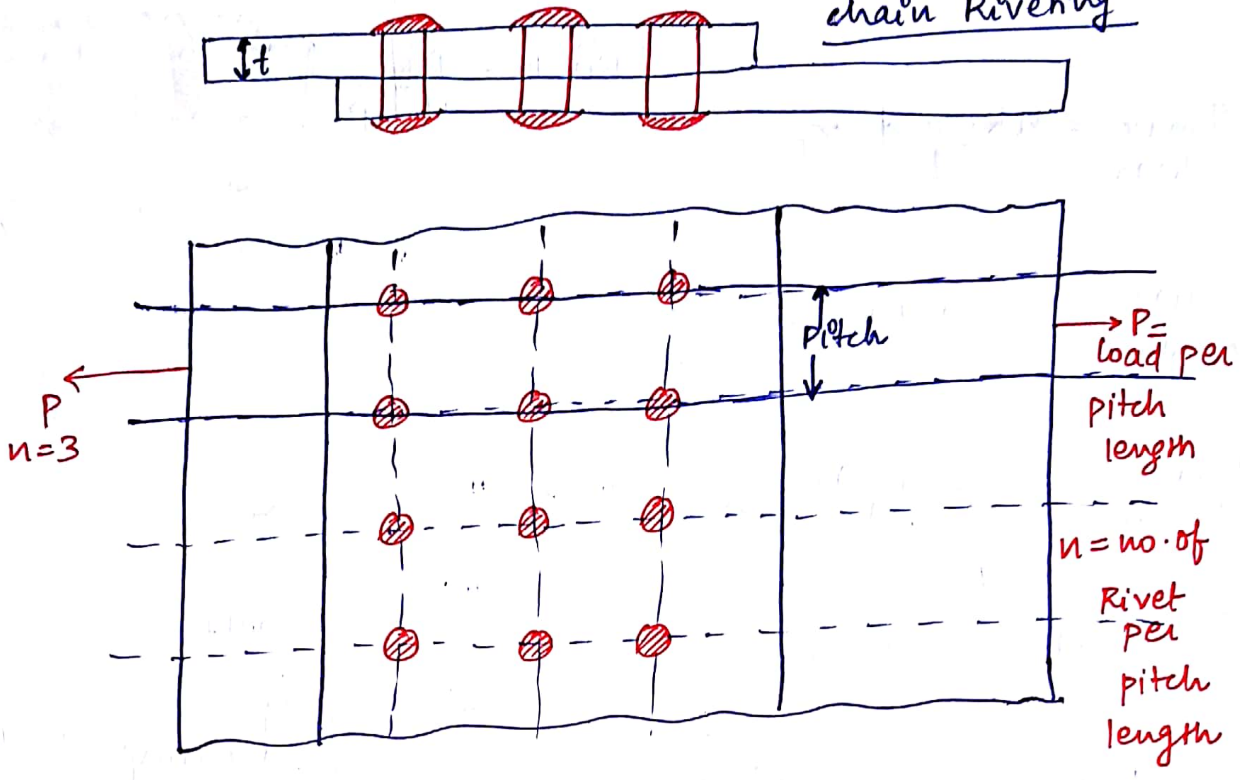
$$\frac{19/3}{\frac{\pi}{4} d^2} \leq \frac{200}{2} \times 10^{-3}$$

