Helical Gears:-

Types of Helical Gears:-

1) Parallel helical gears:-

i) They operate on two parallel shafts.

ii) The Magnitude of the helix angle is the same for the pinion and the gear.

iii) They have opposite hand of the helix.

2) crossed-helical (or spiral) gears:-

i) The operate on two non-parallel shafts.

ii) They have the same (or) opposite hand of the helix.

VIRTUAL OR FORMATIVE NUMBER OF TEETH:-

In the design of helical gears, an imaginary (virtual) spur gear is considered in normal plane. This spur gear is called a virtual (or) 'formative' spur gear. The number of teeth on the virtual spur gear in the normal plane is known as virtual (or) formative (or) equivalent no. of teeth.

$$Z_{eq} = \frac{Z}{\cos^3 \beta}$$

Z → Actual no. of teeth on a helical gear.

β ⇒ Helix angle.

Face width OF Helical gears :- (b) :-

For smooth and continuous operation of the helical gears, It is recommended that the face width is greater than axial pitch.

$$b \geq P_a \text{ (or) } b \geq \frac{\pi . m_n}{\sin \beta .}$$

$b = 10 \; m_n \approx 3 \; P_n$ is generally used in the initial stages of the design.

Tooth Properties of Helical gears:-

- ✓ Normal Pressure angle $(\alpha_n) = 15°$ to $25°$
- ✓ Helix angle (β) = 8° to 25° for helical
- ✓ => 25° to 40° for herringbone.

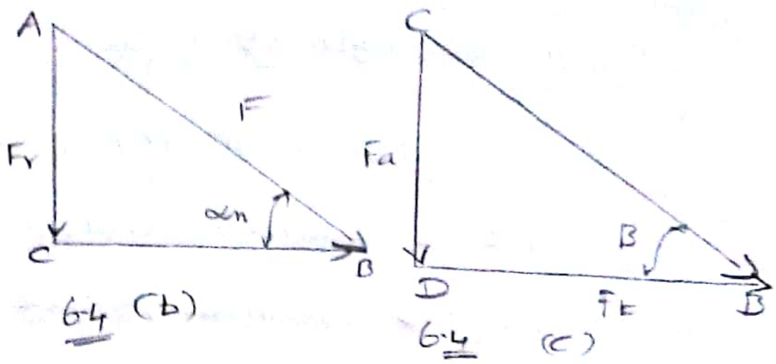
- ✓ Addendum, Maximum => $0.8 \; m_n$
- ✓ Dedendum, Minimum => m_n
- ✓ Tooth depth => $2.25 \; m_n$
- ✓ Minimum Clearance => $0.2 \; m_n$
- ✓ Thickness of tooth => $1.5708 \; m_n$.

Basic Dimensions of helical and Herringbone Gears:-

Basic Dimensions of helical gears
(from data book, P.g No: 8.22).

Force Analysis of helical gears:-

The three dimensional view of the forces acting against a helical gear tooth is shown in fig. The point of application of the forces is in the pitch plane and in the centre of the gear face. The Resultant force (F) whereby one helical gear transmits power to another gear is resolved into 3 perpendicular components.,



From triangles ABC (refer fig 6.4 (b))

$$F_r = F \cdot \sin\alpha_n$$

$$\vec{BC} = F \cdot \cos\alpha_n .$$

From triangle CBD (refer fig 6.4 (c))

$$F_a = \vec{BC} \times \sin\beta = F \cdot \cos\alpha_n \cdot \sin\beta$$

$$F_t = \vec{BC} \times \cos\beta = F \times \cos\alpha_n \cdot \cos\beta .$$

Dividing equation (iii) by (iv) we get.,

$$\frac{F_a}{F_t} = \frac{F \cdot \cos\alpha_n \cdot \sin\beta}{F \cdot \cos\alpha_n \cdot \cos\beta} = \tan\beta .$$

Axial force $F_a = F_t \times \tan\beta .$

$$F = F_t / \cos\alpha_n \cdot \cos\beta .$$

$$F_r = \frac{F_t}{\cos\alpha_n \cdot \cos\beta} \times \sin\alpha_n$$

$$F_r = F_t \left[\frac{\tan\alpha_n}{\cos\beta} \right]$$

$$F_t \times \frac{d}{2} = M_t .$$

Tangential force $F_t = \frac{2 \cdot M_t}{d}$

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

Design of Helical gears:-

1) Beam strength of Helical gears [Lewis equation]

$$F_s = \pi \cdot m_n \cdot b\ [\sigma_b] \cdot y'$$

$F_s \rightarrow$ Beam strength of helical gears.

