... then the design is safe and satisfactory.

Example 5.20 *It is desired to determine the proportions of a spur gear drive to transmit 8 kW from a shaft rotating at 1200 r.p.m. to a low speed shaft, with a reduction of 3 : 1. Assume that the teeth are 20° full depth involute, with 24 teeth on the pinion. The pinion is to be of 40 C 8 normalized steel and gear of 30 C 8 normalized steel. Assume that the starting torque is 130% of the rated torque.*

Given Data : $P = 8$ kW; $N_1 = 1200$ r.p.m. ; $i = 3$; $\phi = 20°$; $z_1 = 24$;

Starting torque = 1.3 × rated torque.

To find : Design a spur gear.

Solution :

1. Gear ratio : $i = 3$ ... (Given)

2. Material selection : Pinion = 40 C 8 normalized steel ; and

Gear = 30 C 8 normalized steel. ... (Given)

3. Gear life : Assume 20,000 hours.

$\therefore \; N = 20000 \times 60 \times 1200 = 144 \times 10^7$ cycles

4. Design torque $[M_t]$:

$$[M_t] = M_t \times K \times K_d$$

where

$$M_t = \frac{60 \times P}{2\pi N_1} = \frac{60 \times 8 \times 10^3}{2\pi \times 1200} = 63.66 \text{ N-m, and}$$

$$K \cdot K_d = 1.3 \qquad \text{... (Assume)}$$

$$\therefore \quad [M_t] = 63.66 \times 1.3 = 82.76 \text{ N-m}$$

5. Calculation of E_{eq}, $[\sigma_b]$ and $[\sigma_c]$:

(i) To find E_{eq} : From Table 5.20, $E_{eq} = 2.15 \times 10^5$ N/mm² for steel.

(ii) To find $[\sigma_b]$: Design bending stress, $[M_t] = \dfrac{1.4 \times K_{bl}}{n \cdot K_\sigma} \times \sigma_{-1}$

✓ K_{bl} = 1, for steel HB ≤ 350 and N ≥ 10^7, from Table 5.14,

✓ n = 2, for steel normalized, from Table 5.17,

✓ K_σ = 1.5, for steel normalized, from Table 5.15,

✓ σ_{-1} = $0.35\,\sigma_u + 120$, for alloy steel, from Table 5.16

$= 0.35 \times 720 + 120 = 372 \text{ N/mm}^2.$ ... [$\because \sigma_u = 720$ N/mm²]

$$\therefore \quad [\sigma_b] = \frac{1.4 \times 1}{2 \times 1.5} \times 372 = 173.6 \text{ N/mm}^2$$

(iii) To find [σ_c] : Design contact stress, $[\sigma_c] = C_B \times HB \times K_{cl}$

where C_B = 2.5, for alloy steel normalized, from Table 5.18,

HB ≤ 350, from Table 5.18, and

K_{cl} = 1, for steel, HB ≤ 350 and N ≥ 10^7, from Table 5.19.

$$\therefore \quad [\sigma_c] = 2.5 \times 300 \times 1 = 750 \text{ N/mm}^2$$

6. *Centre distance (a) :* Assume ψ = 0.3.

Design 8.13 ①

$$a \geq (i+1)\sqrt[3]{\left(\frac{0.74}{[\sigma_c]}\right)^2 \times \frac{E_{eq}[M_t]}{i\,\psi}}$$

$$\geq (3+1)\sqrt[3]{\left(\frac{0.74}{750}\right)^2 \times \frac{2.15 \times 10^5 \times 82.76 \times 10^3}{3 \times 0.3}}$$

≥ 107.2 or $a = 110$ mm

7. Given that $z_1 = 24$. $\therefore z_2 = i\,z_1 = 3 \times 24 = 72.$

8. *Module (m) :* 8.22 $m = \dfrac{2a}{z_1 + z_2} = \dfrac{2 \times 110}{24 + 72} = 2.29$ mm

From Table 5.8, the nearest higher standard module, $m = 2.5$ mm.

9. *Revised centre distance :* 8.22 $a = \dfrac{m(z_1 + z_2)}{2} = \dfrac{2.5\,(24 + 72)}{2} = 120$ mm.

10. *Calculation of b, d, v and ψ_p :*

✓ Face width (b) : 8.14 $b = \psi \times a = 0.3 \times 120 = 36$ mm

✓ Pitch diameter of pinion (d_1) : $d_1 = m \cdot z_1$ 8.22 $= 2.5 \times 24 = 60$ mm.

✓ Pitch line velocity (v) : $v = \dfrac{\pi\, d_1\, N_1}{60}$ 8.15 $= \dfrac{\pi \times 60 \times 10^{-3} \times 1200}{60} = 3.77$ m/s

✓ 8.15 $\psi_p = \dfrac{b}{d_1} = \dfrac{36}{60} = 0.6$

11. *Quality of gear :* From Table 5.22, for v = 3.77 m/s, IS quality 8 gears are selected.

