1.

(i)	d	(vi)	p
(ii)	c	(vii)	a
(iii)	d	(viii)	d
(iv)	a	(ix)	a
(v)	~~a~~ c	(x)	a

2. Given $S_{yt} = 380\ N/mm^2$

$$\sigma_{max} = \frac{380}{3.5} = 108.57\ N/mm^2$$

Calculation of eccentricity :-

$$R_N = \frac{\left(\frac{b_i + b_o}{2}\right) h}{\left(\frac{b_i R_o - b_o R_i}{h}\right) \ln\left(\frac{R_o}{R_i}\right) - (b_i - b_o)}$$

$$R_N = \frac{\left(\frac{90+30}{2}\right)(120)}{\left(\frac{90\times170 - 30\times50}{120}\right) \ln\left(\frac{170}{50}\right) - (90-30)}$$

$$= 89.1816\ mm.$$

$$R = R_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)}$$

$$= 50 + \frac{120\ (90 + 2\times30)}{3(90+30)} = 100\ mm$$

Calculation of bending stress :-

$$h_i = R_N - R_i = 39.1816\ mm$$

$$A = \frac{1}{2}[h(b_i + b_o)] = 7200\ mm^2$$

$$M_b = PR = (100P)\ N\cdot mm$$

Bending stress ;

$$\sigma_{bi} = \frac{M_b h_i}{A e R_i} = \frac{(7.2435)P}{7200} \ N/mm^2$$

Calculation of direct Tensile stress :-

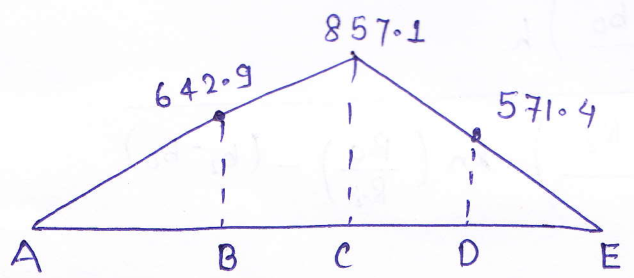
$$\sigma_t = \frac{P}{A} = \frac{P}{7200} \ N/mm^2$$

$$\sigma_{bi} + \sigma_t = \sigma_{max}$$

$$\Rightarrow P = 94827.95 \ N$$

Or,

2. the bending moment diagram is →



Construction of S-N Diagram :-

At B ;

$$S_f = \sigma_b = \frac{32 M_b}{\pi d^3} = 242.54 \ N/mm^2$$

$$S_e' = .5 S_{ut} = 250 \ N/mm^2$$

$K_a = 0.79$

For 30 mm dia, $K_b = .85$

$K_c = .897$

Since $\frac{r}{d} = \frac{3}{30} = .1$; $\frac{D}{d} = 1.5$

$K_t = 1.72$

$q = .78$

$$K_d = \frac{1}{K_f} = \frac{1}{1 + q(K_t - 1)} = \frac{1}{1.5616} = .64$$

$S_e = K_a \cdot K_b \cdot K_c \cdot K_d \cdot S_e'$

$\Rightarrow \cancel{.9S_{ut}} = .79 * .85 * .897 * .64 * 250 = 96.37 \ N/mm^2$

$0.9\, S_{ut} = 0.9 * 500 = 450 \ N/mm^2$

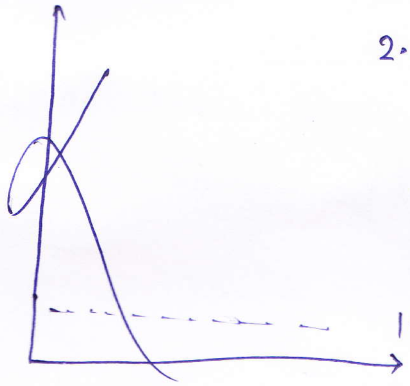
$\log_{10}(0.9\, S_{ut}) = \log_{10}(450) = 2.6532$

