UNIT - III

BEVEL, WARM AND CROSS HELICAL GEARS

BEVEL GEARS

Introduction:

Bevel Gears are used to transmit Power between two intersecting Shafts. Bevel gears are Commonly used in automobile Differential Unit. The Structure of bevel gear is Similar to an Uniformly Serrated frustum of a Cone.

Bevel Gears are mounted on intersecting Shaft angle is most although 90° Shaft angle is most Common. Bevel gears are not interchangeable, Because they are designed and manufactured in Parts.

Classification of Bevel Gears:

1) Based on the Shape of teeth Such as Straight teeth bevel gears and Curved teeth bevel gears Specified as Spiral bevel Gear.

Based on the angle between the Shaft as Based on the angle is known as Shaft angle ascos - This angle is known as Shaft angle denoted as ϕ. The bevel gear drive having Shaft angle, less than 90° is known as external gear drive. Where as it the Shaft angle is more than 90°, Then It may be Called as Internal gear.

Bevel Gear Terminology

1, Pitch Cones.
2, Cone distance
3, Pitch angle
4, Shaft angle
5, Back Cone
6, Blank Cone angle
7, Root angle
8, Addendum angle
9, Deddendum angle
10, Addendum
11, Deddendum
12, Tooth height
13, face width.

Design Procedure :

1, Select the materials
2, Calculate the required minimum Centre distance, Determine M_c, E, σ_c,
3, Decide the minimum average module
4, Find out the transverse module.
5, Standard the transverse module m using the table 7.19 (PSG 8.2)
6, Correct the number of teeth.
7, Find Out the referance diameters.
8, Evaluate the induced Surface Compressive Stress σ_c.
9, If σ_c and σ_b are not with in allowable Limits.

10, Calculate the other essential Parameters

1, Tip dia for Pinion as gear.

2, Addendum

3, Deddendum

4, Tip angle for Pitch gear.

5, Root angle for Pinion gear.

11, Draw a neat Sketch of Bevel gear Drive.

Problem:

1, Design a bevel gear drive to transmit 7kw at 1600 rpm for the following data,

Gear ratio = 3

Material for Pinion and gear = C_{45} Steel

Life = 10,000 hours

(A.U may 2010)

Soln:

Since Pinion and gear are made of Same material, Pinion is weaker than gear and its teeth are subjected to more number of Cycle.

Let the tooth Profile is 20° Pressure angle, minimum Cone distance based on Surface, Compressive Strength its given by

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

Design Torque $M_t = M_t \cdot K_d = \frac{60 \times P}{2\pi n} k \cdot k_d$

$$= \frac{60 \times 7 \times 10^3}{2\pi \times 1600} \times 1.5 = 62.7 \, Nm$$

Assume $k \cdot k_d = 1.5$

$= 62.7 \times 10^3$ N-mm

Equivalent Young modulus $= 2.15 \times 10^3$ N/mm²

PSG 8-14

$\sigma_c = 500$ N/mm²

$W = \frac{B}{b} = 3$ for $i = 3$

$$R \geq 3 \times \sqrt{3^2+1} \; \sqrt[3]{\left(\frac{0.72}{3-0.5\times 500}\right)^2 \frac{2.15\times 10^5 \times 62.7\times 10}{3}}$$

> 108 mm

Average module based on beam strength

$$m_{av} \geq 1.26 \times \sqrt[3]{\frac{M_t}{y_v \sigma_b \psi_m Z}}$$

$\sigma_b = 140$ N/mm²

Assume $\psi_m = \frac{b}{m_{av}} = 10$ initials

Assume $Z_1 = 20$ initials

y_v = form factor based on equivalent no of teeth on the vertical cylinder

$Z_v = \frac{Z_1}{\cos \delta_2}$ (for pinion)

$\delta_2 = i = 3$

$\delta_2 = \tan^{-1} i = \tan^{-1} 3 = 71.56$

$\delta_1 = 90 - 71.56 = 18.43$

Since $(\delta_1 + \delta_2 = 90^\circ)$

$$Z_V = \frac{Z_1}{\cos \delta_1} = \frac{20}{\cos 18.43} = \frac{20}{0.96} = 21$$

$y_v = 0.396$

$m_v = 1.26 \sqrt[3]{\dfrac{62.7 \times 10^3}{0.396 \times 140 \times 10 \times 20}}$

$= 2.3\,mm$

Transverse module $m_t = m_{av} \times \dfrac{\psi_y}{\psi_y - 0.5}$

$= 2.3 \times \dfrac{3}{3-0.5} = 2.3 \times \dfrac{3}{2.5} = 2.76\,mm$

Nearest higher standard module $= 3mm$

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

$Z_1 = \dfrac{R}{0.5 m_t \sqrt{i+1}} = \dfrac{108}{0.5 \times 3 \sqrt{3^2+1}} = 22.8$