$m_n \rightarrow$ Normal Module

$b \rightarrow$ Face width $= 2p_n$ to $4p_n$ (or) $10m_n$.

2) Buckingham's equation for dynamic load:-

$$\text{Dynamic Load } F_d = F_t + \frac{21V\,(Cb \cdot \cos^2\beta + F_t) \cdot \cos\beta}{21V + \sqrt{Cb \cdot \cos^2\beta + F_t}}$$

$F_t \rightarrow$ Tangential load neglecting service factor $= P/V$

$V =$ Pitch line velocity

$b =$ Face width

$C =$ Deformation (or) dynamic factor in N/mm.

$\beta =$ Helix angle.

Design Procedure:-

1) Selection of material:-

If not given, select a suitable pinion and gear materials.

2) Calculation of z_1 and z_2:-

✓ If not given assume no. of teeth on pinion $z_1 \geq 17, 18$.

✓ Then no. of teeth on gear $z_2 = i \times z_1$

Where $i \rightarrow$ gear ratio.

Calculation of tangential load on tooth (F_t):-

$$F_t = P/V \times k_o$$

$P \Rightarrow$ Power transmitted in watts.

$V \Rightarrow$ pitch line velocity $= \frac{\pi d N}{60}$ in m/s.

$k_o \Rightarrow$ Service / shock factor from Table 5.6.

4) Calculation of Initial dynamic load (F_d):-

Calculate the preliminary value of dynamic load F_d using the relation..,

$$F_d = \frac{F_t}{C_v}$$

C_v = velocity factor.

5) Calculation of beam strength:- (F_s):-

$$F_s = \pi \cdot m_n \cdot b \, [\sigma_b] \, y'$$

m_n = Normal Module in mm.

b = Face width in mm, initially assume $b = 10 m_n$.

$[\sigma_b]$ = Design bending stress.

$y' \Rightarrow$ Form factor based on virtual number of teeth.

6) Calculation of normal Module (m_n):-

$F_s \geq F_d$. F_s and F_d find the normal module.

Then select the nearest higher standard module value from table 5.8.

7) Calculation of b, d, and v :-

✓ Find Face width (b) $b = 10 m_n$.

✓ Find pitch circle diameter $d_1 = \frac{m_n}{\cos\beta} \times z_1$

✓ Find Pitch line Velocity = $v \Rightarrow \frac{\pi d_1 N_1}{60}$.

8) Recalculation of the beam strength (Fs) :-

Recalculate the beam strength of the gear tooth using the relation,

$$F_s = \pi . m_n . b [\sigma_b] y'$$

9) calculation of accurate dynamic load (Fd) :-

$$F_d = F_t + F_I$$

$$\Rightarrow F_t + \frac{21v (C_b . \cos^2\beta + F_t).\cos\beta}{21v + \sqrt{C_b \cos^2\beta + F_t}}$$

c = Deformation factor

$$F_t = \frac{P}{V}$$

10) check for beam strength :-

✓ compare Fd and Fs.

✓ If $F_d \le F_s$, the gear tooth has adequate beam strength and it will not fail by breakage.

✓ If $F_d > F_s$, then change face width, module or both. usually to reduce the dynamic load (Fd), the gear should be carefully cut

Even for Precision gears, if $F_d > F_s$ then increase the face width, till $F_d < F_s$.

11) Calculation of maximum wear load (Fw) :-

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$$F_w = \frac{d_1 \times b \times Q \times K_w}{\cos^2 \beta}.$$

Q = Ratio factor $= \frac{2i}{i+1} = \frac{2z_2}{z_1+z_2}$ and

K_w = Load stress factor, from table 5.9.

12) Check for wear:-

✓ compare the calculated values of dynamic load (F_d) & wear strength (F_w).

✓ If $F_d < F_w$, the gear tooth has adequate wear capacity and will not wear out. Then the design is safe and satisfactory.

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

Finally, calculate the basic dimensions of pinion and gear by consulting. Table 6.1