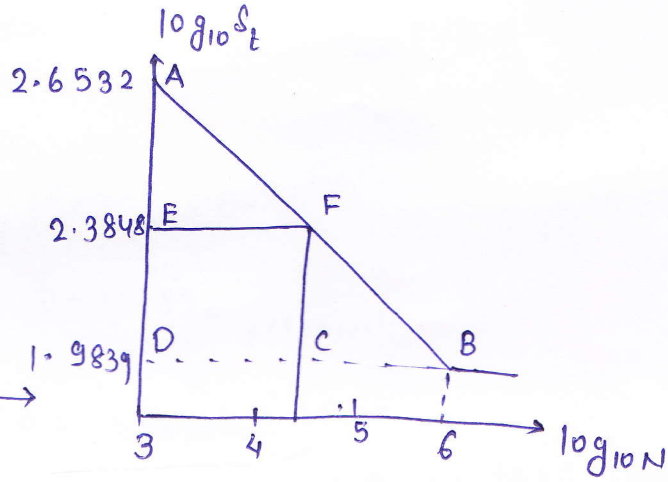
$\log_{10}(S_e) = \log_{10}(96.37) = 1.9839$

$\log_{10}(S_f) = \log_{10}(242.54) = 2.3848$

$\log_{10}(10)^3 = 3$ & $\log_{10}(10^6) = 6$

S-N curve :-





Fatigue life of shaft

$$\overline{EF} = \frac{\overline{DB} * \overline{AE}}{\overline{AD}} = \frac{(6-3) * (2.6532 - 2.3848)}{(2.6532 - 1.9839)}$$

$$= 1.2030$$

$\therefore \log_{10} N = 3 + \overline{EF} = 3 + 1.2030 = 4.2030$

$N = 15958.79$ cycles.

$$\tau = \frac{S_{sy}}{(fs)} = 76\ N/mm^2$$

Direct shear stress on bolt

$$P_1' = P_2' = \frac{\cancel{P}}{(\cancel{N}o} \quad \frac{P}{4} = 6250\ N$$

$\therefore$ Direct shear stress $\Rightarrow$ $\tau = \left(\frac{6250}{A}\right)\ N/mm^2$

Tensile stress on bolt

$$P_1'' = \frac{P e l_1}{2(l_1^2 + l_2^2)} = 8108.11\ N$$

$\therefore$ Tensile stress $\sigma_t = \frac{P_1''}{A} = \frac{8108.11}{A}$

Principal shear stress

$$\tau_{max} = \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + (\tau)^2}$$

$$76 = \frac{7449.69}{A}\ N/mm^2$$

$\therefore$ $A = 98.02\ mm^2$

std. size of coarse thread is M16.

OR

Given $P = 60\ KN$, $e = 200mm$, $\tau = 150\ MPa$
$= 150\ N/mm^2$

Step 1:- Primary shear force

$$P_2' = P_4'' = P_6' = \frac{P}{No.\ of\ \cancel{bolts}} = \frac{60 \times 10^3}{6}$$

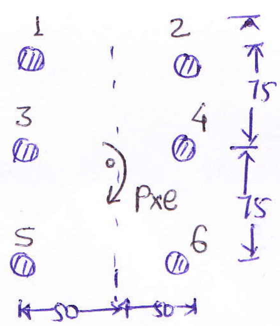
$$P_2' = \cancel{10} \times 10^3\ N$$

Step 2:- Secondary shear force —

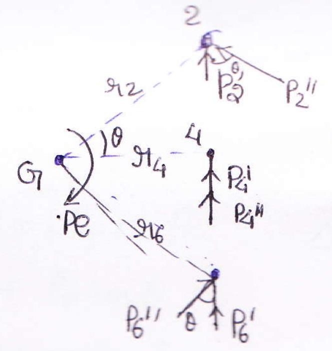
$$r_1 = r_2 = r_3 = r_4 = r_5 = r_6$$

$$= \sqrt{75^2 + 50^2} = 90.14mm$$
$$\simeq 90mm$$

& $r_3 = r_4 = 50mm$



rivets 3, 6, 9 are shown



$$\tan\theta = \cancel{\frac{150}{100}}\ \frac{75}{50} = 56.31^\circ$$

Now $\quad C = \dfrac{P \times e}{(r_1^2 + r_2^2 + \cdots r_6^2)}$

$$\therefore \quad C = \frac{60 \times 1000 \times 200}{4 \times 90^2 + 2 \times 50^2} = 320$$

$\therefore$ Hence $\quad P_2'' = P_6'' = C \cdot r_2 = 320 \times 90$

$P_2'' = 28800\,N$

& $\quad P_4'' = P_3'' = C \times r_4 = 320 \times 50$

$= 16000\,N$

Step 3 :- The Resultant force :-

$$P_2 = \sqrt{(10 \times 10^3 + 28800 \cos 56.31^\circ)^2 + (28800 \sin 56.31^\circ)^2}$$

$$P_2 = 35340.44\,N = P_6$$

Now $\quad P_4 = P_4' + P_4''$

$= 10 \times 10^3 + 16000 = 26000\,N.$

Therefore rivets 2 & 6 are subjected to maximum shear force.