$Z_1 = 24$ and $Z_2 = i Z_1 = 3 \times 24 = 72$

Now find Cone distance

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

$= 0.5 \times 3 \times 24 \sqrt{3^2+1}$

$= 114\,mm$

Face width $b = \dfrac{R}{\psi_y} = \dfrac{114}{3} = 38\,mm$

take $b = 40\,mm$

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

$= \dfrac{0.72}{114 - 0.5 \times 40} \sqrt{\dfrac{(3^2+1)^{3/2} \times 2.15 \times 10^5 \times 62.7 \times 10^3}{3 \times 40}}$

$= 457\,N/mm^2$

$\sigma_c = 457\,N/mm^2$

$\sigma_c = 457 < \sigma_c = 500\,N/mm^2$

Gear design is Safe

$$\sigma_b = R\sqrt{i^2+1}\ m_t \times \frac{1}{\cos\alpha} \leq \sigma_b$$

$$= \frac{114\sqrt{3^2+1} \times 62.7 \times 10^3}{(114-0.5\times 40)^2 \times 40 \times 3 \times 0.396} \times \frac{1}{\cos 20}$$

$$= 57.2\ N/mm^2 \leq \sigma_b = 140\ N/mm^2$$

Our design is Safe

Pitch Circular diameter

for Pinion $d_1 = m_1 z_1 = 3\times 24 = 72mm$

$d_2 = m_1 z_1 = 3\times 72 = 216mm$

Tip Circle diameter

$$da_1 = m_1 (z_1 + 2\cos\delta_1$$

$$= 3(24 + 2\cos 18.40)$$

$$= 77.7\ mm$$

$$= 78mm$$

$$da_2 = m_1 (z_2 + 2\cos\delta_2)$$

$$= 3(72 + 2\cos 71.56) = 218mm$$

Addendum angle θa

$$\theta a_1 = \theta a_2 = \tan^{-1}\left(\frac{m_1\ \theta_0}{R}\right)$$

$$= \tan^{-1}\left(\frac{0.3\times 1 + 0.2}{11.4}\right) = 1.8$$

Tip angle δ_2

Pinion $\delta a_1 = \delta_1 + \theta a_1 = 18.43 + 1.5$

$= 19.93$

Gear $\delta a_2 = \delta_2 + \theta a_2 = 71.56 + 1.5 = 73.06$

Root angle :

For Pinion $\delta_{f1} = \delta_1 - \theta_{f1} = 18.43 + 08 = 16.63$

for gear $\delta_2 = \delta_2 - \theta_2$

$= 71.56 - 18 = 69.76°$

Addendum $= h_a = m_a = 32mm$

Deddendum $= h_1 = 123.68 \times m_1 = 1.1228 \times 3$

$= 3.4mm$

Tooth hight $h = h_2 + h_1 = 3 + 3.4$

$= 6.44$

Specification :

SL No	Description	Pinion	Gear
1,	Material	C45 Steel	C45 Steel
2,	Cone distance	114mm	114 mm
3,	Module	3mm	3mm
4,	No. of Teeth	24	72
5,	Face width	40mm	40mm
6,	Semi cube angle	18.43°	71.56°
7,	Addendum	3mm	3mm
8,	Deddendum	34mm	34mm
9,	Pitch Circle dia	72 mm	216mm
10,	Tip circle dia	78mm	218mm
11,	Tip angle	19.93°	73.06
12,	Root angle	16.63°	68.76°
13,	Addendum angle	1.5°	1.5°
14,	Deddendum angle	1.8°	1.8°

Worm Gear

Worm Gear drive, the driving member is Similar to an Archimedian Screw and the driver member is Similar to a helical gear with Curved teeth and also the pitch Surface of this helical gear is Slightly Con Cormed.

Worm drive, the drawing member is known as worm is stead of Calling as Pinion and the driver member is known as Worm gear or worm wheel and the Power is transmitted from worm to worm wheel by Sliding Contact.

Materials for worm Gear:

Usually Carbon or alloy Steels may be used to make worm and gray Cast worm are brace may be Selected for making Worm wheels.

Applications:

1. Steering mechanism of automobile Vehicle.
2. Processes; rolling mills, machines
3. Milling heads, rotary tables, tilting furnace
4. Lifts or elevators and escalator drive.

Worm Gear Terminology:

The technical terms related to Worm Gear design are as follows

1, Axial pitch (or) Pitch of wear

2, Lead
3, Lead angle.
4, Helix angle
5, Tooth Pressure angle
6, Normal pitch
7, Centre distance
8, Speed ratio (or) Gear ratio.
9, Axial module.
10, Diameter factor.
11, Relation b/w Lead angle or diameter factor
12, Relation b/w sliding Velocity and dia factor.

