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CMRIT

Internal Assessment Test 2 – May. 2023

Sub:	Design of Machine Elements - 2					Sub Code:	18ME62	Branch:	Mech
Date:	24.5.23	Duration:	90 min's	Max Marks:	50	Sem / Sec:	VI/A		OBE

Answer All the Questions

Usage of Machine Design Data Handbook is permitted

		MARKS	CO	RBT
1	A pair of straight tooth Bevel gears at right angles is to transmit 5 kW at 1200 rpm of the pinion. The diameter of the pinion is 75 mm and the velocity ratio is 3.5 to 1. The tooth form is 14 ½ ° composite type. Both pinion and gear are made of cast iron (σ_0 = 55 MPa). Design the gear pair.	[20]	CO3	L3
2	A full load journal bearing 50 mm in diameter and 50 mm long operates at 1000 rpm and carries a load of 5 kN. The radial clearance is 0.025 mm. The bearing is lubricated with SAE 30 oil and the operating temperature is 80°C. Assume attitude angle as 60°, determine (i) Bearing pressure (ii) Sommerfeld number (iii) Attitude (iv) minimum film thickness (v) Heat generated (vi) Heat dissipated if the ambient temperature is 20° C and (vii) Amount of artificial cooling required if necessary. Use McKee's and Pederson's equations.	[20]	CO5	L
3	Derive Petroff's equation for journal bearing.	[10]	CO5	L

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Scheme of Evaluation

1) Tangential tooth load → 5

Dimensions → 5

Dynamic load → 5

Wear load and BHN → 5

2) Each parameter 3 marks → (20)

3) Derivation → 8 marks

Figure → 2 marks

Qn: 1

Data : $\Sigma = 90^\circ$, $N = 5$ kW, $n_1 = 1200$ rpm, $d_1 = 80$ mm, $i = 3.5$, $\alpha = 14\frac{1}{2}^\circ$, $\sigma_{o1} = \sigma_{o2} = 55$ N/mm^2

Solution :

Transmission ratio $$i = \frac{n_1}{n_2} = \frac{d_2}{d_1} \quad \text{..... (2.393)}$$

$\therefore$ Speed of the gear $$n_2 = \frac{n_1}{i} = \frac{1200}{3.5} = 342.857 \text{ rpm}$$

Pitch diameter of gear $d_2 = i\, d_1 = 3.5 \times 80 = 280$ mm

The pinion and gear are made of same material, the pinion is the weaker member. Therefore the design is based on pinion strength.

Pitch cone angle of pinion, $$\tan \delta_1 = \frac{1}{i} = \frac{1}{3.5} \quad \text{..... (2.402)}$$

$\therefore$ $\delta_1 = 15.95^\circ$

Pitch cone angle of gear $\delta_2 = \Sigma - \delta_1 = 90 - 15.95 = 74.05^\circ$

Formative number of teeth on pinion $$z_{v1} = \frac{z_1}{\cos \delta_1} \quad \text{..... (2.418)}$$

$$= \frac{d_1}{m \cos \delta_1} = \frac{80}{m \cos 15.95} = \frac{83.203}{m} \qquad \left(\because z_1 = \frac{d_1}{m}\right)$$

Lewis form factor for $14\frac{1}{2}^\circ$ tooth form $$y = 0.124 - \frac{0.684}{z_v} \quad \text{..... (2.97)}$$

$$\therefore \quad y_1 = 0.124 - \frac{0.684 \times m}{83.203} = 0.124 - 8.2209 \times 10^{-3}\, m$$

Torque on pinion $M_{t1} = \dfrac{9550\,N}{n_1}$

$$= \frac{9550 \times 5}{1200} = 39.792 \text{ N-m} = 39792 \text{ N-mm}$$

Tangential force $F_t = \dfrac{2\,M_{t1}}{d_1} = \dfrac{2 \times 39792}{80} = 994.8 \text{ N}$

Cone distance $R = \dfrac{1}{2}\sqrt{d_1^2 + d_2^2}$ ..... (2.414)

$$= \frac{1}{2}\sqrt{80^2 + 280^2} = 145.602 \text{ mm}$$

Let the face width $b = \dfrac{R}{3} = \dfrac{145.602}{3} = 48.534 \text{ mm}$

Take the face width $b = 48 \text{ mm}$ $\left(\because \dfrac{R}{4} \le b \le \dfrac{R}{3}\right)$