12. *Revised design torque* $[M_t]$:

From Table 5.11, for $\psi = 0.6$, $K = 1.03$.

From Table 5.12, for IS quality 8, HB $\leq$ 350 and $v = 3.77$ m/s, $K_d = 1.55$.

$$\therefore \quad [M_t] = M_t \cdot K \cdot K_d$$

$$= 63.66 \times 1.03 \times 1.55 = 101.63 \text{ N-m}$$

13. *Check for bending* :

Induced bending stress, $\sigma_b = \dfrac{i+1}{a \cdot m \cdot b \cdot y}[M_t]$

where $y = 0.414$, for $z_1 = 24$, from Table 5.13.

$$\therefore \quad \sigma_b = \frac{(3+1)}{120 \times 2.5 \times 36 \times 0.414} \times 101.63 \times 10^3 = 90.9 \text{ N/mm}^2$$

We find $\sigma_b < [\sigma_b]$. Thus ***the design is satisfactory.***

14. *Check for wear strength* : Induced contact stress is given by

$$\sigma_c = 0.74 \frac{i+1}{a} \sqrt{\frac{i+1}{i\,b} \times E_{eq}\,[M_t]}$$

$$= 0.74 \left(\frac{3+1}{120}\right) \sqrt{\left(\frac{3+1}{3 \times 36}\right) \times 2.15 \times 10^5 \times 101.63 \times 10^3}$$

$$= 701.71 \text{ N/mm}^2$$

We find $\sigma_c < [\sigma_c]$, thus ***the design is safe and satisfactory.***

15. *Check for plastic deformation* :

M_t = Rated torque = 63.66 N-m ... (already calculated)

Given that starting torque is 130% of rated torque.

$$\therefore \quad [M_t]_{max} = \text{Maximum instantaneous torque} = 1.3 \times M_t$$

$$= 1.3 \times 63.66 = 82.758 \text{ N-m}$$

(i) Check for bending : Induced bending stress due to maximum instantaneous torque is given by

$$\sigma_{b\,max} = \sigma_b \frac{[M_t]_{max}}{M_t} = 90.9 \times \frac{82.758}{63.66} = 118.17 \text{ N/mm}^2 \qquad [\because \sigma_b = 90.9 \text{ N/mm}^2]$$

From Table 5.23, for steel HB $\leq$ 350, permissible bending stress is given by

$$[\sigma_b]_{max} = 0.8\,\sigma_y = 0.8 \times 540 = 432 \text{ N/mm}^2$$

Since $\sigma_{b\,max} < [\sigma_b]_{max}$, ***the design is satisfactory.*** $[\because \sigma_y = 540 \text{ N/mm}^2]$

(ii) Check for wear strength : Induced contact stress due to maximum instantaneous torque is given by

$$\sigma_{c\,max} = \sigma_c \times \frac{[M_t]_{max}}{M_t}$$

$$= 701.71 \times \frac{82.758}{63.66} = 912.22 \text{ N/mm}^2 \qquad [\because \sigma_c = 701.71 \text{ N/mm}^2]$$

From Table 5.24, for steel HB ≤ 350, permissible contact stress is given by

$$[\sigma_c]_{max} = 3.1\,\sigma_y = 3.1 \times 540 = 1674 \text{ N/mm}^2$$

Since $\sigma_{c\,max} < [\sigma_c]_{max}$, ***the design is safe and satisfactory against plastic deformation also.***

16. ***Basic dimensions of pinion and gear :*** Refer Table 5.10.

- ✓ Module : $m = \mathbf{2.5\ mm}$
- ✓ Face width : $b = \mathbf{36\ mm}$
- ✓ Height factor : $f_0 = \mathbf{1}$
- ✓ Bottom clearance : $c = 0.25\,m = 0.25 \times 2.5 = \mathbf{0.625\ mm}$
- ✓ Tooth depth : $h = 2.25\text{ m} = 2.25 \times 2.5 = \mathbf{5.625\ mm}$
- ✓ Pitch circle diameter : $d_1 = m \cdot z_1 = 2.5 \times 24 = \mathbf{60\ mm}$; and

  $d_2 = m \cdot z_2 = 2.5 \times 72 = \mathbf{180\ mm.}$
- ✓ Tip diameter : $d_{a1} = (z_1 + 2\,f_0)\,m = (24 + 2 \times 1)\,2.5 = \mathbf{65\ mm}$; and

  $d_{a2} = (z_2 + 2\,f_0)\,m = (72 + 2 \times 1)\,2.5 = \mathbf{185\ mm.}$
- ✓ Root diameter : $d_{f1} = (z_1 - 2\,f_0)\,m - 2\,c$

  $= (24 - 2 \times 1)\,2.5 - 2 \times 0.625 = \mathbf{53.75\ mm}$; and

  $d_{f2} = (z_2 - 2\,f_0)\,m - 2\,c = (72 - 2 \times 1)\,2.5 - 2 \times 0.625$

  $= \mathbf{173.75\ mm.}$