Design Procedure:

1, Select the material
2, Estimate the minimum Centre distance
3, Determine the minimum axial module.
4, Standard's this module.
5, Find out the dia factor
6, Estimate the actual Sliding Velocity
7, Calculate the induced Surface Compressive Stress bending Stress.
8, Determine the length of wormas.
9, Decide the number of teeth on worm

$$\lambda = \frac{L_1}{\pi m_x}$$

10, Determine the face width of worm wheel
11, Compute the Parameters of worm and worm wheel such as their referance Pitch, diameters, tip dia, root, dia etc.
12, calculate the efficiency of the diameter
13, Draw the neat Sketch of worm gear drive

Efficiency of Worm Gearing

The Efficiency of the Worm gear drive is given by

$$\eta = \frac{\text{Power output}}{\text{Power input}}$$

$$\eta = \frac{F_{t2} \times V_2}{F_{t1} \times V_1}$$

$$= \frac{\cos\alpha - \mu \tan\gamma}{\cos\alpha + \mu \cos\gamma}$$

V_1 and V_2 = Pitch Low Velocity.

$$V_s = \frac{V_1}{\cos\gamma} = \frac{V_2}{\sin\gamma}$$

Due to friction in Sliding
due to the Clearing the Splashing of Lubricatio

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

ρ = angle of friction

μ = Co-efficient of friction

$$\eta = (0.95 - .96)\frac{\tan\gamma}{\tan(\gamma+\rho)}$$

Thermal Rolling of Worm Gearing

Heat generated (H_g) = Heat dissipated to the atmosphere (H_d)

$H_g = (1-\eta) \times$ input power

$H_d = k_t \times A(t_o - t_a)$

k_t = Heat transfer Co-efficient

A = Effective Surface area

t_o = temp of Lubrication

t_a = Temp of atmosphere

$(1-\eta) \times \text{input} \; P = K_t \times A \times (t_o - t_a)$

$$P = \frac{K_t \times A (t_o - t_a)}{(1-\eta)}$$

Solved Problem by Harty and Belag ar Relation:

Problem:

The Input to worm gear shaft is 18kw and 600rpm. Speed ratio is 20, The worm is to be of hardened steel and the wheel is made of chilled phosphare bronze. Considering wear and straight design worm and worm wheel.

Solution:

$P = 18$ kw

$n = 600$ rpm

$c = 20$

Worm = hardened steel.

Wheel = Phosphare Bronze

1) Centre distance:

$$a \geq \left(\frac{z_2}{q} + 1\right) \sqrt[3]{\left(\frac{540}{\frac{z_2}{q}(\sigma_c)}\right)^2 M_t \; C_w}$$

$M_t = 97420 \times \frac{kw}{n}$, PSG 8.44

$C = \frac{Z_2}{Z_1} = 20$, $q = 11$

$Z_1 = 3$

$\eta = 0.86$

$Z_2 = 3 \times 20 = 60$

$M_t = 97420 \times \frac{18}{600} \times 20 \times 0.86$

$M_t = 50270$ kgf.cm

$M_t = m_t \cdot k \cdot k_d$

$= 50270 \times 1 \times 1 = 50270$ kgf.cm.

$\sigma_c = 1590 \, kgf/cm^2$, $V_3 = 3 m/s$

$\sigma_b = 550 \, kgf/cm^2$

$$a \geq \left(\frac{60}{11}\right)+1 \times \sqrt[3]{\left(\frac{540}{\left(\frac{60}{11}\right)1590}\right)^2 \times 50270} \geq 37.4 \, cm$$

2) Determination of axial module

$$m_x \geq 1.24 \sqrt[3]{\frac{M_t}{z_2 \, q \, y_v \, \sigma_b}}$$

$y_v = z_{v_2}$

$$z_{v_2} = \frac{z_2}{\cos^3 \gamma}$$

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right) = \tan^{-1}\frac{3}{11} = 15° 15' 18''$$