$\therefore$ Ratio $\dfrac{R-b}{R} = \dfrac{145.602 - 48}{145.602} = 0.6703$

Velocity $v_m = \dfrac{\pi\, d_1 n_1}{60 \times 1000} = \dfrac{\pi \times 80 \times 1200}{60 \times 1000} = 5.026 \text{ m/sec}$

For generated teeth, velocity factor $C_v = \dfrac{5.55}{5.55 + v_m}$ ..... (2.429)

$$= \frac{5.55}{5.55 + 5.026} = 0.5248$$

By Lewis equation, tangential force $F_t = \sigma_o\, C_v\, b\, Y\, m \left(\dfrac{R-b}{R}\right)$ ..... (2.426a)

i.e., $994.8 = 55 \times 0.5248 \times 48 \times \pi(0.124 - 8.2209 \times 10^{-3}\, m) \times m \times 0.6703$

$8.2209 \times 10^{-3}\, m^2 - 0.124\, m + 0.341 = 0$

$\therefore$ Module $m = \dfrac{0.124 \pm \sqrt{0.124^2 - 4 \times 8.2209 \times 10^{-3} \times 0.341}}{2 \times 8.2209 \times 10^{-3}}$

$$= 11.47 \text{ mm or } 3.618 \text{ mm}$$

Take $m = 3.618 \text{ mm}$

From table (2.84), the standard module m is 4 mm.

Allowable stress $\sigma_{all} = \sigma_{o1}\, C_v = 55 \times 0.5248 = 28.864 \text{ N/mm}^2$

Tangential force $F_t = \sigma_{o1}\, C_v\, b\, Y m \left(\frac{R-b}{R}\right) = \sigma_{in}\, b\, \pi\, y\, m \left(\frac{R-b}{R}\right)$ ..... (2.426a)

i.e., $994.8 = \sigma_{in} \times 48 \times \pi\,(0.124 - 8.2209 \times 10^{-3} \times 4) \times 4 \times 0.6703$

∴ Induced stress $\sigma_{in} = 27 \text{ N/mm}^2$

Since the induced stress is less than the allowable stress, the design is safe.

Number of teeth on pinion $z_1 = \frac{d_1}{m} = \frac{80}{4} = 20$

Number of teeth on gear $z_2 = \frac{d_2}{m} = \frac{280}{4} = 70$

Dynamic load :

Assume the drive uses carefully cut gears, from fig. (2.29), corresponding to 4 mm module, the expected error f is 0.025 mm. From table (2.35), for $f = 0.025$ mm, the value $C = 139.7$ kN/m = 139.7 N/mm

Dynamic load $F_d = F_t + \frac{21 v_m (F_t + bC)}{21 v_m + \sqrt{F_t + bC}}$ ..... (2.148a)

$$= 994.8 + \frac{21 \times 5.026(994.8 + 48 \times 139.7)}{21 \times 5.026 + \sqrt{994.8 + 48 \times 139.7}} = 5199.43 \text{ N}$$

Wear load :

Wear load $F_w = \frac{d_1 b Q K}{\cos\delta_1}$ ..... (2.441a)

Formative number of teeth on pinion $z_{v1} = \frac{z_1}{\cos\delta_1} = \frac{20}{\cos 15.95} = 20.8$

Formative number of teeth on gear $z_{v2} = \frac{z_2}{\cos\delta_2} = \frac{70}{\cos 74.05} = 254.73$

Ratio factor $Q = \frac{2 z_{v2}}{z_{v2} + z_{v1}} = \frac{2 \times 254.73}{254.73 + 20.8} = 1.849$

From table (2.40), for 14½° cast iron gears, the load stress factor $K = 1.0487$

∴ Wear load $F_w = \frac{80 \times 48 \times 1.849 \times 1.0487}{\cos 15.95} = 7744.07 \text{ N}$

Since $F_w > F_d$, the design is safe.

2)

Solution :

Referring table (11.12) of volume I, diameter of shaft for e9 fit is 60^{-60}_{-134}

$\therefore$ Minimum diameter of shaft $d = 60 - 0.134 = 59.866$ mm

Referring table (11.13) of volume I, diameter of bearing for H9 fit is 60^{+74}_{+0}

$\therefore$ Maximum diameter of bearing $D = 60 + 0.074 = 60.074$ mm

Diametral clearance $c = D - d = 60.074 - 59.866 = 0.208$ mm

Clearance ratio $$\psi = \frac{c}{d} = \frac{0.208}{59.866} = 3.4744 \times 10^{-3} \quad \text{..... (1.29)}$$