$d \geq 8.9$ mm

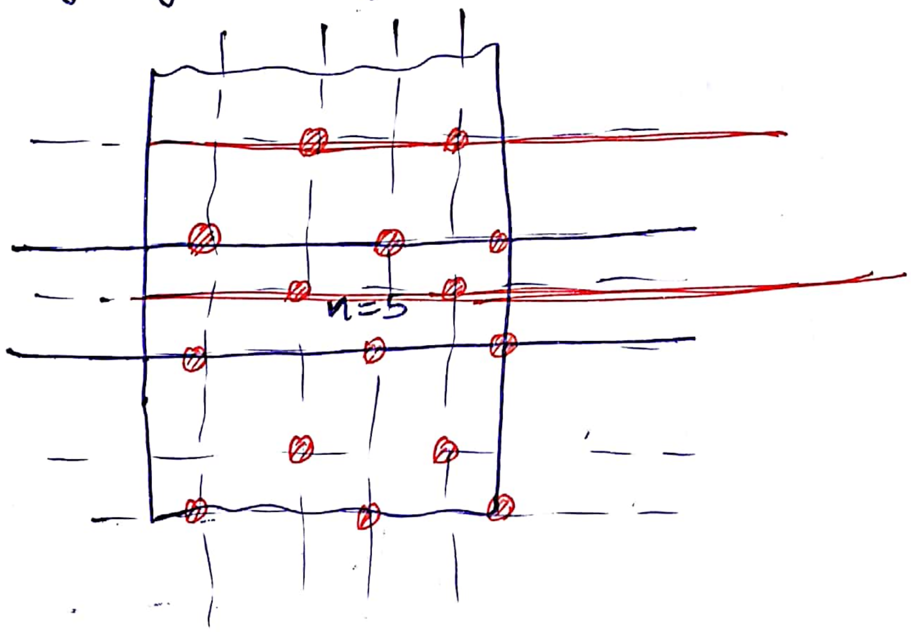
$\boxed{d = 10 \text{mm}}$

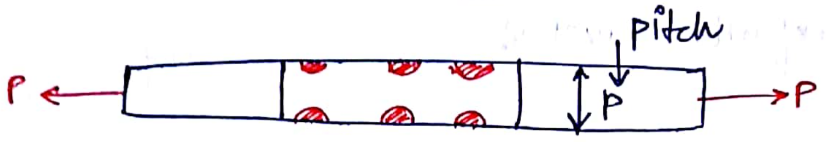
## Case II → Case of Infinite Riveting

chain Riveting



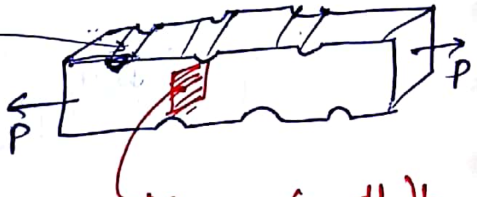
zig-zag Riveting





$n = 3$

$F_{Rivet} = \frac{P}{n}$



$A_{min} = (p-d_h)t.$

$$P_{max\,shear} = nk\cdot\frac{\pi}{4}d^2\tau_{pr}$$

$$P_{max\,crushing} = n\cdot d\cdot t\cdot\sigma_{pr}$$

$$P_{max\,Tearing} = (p-d_h)t\cdot\sigma_{pr}$$

$$P_{max\,solid} = p\cdot t\cdot\sigma_{pr}$$

$$\eta_{shear} = \frac{nk\frac{\pi}{4}d^2\tau_{pr}}{Pt\cdot\sigma_{pr}}$$

$$\eta_{crushing} = \frac{n\,d\,t\cdot\sigma_{pr}}{P\,t\cdot\sigma_{pr}}$$

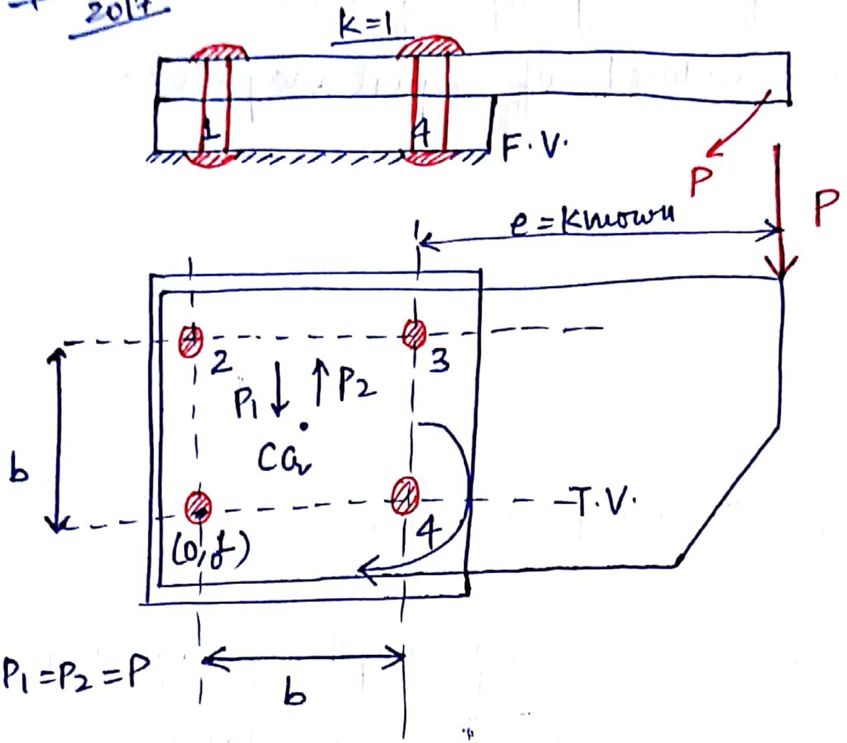
$$\eta_{tearing} = \frac{(p-d_h)\,t\cdot\sigma_{pr}}{Pt\cdot\sigma_{pr}}$$

ESE-2017

$$\boxed{\eta_{Tearing} = 1-\frac{d_h}{p}}$$

very very imp. Topic for gate 2017

# -: Design of Riveted / Bolted Joint :- under eccentric loading.



$P_1 = P_2 = P$

**Solution →** **step 1 →** Find out the C.G. of the group of Rivet / bolt.

$$\bar{X} = \frac{X_1 + X_2 + X_3 + X_4 + \text{-----}}{n} \Rightarrow \bar{X} = \frac{0+0+b+b}{4} = \frac{b}{2}$$

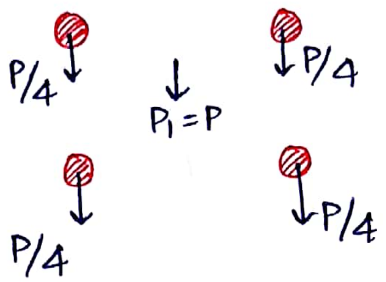
$$\bar{Y} = \frac{Y_1 + Y_2 + Y_3 + Y_4 + \text{----}}{n} \Rightarrow \bar{y} = \frac{0+b+b+0}{4} = \frac{b}{2}$$

**Step 2 →** Find out Eccentricity → is the distance b/w C.G. of the group of Rivet to the line of action of the load.

**step 3 →** apply 2 equal & opposite forces passing through C.G. of the group of Rivet such that $P_1 = P_2 = P$

## Step 4.→ Effect of $P_1$

$P_1$ Passes through c.G. of the group of Rivet / Bolt. it result primary shear force induced of equal magnitude in each rivet / Bolt.

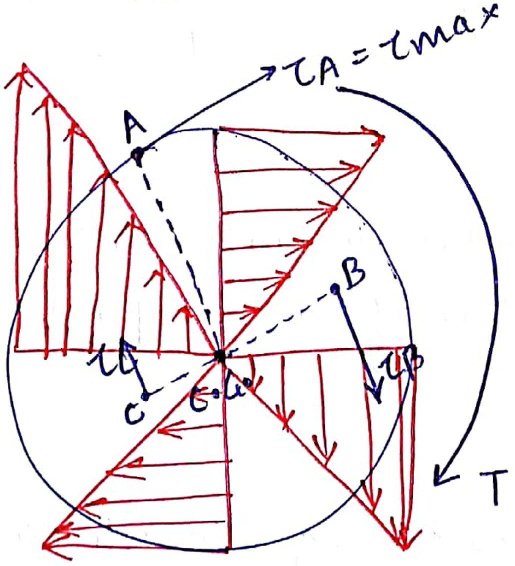


P/4 = Primary shear force.

Step5.→ $P_2$ and P causes a constant Twisting couple of magnitude $P \times e$ in clockwise direction about the c.G. of the group of rivet / Bolt which result twisting in the rivets.

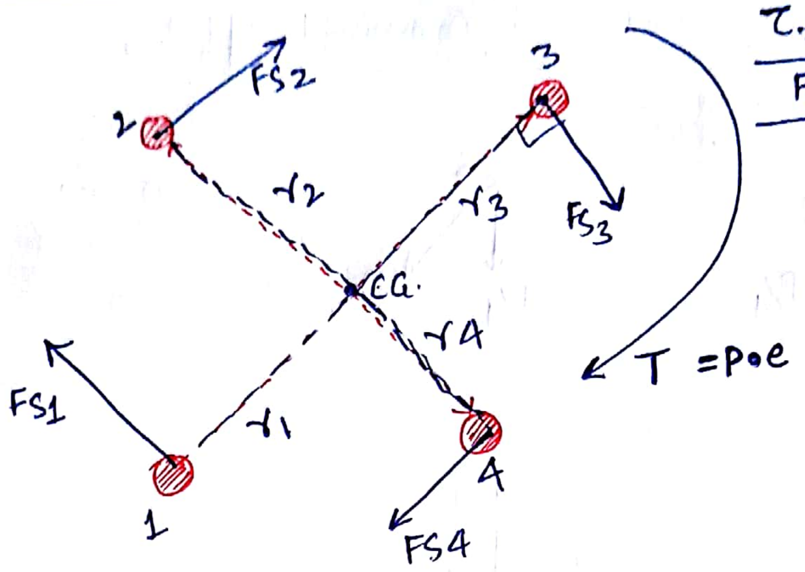
## Twisting in shaft

$\tau_{torsional} \propto r$.