Step 4 :- Diameter of Rivets :- According to max. shear stress theory –

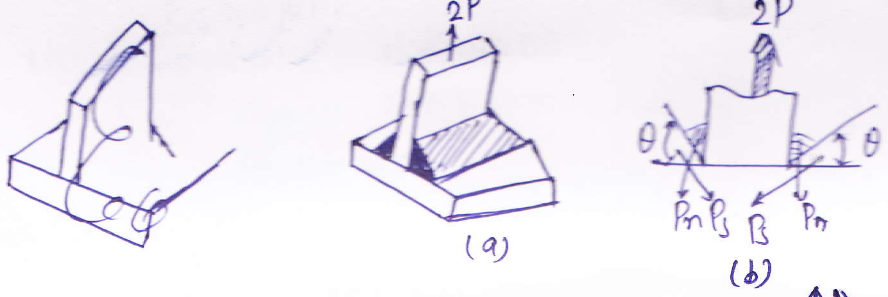
$$P_2 = \frac{\pi}{4} \cdot d^2 \cdot \tau$$

$$35340.45 = \frac{\pi}{4} \times d^2 \times 150$$

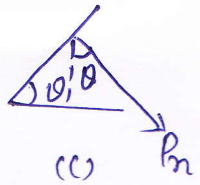
$$\therefore \quad d^2 = 299.98$$

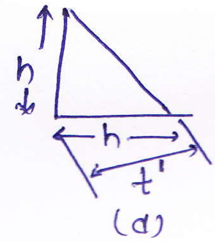
$\Rightarrow \boxed{d = 17.32\,mm.}$ Ans

Q.4



From (b) & (c)





$$2P = 2P_s \sin\theta + 2P_n \cos\theta$$

$$P = P_s \sin\theta + P_n \cos\theta$$

As P_s & P_n are vertical $\therefore$ Σ

$\therefore P_s \cos\theta = P_n \sin\theta$

$$P_n = \frac{P_s \cos\theta}{\sin\theta} \qquad \therefore P = P_s \sin\theta + \frac{P_s \cos\theta \cos\theta}{\sin\theta}$$

$$\Rightarrow \boxed{P \sin\theta = P_s} \quad \text{— (1)}$$

$$t' = \frac{h}{\sin\theta + \cos\theta}$$

$$\tau = \frac{P_s}{t' \cdot l} \Rightarrow \tau = \frac{P \sin\theta (\sin\theta + \cos\theta)}{hl}$$

For minimum shear stress.

$$\frac{\partial \tau}{\partial \theta} = 0$$

$$\Rightarrow \sin 2\theta + \cos 2\theta = 0$$

$$\Rightarrow \theta = 67.5^\circ$$

$\therefore$ Cond^n for plane with max. shear stress is $\theta = 67.5^\circ$

$$\tau_{max} = \frac{1.21 P}{hl}$$

$l = 1$ mm

$$P_{all} = \frac{h \tau_{max}}{1.21} \qquad [\text{for transverse load}]$$

& P_{all} for ||el weld $= .707 h \tau_{max}$

$$\therefore \frac{P_{all}.\ \text{transverse load}}{P_{all}.\ \text{Parallel load}} = 1.17$$

OR

$P = 200\ kN$

Part-I : Let the material is 40 C8 -

$\therefore\ \sigma_{YTS} = 380\ N/mm^2$

fos = 2.5

Let $S_{YC} = 2S_{YT}$

Part-II :- Permissible stresses -

$$\sigma_t = \frac{S_{YT}}{fos} = \frac{380}{2.5} = 152\ N/mm^2$$

$$\sigma_c = \frac{S_{YC}}{fos} = \frac{2 \cdot S_{YT}}{fos} = 304\ N/mm^2$$

$$\tau = \frac{S_{SY}}{(fos)} = \frac{0.5 \times S_{YT}}{2.5} = 76\ N/mm^2$$

Part III - Calculation of dimensions -

Step① - Diameter of Rod -

$$D = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4(200 \times 10^3)}{\pi \times 152}} = 40.93\ mm \simeq 45\ mm$$