Surface velocity of journal $$v = \frac{\pi dn}{60} = \frac{\pi \times 59.866 \times 500}{60} = 1.567 \text{ m/sec}$$

By McKee, coefficient of friction $$\mu = K_a \left(\frac{\eta n'}{P}\right)\left(\frac{1}{\psi}\right) 10^{-10} + \Delta\mu \quad \text{.... (1.22)}$$

where $K_a = 1.95 \times 10^{11}$ for full journal bearing and $\Delta\mu = 0.002$ for $0.75 < L/d < 2.8$.

$$\therefore \quad \mu = \frac{1.95 \times 10^{11} \times 0.099 \times 500 \times 10^{-10}}{2 \times 10^6 \times 60 \times 3.4744 \times 10^{-3}} + 0.002 = 0.004315$$

For heat balance, equate the heat generated H_g and heat dissipated H_d, we get

$$(\Delta T + 18)^2 = K' \mu P v \quad \text{..... (1.69c)}$$

where $K' = 0.475$ for light bearings located in still air.

i.e., $$(\Delta T + 18)^2 = 0.475 \times 0.004315 \times 2 \times 10^6 \times 1.567$$

$\therefore$ Difference in temperature $\Delta T = 62.147$°C

Also, $$\Delta T = t_b - t_a$$

i.e., $$62.147 = t_b - 20$$

$\therefore$ Temperature of bearing surface $t_b = 82.147$°C

Also $$t_b - t_a = \frac{t_o - t_a}{2}$$

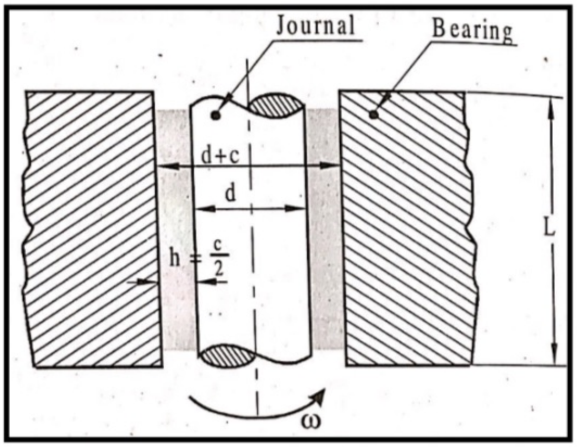
i.e., $$82.147 - 20 = \frac{t_o - 20}{2}$$

$\therefore$ Temperature of oil $t_o = 144.294$°C

3) Pettroff's Equation

1. Petroff's Equation

Petroff's equation is used to find the coefficient of friction in journal bearings.



Consider a vertical shaft rotating in a guide bearing as shown in figure.
Assumptions:

1. The bearing is lightly loaded.
2. The clearance 'c' is completely filled with oil.
3. There is no end leakage.
4. Viscosity of oil used is very high.
5. The journal rotates at very high speed.
6. There is no eccentricity between the journal and bearing.

Let

d = diameter of journal or shaft
c = diametral clearance
n' = Speed of the journal or shaft in rev/sec

$$n' = \frac{n}{60}$$

L = Length of the bearing

$\psi = \frac{c}{d}$ = diametral clearance ratio

η = Viscosity of oil , Pas

v = velocity = $\pi d n'$, m/s

shear stress, $\tau = \eta . \frac{v}{h} = \eta . \frac{\pi d n'}{\frac{c}{2}} = \frac{2\pi d \eta . n'}{c}$

Surface area A = πdL

Therefore,

Force, F = τA = $\frac{2\pi^2 d^2 n' \eta . L}{c}$

Torque, M_t = F x r

$= F.(\frac{d}{2})$

$= \frac{\pi^2 d^2 n' \eta . L}{(\frac{c}{d})}$

$$M_t = \frac{\pi^2 d^2 n' \eta . L}{\psi} \longrightarrow (a)$$

But, $M_t = (\mu . W)(\frac{d}{2})$ $((\mu . W)$ = Frictional Force)

and W = PA = P.L.d = Load

where P = Bearing Pressure in Pa

$$M_t = (\mu . PL . d)(\frac{d}{2}) \longrightarrow (b)$$

Equating (a) and (b)

$$(\mu . PL . d)(\frac{d}{2}) = \frac{\pi^2 d^2 n' \eta . L}{\psi}$$

$$\mu = 2\pi^2 (\frac{\eta . n'}{P}) . \frac{1}{\psi}$$

The above equation is Petroff's Equation