B.E.

Sixth Semester Examination, December-2008

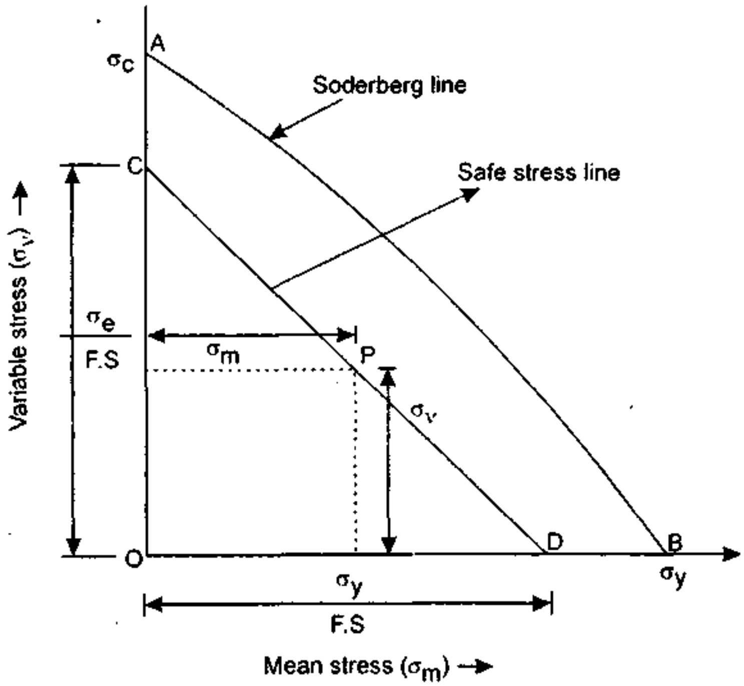
Machine Design-II (ME-304-E)

Note : Attempt any **five** questions. All questions carry equal marks.

Q. 1. (a) Write Soderberg's equation and state its application to different types of loadings.

Ans. A straight line connecting the endurance limit (σ_e) and the yield strength (σ_y), as shown by line *AB*, follows the suggestion of Soderberg line. This line is used when the design is based on yield strength.

The line *AB* connecting σ_e and σ_y is called Soderberg's failure stress line.



Let us consider a design point *P* on the line *CD*. Now from similar triangles *COD* and *PQD*,

$$\frac{PQ}{CO} = \frac{QD}{OD} = \frac{OD - OQ}{OD}$$

$$= 1 - \frac{OQ}{OD} \qquad (\because QD = OD - OQ)$$

$$\therefore \qquad \frac{\sigma_v}{\sigma_e / F.S} = 1 - \frac{\sigma_m}{\sigma_y / F.S}$$

or $$\sigma_v = \frac{\sigma_e}{F.S}\left[1 - \frac{\sigma_m}{\sigma_y / F.S}\right] = \sigma_e\left[\frac{1}{F.S.} - \frac{\sigma_m}{\sigma_y}\right]$$

$$\therefore \quad \frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_\upsilon}{\sigma_e}$$

For machine parts subjected to fatigue loading, the fatigue stress concentration factor (K_p) should be applied to only variable stress (σ_υ). Thus, the equation may be written as

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_\upsilon K_p}{\sigma_e}$$

Considering the load factor, surface finish factor and size factor may be written as

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_\upsilon \times K_p}{\sigma_{eb} \times K_{sur} \times K_{sz}}$$

Since $\sigma_{e_b} = \sigma_e \times K_b$ and $K_b = 1$ for reversed bending load, therefore, $\sigma_{e_b} = \sigma_e$ may be substituted in above equation.

Application of Soderberg Equation

In case of axial loading, we know that mean or average stress, $\sigma_m = w_m / A$

$\sigma_\upsilon = w_\upsilon / A$

w_m = Mean or average load

w_υ = Variable load and A = cross rectional area

$$\text{Working or design stress} = \frac{w_m}{A} + \left(\frac{\sigma_y}{\sigma_e}\right) K_p \times \frac{w_\upsilon}{A}$$

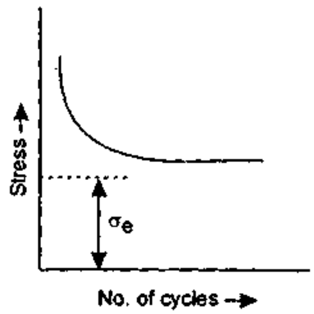
$$= \frac{w_m + \left(\frac{\sigma_y}{\sigma_e}\right) K_p \times w_\upsilon}{A}$$

$$\therefore \quad F.S. = \frac{\sigma_y \times A}{w_m + \left(\frac{\sigma_y}{\sigma_e}\right) K_p \times w_\upsilon} \quad \textbf{Ans.}$$

Q. 1. (b) Define and Discuss :

Endurance limit; Size factor; Surface finish factor; Notch Sensitivity.

