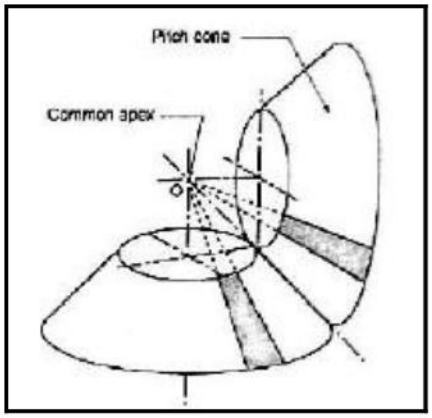
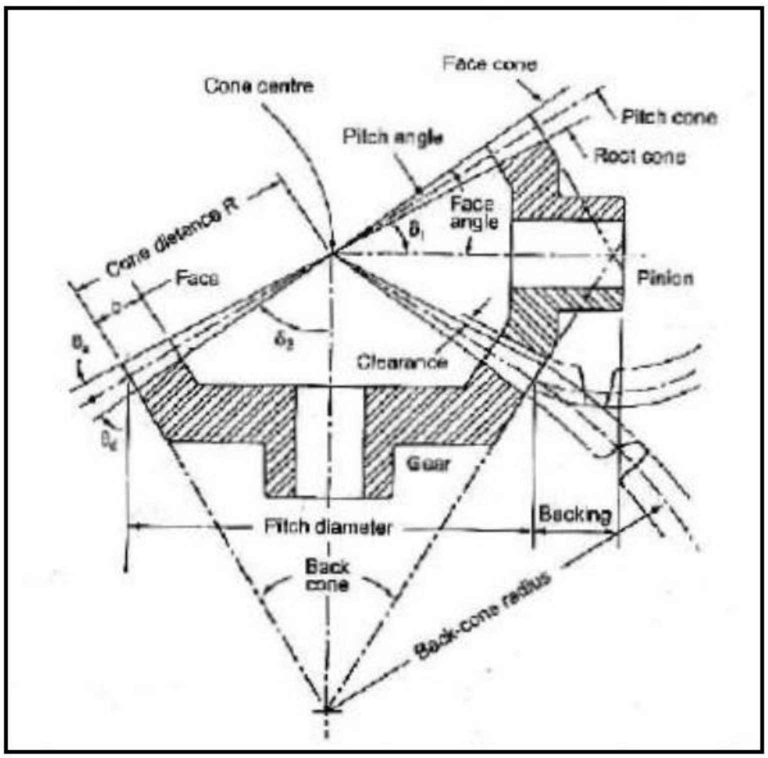
UNIT-III

BEVEL, WORM AND CROSS HELICAL GEARS

Bevel Gears



BEVEL GEAR NOMENCLATURE



DESIGN PROCEDURE FOR BEVEL GEAR

1. **Selection of material:** Select a suitable pinion and gear materials.

2. **Calculation of z_1 and z_2 :**
    1. Assume z_1 =17

2. $Z_2 = i*z_1$. Where **i** = gear ratio.

3. **Calculate the pitch angles** (i,e., δ_1 and δ_2) and the virtual number of teeth (i.e., z_{v1} and z_{v2}) using the following relations.
   Pitch angles: **tan δ_2 = i and δ_1= 90~- δ_2,…..(From data book page no. 8.39)**

$$z_{v1}=\frac{z_1}{\cos\delta_1} \text{ and } z_{v2}=\frac{z_2}{\cos\delta_2}$$ …… **(From data book page no. 8.52)**

4. **Calculation of tangential load on tooth (Ft):**
   1. $F_t = \left(\frac{P*K_0}{V}\right)$

   P= Transmitter power in watts.
   V= Pitch line velocity in m/s.
   K_0 = Service factor (Assume 1.25).

