Design of Machine Elements DME/S6

Design of cast Iron (C.I.) Pulley

Prob:- Design a C.I. Pulley (Driver) which transmit a power of 15KW. Speed of the motor shaft 500 rev./min. Speed ratio is 4 and there is considerable variation of speed. For material for shaft allowable shear stress is 50 N/mm², for key material, allowable shear stress = 40 N/mm² allowable crushing stress = 100 N/mm², for cast ~~ir~~ iron, allowable tensile stress is 20 N/mm² and allowable shear stress is 15 N/mm² Assume any data if reqd.

Solⁿ:

$P = 15 KW.$

$N = 500 rev/min$ Speed ratio = 4

For shaft. ~~For~~ Key material

$\tau_s = 50 N/mm^2$ $\tau_s = 40 N/mm^2$

$\sigma_c = 100 N/mm^2$

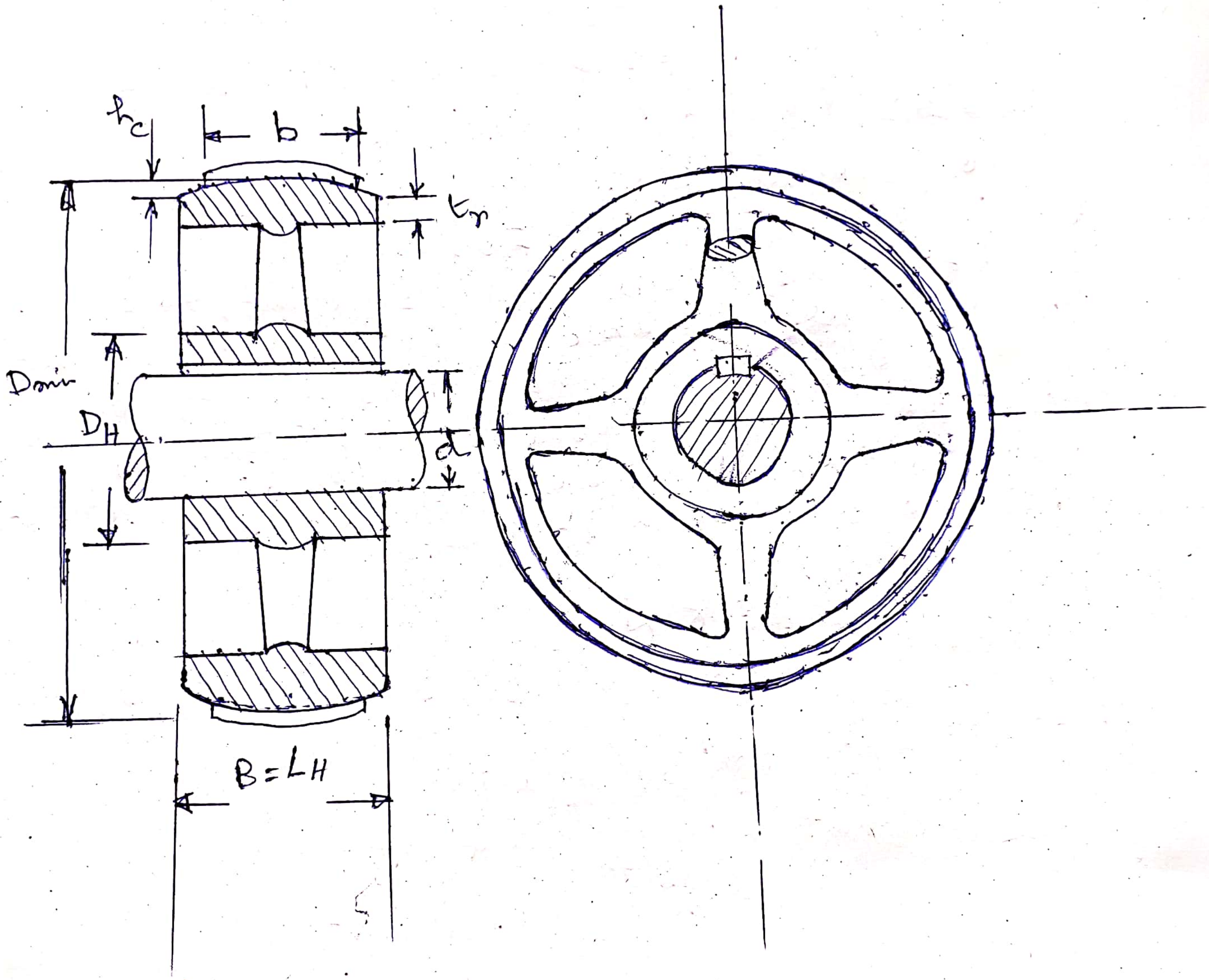
For C.I.