Ans. Endurance Limit : It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite no. of cycles (usually 10^7 cycles).



Size Factor : If the size of standard specimen will increase, then the endurance limit of the material will decrease. This is due the fact that a longer specimen will have more defects than a smaller one.

Let K_{sz} = size factor

$\therefore$ Endurance limit,

$$\sigma_{ez} = \sigma_e \times K_{ez}$$
$$= \sigma_{eb} . K_{sur} . K_{sz} = \sigma_e . K_b . K_{sur} . K_{sz} = \sigma_e . K_{sur} . K_{sz} \qquad (\because K_b = 1)$$
$$= \sigma_{ea} . K_{sur} . K_{sz} = \sigma_e . K_a . K_{sur} . K_{sz} \qquad \text{(for reversed axial load)}$$
$$= \tau_e . K_{sur} . K_{sz} = \sigma_e . K_s . K_{sur} . K_{sz} \qquad \text{(for reversed torsional or shear load)}$$

Surface Finish Factor : When a machine member is subjected to variable loads, the endurance limit of the material for that member depends upon the surface conditions. When the surface finish factor is known, than the endurance limit for the material of the machine member may be obtained by multiplying the endurance limit and surface finish factor.

Let K_{sur} = surface finish factor

$\therefore$ Endurance limit,

$$\sigma_{e_1} = \sigma_{eb} . K_{sur} = \sigma_e . K . K_{sur} = \sigma_e . K_{sur} \because K_b = 1 \qquad \text{(For reversed bending load)}$$
$$= \sigma_{ea} . K_{sur} = \sigma_e . K_a . K_{sur} \qquad \text{(For reversed axial load)}$$
$$= \tau_e . K_{sur} = \sigma_e . K_s . K_{sur} \qquad \text{(For reversed torsional or shear loads)}$$

Notch Sensitivity : It may be defined as the degree to which the theoretical effect of stress concentration is actually reached. The stress gradient depends mainly on the radius of the notch, hole or fillet and on the grain size of material.

Q. 2. A shaft is supported on bearings A and B, 800 mm between centres. A 20° straight tooth spur gear having 600mm pitch diameter, is located 200mm to the right of the left hand bearing A and a 700mm diameter pulley is mounted 250mm towards the left of bearing B. The gear is driven by a pinion with a downward tangential force while the pulley drives a horizontal belt having 180° angle of wrap. The pulley also serves as a flywheel and weigh 2000N. The maximum belt tension is 3000N and the tension ratio is 3 : 1. Determine the maximum bending moment and the necessary shaft diameter if the allowable shear stress of the material is 40MPa.

Ans. Given $AB = 800$ mm; $\alpha_c = 20°$, $D_c = 600$ mm or $R_c = 300$ mm; $AC = 200$ mm; $D_D = 700$ mm or $R_D = 350$ mm; $D_B = 250$ mm; $\theta = 180° = \pi$ rad; $w = 2000$ N; $T_1 = 3000$ N; $T_1/T_2 = 3$; $\tau = 40$ MPa $= 40$ N/mm^2.

We know that the torque acting on the shaft at D,

$$T = (T_1 - T_2)\, R_D = T_1\left(1 - \frac{T_2}{T_1}\right) R_D$$
$$= 3000\left(1 - \frac{1}{3}\right) 350 = 700 \times 10^3 \text{ N-mm} \qquad (\because T_1/T_2 = 3)$$

Assuming that the torque at D is equal to the torque at C, therefore tangential force acting on the gear C.

$$F_{tc} = \frac{T}{R_c} = \frac{700 \times 10^3}{300} = 2333 \text{ N}$$

& the normal load acting on the tooth of gear C,

$$w_c = \frac{F_{tc}}{\cos \alpha_c} = \frac{2333}{\cos 20°} = \frac{2333}{0.9397} = 2483 \text{ N}$$

Vertical component of w_c

$$w_{cv} = w_c \cos 20° = 2483 \times 0.9397$$
$$= 2333 \text{ N}$$

Horizontal component of w_c

$$w_{CH} = w_c \sin 20°$$
$$= 2483 \times 0.342 = 84 \text{ NN}$$

Since $\quad T_1/T_2 = 3 \quad$ and $\quad T_1 = 3000$ N,

$\therefore \quad T_2 = \dfrac{T_1}{3} = 1000$ N

Let us find the maximum bending moment for vertical and horizontal loading.

