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Design Procedure:-

1) Calculation of gear ratio:-

$$i = \frac{N_1}{N_2} = \frac{Z_2}{Z_1}.$$

In case of multistage speed reducers, the speed ratio may be selected from Reo series.

2) Selection of materials:-

Consulting Table 5.3, choose the suitable combination of materials for pinion and wheel.

3) If not given, assume gear life (say 20,000 hrs).

4) Calculation of initial Design Torque):-

$$[M_t] = M_t \cdot K \cdot K_d.$$

$$[M_t] \Rightarrow \text{Transmitted Torque} \Rightarrow \frac{60 \times P}{2\pi N}$$

5) calculation of E_{eq}, (σ_b), and (σ_c):-

✓ consulting Table 5.20, calculate the equivalent Young's Modulus (E_{eq}),

✓ Calculate the design bending stress $[\sigma_b]$,

To find $[\sigma_c]$

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

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

6) calculation of centre distance:-

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

$\psi = b/a \Rightarrow$ width to centre distance ratio.

7) Selection of number of teeth on pinion (z_1) and gear (z_2):-

i) No. of teeth on pinion $z_1 \geq 17$ say 18.

ii) No. of teeth on gear z_2 : $z_2 = i \times z_1$

8) Calculation of Module:-

$$m = \frac{2a}{(z_1+z_2)}.$$

9) Revision of centre Distance:- (a):-

$$a = \frac{m(z_1+z_2)}{2}.$$

10) Calculation of b, d_1, v and ψp :-

Calculate face width $b = b = \psi a$.

Calculate the pitch dia of pinion $d_1 = d_1 = m.z_1$

Calculate pitch line velocity $V = \frac{\pi d_1 N_1}{60}.$

$$\psi p = \frac{b}{d_1}$$

11) Selection of quality of gears :-

Peripheral speed of gear (from databook P.g No 8.3).

12) Revision of Design Torque:- $[M_t]$:-

Revise k:- using the calculated value of ψp, revise the value of load concentration factor (k) from table 5.11,

Revise kd:- using the selected quality of Gear and calculated pitch line velocity revise the value of dynamic load factor (kd).

Revise [Mt]:

using the revised values of k and kd,

$$[M_t] \Rightarrow M_t \cdot k \cdot k_d.$$

check for bending:-

$$\sigma_b \Rightarrow \frac{(i \pm 1)}{a m \cdot b \cdot y} [M_t]$$

from data book
P.g No 8.13.

14) check for wear strength:-

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

from data book
P.g No: 8.13

15) If the design is not satisfactory, the increases the module (or) face width value (or) change the gear material.

16) check for gear:-

i) check for bending:-

$$\sigma_{b1} y_1 \text{ and } \sigma_{b2} y_2 \quad (\text{or}) \quad \sigma_{b2} = \frac{\sigma_{b1} y_1}{y_2}.$$

σ_{b1} and σ_{b2} => Induced bending stress in the Pinion and gear respectively.

y_1 and y_2 => Form factors of Pinion and gear respectively.

If $\sigma_{b2} \leq [\sigma_{b2}]$

ii) check for wear strength:-

-> calculate the induced contact stress σ_{c2} for gear using the equation In fact, the induced contact stress will be same for pinion and wheel.

→ calculate the design contact stress for gear $[\sigma_{c2}]$ as discussed in step 5.

→ compare the induced bending stress σ_{c2} and the design bending stress $[\sigma_{c2}]$. If $\sigma_{c2} \leq [\sigma_{c2}]$. then the design is safe and satisfactory.

17) Calculation of basic dimensions of pinion and gear:-

Calculate all the basic dimensions of pinion and gear using the relations listed in Table 5.10.