D. Design of Bevel gear

Design a bevel gear drive to transmit 7.36 kW at 1440 rpm for the following data. Gear ratio = 3. Pinion teeth Number = z_1 = ... material for pinion and gear C45 surface hardened (~~HB ≤ 350~~) (HB > ... ~~with~~ to 55 RC, ~~core hardness ≤ 350 BHN~~

~~Step 1~~ Assume the life of the gear = 10,000 hrs.

Step ①

Material selection:

P.NO:- ~~Table 5~~ ~~i.e., HB ≤ 350, C45~~

Table 7, P.NO:- 8.5 steel C45; m up to 6.

$[\sigma_b] = 1400$ kgf/cm²; $[\sigma_c] = 5000$ kgf/cm².

$[\sigma_b] = 140$ N/mm²; $[\sigma_c] = 500$ N/mm².

$\sigma_u = 70$ kgf/mm². min $\sigma_{-1} = 27$ kgf/mm² 1 cm = 10 mm. 1 cm² = 100 mm²

$\boxed{\sigma_u = 700\ N/mm^2}$ $\boxed{\text{Take } \sigma_{-1} = 300\ N/mm^2}$

$\frac{1400 \times 10}{100}$ N/... 140 N/mm ... ~~70×0~~

$[\sigma_b]$ = Design Bending stress = 140 N/mm²
= 1400 kgf/cm².

10 mm = 1 cm
100 mm² = 1 ...

$[\sigma_c]$ = Design Contact Stress = 5000 kgf/cm²
= 500 N/mm².

σ_{-1} = Endurance limit stress = min 27 kgf/mm²; Take = 30 kgf/mm² = 300 N...

σ_u = Ultimate tensile strength = 70 kgf/mm² = 700 N/mm²

step ② Assume life of the gear:

P.NO:- 8.17;

~~Life~~ for Constant loading

life in cycle $N = 60 \times n \times T$,
$= 60 \times 1440 \times 10{,}000$.

$\boxed{N = 86.4 \times 10^7 \text{ Cycles}}$

n = Speed of the pinion in rpm.

T = life in hors.

Step (3)

Determination of cone distance (R).

Page NO :- 8.13 ; Table 8 ;

Cone distance $R \geq \psi_y \sqrt{i^2+1} \sqrt[3]{\left[\frac{0.72}{(\psi_y - 0.5)[\sigma_c]}\right]^2 \frac{E[M_t]}{i}}$

P.NO :- 8.1 $\psi_y = R/b$ (Table - 13).

From P.NO : 8.15 ; Table 13 for $i = 1$ to 4 : $(i=3)$.

$$\boxed{\psi_y = R/b = 3}$$

Design twisting Moment : $[M_t]$ (P.NO : 8.15).

$[M_t] = M_t \, k_d \, k$.

for intial Calculation assume $k_d k = 1.3$.

Power $P = \frac{2\pi N M_t}{60}$.

$M_t = \frac{7.36 \times 10^3 \times 60}{2 \times \pi \times 1440}$.

$$\boxed{M_t = 48.807 \text{ N-m}}$$

Design Torque $[M_t] = M_t \, k_d \, k$.

$= 48.807 \times 1.3$

$$\boxed{[M_t] = 63.45 \text{ N-m}}$$

$[M_t] = 63.45 \times 10^3$ N-mm.

Young's Modulus :- (E)

From P.NO :- 8.14 ; Table (9) (Pinion & gears are same material C45 steel).

Equivalent young's Modulus $E = 2.15 \times 10^6$ kgf/cm².

$E = \frac{2.15 \times 10^6 \times 10}{100}$

$$\boxed{E = 2.15 \times 10^5 \text{ N/mm}^2}$$

$\therefore$ Cone distance $R \geq \psi_y \sqrt{i^2+1}\ \sqrt[3]{\left[\frac{0.72}{(\psi_y-0.5)[\sigma_c]}\right]^2 \frac{E[M_t]}{i}}$

$= 3\sqrt{(3^2+1)}\ \sqrt[3]{\left(\frac{0.72}{(3-0.5)(500)}\right)^2 \frac{2.15\times10^5 \times 63.45\times10^3}{3}}$

$\boxed{R = 108.8\text{ mm}}$

Step ④ Determination of Module (m_t).