$\tau_{torsional} \propto r$

$F_S \propto r$



Secondaly sheas force ⟶ Fs

$$F_{S1} l_1 + F_{S2} l_2 + F_{S3} l_3 + F_{S4} l_4 = P \cdot e$$

$F_S \propto r$

$$\frac{F_{S1}}{r_1} = \frac{F_{S2}}{r_2} = \frac{F_{S3}}{r_3} = \frac{F_{S4}}{r_4}$$

$$\frac{F_{S1}}{r_1}\left[r_1^2 + r_2^2 + r_3^2 + r_4^2\right] = P.e$$

$F_{S1}$, $F_{S2}$, $F_{S3}$, $F_{S4}$ are known

if $r_1 = r_2 = r_3 = r_4 = r$

$$F_{S1} = F_{S2} = F_{S3} = F_{S4} = \frac{P.e}{4.r}$$

step 6 →

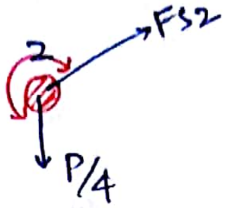
N.P.

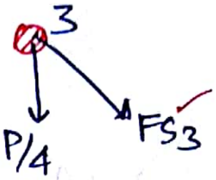
Scanned by CamScanner

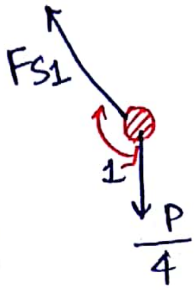
## Combined Effect

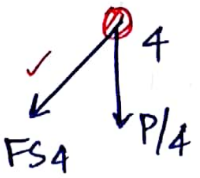
$R^2 = P^2 + Q^2 + 2PQ\cos\theta$

$\theta \downarrow \rightarrow R \uparrow$

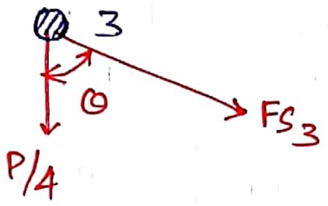








## Critical Rivet or worst loaded Rivet



$$R_3 = \sqrt{\left(\frac{P}{4}\right)^2 + (F_{S3})^2 + 2\left(\frac{P}{4}\right)(F_{S3})\cos\theta}$$

maxm. Shear Force

$$\tau_{max} = \frac{R_3}{\frac{\pi}{4}d^2}$$

Safe condn

$$\frac{R_3}{\frac{\pi}{4}d^2} \leq \tau_{pr}$$

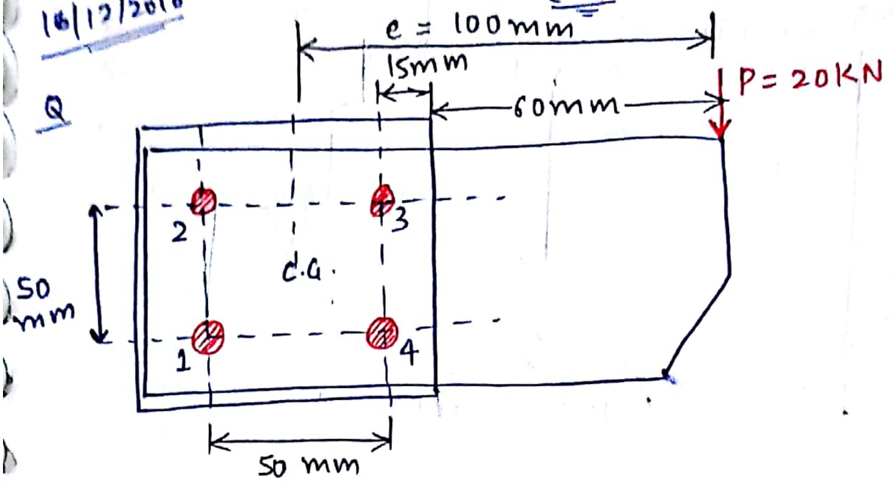
$$\boxed{d \geq \_\_}$$

Answer

"(माँ)"

16/12/2016

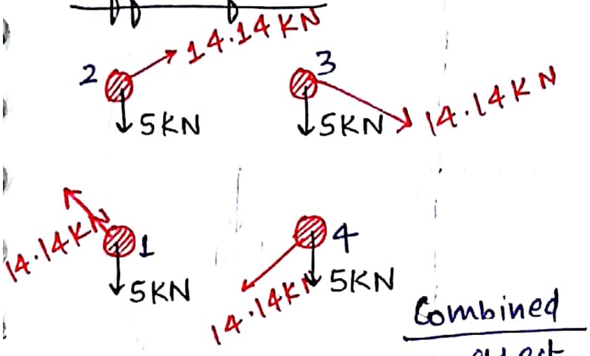
**Q**



$\tau_{per} = 70\,MPa$

d = ?

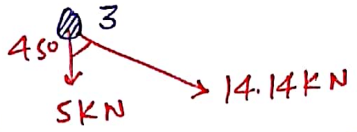
### Effect of $P_1$



Combined effect

Critical Rivet 3, 4

safe condn.



$$\frac{18.02}{\frac{\pi}{4}d^2} \le 70\times10^{-3}$$

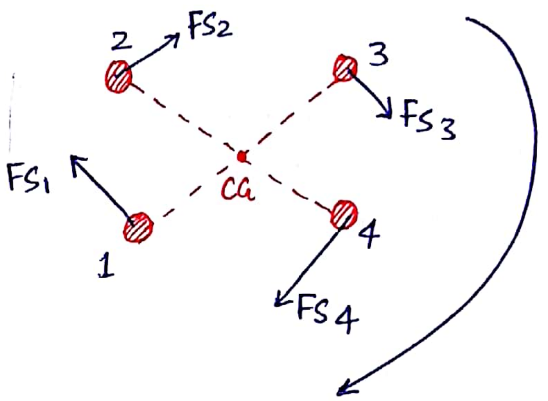
$R_3 = \sqrt{5^2 + 14.14^2 + 2(5)(14.14)\cos 45}$

