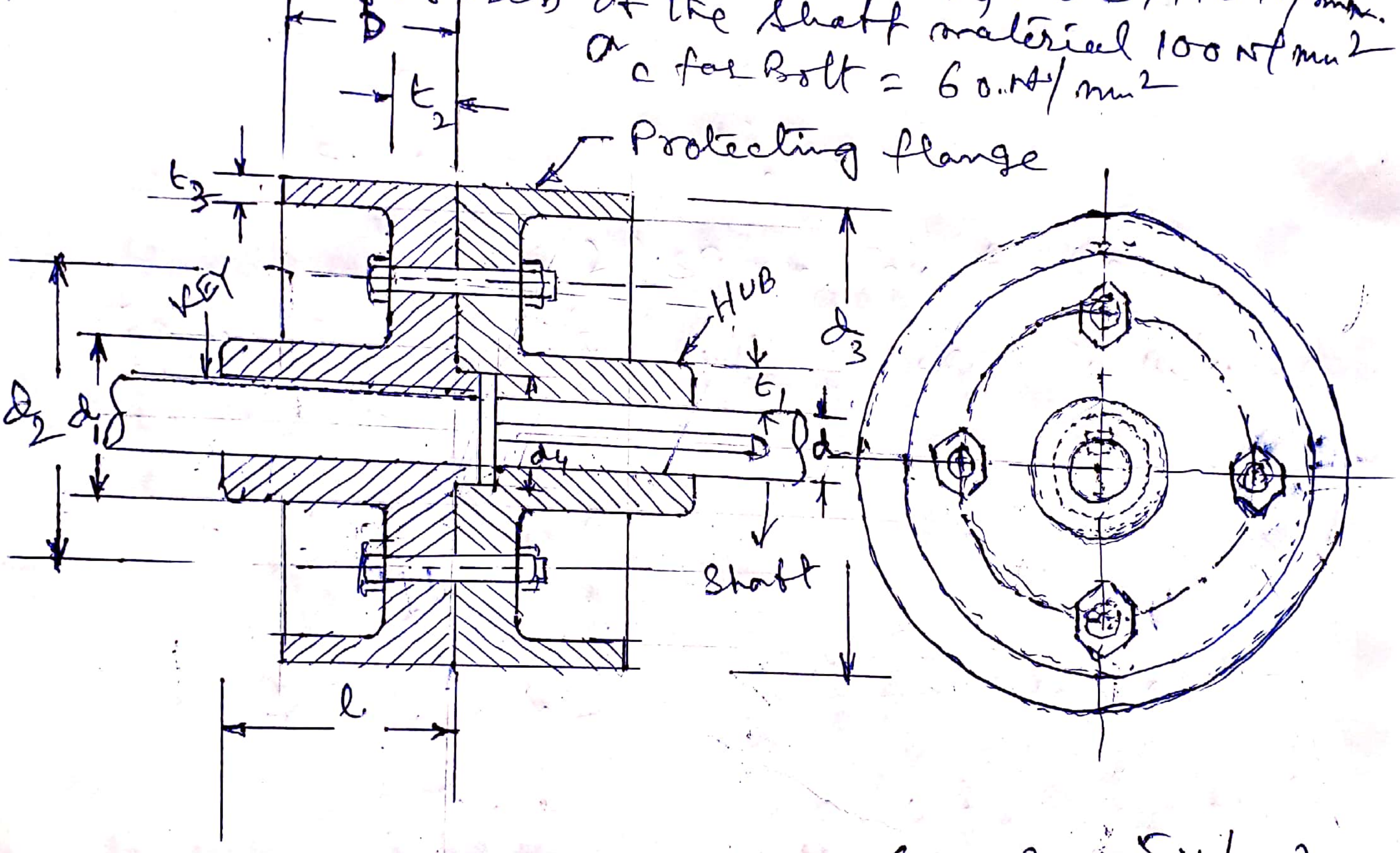
Design of Protected type flange coupling

Prob:- Design a protected type flange coupling to transmit a power of 15 KW at 750 r.p.m. The allowable ~~stress~~ shear stress in the shaft material is 50 N/mm² and the coupling bolt is 30 N/mm² and the angle of twist is not to exceed 0.75 degrees in a length of 20 times diameter. Assume $G = 0.84 \times 10^5 \, N/mm^2$. Crushing stress of the shaft material 100 N/mm²

σ_c for Bolt = 60 N/mm²



$P = 15\,KW$, $N = 750\,r.p.m$, $G = 0.8 \times 10^5 \, N/mm^2$

For shaft material	For Bolt material:
$\sigma_c = 100\,N/mm^2$	$\tau_s = 30\,N/mm^2$
$\tau_s = 50\,N/mm^2$	$\sigma_c = 60\,N/mm^2$

$\theta_{max} = 0.75$ deg. in a length of 20 dia.

$$P = \frac{2\pi N T_{av}}{60\times1000}$$ as usual notations

$$15 = \frac{2\pi\times750\times T_{av}}{60\times1000} \quad \text{or} \quad T_{av} = \frac{15\times60\times1000}{2\pi\times750}$$
$$= 191.08 \text{ N-m.}$$

Assume maximum Torque (T_{max}) is 20% greater than the Average Torque (T_{av})

So. $T_{max} = 1.2\,T_{av} = 1.2\times191.08 = 229.30$ N-m.

Let 'd' be the diameter of the Shaft, then

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

$$229.30\times10^3 = \frac{\pi}{16} d^3 \times 50$$

or, $d = 28.59$ mm. Say, $d = 30$ mm.