5. **Calculation of initial dynamic load (Fd):**
   1. $F_d = \left(\frac{Ft}{C_v}\right)$……… **(From data book page no. 8.50)**
   2. $C_v = \frac{5.6}{5.6+\sqrt{V}}$ (Assume V=5)

6. **Calculation of beam strength (Fs):**

   Fs= $\pi.m_t.b.[\sigma_b].y'.\left(\frac{R-b}{R}\right)$…. (From data book page no. 8.52)

   m_t= Transverse module in mm.
   Fs = Strength of gear tooth.
   $[\sigma_b]$ = Allowable static stress.
   b = Face width.
   y' = Form factor = **$(0.154 - \frac{0.912}{z_{v1}})$for 20° involute….(Pg no .8.50)**

   $$R=\sqrt{\left(\frac{d1}{2}\right)^2+\left(\frac{d2}{2}\right)^2}$$

7. **Calculation of module (m_t):**
   1. **Fs $\geq$ F_d**

   Calculate the value of m_t and select the nearest standard module value **from data book page no. 8.2**

8. **Calculation of b, d and v :**
   1. Face width **b = $10m_t$.**
   2. Pitch circle diameter **$d_1 = m_t * z_1$**
   3. Pitch line velocity **v** = $\left(\frac{\pi d_1 N_1}{60}\right)$

9. **Recalculation of beam strength (F_s):**
   1. **Fs= $\pi.m_t.b.[\sigma_b].y'.\left(\frac{R-b}{R}\right)$**

10. Calculation of accurate dynamic load (F_d):

$$F_d = F_t + \frac{21v(cb.+F_t)}{21v+\sqrt{bc+F_t}}$$

F_d= Total dynamic load on gear tooth.

F_t = Transmitted load.

c = Deformation factor ……**(From data book page no. 8.53)**

11. Check for Beam Strength :

i. Compare F_d and F_s

ii. If $F_s \geq F_d$ Design is safe and satisfactory.

12. Calculation of maximum wear load (F_w) :

$$F_w = \left(\frac{0.75.d_1.b.Q'.Kw}{\cos\delta 1}\right)$$

Q = Ratio factor = $\left(\frac{2*i}{i+1}\right)$ ….. **(From data book page no. 8.51)**

F_w= Maximum wear load.

$$K_w = \frac{f^2 \sin\alpha}{1.4}\left(\frac{1}{E_p} + \frac{1}{E_g}\right)$$ …..**(From data book page no. 8.51)**

f = (2.8 * BHN – 70) N/mm^2

d_1 = Pitch circle diameter.
B = Face width.

13. Check for wear:

i. Compare F_d and F_w

ii. If $F_w \geq F_d$ Design is safe and satisfactory.

14. Calculation Basic Dimensions of:
Basic dimensions of bevel gear ……... **(From data book page no. 8.38)**

DESIGN PROCEDURE FOR BEVEL GEAR WITH GEAR LIFE

INDIAN STANDARD

1. **Selection of Material:** Select a suitable pinion and gear materials.
2. **Gear Ratio:**

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

$$i = \tan\delta_2$$

$$\delta_1 + \delta_2 = 90^0$$

3. **Gear Life:**

$$N_{cycle} = N_{(in\ hrs)} \text{ x } 60 \text{ x rpm}$$

$$N_{cycle} = N_{(in\ mins)} \text{ x rpm}$$

4. **Calculation of Initial design Torque**: $[M_t]$

$$[M_t] = M_t \text{ x } K \text{ x } K_d$$

$$K \text{ x } K_d = 1.3$$

M_t =Transmitted torque = $\frac{60 * p}{2\pi N}$

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

   g. $[E_{eq}]$ = Equivalent young's modulus….[**PSG data book page no:8.14**]

   h. $[\sigma_b] = (\frac{1.4\, K_{bl}}{n K_\sigma})\sigma_{-1}$ …….[**PSG data book page no:8.18**]

   K_{bl}= Life factor for bending……….[**PSG data book page no:8.20**]

   K_σ = Stress concentration factor …….[**PSG data book page no:8.19**]

   n = Factor of safety…………………..[**PSG data book page no:8.19**]

   σ_{-1} = Endurance limit stress…………..[**PSG data book page no: 8.19**]

   i. $[\sigma_c] = C_B.\ HB.\ K_{cl}$

   $C_B = \frac{C_B}{10}$ ………… [**PSG data book page no: 8.16**]