From P.NO:- 8.38, Table 31

Cone Distance $R = 0.5 m_t Z_1 \sqrt{i^2+1}$

~~Transverse module m_t~~ $\boxed{Z_1 = 18}$

$108.8 = 0.5\, m_t \times 18 \times \sqrt{3^2+1}$

$\boxed{m_t = 3.822}$

from P.NO: 8.2, Table 1

the Standard Module $\boxed{m_t = 4}$

Step ⑤ Revis the Cone distance and face wide based on new standard Module $m_t = 4$.

$R = 0.5\, m_t Z_1 \sqrt{i^2+1}$

$R = 0.5 \times 4 \times 18 \sqrt{3^2+1}$

Cone distance:

$\boxed{R = 113.84\text{ mm}}$

$\psi_y = R/b = 3$

$b = \frac{113.84}{3}$

face width $\boxed{b = 37.95\text{ mm}}$

step ⑥ Revise ~~design~~ design Torque $[M_t]$, k_d, & k.

from P.NO:- 8.38. Table 3,

Transverse Module $m_t = m_m + \frac{b \sin\delta_1}{z_1}$ (for pinion use z_1).

$\therefore$ Mean Module $m_m = m_t - \frac{b \sin \delta_1}{z_1}$

From P.NO:- 8.39. Reference angle

$\tan\delta_2 = i$; $\delta_1 = 90° - \delta_2$.

$\tan\delta_2 = 3$;

$\boxed{\delta_2 = 71.56°}$ $\delta_1 = 90° - 71.56$

$\boxed{\delta_1 = 18.44°}$

$\therefore m_m = 4 - \frac{37.95 \sin 18.44}{18}$

$\boxed{m_m = 3.333 \text{ mm}}$

Pitch line velocity. (P.NO: 8.15; below Table 14).

$V = \frac{\pi d_{1av} n_1}{60 \times 1000}$ m/sec. n_1 = Speed in rpm.

Reference dia $d_1 = m_t z_1$ (P.NO:- 8.38, Table 3)

~~It~~ Similarly avg reference dial d_{1av}.

$d_{1av} = m_m z_1$

$= 3.333 \times 18$

$= 59.9945$

$\boxed{d_{1av} = 60 \text{ mm}}$

Pitch line velocity $V = \frac{\pi \times 60 \times 1440}{60 \times 1000}$.

$\boxed{V = 4.524 \text{ m/sec}}$

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From ~~Table~~ P.NO:- 8.3; Table 2 (Bevel gear).

IS Quality grade for $V = 4.52$ m/sec. (straight bevel gear)

speed up to 6 m/sec $\Rightarrow$ $\boxed{\text{Quality grade is } 6}$

Load concentration factor (k) (From P.NO:- 8.15, Table 14).

HB $\not<$ 350; $b/d_{iav} = \frac{37.95}{60} = 0.6325$.

Surface hardened HB > 350. $b/d_{iav} \leq 1$ (Bevel Gear).

$$\boxed{k = 1.6}$$

Dynamic Load factor (kd)

From P.NO:- 8.16; Table 15.

Conical gear / IS Quality ⑥; pitch line velocity up 8 m/sec ($V = 4.25$ m/sec), HB > 350.

$$\boxed{k_d = 1.4}$$

$\therefore$ Revised $[M_t] = M_t\, k_d\, k$.

$$= 48.807 \times 1.4 \times 1.6$$

$$\boxed{[M_t] = 109.33 \text{ N-m}}$$

$$\boxed{[M_t] = 109.33 \times 10^3 \text{ N-mm}}$$

Step ⑦. Calculation of Induced stress.

Step (7.1) Induced Bending stresses.

P.NO:- 8.13A (Table 8 Contd).

For straight bevel gear:

$$\sigma_b = \frac{R\sqrt{i^2+1}\,[M_t]}{(R-0.5b)^2\, b\, m_t\, y_v} \times \frac{1}{\cos\alpha} \leq \sigma_b.$$

$\alpha = 20°$; Pressure angle.

Virtual Number of teeth. (Z_v) (Tredgold's approximation)

$$Z_v = \frac{Z}{\cos \delta_1} = \frac{18}{\cos 18.44} = 18.97$$

$$\boxed{Z_v = 19 \text{ teeth.}}$$

for Calculating tooth form factor $\alpha = 20°$

From P.NO: 8.18; Table 18. [Addendum Modification $X = 0$].

Form factor y_v.