d ≥ 18.1mm

d = 19mm

$R_3 = 18.02\,KN$

### Effect of $P_2$ and P



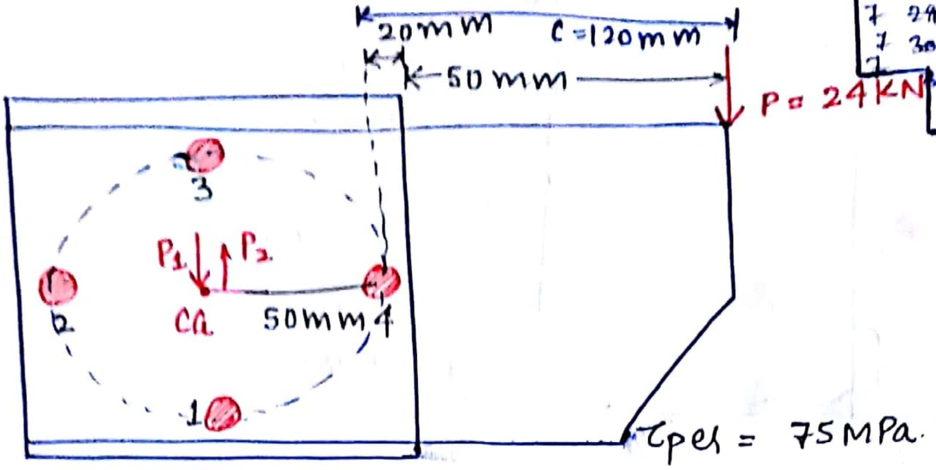
P = 20×100 KN-mm

$r_1 = r_2 = r_3 = r_4 = 25\sqrt{2}$

$$FS_1 = FS_2 = FS_3 = FS_4 = \frac{20\times100}{4\times25\sqrt{2}}$$

$= 14.14\,KN$

**Q.**



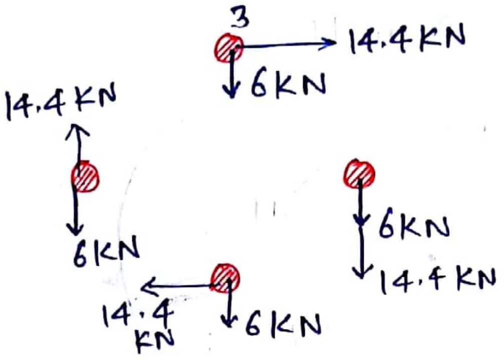
46 × 16

30 × 16

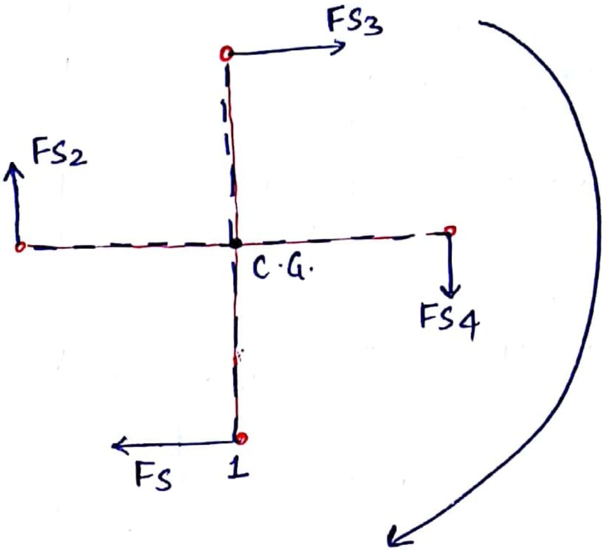
46 × 16 = 276 + 460 = 736

**Sol.**

## ~~Effect of P₁~~ combined



## Effect of P₂ and P



Critical Rivet ⇒ 4

$$\frac{20.4}{\frac{\pi}{4}d^2} \leq 75\times10^{-3}$$

R4 = 20.4 KN

$d \geq 18.6$

$\Rightarrow \boxed{d = 19\,mm}$

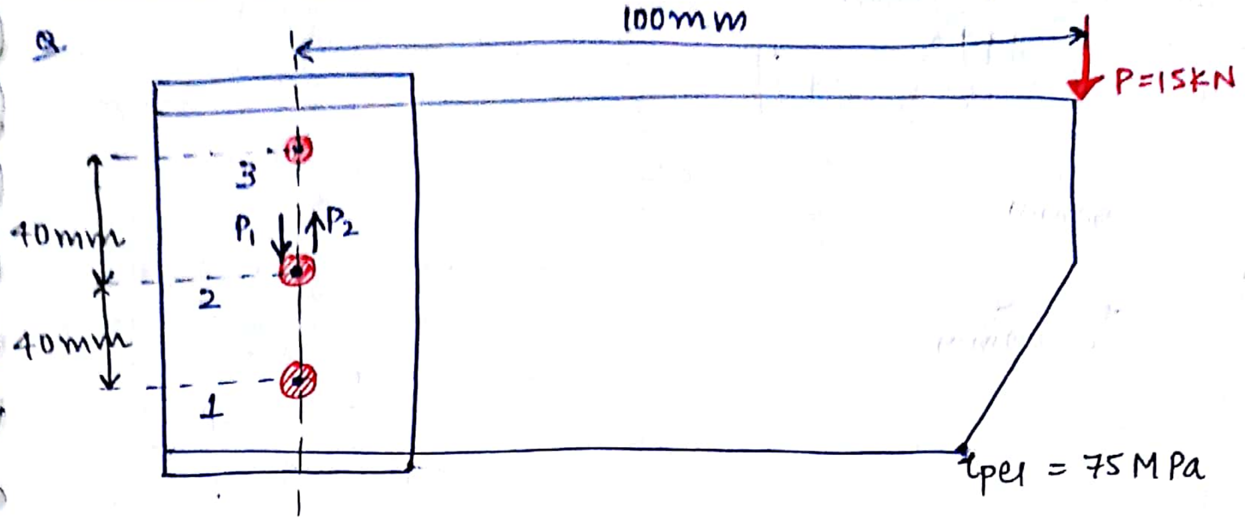
T = 24 × 120 KN-mm

$r_1 = r_2 = r_3 = r_4 = 50$

$F_{S1} = F_{S2} = F_{S3} = F_{S4} =$

$$\frac{24\times120}{9\times50} = 14.4\,KN$$

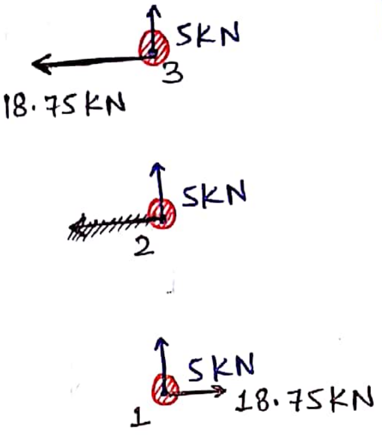
Q.



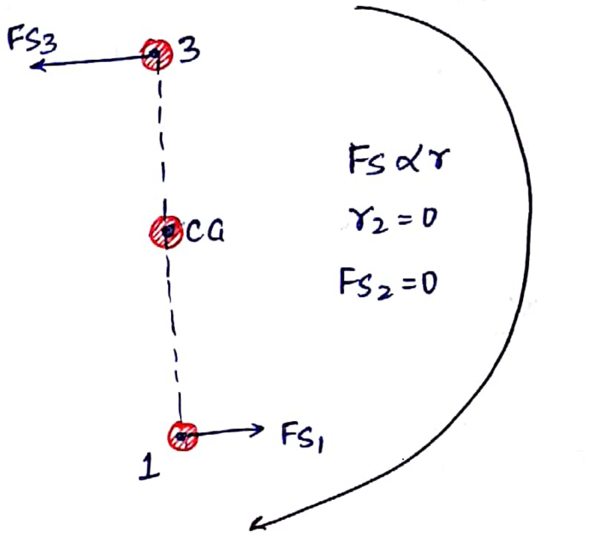
Find out the critical Rivet, Resultant force on each Rivet, safe diameter of the Rivet.