$\sigma_t = 20 N/mm^2$

$\tau_s = 15 N/mm^2$

Taking service factor is 1.3

1 A



Dia. of the Pulley

The dia. of the motor (driver) shaft pulley can be found by the ration

$$D_{min} = 1{\cdot}10 \text{ to } 1{\cdot}33 \sqrt[3]{\frac{P}{N}}$$

where, P is the power in kW and N is the r.p.m. of the shaft.

We take $D_{min} = 1{\cdot}3 \sqrt[3]{\frac{P}{N_{max}}}$

Since there is considerable variation of load, so let us take the service factor as 1·3

Therefore the design power

P = Power ~~to~~ to be transmitted × service factor

$= 15 \times 1{\cdot}3 = 19{\cdot}5$ kW.

$N_{max} = 500$ rev./min

So, $D_{min} = 1{\cdot}3 \sqrt[3]{\frac{19{\cdot}5}{500}} = 0{\cdot}4408$ mm.

say 0·45 m. or. 45 cm.

So, the dia. of the driven pulley is 180 cm = 1·80 m.

neglecting slip between belt and the pulley

Belt speed is V,

So, $V = \frac{\pi D N}{60} = \frac{\pi \times 0{\cdot}45 \times 500}{60} = 11{\cdot}775$ m/sec.

Since the belt speed is more than 10 m/sec.

therefore the centrifugal force will be considered.

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The angle of contact between the belt and the smaller pulley is θ.

So, $\theta = 180 - 2\sin^{-1}\dfrac{D_1 - D}{2c}$

where, D_1 is the dia. of the larger (driven) pulley

c is the centre distance between two pulley

then $c \geq 3.5 D_1$ i.e. 6.3 m.

$\theta = 180 - 2\sin^{-1}\dfrac{1.8 - 0.45}{2 \times 6.3} = 167.7^\circ = 2.92$ rad.

Now, Power, $P = \dfrac{(T_1 - T_2) \times V}{1000}$ where, T_1 & T_2 are the tension on tight side & tension on slack side of the belt respectively.

$19.5 = \dfrac{(T_1 - T_2) \times 11.775}{1000}$

or $T_1 - T_2 = \dfrac{19.5 \times 1000}{11.775} = 1656.05$ N ...(1)

Now, $\dfrac{T_1}{T_2} = e^{\mu\theta}$

where, μ = co-efficient of friction between leather belt & pulley & may be taken as 0.35

So, $\dfrac{T_1}{T_2} = e^{0.35 \times 2.92} = 2.77$ ...(2)

From. Eqn. (1) & (2)

$T_1 = 2591.67$ N $T_2 = 935.62$ N

We know, the centrifugal tension of a belt is given by $T_c = m v^2$ Newton.

where, $m = \rho l b t$

m = mass of belt in kg per meter length of belt

ρ = density of belt material = 1000 kg/m³ for leather belt.

l = length of belt = 1 m.

b = width of belt in 'm'

t = thickness of belt in 'm'

So, taking thickness of belt $t = 0.03 D_{min}$

$= 13.5$ mm say 14 mm.

So, $T_c = 1000 \times 1 \times b \times 0.014 \times (11.775)^2$

$= 1941.10\, b$ Newton

Now, the working stress of leather belt is varying from 2.1 MPa to 3.15 MPa.

So we take, 3.0 MPa is the working stress of leather belt.

Now, $\sigma_{max} = \dfrac{T_1 + T_c}{b \times t}$

$\therefore\ 3 \times 10^6 = \dfrac{2591.67 + 1941.10 b}{b \times 0.014}$

$\therefore\ 42{,}000 b = 2591.67 + 1941.10 b$

$\therefore\ b = 0.06469$ m. $= 64.69$ mm say 65 mm.

width of the pulley face, $B = 1.25 b = 1.25 \times 65$

$= 81.25$ mm.

Say 82 mm.

According to empirical formula, the rim thickness of the pulley, t_r

So, $t_r = \dfrac{D_{min}}{300} + 2$ mm.

$= \dfrac{450}{300} + 2 = 3.5$ mm.

Another relation of the rim thickness

$t_r = (0.005 D_{min} + 0.003)$ m

$= (0.005 \times 0.45 + 0.003) = 0.00525$ m.

$= 5.25$ mm.

We take $t_r = 6$ mm.

Crown height $= 1.5$ mm per metre of face width.

i.e. about 1.5 mm. taken, $h_c = 1.5$ mm.

Design of hub

d = dia. of the shaft = Inner dia. of the hub

D_H = outer dia. of the hub.

L_H = Length of the hub

P = Power transmitted by the shaft taken with service factor

$= 19.5$ KW

So, $P = \dfrac{2\pi T_{max} \cdot N}{60 \times 1000}$

$19.5 = \dfrac{2\pi \times 500 \times T_{max}}{60 \times 1000}$

$\therefore T_{max} = 372.61$ N-m.