First of all considering the vertical loading at C and D.

Let R_{AV} and R_{BV} be the reactions

$$R_{AV} + R_{BV} = 2333 + 2000 = 4333 \text{ N}$$

Taking moment about A, we get

$$R_{BV} \times 800 = 2000\,(800 - 250) + 2333 \times 200$$
$$= 1566600$$
$$R_{BV} = 1566600/800 = 1958 \text{ N}$$
$$R_{AV} = 4333 - 1958 = 2375 \text{ N}$$

We know that B.M. at A and B

$$M_{AV} = M_{BV} = 0$$

B.M at C $\quad M_{CV} = R_{AV} \times 200 = 2375 \times 200$

$$= 475 \times 10^3 \text{ N-mm}$$

B.M at D, $\quad M_{DV} = R_{BV} \times 250 = 1958 \times 250$

$$= 489.5 \times 10^3 \text{ N-mm}$$

Now consider the horizontal loading at C and D. Let R_{AH} and R_{BH} be the reactions at the bearings A and B respectively.

$$R_{AH} + R_{BH} = 849 + 4000 = 4849 \text{ N}$$

Taking moments about A, we get

$$R_{BH} \times 800 = 4000\,(800 - 250) + 849 \times 200 = 2369800$$
$$R_{BH} = 2369800/800 = 2963 \text{ N}$$
$$R_{AH} = 4849 - 2963 = 1886 \text{ N}$$

We know that B.M. at A and B

$$M_{AH} = M_{BH} = 0$$

B.M at C, $\quad M_{CH} = R_{AH} \times 200 = 1886 \times 200 = 377\,200$ N-mm

B.M at D, $\quad M_{DH} = R_{BH} \times 250 = 2963 \times 250 = 740750$ N-mm

We know that resultant B.M. at C,

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2} = \sqrt{(475 \times 10^3)^2 + (377200)^2}$$
$$= 606552 \text{ N-mm}$$

Resultant B.M. at D,

$$M_D = \sqrt{(M_{DV})^2 + (M_{DH})^2} = \sqrt{(489.5 \times 10^3)^2 + (740750)^2}$$

$$= 887874 \text{ N-mm}$$

Maximum B.M. $\quad M = M_D = 887874$ N-mm **Ans.**

Diameter of the shaft

Equivalent twisting moment

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(887874)^2 + (700 \times 10^3)^2}$$

$$= 1131 \times 10^3 \text{ N-mm}$$

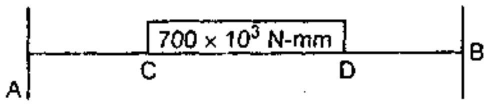
We also know that equivalent twisting moment (T_e)