   H_B = Brinell hardness number….. [PS**G data book page no: 8.16**]

   K_{cl}= Life factor for surface strength….[**PSG data book page no: 8.17**]

6. **Calculation of cone radius:**

$$R \geq \varphi_y\sqrt{i^2+1}\sqrt[3]{[\frac{0.72}{(\varphi_y-0.5)[\sigma_c]}]^2 \times \frac{E_{eq}[M_t]}{i}}$$ ….[**PSG data book page no: 8.13**]

Assume $\varphi_y = \frac{R}{b} = 3$

7. **Selection of number of teeth:**

   Assume $Z_1 = 17$

   $Z_2 = i \times Z_1$

   $Z_{v1} = \frac{Z_1}{cos\delta_1}$ and $Z_{v2} = \frac{Z_2}{cos\delta_2}$

8. **Transverse module:**

$$m_t = \frac{R}{0.5\ \sqrt{Z1^2+Z2^2}}$$ ……….[**PSG data book page no: 8.38**]

Standard transverse module in **PSG data book pg no: 8.2**

9. **Revision of cone distance:**

   $R = 0.5\, m_t\sqrt{Z1^2+Z2^2}$ …..[**PSG data book page no: 8.38**]

10. **Calculation of b, $m_{av}, d_{1av}, v, \varphi_y$:**

   Face width, $b = \frac{R}{\varphi_y}$

   Average module, $m_{av} = m_t - \frac{b\, sin\delta_1}{Z_1}$

   Average pitch circle dia, $d_{1av} = m_{av} \times Z_1$

Pitch line velocity, $v = \frac{\pi \times d_{1av} \times N_1}{60}$

$\varphi_y = \frac{b}{d_{1av}}$

11. ***Selection of quality of gears*:**

Quality of gears can be selected from the **PSG data book page number 8.3**

12. Revision of design Torque $[M_t]$:

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

K= Load concentration factor ……[**PSG data book page no: 8.15]**

K_d= Dynamic Load Factor………..[**PSG data book page no:8.16]**

13. Check for Bending:

$\sigma_b = \frac{R\sqrt{i^2+1}\,[M_t]}{(R-0.5b)^2 \times b \times m_t \times Y_v}$………..[**PSG data book page no:8.13A]**

Y_v from **PSG data book page no: 8.18** for value of Z_{v1}

$\sigma_b < [\sigma_b]$ – Design is safe and satisfactory.

14. Check for wear strength:

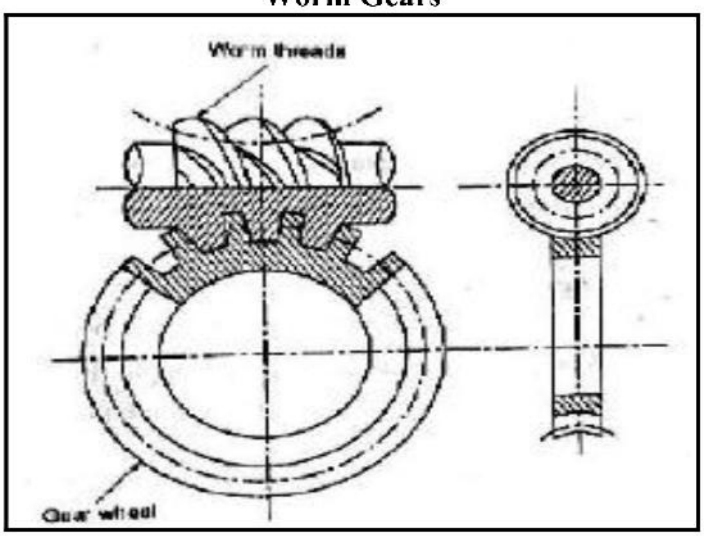
$\sigma_c = \frac{0.72}{(R-0.5b)}\sqrt{\left[\frac{\sqrt{(i^2 \pm 1)^3}}{i \times b} \times E_{eq}\,[M_t]\right]}$ ………..[**PSG data book page no:8.13]**

$\sigma_c < [\sigma_c]$ - Design is safe and satisfactory.