$= 15.255$

$$z_{v_2} = \frac{60}{\cos^3 15.255} = 66.8 = 67$$

$y_v = 0.493$

$\sigma_b = 550 \, kgf/cm^2$

$$m_x \geq 1.24 \sqrt{\frac{50270}{60 \times 11 \times 0.493 \times 550}} \geq$$

$0.82 \, cm \geq 8.2 \, mm$

$m_x = 10 \, mm$

$a = 0.5 \, m_x (q + z_2 + 2x)$

$= 0.5 \times 10 (11 \times 60)$

$= 355 \, mm$

$m_x = 12 \, mm$

$a = 0.5 \times 12 \, (11 \times 60) = 426 \, mm = 42.6 \, cm$

$$V_s = \frac{\pi d_1 n_1}{60 \times 1000 \times \cos \gamma}$$

$d_1 = q \, m_x = 11 \times 12 = 132 \, mm$

$\gamma = 15.255$

$$V_s = \frac{\pi \times 132 \times 600}{60 \times 1000 \times \cos 15} = 4.29 \text{ m/s}$$

$$V_s = \frac{m_x u}{19100} \sqrt{z_1^2 + q^2}$$

$$= \frac{12 \times 600}{19100} \sqrt{3^2 + 11^2}$$

$$= 4.29 \text{ m/s}$$

Since adequate data are not available for surface strength for slider wheel

$$\sigma_c = 1490 \text{ kgf/cm}^2$$

3) Induced Stresses

$$\sigma = \frac{540}{\frac{z_2}{q}} \sqrt{\left(\frac{\frac{z_2}{q} + 1}{a}\right)^3 M_t} \text{ kgf/cm}^2$$

$$= \frac{540}{\frac{60}{11}} \sqrt{\left(\frac{\frac{60}{11} + 1}{42.6}\right)^3 50270}$$

$$= 1300 \text{ kgf/cm}^2 \leq \sigma_c = 1490$$

$$\sigma_b = \frac{1.9 M_t}{m^3 \times q z_2 y_v} = \frac{1.9 \times 50270}{1.23 \times 11 \times 60 \times 493}$$

$$= 170 \text{ kgf/cm}^2 < \sigma_b = 550 \text{ kgf/cm}^2$$

Gear design is safe

$$L \geq (12.5 + 0.09 z_2) m_x$$

$$L \geq (12.5 + 0.09 \times 60) 12 \geq 214.8 \text{ mm} \geq 215 \text{ mm}$$

The increased length L_1 for grand worm is

$$L_1 = L + 35 \text{ mm} = 215 + 35 = 250 \text{ mm}$$

$$\lambda = \frac{L_1}{\pi m_x} = \frac{250}{\pi \times 12} = 6.63$$

Length of worm $= 7 \times \pi \times m_x = 7 \times \pi \times 12 = 264$

Breadth of worm wheel (face width) $= 0.7 d_1$

$= 0.75 \times 132 = 99 \text{ mm} = 100 \text{ mm}$

4, Parameters of Worm:

$d_1 = q m_x = 132\,mm$

$d_{a_1} = d_1 + 2 f_o m_x$

$= 132 + (2 \times 1 \times 12) = 156\,mm$

$d_{f_1} = d_1 - 2 f_o m_x - 2c$

$= 132 - 2 \times 1 \times 12 - 2 \times 0.2 \times 12$

$= 132 - 2 \times 12\,(1 + 0.2) = 103\,mm$

$d_1 = m_x\,(q + 2x)$

$= 12\,(11 + 2 \times 0)$ Assuming $x = 0$

$= 132\,mm$

5, Parameters of wheel:

Referance dia $d_2 = z_2 m_x$

$= 60 \times 12 = 720\,mm$

$d_{a_2} = (z_2 + 2 f_o + 2x) m_x$

$= (60 + 2)\,12 = 744\,mm$

$d_{f_2} = (z_2 - 2 f_o)\,m_x - 2c$

$= (60 - 2)\,12 - (2 \times 0.2 \times 12) = 691\,mm$

$d_g = d_2 = 720\,mm$

6, Efficiency of worm gear

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

$\tan \rho = \mu$ = friction Coefficient

$= 0.03$ for Sliding velocity $V_s = 4.29\,m/sec$

$\rho = \tan^{-1} 0.03 = 1.72^\circ$

$\eta = \tan^{-1} 0.03 = 1.72^\circ$

$$\eta = \frac{\tan(15.255)}{\tan(15.255 + 1.72)} = 0.893$$

$= 89.3\%$

Specification:

SL NO	Description	Worm	wheel
1,	Material	Steel	Phosphor Bronze
2,	No. of. teeth	0.6	60
3,	Module	12 mm	12mm
4,	Refrence Dia	132mm	720mm
5,	Tip dia	156mm	744mm
6,	Root dia	103mm	691mm
7,	Length of wheel	264mm	–
8,	Face width of wheel	–	100mm
9,	Centre distance	426	
10,	efficiency of drives	89.3.	

Crossed - helical of Spiral or Screwer Skew Gears:

For Connecting non parallel and non intersecting shafts. Crossed helical gears are used:

Crossed helical Gears transmit relatively Small amounts of Power because of Point of Contact b/w teeth.

Kinematics of Spiral Gears:

Shaft angle $Q = B_1 + B_2$, when both gears of Same hand.

— x — x — x —