$$1131 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 40 \times d^3$$

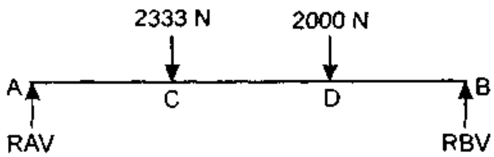
$$= 7.86\, d^3$$

$\therefore \quad d^3 = 1131 \times 10^3 / 7.86 = 144 \times 10^3$

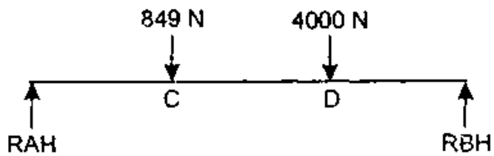
or $\quad d = 52.4$ say 55 nm **Ans.**



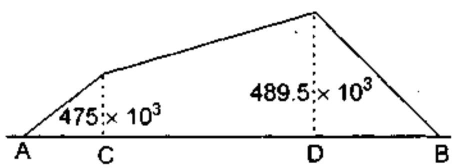
Torque diagram



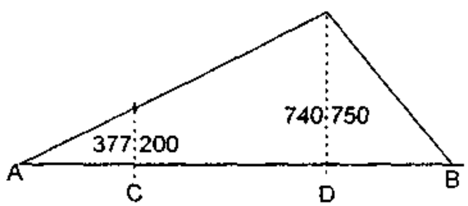
Vertical load diagram



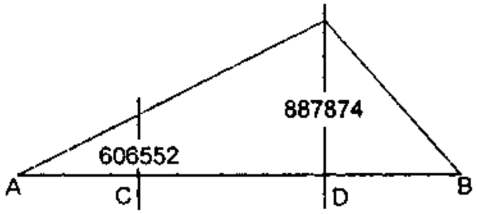
Horizontal load diagram



Vertical B.M. diagram



Horizontal B.M. diagram



Resultant B.M. diagram

Q. 3. A semi elliptical spring has ten leaves in all, with the two full length leaves extending 625 mm. It is 62.5 mm wide and 6.25 mm thick. Design a helical spring with mean diameter of coil 100mm which will have approximately the same induced stress and deflection for any load. The Young's modulus for the material of the semi-elliptical spring may be taken as 200 KN/mm^2 and modulus of rigidity for the material of the spring is 80 KN/mm^2.

Ans. Assume 2P be the load acting at the centre of the semi-elliptical or helical spring. The maximum stress induced in the leaf spring is given by

$$f = \frac{6PL}{Zbt^2},\ f = \frac{6 \times P \times 625}{2 \times 10 \times 62.5 \times (6)^2} = 0.0833P$$

The maximum shear stress induced in the helical spring is given by

$$f = \frac{8PP}{\pi d^3},\ f = \frac{8 \times (2P) \times 100}{\pi d^3} = \frac{500P}{d^3}$$

If the value of induced stress is to be same, then

$$0.0833\,P = \frac{500P}{d^3},\ \text{gives } d = 18.5 \text{ mm}$$

From table, the standard wire size well be 19 mm

Outside coil diameter $= 100 + 19 = 119$ mm

Inside coil diameter $= 100 - 19 = 81$ mm

The deflection of a leaf spring will be

$$\delta = \frac{12PL^3}{Ebt^3\ (2zg + 3zf)} = \frac{12 \times P \times (312.5)^3}{E \times 62.5 \times 6^3\ (2 \times 8 + 3 \times 2)}$$

$$\delta = \frac{1240P}{E}$$

The deflection of the helical spring will be

$$\delta = \frac{8PD^3Z}{Cd^4},\ \delta = \frac{8 \times 2P \times (100)^3 \times Z}{C \times 19^4}$$

$$\delta = \frac{0.123\,P_2}{C} \times (10)^3$$

If the deflection is to be same, then

$$\frac{1240P}{E} = \frac{0.123\,P_2}{C} \times 10^3,\ Z = 10 \times \frac{C}{E}$$

The ratio of $\frac{E}{C}$ may be taken as 2.5

Then $$Z = \frac{10}{2.5} = 4.0 \text{ coils}$$

The total no. of coils may be taken as 6, to allow for two end coils.

Q. 4. A full journal bearing of 50 mm diameter and 100 mm long has a bearing pressure of 1.4 N/mm^2. The speed of the journal is 900 rpm and the ratio of journal diameter to the diametral clearance is 1000. The bearing is lubricated with oil whose absolute viscosity at the operating temperature of 75°C may be taken as 0.011 kg/m-s. The room temperature is 35°C. Find : (1) The amount of artificial cooling required; (2) The mass of the lubricating oil required if the difference between the outlet and the inlet temperature of the oil is 10°C. Take specific heat of the oil as 1850 J/kg°C.

Ans. Given $d = 50$ mm $= 0.05$ m; $l = 100$ mm $= 0.1$ m. $P = 1.4$ N/mm^2; $N = 900$ rpm; $d/C = 1000$; $Z = 0.011$, $t_0 = 75°C$; $t_a = 35°C$; $t = 10°C$; $S = 1850$

Amount of artificial cooling required.