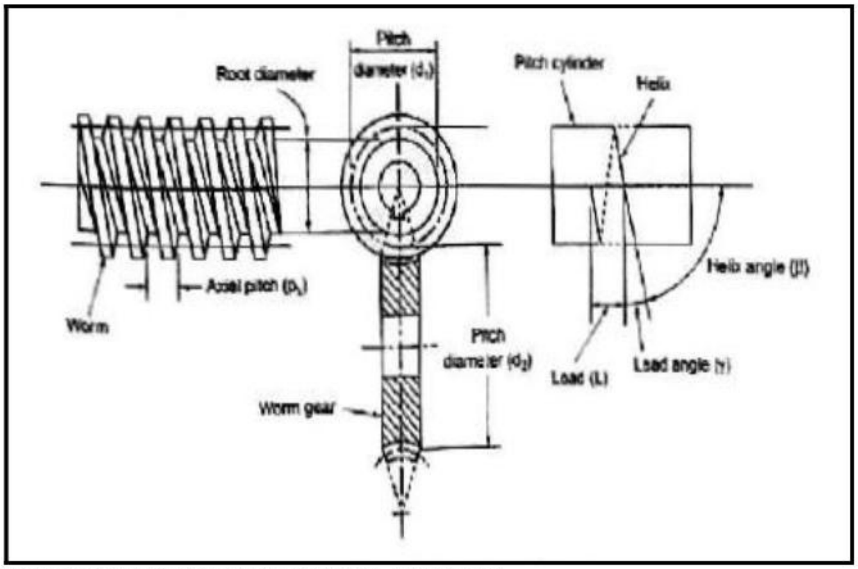
15.**Basic Dimensions:**

For Basic Dimensions of bevel pinion and gear ..[**PSG data book page no:8.38]**

Worm Gears



NOMENCLATURE OF WORM GEARS



TOOTH PROPORTIONS OF WORM GEARS

S.No.	Particulars	Symbol	Unit	Worm	Worm Gear
1.	Addendum	h_a	mm	$h_{a1} = m_x$	$h_{a2} = m_x (2\cos\gamma - 1)$
2.	Dedendum	h_f	mm	$h_{f1} = (2.2 \cos\gamma - 1)\, m_x$	$h_{f2} = m_x (1 + 0.2 \cos\gamma)$
3.	Clearance	c	mm	$C = 0.2\, m_x \cos\gamma$	-
4.	Outside diameter	d_a	mm	d_{a1}-$d_1 + 2h_{a1} = m_x (q + 2)$	d_{a2}-$d_2 + 2h_{a2}$ $= m_x (z_2 + 4 \cos\gamma - z)$
5.	Root diameter	d_f	mm	$d_{f1} = d_1 - 2h_{f1}$ $= m_x (q + 2 - 4.4 \cos\gamma)$	$d_{f2} = d_2 - 2h_{f2}$ $= m_x (z_2 - 2 - 0.4 \cos\gamma)$

MATERIALS FOR WORM AND WORM WHEEL • 1

The following guidelines may be used while selecting the materials for Worm and worm wheel.

S. No.	Condition	Material	
		Worm	Worm Wheel
1.	Light loads and low speed	Steel	Cast iron
2.	Medium service conditions	Case hardened steel of BHN 250	Phosphor bronze
3.	High speeds, heavy loads with shock conditions	Hardened molybdenum steel or chrome vanadium steel	Phosphor bronze (chilled)

SELECTION OF NUMBER OF STARTS IN THE WORM (Z_1):

Table 8.4 shows the approximate efficiencies for the number of starts in the worm (from data book, page no. 8.46

LENGTH OF WORM (OR LEAD), L (from data book, page no. 8.48)
FACE WIDTH OF THE WHEEL (b) (from data book, page no. 8.48)