~~Step② - Thickness of cotter -~~

$\cancel{t = 0.31d = 13.95\ mm \simeq 14\ mm}$

~~Step③ - Diameter (d2) of spigot -~~

Step② - Enlarged dia. D_1 -

$$D_1 = 1.1 D = 49.5\ mm \simeq 50\ mm.$$

Step③ -

$$a = 0.75 D = 33.75 \simeq 34\ mm$$

$$b = 1.25 D = 56.25\ mm \simeq 57\ mm$$

Step④ - Diameter of pen -

$$d = \sqrt{\frac{2P}{\pi \tau}} = \sqrt{\frac{2 \times 200 \times 10^3}{\pi \times 76}} = 40.93\ mm = 42\ mm$$

Also. $$d = \sqrt{\frac{32}{\pi \sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

Step⑤ - Diamensions of d_o & d_1 -

$$d_o = 2d = 2 \times 45 = 90\ mm.$$

$$d_1 = 1.5d = 1.5 \times 40 = 60\ mm$$

Step ⑥ - check of stresses for eye -

$$\sigma_t = \frac{P}{b(d_o - d)} = \frac{200 \times 10^3}{57(90-45)} = 77.97\ N/mm^2 < \sigma_t = 152\ N/mm^2$$

$$\sigma_c = \frac{P}{bd} = \frac{200 \times 10^3}{57 \times 45} = 77.97\ N/mm^2 < 304\ N/mm^2$$

Step⑦ - $$\tau = \frac{P}{b(d_o - d)} = \frac{200 \times 10^3}{57(90-45)} = 77.97\ N/mm^2 < 7.$$

Similarly check for the fork stresses -

Note It is observed that stresses are in limit & the design is safe.

Permissible compressive & shear stresses

$$S_{yc} = S_{yt} = 460\ N/mm^2$$

$$\sigma_c = \frac{S_{yc}}{fs} = 153.33\ N/mm^2$$

A/c to max^m shear stress theory of failure

$$S_{sy} = .5 S_{yt} = 230\ N/mm^2$$

$$\tau = \frac{S_{sy}}{fs} = 76.67\ N/mm^2$$

Torque transmitted by shaft

$$M_t = \cancel{1989} \ 198943.68\ Nmm$$

Key Dimension

$$b = h = \frac{d}{4} = 6.25 \approx 6mm$$

$$l = \frac{2M_t}{\tau d b} = 34.60\ mm$$

$$l = \frac{4M_t}{\sigma_c d h} = 34.60\ mm \approx 35\ mm$$

$\therefore$ dimension $= 6 \times 6 \times 35mm$.

Or

For mat. of shafts, key & bolt

$$\sigma_t = \frac{S_{yt}}{fs} = 80\ N/mm^2 \qquad \sigma_c = \frac{1.5 \times 400}{5} = 120\ N/mm^2$$

$$\tau = \frac{S_{sy}}{(fs)} = 40N/mm^2$$

Dia. of each shaft

$$M_t = \frac{60 \times 10^6\ kW}{2\pi n} = 3978873.58\ Nmm$$

$$\tau = \frac{16 M_t}{\pi d^3} \Rightarrow d = 79.72 \approx 80mm$$

$D = 2.5d = 200\text{ mm}$

$L = 3.5d = 280\text{ mm}$

Dia. of clamping bolts – Friction basis

$$P_1 = \frac{2M_t}{f d n} = 41446.6\text{ N}$$

$$P_1 = \frac{\pi}{4} d_1^2 \sigma_t \Rightarrow d_1 = 25.68 \approx 26\text{ mm.}$$