We know that the coefficient of the friction

$$\mu = \frac{33}{10^8}\left(\frac{ZN}{P}\right)\left(\frac{d}{C}\right) + K$$

$$= \frac{33}{10^8}\left(\frac{0.011 \times 900}{1.4}\right)(1000) + 0.002$$

$$= 0.00233 + 0.002 = 0.00433$$

Load on the bearing

$$w = P.dl = 1.4 \times 50 \times 100 = 7000 \text{ N}$$

& rubbing velocity,

$$V = \frac{\pi dN}{60} = \frac{\pi \times 0.05 \times 900}{60} = 2.36 \text{ m/s}$$

∴ Heat generated

$$Q_g = \mu w V = 0.00433 \times 7000 \times 2.36 = 71.5 \text{ J/s}$$

Let t_0 = Temperature of bearing surface

We know that

$$(t_b - t_a) = \frac{1}{2}(t_0 - t_a) = \frac{1}{2}(75 - 35) = 20°C$$

Let us take $C = 280 w / m^2 /°C$

We know that,

Heat dissipated

$$Q_d = CA(t_b - t_a) = C\, l\, d\,(t_b - t_a)$$

$$= 280 \times 0.05 \times 0.1 \times 20 = 28\, w = 28 \text{ J/s}$$

∴ Amount of artificial cooling required

$$= \text{Heat generated} - \text{Heat dissipated}$$

$$= 71.5 - 28 = 43.5 \text{ J/s or } w$$ **Ans.**

Mass of the lubricating oil required

Let M = Mass of oil required

$$Q_t = m\ S\ t = m \times 1850 \times 10 = 18500 \text{ mJ/s}$$

$$Q_t = Q_g \text{ or } 715 = 18500 \text{ m}$$

$$m = \frac{715}{18500} = 0.23 \text{ kg/min} \quad \textbf{Ans.}$$

Q. 5. A gear drive is required to transmit a maximum power of 22.5 kW. The velocity ratio is 1 : 2 and rpm of the pinion is 200. The approximate centre distance between the shaft may be taken as 600 mm. The teeth have 20° stub involute profiles. The static stress for the gear material may be taken as 60MPa and face width as 10 times the module. Find the module, face width and number of teeth on each gear. Check the design for dynamic and wear loads. The deformation or dynamic factor in the Buckingham equation may be taken as 80 and the material combination factor for the wear as 1.4.

Ans. Given $P = 22.5$ kW $= 22500$w; $V_R = D_G / D_P = 2$, $N_P = 200$ rpm; $L = 600$ mm;

$$\sigma_{op} = \sigma_{oG} = 60 \text{ MPa}; b = 10\text{m}; C = 80; K = 1.4$$

We know that

$$600 = \frac{D_p}{2} + \frac{D_G}{2} = \frac{D_p}{2} + \frac{2D_p}{2} = 1.5\ D_p$$

$$\therefore \quad D_P = \frac{600}{1.5} = 400\text{mm} = .4\text{m}$$

$$D_G = 2D_P = 2 \times 400 = .8\text{m}$$

Since both the gears are mode of same material, therefore, pinion is weaker. Thus, design will be based upon pinion.

$$v = \frac{\pi\ D_P\ N_P}{60} = \frac{\pi \times 0.4 \times 200}{60} = 4.2 \text{ m/s}$$

Since v is less than 12 m/s

$$C_v = \frac{3}{3 + v} = \frac{3}{3 + 4.2} = 0.417$$

We know that no. of teeth on the pinion

$$T_P = \frac{D_P}{m} = 400/\text{m}$$

$\therefore$ Tooth form factor for the pinion;

$$T_P = 0.175 - \frac{0.841}{T_P} = 0.175 - \frac{.841 \times \text{m}}{400}$$

$$= 0.175 - 0.0021\text{ m}$$

Service factor $C_s = 1$

$$w_T = \frac{P}{v} \times C_s = 5357 \text{ N}$$

Tangential tooth load (ω_T)

$$5357 = \sigma_{wp} \times b \times \pi\text{ m} \times \gamma_P$$

$$= (\sigma_{OP} \times C_v)\ G\ \pi\text{ m}\ \gamma_P$$

$$= (60 \times 0.417)\ 10\text{m} \times \pi\text{ m}\ (0.175 - 0.0021\text{ m})$$