EFFICIENCY

The efficiency of the worm gearing considering only the gearing losses is given by

$$\eta = \frac{\tan\alpha}{\tan(\gamma + \rho)}$$

Where ρ = Angle of friction = $\tan^{-1}(\mu)$, and
μ = Coefficient of friction,

The efficiency of the worm gearing taking into account all the losses is given by

$$\eta = (0.95 - 0.96)\frac{\tan\gamma}{\tan(\gamma + \rho)}$$

THERMAL RATING OF WORM GEARING

Where $H_g = (1 - \eta)$ x Input power,
$H_d = K_t \times A \times (t_o - t_a)$
K_t = Heat transfer coefficient of housing walls ($W/m^2 8C$),
A = Effective surface area of the housing (m^2),
t_o = Temperature of lubricating oil (8C), and
t_a = Temperature of the atmospheric air (8C).

Therefore **$(1 - \eta)$ x Input power = $K_t \times A \times (t_o - t_a)$**

DESIGN PROCEDURE FOR WORM AND WORM WHEEL

1. Selection of material: Select a suitable material for worm and worm wheel.

2. Calculation of z_1 and z_2 :

1. Depending upon the efficiency requirement, select the no. of starts (Z_1) in the worm .....**(From data book page no. 8.46)**
2. $Z_2 = i*z_1$. Where **i** = gear ratio.

3. Calculate the diameter factor (q) and lead angle (γ):

Diameter factor, $q = \frac{d1}{m_x}$. If not assume **q = 11**

Lead angle, $\gamma = tan^{-1}(\frac{z1}{q})$

4. Calculation of tangential load on wheel (Ft):

1. $F_t = \left(\frac{P * K_0}{V}\right)$

P= Transmitter power in watts.
V= Pitch line velocity in m/s.
K_0 = Service factor (Assume 1.25).

5. **Calculation of initial dynamic load (Fd):**
   1. **$F_d = \left(\frac{Ft}{C_v}\right)$……… (From data book page no. 8.50)**
   2. **$C_v = \frac{6}{6+V}$** (Assume V=5)… **(From data book page no. 8.51)**

6. **Calculation of beam strength (Fs):**

   $Fs= \pi.m_x.b.[\sigma_b].y'$ …. (From data book page no. 8.52)

   m_x= Axial module in mm.
   Fs = Strength of gear tooth.
   $[\sigma_b]$ = Allowable static stress.
   b = Face width = **$8.25m_x$**
   y' = Form factor = $\left(\frac{Y}{\pi}\right)$ ….**(From data book page no. 8.52)**

7. **Calculation of axial module (m_x):**
   1. **$Fs \geq F_d$**

   Calculate the value of **m_x** and select the nearest standard module value**from data book page no. 8.2**

8. **Calculation of b, d and v :**
   1. Face width **$b = 8.25m_x$.**
   2. Pitch circle diameter **$d_1= m_x * z_2$**
   3. Pitch line velocity **$v = \left(\frac{\pi d_1 N_1}{60}\right)$**

9. **Recalculation of beam strength (F_s):**

   $Fs= \pi.m_x.b.[\sigma_b].y'$

10. **Calculation of accurate dynamic load (F_d):**

    F_d $=\left(\frac{Ft}{C_v}\right)$

    F_d= Total dynamic load on gear tooth.

    F_t = Transmitted load.

11. **Check for Beam Strength :**
    i. Compare $F_d$and F_s
    ii. If $F_s \geq F_d$ Design is safe and satisfactory.

12. **Calculation of maximum wear load (F_w) :**

    $F_w = d_2 * b * K_w$……. (From data book page no. 8.52)
    F_w= Maximum wear load.
    K_w=($\frac{Pressure\ angle}{10}$)…....(From data book page no. 8.54)
    b = Face width.

13. **Check for wear:**
    i. Compare $F_d$and F_w

ii. If $F_w \geq F_d$ Design is safe and satisfactory.

14. Calculate the Efficiency:

i. **$\eta = 0.96 \left(\frac{\tan\gamma}{\tan(\gamma+\rho)}\right)$……….(From data book page no. 8.49)**

ii. **$\mu = \tan\rho$**

15. Calculation of power loss and area required:

i. Heat generated = Heat dissipated…….**(From data book page no. 8.52)**

$(1-\eta) * P = K_t*(\Delta t) *A$

(Δt) = Oil temp – Air temp

K_t= Heat transfer coeff.