Equivalent torque on the shaft is given by

$$T_E = \sqrt{M^2 + T_{max}^2}$$

where, M = Maximum bending moment on the shaft is taken as zero, since weight of the pulley is to be neglected

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

or, $372.61 \times 10^3 = \frac{\pi}{16} d^3 \times 50$

or, $d = 33.60$ mm. say 34 mm.

According to emperical relation, hub dia D_H

$D_H = 1.75d + 7$ mm.

$= 1.75 \times 34 + 7 = 66.5$ mm.

say 68 mm.

As other emperical relation is $D_H = 2d$.

L_H = length of the hub

$= \frac{2}{3} B$ to B

$L_H = B = 82$ mm.

Checking for shear stress of the hub material which is made of cast iron.

$$T_{max} = \frac{\pi}{16} \frac{(D_H^4 - d^4)}{D_H} \tau_{s\,CI}$$

$$372.61 \times 10^3 = \frac{\pi}{16} \frac{\{(68)^4 - (34)^4\}}{68} \times \tau_{s\,CI}$$

$$= 6.44 \text{ N/mm}^2$$

Scanned with CamScanner

which is less than the allowable shear stress $15 N/mm^2$. Hence safe

Design of key

W_K = width of key $= \frac{d}{4} = \frac{34}{4} = 8.5$ mm say 9 mm.

d_K = depth of key $= \frac{d}{6} = \frac{34}{6} = 5.66$ mm say 6 mm.

Shear failure of key

$T_{max} = W_K \times l_K \times \tau_{sK} \times \frac{d}{2}$

where, l_K = length of key

τ_{sK} = Allowable shear stress of key material $= 40 N/mm^2$

So, According to relation

$372.61 \times 10^3 = 9 \times l_K \times 40 \times \frac{34}{2}$

or $l_K = 60.88$ mm. Say 61 mm.

Crushing failure of the key

$T_{max} = l_K \times \frac{d_K}{2} \times \sigma_{cK} \times \frac{d}{2}$

where, σ_{cK} = allowable crushing stress of the key material $= 100 N/mm^2$

So, according to relation

$372.61 \times 10^3 = l_K \times \frac{6}{2} \times 100 \times \frac{34}{2}$

or, $l_K = 73.06$ mm.

So, we take $l_K = 74$ mm.

So, Actual length of is equal to l_K plus 12 mm. for quick fitting & dismantling

So, $l_{Kac} = 74 + 12 = 86$ mm.

Design of arms.

Since the dia. of the pulley is 45cm, we should take the no. of arms is 4 and the section of the arm are eliptical.

Let, a_1 = length of the major axis near the hub

b_1 = length of the minor axis near the hub

Then, we know that the maximum bending stress set up in the arm material (C.I) is given by

$$\frac{M}{I} = \frac{\sigma}{y}$$

where, M = Maximum Bending moment per arm $= \frac{T_{max}}{n/2}$

n = no. of arm = 4.

$\frac{n}{2}$ = no. of effective arm.

I = Moment of Inertia of the elipse at hub end about neutral axis.

$$= \frac{\pi}{64} b_1 a_1^3$$

length of minor axis b_1 is half of the length of major axis a_1

So, $b_1 = \frac{a_1}{2}$

∴ maximum bending stress set up in the pulley.

So,
$$\frac{\frac{T_{max}}{n/2}}{\frac{\pi}{64} b_1 a_1^3} = \frac{\sigma}{\frac{a_1}{2}} \quad \text{or,} \quad \sigma = \frac{\frac{2T_{max}}{n}}{\frac{\pi}{64} b_1 a_1^3} \times \frac{a_1}{2}$$

(9)

$$a_f = \frac{128\, T_{max}}{\pi\, \pi\, a_1^3}$$

$a = 20\, N/mm^2$

So, $20 = \dfrac{128 \times 372.61 \times 10^3}{4 \times \pi \times a_1^3}$

or, $a_1 = 57.46$ mm. Say 58 mm.

$b_1 = \dfrac{58}{2} = 29$ mm.

Taper of the arms = 1 : 25

Length of each arm is l_{arm}.

So. $l_{arm} = \left(\dfrac{D_{min} - 2 \times \text{crown height} - 2 \times \text{rim thickness} - D_H}{2}\right)$

$= \left(\dfrac{450 - 2 \times 1.5 - 2 \times 6 - 68}{2}\right)$

$= 183.5$ mm.

So, a_2 = length of major axis at rim end.

b_2 = length of minor axis at rim end.

$a_2 = 43$ mm. $b_2 = 22$ mm.

Final Data

$D_{m} = 450$ mm. $t_r = 6$ mm $a_1 = 58$ mm.

$b = 65$ mm. $h_c = 1.5$ mm. $b_1 = 29$ mm.

$B = 82$ mm. $w_k = 9$ mm. $a_2 = 43$ mm

$d = 34$ mm. $d_k = 6$ mm $b_2 = 22$ mm.

$D_H = 68$ mm, $l_k = 86$ mm. Taper of arm 1:25

$L_H = 82$ mm.