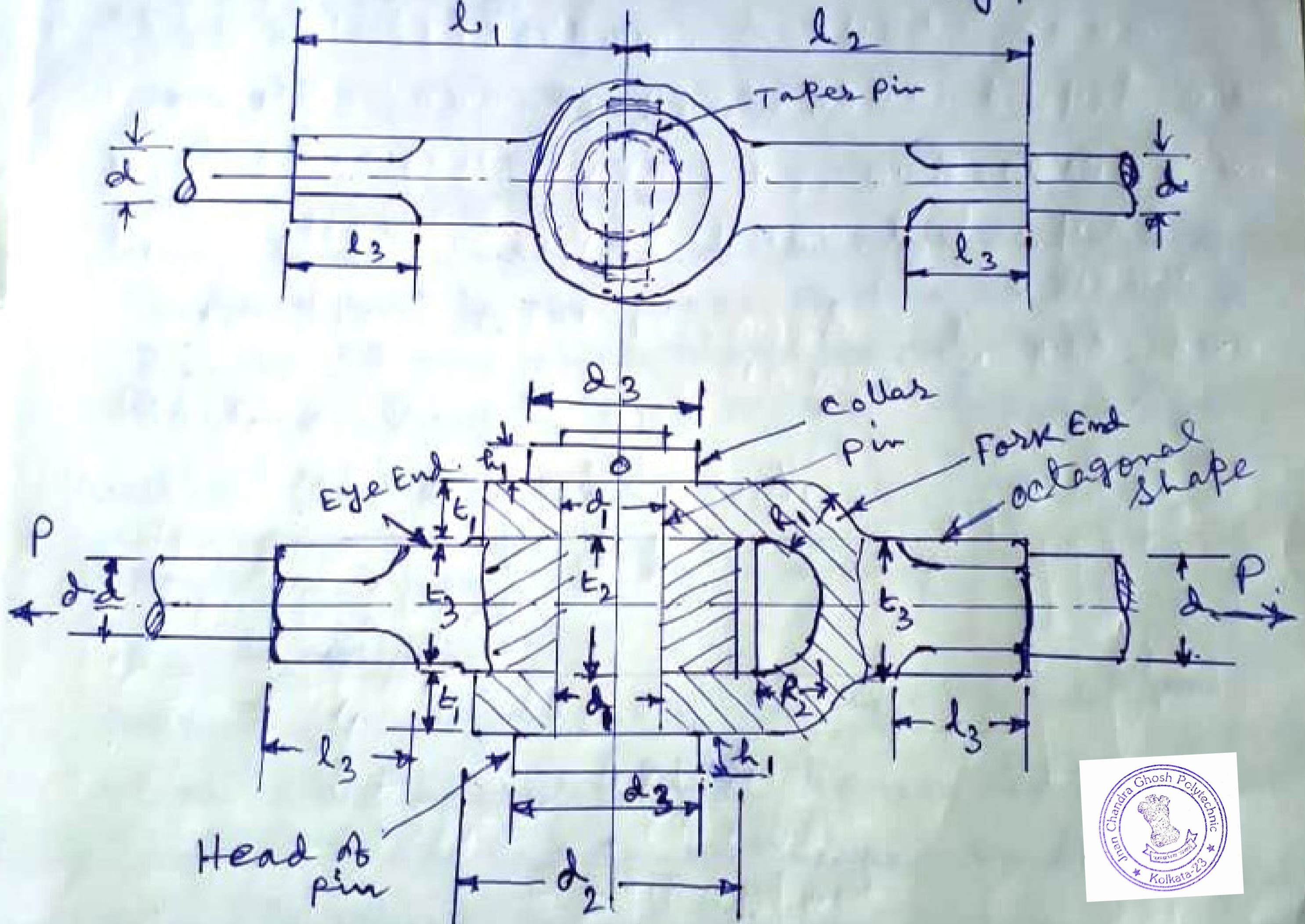
Machine Design DME/S6

Design of Knuckle Joint.

Prob: Design a knuckle joint to transmit an axial load of 100 kN, assuming the working stresses for both pin & rod material as 80 N/mm² in tension, 60 N/mm² in shear, and 120 N/mm² in crushing. Assume that the rods to be connected are not very long.



$F = 100\ kN$, $\sigma_t = 80\ N/mm^2$, $\sigma_c = 120\ N/mm^2$

$\tau_s = 60\ N/mm^2$

$\sigma_t = f_t \quad \sigma_c = f_c \quad \tau_s = f_s$

Design of rod.

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

$d = \sqrt{\frac{4P}{\pi \sigma_t}} = \sqrt{\frac{4 \times 100 \times 10^3}{\pi \times 80}} = 39.90$ mm.

we take, $d = 40$ mm.

We adopt Emperical relation

$d_1 = d = 40$ mm.

$d_2 = 2d = 2 \times 40 = 80$ mm., $d_3 = 1.5d = 60$ mm.

$t_1 = 0.75d = 30$ mm, $t_2 = 1.25d = 50$ mm, $t_3 = 1.2d = 48$ mm.

$R_1 = 0.6d = 0.6 \times 40 = 24$ mm. $R_2 = 0.8d = 0.8 \times 40 = 32$ mm.

$l_1 = 4d = 160$ mm, $l_2 = 4.5d = 180$ mm, $l_3 = t_3 = 1.2d = 48$ mm.

mean dia. of Taper Pin $= 0.25d = 10$ mm. $h_1 = 0.5d = 20$ mm.

