8

~~Design of spur gear using Manufacturer's data: (Type-2).~~

(or).

Design of Spur gear based on Gear life:

(1) In a spur gear drive for a stone crusher, the gears are made of C40 steel. The pinion is transmitting 30kW at 1200 rpm. The gear ratio is 3. Gear is to work 8 hours/day six days/week and for 3 years. Design the drive.

Given data:

Gear & pinion materials = C40 Steel.

power transmitted by pinion P = 30kW.

Speed of pinion N_1 = 1200 rpm.

Gear ratio i = 3.

Gear life = 8 hours/day

Solution:

Since both gear and pinion are of same material the pinion will be weaker and we have to design the pinion only.

Design procedure.

1. Gear ratio:

The gear ratio $i = 3$ (given).

2. Selection of Material:

The Gear & pinion are of same material : (C_{40} steel) and Assume that, Surface hardness is > 350 BHN.

1.12 | PSGDB | 8.4, 8.5, 8.16

3
π

1

3. Calculation of gear life.

Given that 8hrs/day, six days/week and for 3 yrs.

∴ Gear life in terms of hours = $8 \times (52 \times 6) \times 3$.

[Gear life = 7488 hrs = 449280 min.]

Gear life in Number of cycles: (N).

N = Gear life in min × N_1

N = 449280 × 1200

[∴ Gear life in Number of cycles N = 53.9×10^7 cycles]

4. Calculation of initial design torque $[M_t]$:

From PSG: 8.15, Table: 2.

∴ Design Torque $[M_t] = M_t \times k_d \times k$., $k_d \cdot k = 1.3$

where, M_t = Nominal torque transmitted by pinion. $\left[P = \frac{2\pi N T}{60}\right]$

$M_t = \frac{60 \times P}{2\pi N_1} = \frac{60 \times 30 \times 10^3}{2\pi \times 1200} = \boxed{238.73 \text{ N.m}}$

Design Torque $[M_t]$ = 238.73 × 1.3

$\boxed{[M_t] = 310.34 \text{ N.m}}$

3. Calculation of Equivalent Young's Modulus E_{eq}
From PSG 8.14 Table 9
The equivalent young's modulus $E_{eq} = 2.15 \times 10^5$ N/mm^2 for C-Steel

4. Calculation of Design Bending Stress $[\sigma_b]$
From PSG 8.18, Table 2.

$$[\sigma_b] = \frac{1.4\, k_{bl}}{n \cdot k_\sigma} \cdot \sigma_{-1}$$ (Assuming unidirectional rotation)

Where,

k_{bl} = Life factor for bending (PSG 8.20 Table 22)
n = Factor of Safety (PSG 8.19 Table 20)
k_σ = Stress concentration factor (PSG 8.19 Table 21)
σ_{-1} = Endurance Limit Stress (PSG 8.19 Table 19)

$\therefore$ $k_{bl} = 0.7$ for BHN > 350 and $N \geq 25 \times 10^7$ cycles
$n = 2$, for C40 steel
$k_\sigma = 1.5$, for steel
$\sigma_{-1} = 0.35\,\sigma_u + 120$ (for C40 steel) ($0.35\,\sigma_u + 1200$ kgf/cm^2)

$\sigma_u = 630$ N/mm^2 (From PSG 1.40) — C45, C55 is given in PSG 1.40 ... for C40

$\therefore$ $\sigma_{-1} = 0.35 \times 630 + 120$

$$\boxed{\sigma_{-1} = 340.5\ \text{N/mm}^2}$$

Hence $$[\sigma_b] = \frac{1.4 \times 0.7}{2 \times 1.5} \times 340.5$$

$$\boxed{\text{Design bending stress } [\sigma_b] = 111.23\ \text{N/mm}^2}$$

7. Calculation of design contact stress $[\sigma_c]$
From PSG 8.16 Table:2.

$$[\sigma_c] = C_B \cdot HB \cdot K_{cl} \quad (or) \quad [\sigma_c] = C_R \cdot HRC \cdot K_{cl}.$$

where,

CB or C_R ⇒ Co-efficient of surface hardness. (PSG 8.16, Table: 11
HB or HRC ⇒ Hardness Number. (PSG 8.16, Table: 16).
Kcl ⇒ Life factor (PSG 8.17, table: 17).

$$\therefore [\sigma_c] = C_R \cdot HRC \cdot K_{cl}.$$

$$[\sigma_c] = 26.5 \times 55 \times 0.585.$$ (C_R = 265, for Alloyed steel → but kgf/mm² ⇒ 26.5 N)

$$\boxed{\text{Design contact stress } [\sigma_c] = 852.64\ N/mm^2}$$

8. Calculation of centre distance (a).
From PSG. 8.13.

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

E_{eq} → eq. youngs mod

ψ → (width to centre ratio).