Sol

Effect of $P_1$



Effect of $P_2$ and P



critical Rivet 1,3

$$R_1 = R_3 = \sqrt{5^2 + 18.75^2}$$

$$R_1 = R_3 = 19.4 KN$$

$$R_2 = 5 KN$$

Safe condn.

$$\frac{19.4}{\pi/4 d^2} \leq 75\times10^{-3}$$

$$d \geqslant 18.15$$

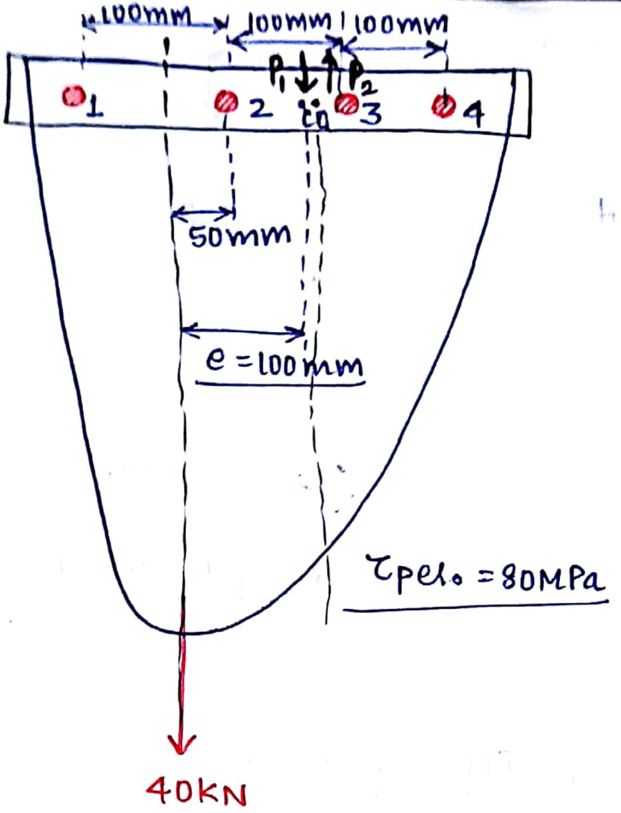
$$\boxed{d = 19 mm}$$

T = 15x100 KN-mm

$$\frac{Fs_1}{40}\left[40^2 + 0^2 + 40^2\right] = 15\times100$$

$$Fs_3 = Fs_1 = 18.75 KN$$

Q



| | | | |
|---|---|---|---|
| ~~17~~ | 1 | 16 | 31 |
| ~~18~~ | 2 | ~~17~~ | ~~1~~ |
| ~~19~~ | 3 | 18 | 2 |
| ~~20~~ | 4 | 19 | 3 |
| ~~21~~ | 5 | 20 | |
| ~~22~~ | 6 | 21 | |
| ~~23~~ | ~~7~~ | ~~22~~ | |
| ~~24~~ | 8 | 23 | |
| ~~25~~ | ~~9~~ | ~~24~~ | |
| ~~26~~ | 10 | ~~25~~ | |
| ~~27~~ | ~~11~~ | ~~26~~ | |
| ~~28~~ | ~~12~~ | ~~27~~ | |
| ~~29~~ | 13 | ~~28~~ | |
| ~~30~~ | 14 | ~~29~~ | |
| ~~31~~ | ~~15~~ | ~~30~~ | |

4 → Paper

49 days

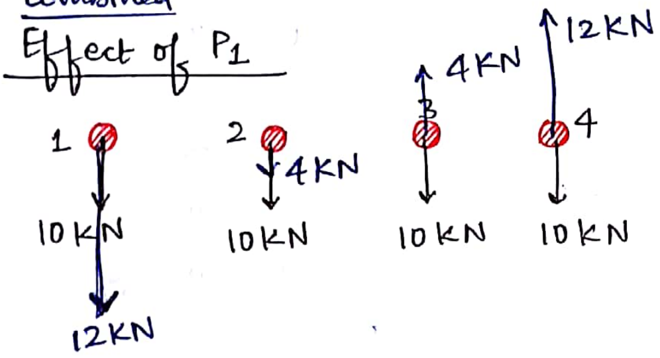
⑬

19mm

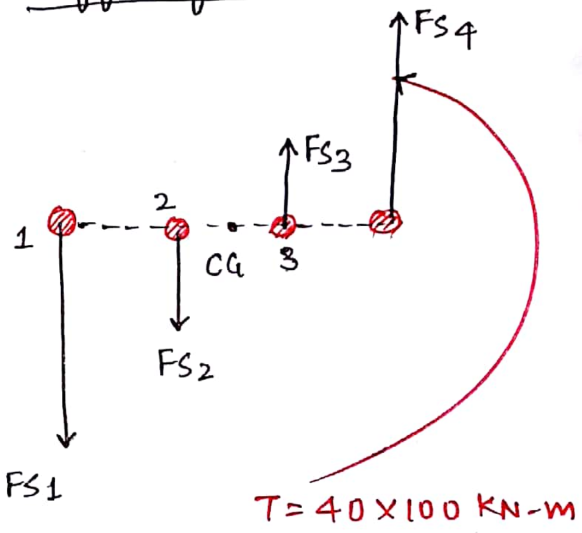
4th is critical.

Combined

Effect of $P_1$



Effect of $P_2$ and P



$R_1 = 22KN$ → critical Rivet

$R_2 = 14KN$

$R_3 = 6KN$

$R_4 = 2KN$

$$\frac{F_{S1}}{150}\left[150^2 + 50^2 + 50^2 + 150^2\right] = 40 \times 100$$

$$F_{S1} = F_{S4} = 12\,KN$$

$$F_{S2} = F_{S3} = 4KN$$

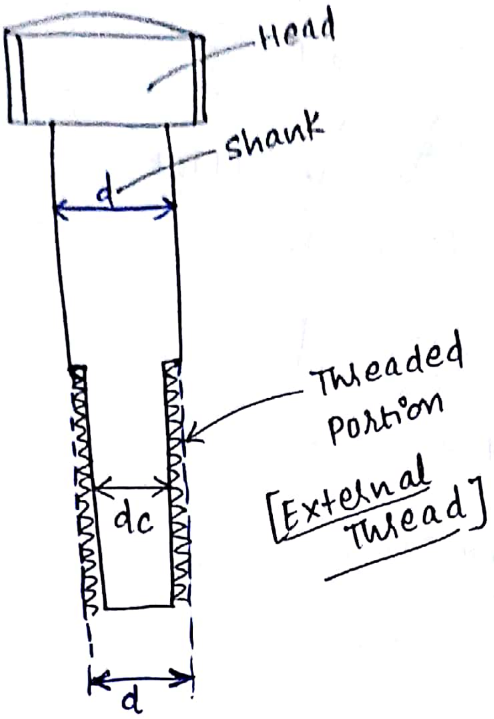
$$\frac{22}{\frac{\pi}{4}d^2} \leq 80 \times 10^{-3}$$