16. Calculation Basic Dimensions:

Basic dimensions of worm and worm wheel ... **(From data book page no. 8.43)**

DESIGN PROCEDURE FOR WORM GEAR WITH GEAR LIFE

INDIAN STANDARD

1. **Selection of Material**: Select a suitable material for worm and worm wheel
2. **Calculation of Initial design Torque: $[M_t]$**

$[M_t] = M_t \times K \times K_d$ ……….[**PSG data book page no:8.13**]

Assume $K \times K_d = 1$

M_t =Transmitted torque = $\frac{60 * p}{2\pi N}$

3. **Selection of Z_1 and Z_2:**

Z_1 value is selected for PSG **data book page no: 8.46.**

$Z_2 = i \times Z_1$.

4. **Selection of $[\sigma_b]$, $[\sigma_c]$:**

$[\sigma_b]$ from **PSG data book page no: 8.45**

$[\sigma_c]$ from **PSG data book page no: 8.45** for V_s

5. **Calculation of Centre Distance(a):**

$$\mathbf{a} = \left[\left(\frac{Z_2}{q}\right) + 1\right] \sqrt[3]{\left[\frac{540}{\left(\frac{Z_2}{q}\right)[\sigma_c]}\right]^2 \frac{[M_t]}{10}}$$ …….[**PSG data book page no:8.44**]

Assume q= 11

6. **Calculation of axial module:**

$$m_a = \frac{2a}{q+Z_2}$$

Standard axial module in **PSG data book pg no:** 8.2

7. Revision of centre distance:

a = 0.5 $m_x\,(q + Z_2)$……….**[PSG data book page no: 8.43]**

8. Calculation of b, d, v, γ, v_3:

Pitch diameter: $d_1 = q \times m_x$

$d_2 = Z_2 \times m_x$

Pitch Velocity: $v_1 = \frac{\pi d_1 N_1}{60}$

$v_2 = \frac{\pi d_2 N_2}{60}$

Lead angle: $\gamma = tan^{-1}\left(\frac{Z_1}{q}\right)$

Assume q = 11

Sliding velocity $v_s = \frac{v_1}{cos\gamma}$

9. ***Recal*****culation of design contact stress** $[\sigma_c]$

for sliding velocity v_s, Find$[\sigma_c]$ from **PSG data book page no:8.45**

10. Revised $[M_t]$**:**

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

K= Load concentration factor ……**[PSG data book page no: 8.15]**

K_d= Dynamic Load Factor………..**[PSG data book page no:8.16]**

11. Check for Bending:

$\sigma_b = \frac{1.9[Mt]}{m_x^3 \times q \times Z_2 \times y_v}$……….**[PSG data book page no:8.44]**

Yv from **PSG data book page no: 8.52** for Z_v

$Z_v = \frac{Z}{cos^3\gamma}$

$\sigma_b < [\sigma_b]$ – Design is safe and satisfactory.

12. Check for wear strength:

$\sigma_c = \frac{540}{\left(\frac{Z_2}{q}\right)}\sqrt{\left[\frac{\left(\frac{Z_2}{q}\right)+1}{a}\right]^3 \times \frac{[M_t]}{10}}$……….**[PSG data book page no:8.44]**

$\sigma_c < [\sigma_c]$ - Design is safe and satisfactory.

13. ***Check for e*****fficiency:**

$$\eta_{actual} = 0.95 \times \frac{tan\gamma}{\tan(\gamma + \rho)}$$

$$\rho = tan^{-1}\ (\mu)$$

14. Calculation of cooling area:

i. Heat generated = Heat dissipated…….**(PSG data book page no. 8.52)**

$(1-\eta) * P = K_t*(\Delta t) *A$

(Δt) = Oil temp – Air temp

K_t= Heat transfer coeff.

15. Basic Dimensions:

For Basic Dimensions of worm and worm wheel..[**PSG data book page no:8.43]**