UNIT -II
SPUR GEARS AND PARALLEL AXIS HELICAL GEARS

Spur Gears

(i). Classification of based on the relative position of their shaft axes:

(i) Parallel shafts

Examples: Spur gears, helical gears, rack and pinion, herringbone gears and internal gears.

(ii) Intersecting shafts

Examples: Bevel gears and spiral gears.

(iii) Non-parallel, non-intersecting shafts

Examples: Worm, hypoid and spiral gears.

2. Classification based on the relative motion of the shafts:

(i) Row gears: In this type, the motion of the shafts relative to each other is fixed.

(ii) Planetary and differential gears

3. Classification based on peripheral speed (v):

(i) Low velocity gears v < 3 m / s

(ii) Medium velocity gears v= 3 m / s

(iii) High velocity gears v > 15 m / s

4. Classification based on the position of teeth on the wheel:
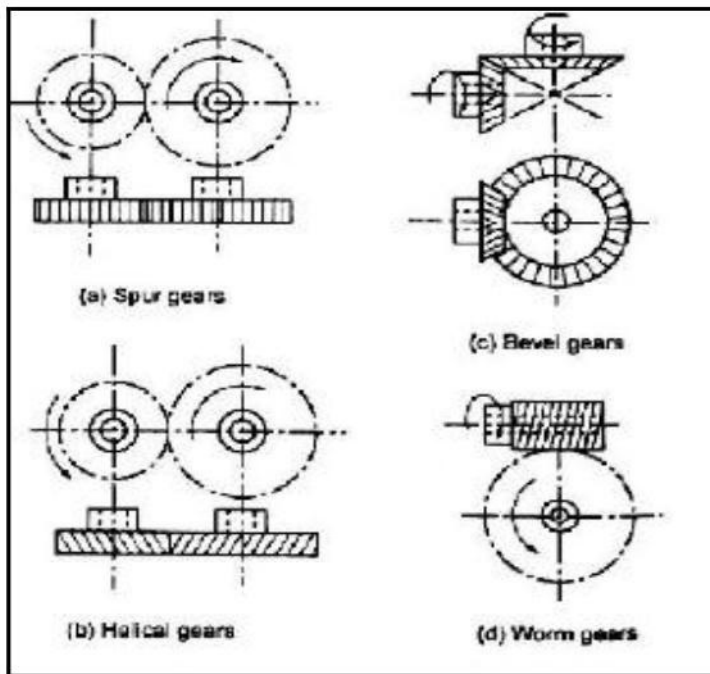
(i) Straight gears (ii) Helical gears

(iii) Herringbone gears

(iv) Curved teeth gears

5. Classification based on the type of gearing:
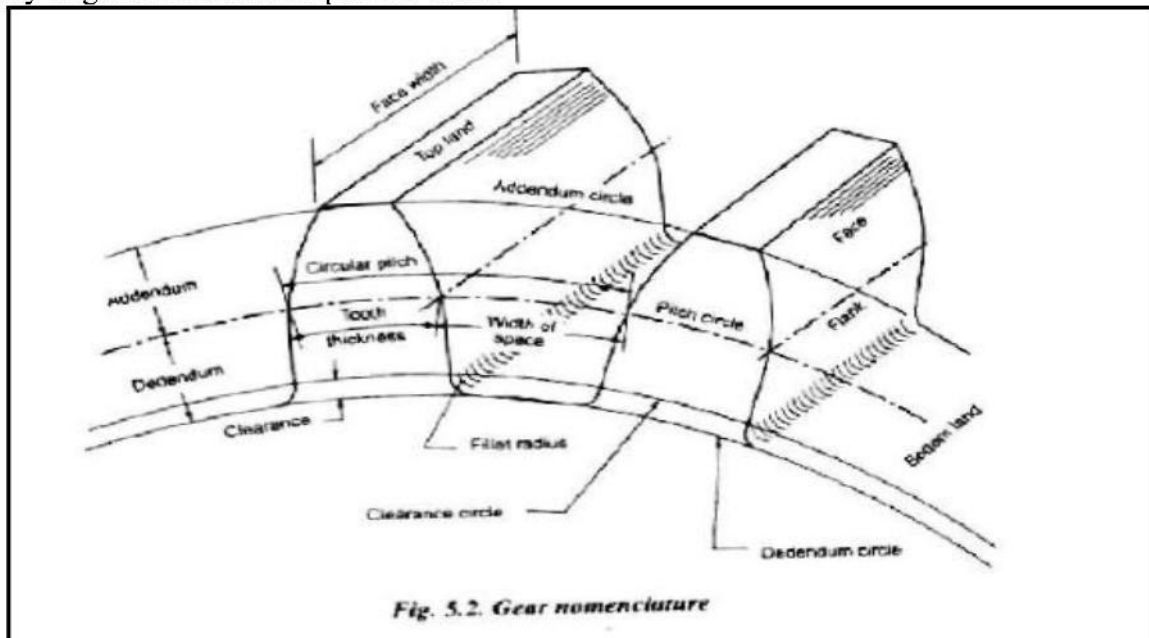
(i) External gearing

(ii) Internal gearing

(iii) Rack and pinion

(a) Spur gears
(c) Bevel gears
(b) Helical gears
(d) Worm gears

SPURGEARS

Terminology Used in Gears (Gear Nomenclature)