	Z_v	y_v
for	18 →	0.377
for	20 →	0.389

$\boxed{\text{for } Z_v = 19} \Rightarrow y_v = \frac{0.377 + 0.389}{2} = 0.383$.

$$\boxed{y_v = 0.383}$$

Bending stress $\sigma_b = \frac{R\sqrt{i^2+1}\,[M_t]}{(R-0.5b)^2\, b\, m_t\, y_v \cos\alpha}$.

$$= \frac{113.84\sqrt{3^2+1} \times 109.33 \times 10^3}{[113.84-(0.5 \times 37.95)]^2 \times 37.95 \times 4 \times 0.383 \times \cos 20}$$

$$= \frac{39358110}{491663.9705}$$

$\boxed{\sigma_b = 80 \text{ N/mm}^2}$. $\sigma_b = 80.05 \text{ N/mm}^2$.

$\sigma_b = 80 \text{ N/mm}^2 < [\sigma_b] = 140 \text{ N/mm}^2$.

The induced bending stress $\sigma_b = 80 \text{ N/mm}^2$ is less than the Material bending strength $[\sigma_b] = 140 \text{ N/mm}^2$. So the bevel gear is safe for bending strength.

Step 7.2. Induced Contact Stress.

P.NO:- 8.13, Table 8.

$$\sigma_c = \frac{0.72}{(R-0.5b)} \sqrt{\frac{\sqrt{(i^2+1)^3}}{i\,b} E[M_t]}$$

$$= \frac{0.72}{[113.84 - (0.5 \times 37.95)]} \cdot \sqrt{\frac{\sqrt{(3^2+1)^3}}{3 \times 37.95} \times 2.15 \times 10^5 \times 109.33 \times 10^3}$$

$$= \frac{0.72}{94.865} \cdot \sqrt{\frac{\sqrt{1000}}{113.85} \times 2.35 \times 10^{10}}$$

$\sigma_c = 613.66 \, N/mm^2 > [\sigma_c] = 500 \, N/mm^2$

Design is not safe. becau $\sigma_c > [\sigma_c]$

Trial ②: Revise the transverse module $\boxed{m_t = 5}$

Repeat the step ⑤ and ⑥ for $\boxed{m_t = 5}$

Cone distance

$$R = 0.5 \times m_t \, z_1 \sqrt{i^2+1}$$

$$= 0.5 \times 5 \times 18 \sqrt{3^2+1}$$

$$\boxed{R = 142.302 \text{ mm}}$$

face width

$\psi_y = R/b = 3$; $b = \frac{142.302}{3} = 47.434 \text{ mm}$

mean module $m_m = m_t - \frac{b \sin\delta_1}{z_1}$

$$= 5 - \frac{47.434 \times \sin 18.44}{18}$$

$$\boxed{m_m = 4.166}$$

$d_{1av} = m_m z_1 = 4.166 \times 18 = 74.99 = 75 \text{ mm}$.

Pitch line velocity $= \frac{\pi \times 75 \times 1440}{60 \times 1000} = 5.655 \text{ m/sec}$.

for ~~v=5~~ v upto 6 m/sec ($v = 5.655$) Quality grad = 6.

Load Concentration factor (k)

HB > 350; $b/d_{1av} = 0.632 \leq 1$

$k = 1.6$;

Dynamic load factor (k

No change in k_d.

$\boxed{k_d = 1.4}$

No change in $[M_t]$ also.

$[M_t] = 109.30$ N-m.

$[M_t] = 109.33 \times 10^3$ N-mm.

Step ~~7.1~~ Induced ~~Bending~~ Contact Stress σ_c

$$\sigma_c = \frac{0.72}{(R-0.5b)} \sqrt{\frac{\sqrt{(i^2+1)^3}}{ib} E[M_t]}$$

$$= \frac{0.72}{[142.302-(0.5 \times 47.434)]} \times \sqrt{\frac{\sqrt{(3^2+1)^3}}{3 \times 47.434} \times 2.15 \times 10^3 \times 109.33 \times 10^3}$$

$$= \frac{0.72 \times 72274.218}{118.585}$$

$\sigma_c = 438.82$ N/mm2 $< [\sigma_c] = 500$ N/mm^2.

The induced Contact stress is less than the material Contact stress. so the bevel gear design is safe for Contact strength.