From the rigidity point of view the induced stress in the shaft can be checked.

$$\frac{\tau_s}{d/2} = \frac{G\theta}{l}$$

$$\text{or, } \tau_s = \frac{G\theta}{l}\times\frac{d}{2} = \frac{0.84\times10^5\times30}{20\times2\times30}\times\frac{\pi}{180}\times0.75$$

$= 27.475$ N/mm², which is below the safe stress i.e. 50 N/mm².

Hence, the Shaft is safe against Twisting moment.

Design of Key

From Emperical relation

width of key, $w = \frac{d}{4} = \frac{30}{4} = 7.5mm \simeq 8mm.$

Thickness of key, $t = \frac{d}{4} = \frac{30}{4} = 7.5mm \simeq 8mm.$

Shear stress induced in the key can be checked by the relation.

$$T_{max} = L \times W \times \tau_s \times \frac{d}{2}$$

where,
L = length of the key
$= 1.5d$
$= 1.5 \times 30 = 45mm.$

$$\tau_s = \frac{T_{max} \times 2}{L \times W \times d}$$

$$= \frac{229.30 \times 10^3 \times 2}{45 \times 8 \times 30} = 42.46 N/mm^2$$

which is less than allowable shear stress i.e. 50 N/mm². Hence safe.

Checking for crushing stress induced in the key.
From the relation.

$$T_{max} = L \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

$$\text{or, } \sigma_c = \frac{2 \times 2 \times T_{max}}{L \times t \times d} = \frac{4 \times 229.30 \times 10^3}{45 \times 8 \times 30}$$

$$= 84.92 N/mm^2$$

which is less than the allowable crushing stress i.e. 100 N/mm². Hence safe.

Design of Hub

From the emperical relation the dia. of the hub, $d_1 = 1.75d + 7$ mm.

So, $d_1 = 1.75 \times 30 + 7 = 59.5$ mm $\simeq 60$ mm.

The can be checked from the induced shear stress, by the relation.

$$T_{max} = \frac{\pi}{16} \frac{d_1^4 - d^4}{d_1} \times \tau_s$$

$$\tau_s = \frac{16 T_{max} \times d_1}{\pi (d_1^4 - d^4)} = \frac{16 \times 229.30 \times 10^3 \times 60}{\pi \{(60)^4 - (30)^4\}}$$

$= 5.77$ N/mm^2

which is less than safe allowable shear stress i.e. 50 N/mm^2, Hence safe.

Length of hub $= 1.5d = 30 \times 1.5 = 45$ mm.

So. length of key = length of hub as.

the length of key may be increased by 5 mm. by smooth fitting between the shaft & the hub. So. length of key is equal to $L_1 = 50$ mm.

Design of Bolts

Bolt circle dia, $d_2 = 3d = 3 \times 30 = 90$ mm.

No. of bolts, z, can be found from the relation

$$z = \frac{4d}{150} + 3 = \frac{4 \times 30}{150} + 3 = 3.8 \simeq 4$$

Say, No. of bolts, $z = 4$

The size of bolt can be obtained from the relation.

$$T_{max} = 4 \times \frac{\pi}{4} d_b^2 \times \tau_{s_b} \times \frac{d_2}{2}$$

$$229.30 \times 10^3 = 4 \times \frac{\pi}{4} d_b^2 \times 30 \times \frac{90}{2}$$

$$\text{or, } d_b = \sqrt{\frac{229.30 \times 10^3}{30 \times 45 \times \pi}} = 7.35 \text{ mm.}$$

So, we take M 10×1.5

Thickness of hub, $t_1 = \frac{d_1 - d}{2} = \frac{60 - 30}{2} = 15$ mm.

Thickness of flange, $t_2 = \frac{1}{2} t_1 + 6.5$ mm.

$$= \frac{1}{2} \times 15 + 6.5 = 14 \text{ mm.}$$

The induced crushing stress in the flange of the coupling can be checked by the relation,

$$T_{max} = d_b \times t_f \times 4 \times \sigma_c \times \frac{d_2}{2}$$

$$\text{or, } 229.30 \times 10^3 = 10 \times 14 \times 4 \times \sigma_c \times 45$$

$$\text{or, } \sigma_c = \frac{229.30 \times 10^3}{10 \times 14 \times 4 \times 45} = 9.09 \text{ N/mm}^2$$

which is below the safe stress i.e. 30 N/mm^2

Hence safe.

The thickness of the protecting flange

$$t_3 = \frac{1}{2} t_2 = \frac{1}{2} \times 14 = 7 \text{ mm.}$$

Outer diameter of flanges

Diameter of the head of the socket wrench

$= 1.85 d_b + 8$ mm.

$= 1.85 \times 10 + 8 = 26.5$ mm Say, 27 mm.

Outer diameter of flange, d_3

$d_3 = 2$ [radius of hub + clearance + dia. of head of socket wrench + clearance + thickness of protecting flange]

$= 2[30+5+27+5+7]$

$= 148$ mm Say 150 mm.

Final Data:

$d = 30$ mm

$d_1 = 60$ mm.

$d_2 = 90$ mm.

$d_3 = 150$ mm.

$d_4 = 1.5d$
$= 45$ mm.

$l = 45$ mm.

$t_1 = 15$ mm.

$t_2 = 14$ mm.

$t_3 = 7$ mm.

$b = 2t_2$
$= 28$ mm.

Size of Bolt

M 10×1.5

Z = No. of bolts
$= 4$

Square key

$W = 8$ mm.

$t = 8$ mm.

Length of key l_1:

$l_1 = 50$ mm.