Assume ψ = 0.3, (width to centre ratio), Assumed intially $[\psi = b/a]$

$$\therefore a = (3+1)\sqrt[3]{\left(\frac{0.74}{852.64}\right)^2 \times \frac{2.15 \times 10^5 \times 310.34 \times 10^3}{3 \times 0.3}}$$

$$a = 152.89. \simeq a = 155mm$$

$$\boxed{\therefore \text{Centre distance } a = 155mm}$$

9. Selection of ~~a~~ No of teeth:

i) Assume $\boxed{Z_1 = 17}$ For 20° Full depth.

ii) $Z_2 = i \times Z_1 = 3 \times 17$ $\boxed{Z_2 = 51}$

10. Calculation of module (m)

We know,

$$m = \frac{2a}{Z_1 + Z_2} = \frac{2(155)}{17+51}$$ (psg. 8.22).

$$\boxed{\therefore m = 4.56\,mm}$$

From psg. 8.2; For choice 1, the standard $\boxed{\text{module } m = 5\,mm}$

11. Revision of centre distance. (a).

$$\text{New centre distance } a = \frac{m(Z_1+Z_2)}{2} = \frac{5(17+51)}{2}$$

$\therefore$ New centre distance $\boxed{a = 170\,mm}$ (8.22 psg).

12. Calculation of Face width, pitch ϕ of pinion, pitch line velocity and ψ pinion (width to centre ratio)

i) Face width $b = \psi \cdot a$ (From psg. 8.1).

$$b = 0.3 \times 170 = 51\,mm.$$

$$\boxed{\text{Face width } b = 51\,mm}$$

ii) pitch dia of pinion (d_1) = $\boxed{m \cdot Z_1 = 5 \times 17 = 85\,mm}$

iii) pitch line velocity (V) = $\frac{\pi d_1 N_1}{60}$

$= \frac{\pi \times 85 \times 10^{-3} \times 1200}{60}$

$\boxed{\text{Velocity } (V) = 5.34\ m/s}$

iv) width to centre ratio (ψ) = $\boxed{b/d_1 = \frac{51}{85} = 0.6}$ PSG 8.15

13 Selection of Quality of gears:

From PSG. 8.3, Table: 2

For peripheral speed (or) pitch line velocity for 5.34 m/s the IS Quality '8' is selected for cylindrical gears.

14 Revision of design Torque of gear [M_t].

Revision of load concentration factor "K", PSG. 8.15, Table 14.

For $\psi_p = 0.6$, Bearing close to gear $\boxed{K = 1.03}$

Revision of Dynamic load factor K_d (8.16, Table 15).

For IS Quality 8 HB > 350, to peripheral speed 5.34 m/s.

$\boxed{K_d = 1.4}$

∴ Revised design Torque [M_t] = 238.73 × 1.03 × 1.4

$\boxed{\text{Design Torque } [M_t] = 344.24\ N.m.}$

15. Check for bending:

Calculation of induced bending stress σ_b.

From PSG. 8.13A.

For spur gear, ~~form~~ checking Table:

$$\sigma_b = \frac{i \pm 1}{a m \cdot b \cdot y} \times [M_t].$$

Form factor 'y' = 0.377 for $Z_1 = 17$. (PSG. 8.18, Tab: 18)

As ~~Form~~ height factor = 0.8, Divide by 0.8.

$$\text{i.e.} = \boxed{y = \frac{0.377}{0.8} = 0.471.}$$

$$\sigma_b = \frac{(3+1)}{170 \times 5 \times 51 \times 0.471} \times 344.24 \times 10^3$$

$$\boxed{\sigma_b = 67.43\ N/mm^2}$$

we find $\sigma_b < [\sigma_b]$ =

$$\boxed{67.43 N/mm^2 < 111.23 N/mm^2,}$$

∴ The design is safe & satisfactory.

16 check for wear strength: σ_c.

From PSG. 8.13.

$$\sigma_c = 0.74 \cdot \frac{i+1}{a} \sqrt{\frac{i+1}{ib} \times E_{eq}[M_t]}.$$

$$= 0.74 \times \frac{3+1}{170} \sqrt{\left(\frac{3+1}{3 \times 51}\right) \times 2.15 \times 10^5 \times 344.24 \times 10^3}$$

$$\boxed{\sigma_c = 765.9\ N/mm^2}$$

AS $\sigma_c < [\sigma_c]$; $\boxed{765.9\ N/mm^2 < 852.64\ N/mm^2,}$ the desig is safe & satisfactory.

17 Calculation of basic dimensions of pinion & gear.

From PSG 8.22,

Module $m = 5$ mm

Face width $b = 51$ mm.

hight factor $f_0 = 1$ for full depth.

Bottom clearance $C = 0.25m = 1.25$ mm.

Tooth depth $h = 2.25m = 11.25$ mm.

pitch circle d $d_1 = m.z_1 = 85$ mm

$d_2 = m.z_2 = 255$ mm.

Tip dia $d_{a1} = (17+2\times1)5 = 95$ mm

$d_{a2} = (51+2\times1)5 = 265$ mm.

Root dia $d_{f1} = (17-2\times1)5 - 2\times1.25 = 72.5$ mm.

$d_{f2} = (51-2\times1)5 - 2\times1.25 = 242.5$ mm.