7.2 Induced Bending stress.

$$\sigma_b = \frac{R\sqrt{i^2+1}\,[M_t]}{(R-0.5b)^2\, b\, m_t\, y_v \cos\alpha}$$

$$= \frac{142.302 \times \sqrt{3^2+1} \times 109.33 \times 10^3}{[142.302-(0.5 \times 47.434)]^2 \times 47.434 \times 5 \times 0.383 \times \cos 20}$$

$$= \frac{49198674.7}{1200338.842}$$

$\sigma_c = 40.987$ N/mm2 $< [\sigma_b] = 140$ N/mm2.

The bevel gear design is safe for Bending strength also.

The Pinion is safe for both bending and Contact strength.

Step 8:

Design for Dynamic load:

~~Same~~ P.NO: $b = \frac{10}{Pd}$; $Pd = \frac{10}{47.434} = 0.2$

P.NO:- 8.52

Static load $F_s = \frac{[\sigma_b]\, b\, Y_v\, (1 - b/R)}{Pd}$;

$$= \frac{[140] \times 47.434 \times 0.383 \left[1 - \left(\frac{47.434}{142.302}\right)\right]}{Pd}$$

P.NO: 8.52 $b = \frac{10}{Pd}$; $47.434 = \frac{10}{Pd}$.

$$\boxed{Pd = 0.21082}$$

Static load $F_s = \frac{[140] \times 47.434 \times 0.383\, [1 - 0.333]}{0.21082}$

$$\boxed{F_s = 8042.915\ N}$$

~~Buckingham Dynamic load~~. use

Buckingham's Dynamic load. (F_d)

P.NO 8.52 Not (For Buckingham's dynamic load use the

Equation for spur gear taking V_m for the largest pitch circle.

P.NO:- 8.51;

$$F_d = F_t + \left[\frac{0.164\, V_m\, (cb + F_t)}{0.164\, V_m + 1.485\left(\sqrt{cb + F_t}\right)}\right]$$

for $V_m = \frac{\pi d_1 n_1}{60 \times 1000}$ m/sec.

$d_1 = m_t Z_1 = 5 \times 18 = 90mm$ (largest Value).

$V_m = \frac{\pi \times 90 \times 1440}{60 \times 1000} = 6.786$ m/sec.

$V_m = 6.786$ m/sec.

velocity factor: (C_v) for straight bevel gear.

P.NO: 8.52; $C_v = \frac{3.5 + \sqrt{V_m}}{3.5}$

$$\boxed{C_v = 1.744}$$

$$F_t = \frac{\text{Power}}{\text{velocity}} \times C_v.$$

$$= \frac{7.36 \times 10^3}{6.786} \times 1.744.$$

$$\boxed{F_t = 1891.52 \text{ N}}$$

Calculation of c from 41 (P.NO: 8.53).

for steel. $\alpha = 20^\circ$ full depth.

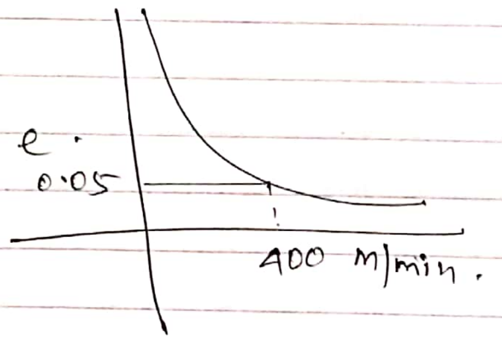
$c = 11860 e.$

Error from graph (P.NO 8.51).

$V_m = 6.786$ m/sec.

$= 6.786 \times 60$ m/min.

$$\boxed{V_m = 407.16 \text{ m/min}}$$



$$\boxed{e = 0.05}$$

$c = 11860 \times 0.05 = 593.$

Dynamic load.

$$F_d = 1891.52 + \frac{0.164 \times 6.786 \left[(593 \times 47.434 \times 10^{-3}) + 1891.52\right]}{(0.164 \times 6.786) + 1.485\left(\sqrt{(593 \times 47.434 \times 10^{-3}) + 1891.52}\right)}$$

in m.

$$= 1891.52 + \frac{2136.384}{66.176}$$

$= 1891.52 + 32.283$

$$\boxed{F_d = 1923.80 \text{ N.}}$$

$8042.91 > 1923.8.$

$F_s > F_d$

Design is safe for Dynamic load.