$$= 137.6\text{ m}^2 - 1.65\text{m}^3$$

Solving this

$$m = 0.65 \text{ say } 8 \text{ mm} \quad \textbf{Ans.}$$

Face width

$$G = 10m = 10 \times 8 = 80 \text{ mm}$$

No. of teeth on the gears

$$T_P = D_P / m = \frac{400}{8} = 50 \quad \textbf{Ans.}$$

No. of teeth on gears

$$T_G = D_G / m = \frac{800}{8} = 100 \textbf{ Ans.}$$

Q. 6. A composite spring has two closed coil helical springs, the outer spring is 15 mm larger than the inner spring. The outer spring has 10 coils of mean diameter 40 mm and wire diameter 4mm. When the spring is subjected to an axial load of 400N, find compression of each spring, load shared by each spring, shear stress induced in each spring. The modulus of rigidity may be taken as 84kN/mm^2.

Ans. Given $\delta_1 = l_1 - l_2 = 15$ mm; $n_1 = 10$; $D_1 = 40$ mm; $d_1 = 5$ mm; $n_2 = 8$; $D_2 = 30$ mm; $d_2 = 4$ mm; $w = 400$ N; $G = 84$ KN/mm^2 $= 84 \times 10^3$ N/mm^2

Compression of each spring :

$$15 = \frac{8P_1 (D_1)^3 n_1}{G (d_1)^4} = \frac{8 \times P_1 (40)^3 \, 10}{84 \times 10^3 \times 5^4} = 0.0975 P_1$$

$$P_1 = 15/0.0975 = 154 \text{ N}$$

$$P_2 = \frac{P_1}{\delta_1} \times \delta_2 = \frac{154}{15} \times \delta_2 = 10.27 \, \delta_2$$

Let w_2 = load taken by the inner spring to compress it by δ_2 mm

$$\delta_2 = \frac{8w_2 (D_2)^3 P_2}{G (d_2)^4} = \frac{8w_2 (30)^3 \, 8}{84 \times 10^3 \times 4^4}$$

$$= 0.08 \, w_2$$

$\therefore \quad w_2 = \delta_2 / 0.08 = 12.5 \, \delta_2$

& $\quad P_2 + w_2 = w - P_1 = 400 - 154 = 246$ N

or $\quad 10.72 \, \delta_2 + 12.5 \, \delta_2 = 246$

$$\delta_2 = 10.8 \text{ mm} \quad \textbf{Ans.}$$

Total compression of outer spring

$$= \delta_1 + \delta_2 = 15 + 10.8 = 25.8 \text{ mm} \quad \textbf{Ans.}$$

Load shared by each spring

$$w_1 = P_1 + P_2 = 154 + 10.27 \, \delta_2 = 265 \text{ N } \textbf{Ans.}$$

$$w_2 = 12.5 \, \delta_2 = 135 \text{ N } \textbf{Ans.}$$

Shear stress induced in each spring

$$C_1 = \frac{D_1}{d_1} = \frac{40}{5} = 8$$

$$C_2 = \frac{D_2}{d_2} = \frac{30}{4} = 7.5$$

$\therefore$ Wahl's stress factor for outer spring

$$K_1 = \frac{4C_1 - 1}{4C_1 - 4} + \frac{0.615}{C_1} = \frac{4 \times 8 - 1}{4 \times 8 - 4} + \frac{0.615}{8}$$

$$= 1.184 \textbf{ Ans.}$$

For inner spring

$$K_2 = \frac{4C_2 - 1}{4C_2 - 4} + \frac{0.615}{C_2}$$

$$= \frac{4 \times 7.5 - 1}{4 \times 7.5 - 4} + \frac{0.615}{7.5} = 1.197 \textbf{ Ans.}$$

We know that stress induced in outer spring

$$\tau = \frac{K_1 \times 8w_1 D_1}{\pi d_1^3} = \frac{1.184 = (8 \times 265 \times 40)}{\pi \times 5^3}$$

$$= 255.6 \text{ N/mm}^2$$

$$e = 255.6 \text{ MPa } \textbf{Ans.}$$

& shear stress induced in the inner spring

$$\tau_2 = K_2 \times \frac{8w_2 D_2}{\pi (d_2)^3}$$

$$= \frac{1.197 \times 8 \times 135 \times 30}{\pi \times 4^3} = 192.86 \text{ N/mm}^2$$

$$= 192.86 \text{ MPa } \textbf{Ans.}$$

Q. 7. A worm drive transmits 15kW at 2000 rpm to a machine carriage at 75rpm. The worm is triple threaded and has 65mm pitch diameter. The worm gear has 90 teeth of 6mm module. The tooth form is to be 20° full depth involute. The coefficient of friction between the mating teeth may be taken as 0.10. Calculate Tangential force acting on the worm; axial thrust and separating force on the worm; efficiency of the worm drive.