$$d \geq 18.7$$

$$\boxed{d = 19mm}$$

# NEW CHAPTER

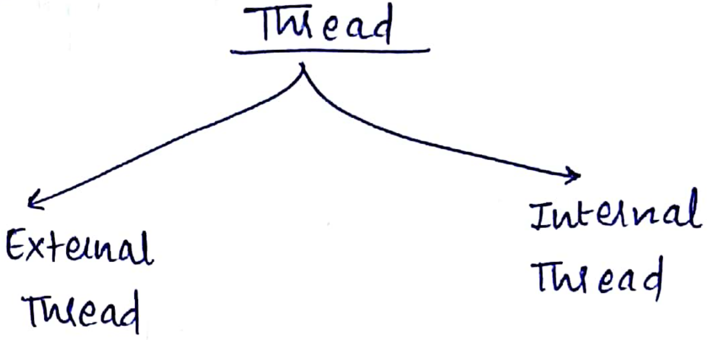
## Bolted Joints

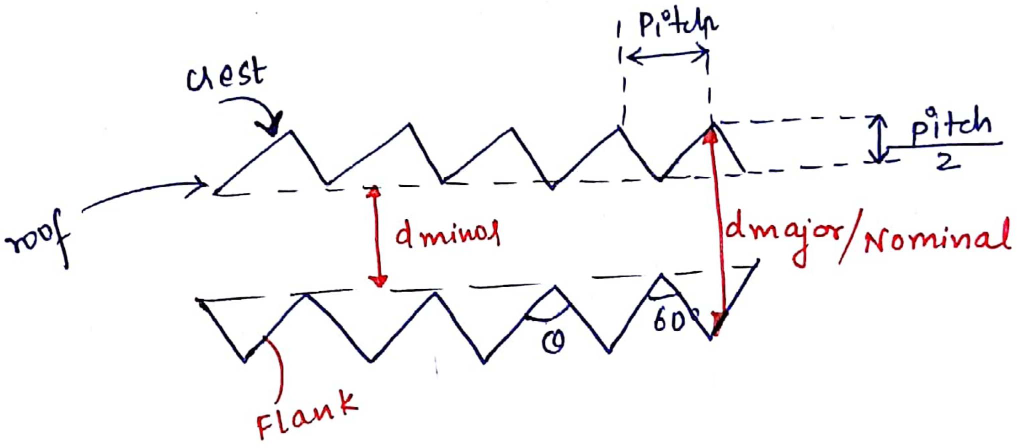


d = Major dia / shank dia / Nominal dia / dia

$d_c$ = core dia / Minor dia

$$\boxed{d_c = .84d}$$

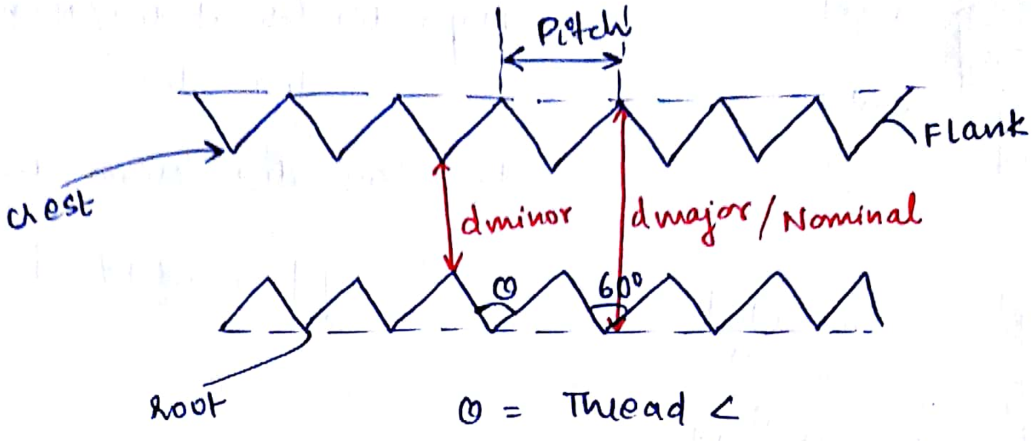




θ = Thread angle

vee- shaped Thread

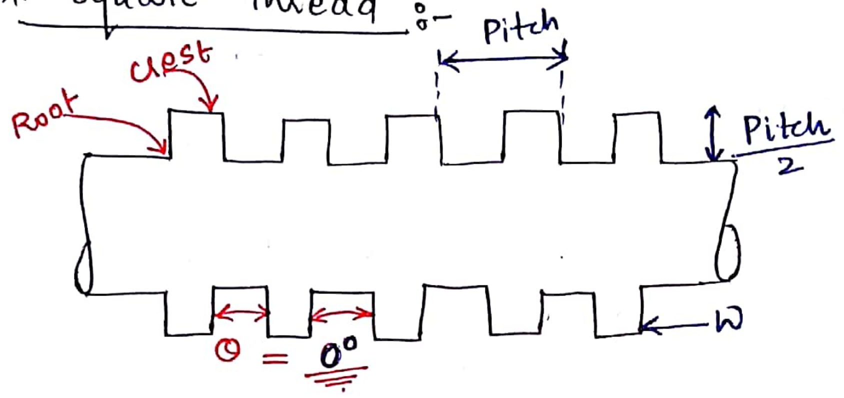
[used for fastening Purpose]



θ = Thread ∠

vee-shape

## * Square Thread :-



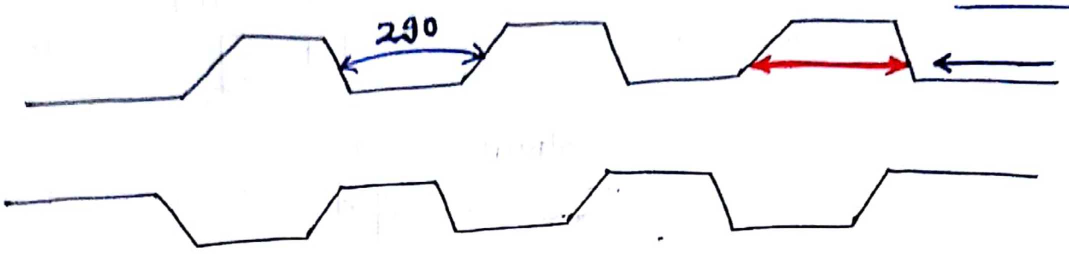
Square Thread

used in screw Jack

① $\eta_{max}$

- (i) Mfg different
- (ii) single Point tool
- (iii) Cost ↑
- (iv) strength ↓

→ strong as compared to square thread.



