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.***

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Given Data : P = 8 kW; N_1 = 1200 r.p.m. ; i = 3 ; ϕ = 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] :*

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

8.15

where

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

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

8.15

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

5. *Calculation of E_{eq}, [σ_b] and [σ_c] :*

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

8.14

(ii) *To find [σ_b] :* Design bending stress,

$$[M_t] = \frac{1.4 \times K_{bl}}{n \cdot K_\sigma} \times \sigma_{-1}$$

8.18

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

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

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

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

$= 0.35 \times 720 + 120 = 372\ \text{N/mm}^2$. 1.40 ... $[\because \sigma_u = 720\ \text{N/mm}^2]$

$\sigma_b = \dfrac{1.4 \times K_{bl}}{n\,K_\sigma}$ 8.18 $\therefore\ [\sigma_b] = \dfrac{1.4 \times 1}{2 \times 1.5} \times 372 = 173.6\ \text{N/mm}^2$

(iii) *To find* $[\sigma_c]$: 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.

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

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

Design 1 8.13

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

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

≥ 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) :*** 8.23 $m = \dfrac{2a}{z_1 + z_2} = \dfrac{2 \times 110}{24 + 72} = 2.29$ 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.

8.13

D → 1
C → 2
C → 1

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 ≤ 350 and $v = 3.77$ m/s, $K_d = 1.55$.

$$\therefore \quad [M_t] = M_t \cdot K \cdot K_d \quad \text{8.15}$$

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

13. Check for bending :

8.13 A

check ②

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

check ①

$$\sigma_c = 0.74 \frac{i+1}{a} \sqrt{\frac{i+1}{i\,b} \times E_{eq} [M_t]} \quad \text{8.13}$$

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

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

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

8.21

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

From Table 5.23, for steel HB ≤ 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

$$\sigma_{c\,max} = \sigma_c \times \frac{[M_t]_{max}}{M_t}$$ 8.21

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

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.***

8.22 **16. *Basic dimensions of pinion and gear :*** Refer Table 5.10. 8.22

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

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

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

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

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

  $=$ **173.75 mm.**

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Example 5.21