Ans. Given $P = 15 \text{ kW} = 15 \times 10^3 \text{ w}$; $N_w = 2000$ rpm; $N_0 = 75$ rpm; $n = 3$; $D_w = 65$ mm; $T_G = 90$; $m = 6$ mm; $\phi = 20°$; $\mu = 0.10$

Tangential force acting on the worm

We know that torque transmitted by the worm

$$= \frac{P \times 60}{2\pi N_w} = \frac{15 \times 10^3 \times 60}{2\pi \times 2000} = 71.6 \text{ N-m}$$

$$= 71600 \text{ N-mm}$$

$\therefore$ Tangential force acting on the worm

$$w_T = \frac{\text{Torque on worm}}{\text{Radius of worm}} = \frac{71600}{65/2} = 2203 \text{ N } \textbf{Ans.}$$

Axial thrust and separating force on worm

Let λ = head angle

$$\lambda = \frac{m.n}{D_w} = \frac{6 \times 3}{65} = 0.277$$

$$\lambda = \tan^{-1}(0.277) = 15.5°$$

$\therefore$ Axial thrust on worm

$$w_A = w_T / \tan\lambda = 2203/0.277$$

$$= 7953 \text{ N} \quad \textbf{Ans.}$$

& separating force on the worm

$$w_R = w_A \tan\phi = 7953 \times \tan 20°$$

$$= 2895 \text{ N } \textbf{Ans.}$$

Efficiency of the worm drive

$$\eta = \tan\lambda \frac{(\cos\phi - \mu \tan\lambda)}{\cos\phi \tan\lambda + \mu}$$

$$= \frac{\tan 15.5° (\cos 20° - 0.10 \times \tan 15.5°)}{\cos 20° \times \tan 15.5° + 0.10}$$

$$= \frac{0.277 (0.9397 - .10 \times 0.277)}{0.9397 \times 0.277 + 0.10} = \frac{0.2526}{0.3603}$$

$$= 0.701 \text{ or } 70.1\% \quad \textbf{Ans.}$$

Q. 8. The rolling contact ball bearing are to be selected a support the overhung countershaft. The shaft speed is 720 rpm, the bearings are to have 99% reliability corresponding to life of 24000 hours. The bearing is subjected to an equivalent radial load of 1KN. Consider life adjustment factors for operating condition and material as 0.9 and 0.85 respectively. Find the basic dynamic load rating of the bearing from the manufacturer's catalogue. specified at 90° reliability.

Ans. Given $N = 720$ rpm; $L_H = 24000$ hrs; $w = 1$KN

We know that life of the bearing corresponding to 99% reliability.

$$L_{99} = 60\text{N } L_H = 60 \times 720 \times 24000 = 1036.8 \times 10^6 \text{ rev}$$

Let L_{90} = Life of the bearing corresponding to 90% reliability.

Considering life adjustment factors for operating condition and material as 0.9 and 0.85 respectively, we have

$$\frac{L_{99}}{L_{90}} = \left\{\frac{[\text{Log}_e (1/R_{99})]}{[\text{Log}_e (1/R_{90})]}\right\}^{1/G} \times 0.9 \times 0.85$$

$$= \left[\frac{\{\log_e (1/0.99)}{\{\log_e (1/0.90)}\right]^{1/1.17} \times 0.9 \times 0.85$$

$$= \left[\frac{0.01005}{0.1054}\right]^{0.8547} \times 0.9 \times 0.85$$

$$= 0.1026$$

$$\therefore \quad L_{90} = L_{99}/0.1026 = 1036.8 \times 10^6/0.1026$$

$$= 10105 \times 10^6 \text{ rev}$$

We know that dynamic load rating

$$C = w\left(\frac{L_{90}}{10^6}\right)^{1/k}$$

$$= 1\left(\frac{10105 \times 10^6}{10^6}\right)^{1/3} \text{ KN} \qquad (\because K = 3, \text{ for ball bearing})$$

$$= 21.62 \text{ KN}$$ **Ans.**