Dia. of clamping bolts — Empirical relⁿ:–

$$d_1 = .15d + 15 = 27\text{ mm}$$

Dimension of key $l = \frac{L}{2} = 140\text{ mm.}$

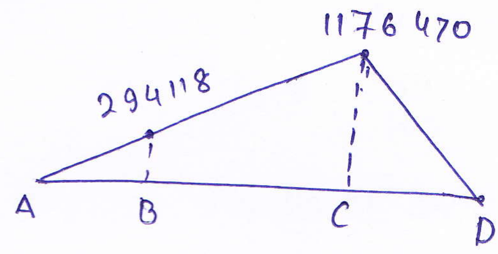
i.e. Dimension (key) = 22 × 14 × 140

<u>Check</u>

$$\tau = \frac{2M_t}{d b l} = 32.3\text{ N/mm}^2 < 40\text{ N/mm}^2$$

$$\sigma_c = \frac{4M_t}{d b l} = 101.5\text{ N/mm}^2 < 120\text{ N/mm}^2.$$

<u>Q.6</u>



Resultant B.M

Permissible shear stress

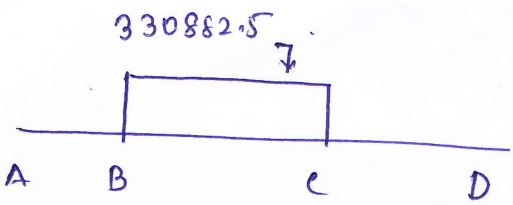
$$\tau = \frac{S_{sy}}{fs} = 66.67\text{ N/mm}^2$$

<u>Torsional moment</u>

$$\frac{P_1}{P_2} = e^{\mu\theta}$$

$P_2 = 1176.47\text{ N}$

$P_1 = 2500\text{ N}$



<u>Torsional Moment</u>

<u>Torque app.</u> $= M_t = 330882.5\text{ N-m}$

Bending moment $(P_3 - P_4)R_2 = M_b$

$$\Rightarrow \frac{P_3}{P_4} = e^{\mu\theta}$$

$P_3 = 5000\text{ N}$

$P_4 = 2352.94\text{ N}$

Resulting Bending moment

$$(M_b) \text{ at } B = \sqrt{(588232)^2 + (294118)^2}$$

$$= 667664.26 \text{ Nmm}$$

$$(M_b) \text{ at } C = \sqrt{(147058)^2 + (1176470)^2}$$

$$= 1185625.45 \text{ Nmm}$$

$\therefore$ stress is max. @ C

$$\therefore \tau_{max} = \frac{16}{\pi d^3} \sqrt{(M_b)^2 + (M_t)^2}$$

$$d = 45.47 \text{ mm}$$

OR

Given:

Power = 10 KW

N = 1440 rpm

$T_{max} = T_m \times 1.5$

$S_{yT} = 380 \text{ N/mm}^2$

& fos = 4

Step ①:- Permissible stresses –

$$\sigma_{yT} = \frac{S_{yT}}{fos} = \frac{380}{4} = 95 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{fos} = \frac{0.5\, S_{yT}}{4} = 47.5 \text{ N/mm}^2$$

Step ②:- Diameter of shaft :-

$$\text{Power} = \frac{2\pi N T_m}{60 \times 10^3} \text{ KW}$$

$$10 = \frac{2\pi \times 1440 \times T_m}{60 \times 10^3}$$

$$T_m = 66.315 \text{ N-m}$$

$$\therefore T_{max} = 1.5 \times T_m$$

$$= 99.47 \text{ N-m}$$

$$= 99.47 \times 10^3 \text{ N-mm}$$

Now from eqn –

$$T_{max} = \frac{\pi}{16} \times d^3 \times \tau$$

$$\therefore d^3 =$$

$$d^3 = \frac{16 \cdot T_{max}}{\pi \times \tau} = \frac{16 \times 99.47 \times 10^3}{47.5\pi}$$

$$d^3 = 10665.2$$

$\therefore$ $d = 22.01$ mm Ans

$\simeq 23$ mm

(i) Diameter of shaft $= 23$ mm.

Now let us suppose the diameter of pump is d',

Hence for max. power transmission – d_c = dia. of centrifugal pump

$$T_{max}|_c = \frac{\pi}{16} \times d_{cp}^3 \times \tau.$$

$\because$ $P_s \propto N_s T_{max}$ & $P_{cp} : \propto N_p \times T_{cp}$

$\therefore$ For same power output –

$$\frac{N_s \cdot T_{max}}{N_p \cdot T_{cp}} = 1$$

$$\Rightarrow \quad N_s \cdot T_{max} = N_p \cdot T_{cp}$$

$$480 \times \frac{\pi}{16} \times d^3 \times 47.5 = 1440 \times 99.47 \times 10^3$$

$$\Rightarrow \quad d^3 = \frac{16 \times 99.47 \times 10^3}{48\pi \times 47.5} = 31995.57$$

$$\Rightarrow \quad \underline{d = 31.74\,mm.}$$

Ans