$\theta = 30^{\circ}$ → Trapezoidal (पहले के Jamane में)

## ACME Thread

- * ① Best thread to transmit Power in both direction.
- * ② Most commonly used
- * ③ Manufacturing simple.
- * ④ cost ↓

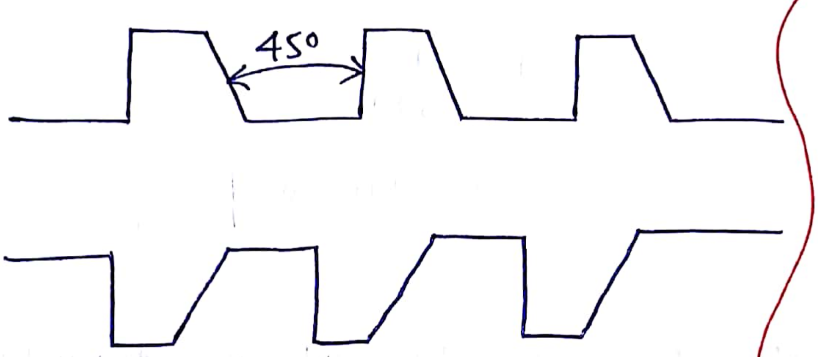
(Lead screw of lathe).

worm and wheel
→ Reversible

① R.R. ↓

② No. of start ↑

③ $\phi < \alpha$ → self lock



## Buttress thread

* Used to transmit power only in one direction.

screw Jack/ Power Press

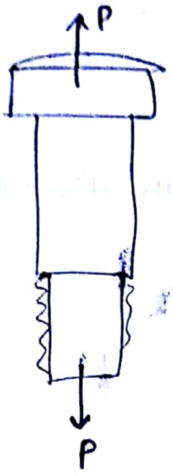
$\mu = \tan\phi$

$$\boxed{\phi = \tan^{-1}(\mu)}$$

**Bolt** → Shear

Aim

"Tension"



Find out safe diameter of Bolt ⇒ Nominal dia

$$\boxed{d = ?}$$

$$\sigma_{shank} = \frac{P}{\frac{\pi}{4} d^2}$$

$$\sigma_{max} = \sigma_{core} = \frac{P}{\frac{\pi}{4} d_c^2}$$

Safe condition

$$\sigma_{max} \le \sigma_{per}$$

$$\frac{P}{\frac{\pi}{4} d_c^2} \le \sigma_{per}$$

$d_c$ = known

$$d = \frac{d_c}{.84}$$

→ This Result is of Databook hence Gate wale.

10mm ≤ [d ~~≤~~ 25mm]

$$\boxed{d = \text{Known}}$$

$d_c = .78d$

$d > 25mm$

Conclusion :→ ⓐ for the safe Design of the Bolt, core diameter '$d_c$' will be taken into consideration becoz core is the weakest portion of the Bolt.

ⓑ GATE

Bolt ≈ Rivet

No ~~$d_c$~~ Ka funda.

WK
Gate book

③.⑱

$$\frac{19/3}{\frac{\pi}{4}dc^2} \le 100$$

$dc = 8.9$

$d = \frac{8.9}{.84} = 11. - -$

M10 - - (M12)

(3.7) ⓐ d से करना ✓

M20×2 ← 2mm Pitch [Fine Pitch]

↓

20mm → Bolt dia / Nominal dia / Magnitude

M20 ← coarse pitch

10mm ✓

↓

20mm = Major dia

| Pitch → | 2mm | 4mm | 6mm | 8mm | 10mm |
|---|---|---|---|---|---|
| 20mm | ✓ | ✓<br>Fine Pitch | ✓ | ✓ | ✓<br>Coarse Pitch |
| 24mm | ✓ | ✗ | ✓ | ✓ | ✗ |

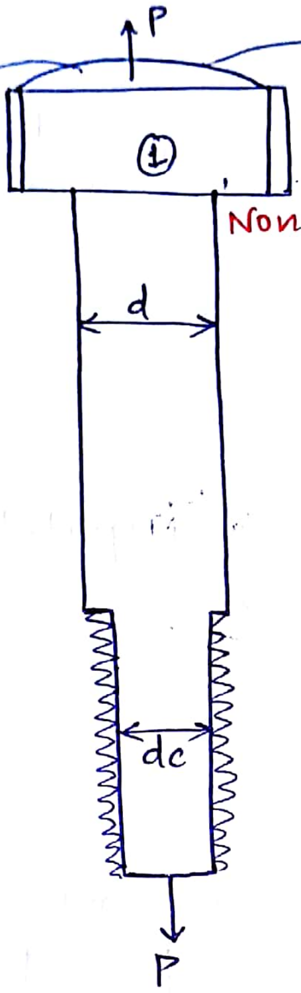
Fine Pitch

coarse pitch

# Uniform strength Bolt :-

A Bolt is said to be uniform strength Bolt when stress induced is equal in each point in the Bolt.

Method-2



Non uniform strength Bolt

$$\sigma_{shank} = \frac{P}{\frac{\pi}{4}d^2}$$

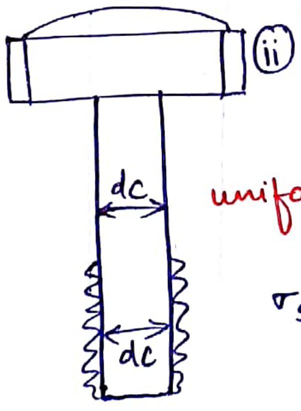
$$\sigma_{core} = \frac{P}{\frac{\pi}{4}d_c^2}$$

$$\sigma_{shank} \neq \sigma_{core}$$

Method-I

making a Bolt of uniform strength

Removal of material

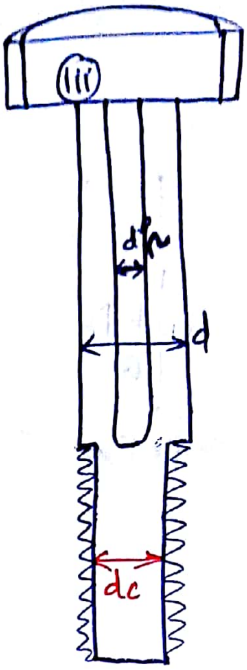


uniform strength bolt

$$\sigma_{shank} = \sigma_{core} = \frac{P}{\frac{\pi}{4}d_c^2}$$

Drilling hole in shank →
Method-2

uniform strength Bolt



$$\frac{\pi}{4}(d^2 - d_h^2) = \frac{\pi}{4} d_c^2$$

$$d^2 - d_h^2 = (.84d)^2$$

$$\boxed{d_h = .542d}$$

$$\boxed{\sigma_{shank} = \sigma_{core}}$$

$\sigma_{impact}$ = I.F. $\sigma_{static}$

$$I.F. = 1 + \sqrt{1 + \frac{2hAE}{WL}}$$

$\sigma_{imp}\downarrow \rightarrow I.F.\downarrow \Rightarrow L\uparrow, E\downarrow, h\downarrow$

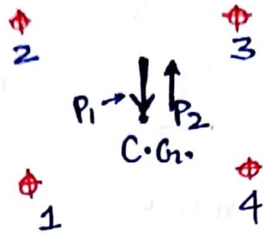
SE ↑ → Can bear more Impact/Fatigue.