Design of Pin

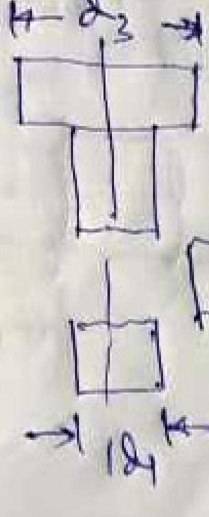
i) for checking shear failure

The pin is subjected to double shear

So, $F = 2 \times \frac{\pi}{4} d_1^2 \times \tau_s$

or, $\tau_s = \frac{4F}{2 \times \pi \times d_1^2} = \frac{4 \times 100 \times 10^3}{2 \times \pi \times (40)^2}$

$= 39.80$ N/mm^2



which is less than allowable shear stress

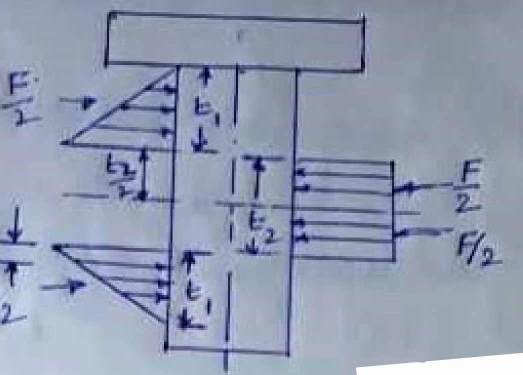
So, it is O.K.

ii) failure of pin due to Bending stress.

From fig

Maximum Bending Moment

$= \frac{F}{2}\left[\left(\frac{t_1}{3}+\frac{t_2}{2}\right)-\frac{t_2}{4}\right] = \frac{F}{2}\left[\frac{t_2}{4}+\frac{t_1}{3}\right]$



Now, from the relation

$\frac{M_b}{I} = \frac{\sigma_b}{y}$ or, $M_b = \frac{I}{y}\sigma_b$

$= \sigma_b \cdot Z$

$\therefore \frac{F}{2}\left[\frac{t_2}{4}+\frac{t_1}{3}\right] = \sigma_b \frac{\pi}{32} d_1^3$

$\frac{100\times10^3}{2}\left[\frac{50}{4}+\frac{30}{3}\right] = \sigma_b \frac{\pi}{32}\times(40)^3$

or, $\sigma_b = 179.14\ N/mm^2$

which is much more than allowable Bending Stress i.e. 80 N/mm², So we adopt the dia of Knuckle Pin is 53 mm, which gives the induced Bending Stress is 77.0 N/mm². Hence Safe.

Design for Eye End.

i) Failure of Eye End due to tearing

$F = (d_2 - d_1)\times t_2 \times \sigma_t$

$100\times10^3 = (80-53)\times50\times\sigma_t$ or, $\sigma_t = 74.07\ N/mm^2$

which is less than 80 N/mm². Hence Safe.

ii) Failure of Eye End can be checked by Shearing

$F = (d_2 - d_1)\times t_2 \times \tau_s$

or, $100\times10^3 = (80-53)\times50\times\tau_s$ or, $\tau_s = 74.07\ N/mm^2$

which is more than allowable Shear stress 60 N/mm².
So, we adopt $d_2 = 90$ mm instead of 80 mm.

which gives the induced Shear stress is 54.05 N/mm^2 Hence Safe.

iii) Failure of Eye end can be checked under crushing

$F = d_1 \times t_2 \times \sigma_c$

or, $100 \times 10^3 = 53 \times 50 \times \sigma_c$ or, $\sigma_c = 37.74$ N/mm^2

which is less than allowable crushing stress 120 N/mm^2

Hence Safe.

Design of Fork End

i) Failure of fork end by tearing

$F = (d_2 - d_1) \times 2 \times t_1 \times \sigma_t$

$100 \times 10^3 = (90-53) \times 2 \times 30 \times \sigma_t$ or, $\sigma_t = 45.04$ N/mm^2

which is less than allowable tensile stress 80 N/mm^2

Hence Safe.

ii) Failure of fork end under shearing stress

$F = (d_2 - d_1) \times 2 \times t_1 \times \tau_s$

$100 \times 10^3 = (90-53) \times 2 \times 30 \times \tau_s$ or $\tau_s = 45.04$ N/mm^2

which is less than allowable shear stress i.e. 60 N/mm^2. Hence Safe.

iii) Failure of fork end under crushing

$F = 2 \times t_1 \times d_1 \times \sigma_c$

$100 \times 10^3 = 2 \times 30 \times 53 \times \sigma_c$ or, $\sigma_c = 31.45$ N/mm^2

which is much below the allowable crushing stress i.e. 120 N/mm^2. Hence it is Safe.

So, Final Dimensions are.

$d = d_1 = 53$ mm, $d_2 = 90$ mm. $d_3 = 60$ mm.

$t_1 = 30$ mm. $t_2 = 50$ mm. $t_3 = 48$ mm.

$R_1 = 24$ mm. $R_2 = 32$ mm

$l_1 = 160$ mm. $l_2 = 180$ mm. $l_3 = t_3 = 48$ mm. $h_1 = 20$ mm.

Mean dia. of Taper Pin = 10 mm.