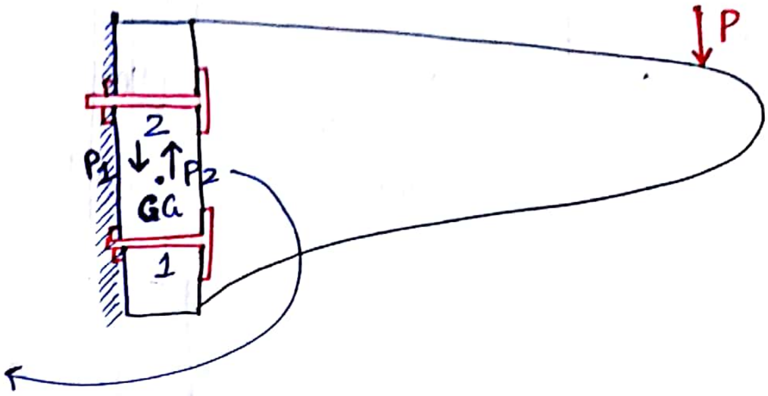
$SE\uparrow = \frac{\sigma^2}{2E}\uparrow$ (more)

Uniform strength bolt ⇒ SE↑↑ as compared to ~~[illegible]~~ normal bolt.

→ ⓐ Uniform strength bolt can bear more impact and fatigue

# * Design of Bolted Joint under eccentric loading:-

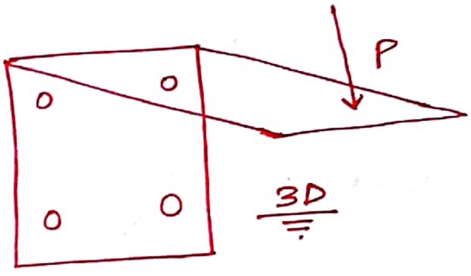


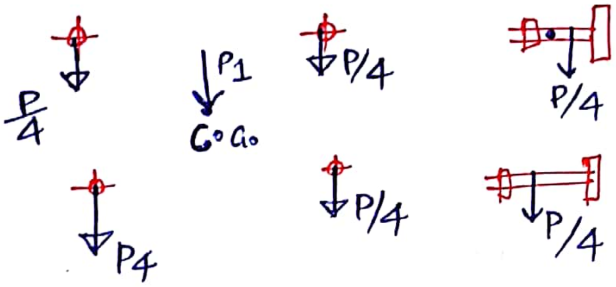


## Effect of $P_1$

⇓

Primary shear force



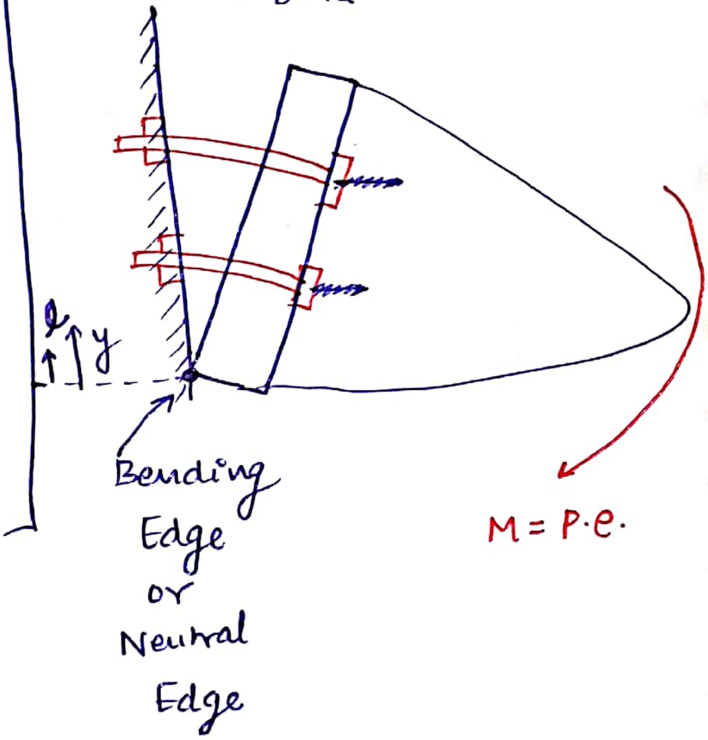


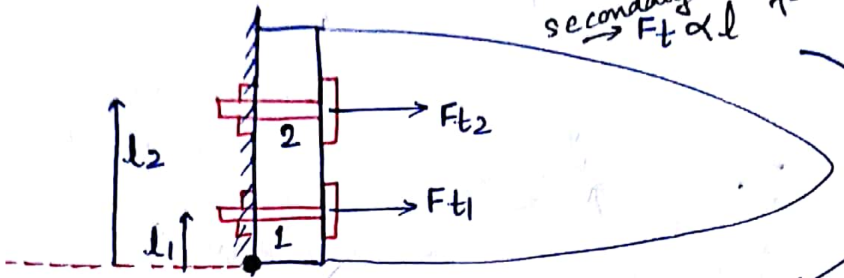
## Effect of $P_2$ and $P$

⇒ Bending couple

$\sigma_b \propto y$

$\sigma_b \propto l$





$\sigma_b \propto l$ secondary tensile force

$\rightarrow F_t \propto l$

$F_{t1} l_1 + F_{t2} l_2 + F_{t3} l_3 + F_{t4} l_4 = P \cdot e$

$F_t \propto l$

$M = P \cdot e$

$\frac{F_{t1}}{l_1}\left[l_1^2 + l_2^2 + l_3^2 + l_4^2\right] = P \cdot e$

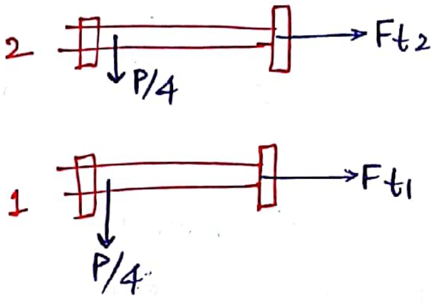
$F_{t1}, F_{t2}, F_{t3}, F_{t4}$ are known

$M = P$

$l_1 = l_4 < l_2 \angle l_3$

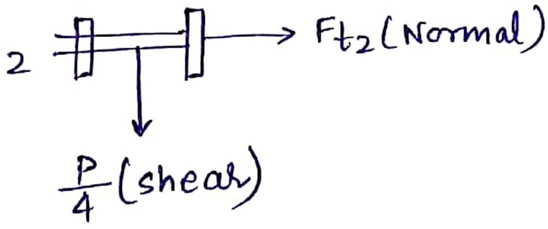
$F_{t1} = F_{t4} \angle F_{t2} = F_{t3}$

## Combined effect



Hence bolt is in combined stress condn.

Critical Bolt ⇒ 2, 3



MSST $\sqrt{\sigma^2 + 4\tau^2} \le \frac{S_{yt}}{N}$

MDET $\sqrt{\sigma^2 + 3t^2} \le \frac{S_{yt}}{N}$

---

MSST $\sqrt{\left(\frac{F_{t2}}{\pi/4\, d_c^2}\right)^2 + 4\left(\frac{P/4}{\pi/4\, d_c^2}\right)^2} \le \frac{S_{yt}}{n}$

$d_c$ = known

$d = d_c/.84$

$\boxed{d = \text{known}}$