The terminology of gear teeth is illustrated in Fig. The various terms used in the study of gears have been explained below

Fig. 5.2. Gear nomenclature

(a) Circular pitch (Pc):

It is the distance measured along the circumference of the pitch circle from a point on one tooth to the corresponding point on the adjacent tooth.

$$\textbf{Circular pitch, } \mathbf{P_c} = \frac{\pi D}{z}$$

Where   D = Diameter of pitch circle, and
                  z = Number of teeth on the wheel

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(b) Diametral pitch (Pd):
It is the ratio of number of teeth to the pitch circle diameter.

$$\textbf{Diametral pitch, } \mathbf{P_d} = \frac{z}{D} = \frac{\pi}{P_C}$$

(c) Module pitch (m):
It is the ratio of the pitch circle diameter to the number of teeth.

$$\textbf{Module, m} = \frac{D}{z}$$

Velocity ratio: It is the ratio of speed of driving gear to the speed of the driven gear.

$$i = \frac{N_A}{N_B} = \frac{Z_B}{Z_A}$$

Where N_A and N_B= Speeds of driver and driven respectively, and,
Z_A and Z_B = Number of teeth on driver and driven respectively,

Contact ratio: The ratio of the length of arc of contact to the circular pitch is known as Contact ratio. The value gives the number of pairs of teeth in contact.

The properties of the various materials used for the gears are given in Table 5.3.

Gear materials and their properties (from data book, page no. 1.40)

FORCE ANALYSIS OF SPUR GEARS
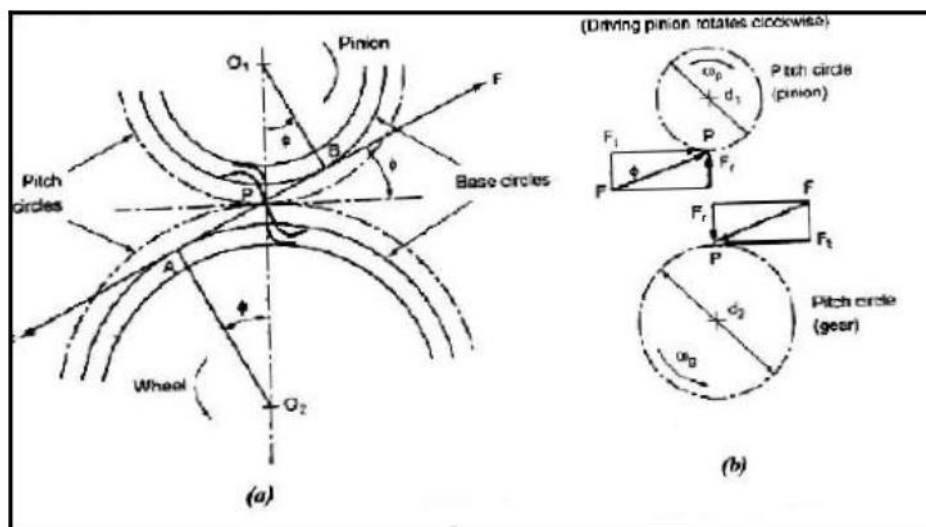
P=Power transmitted by gears in watts,
M_t = Torque transmitted by gears in N-m,
N_1 & N_2 –Speeds of pinion and gear respectively in r.p.m.,
d_1 & d_2 – Pitch circle diameters of pinion and wheel respectively in m, and
$\varnothing$ = Pressure angle.

The torque transmitted by the gears is given by

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

The tangential component F_t acts at the pitch circle radius.

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

Or

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

Radial component, $F_r = F_t \cdot \tan\phi$

Therefore   resultant force, $F = \sqrt{F_t^2 + F_r^2}$

Or

$$F = \frac{F_t}{\cos\phi}$$

Pitch line velocity (v) is given by

$$v = \frac{\pi dN}{60} \; m/s$$

Then the transmitted power is calculated as

$$P = F_t \times v$$

DESIGN PROCEDURE FOR SPUR 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. **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).

4. **Calculation of initial dynamic load (Fd):**
   1. $F_d = \left(\frac{Ft}{C_v} \right)$ .... **(From data book page no. 8.50)**

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2. $C_v = \dfrac{6}{6+v}$ (Assume V=12)… **(From data book page no. 8.51)**

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

$$Fs = \pi.m.b.[\sigma_b].y \ldots\ldots \text{ (From data book page no. 8.50)}$$

m = Module in mm.
Fs = Strength of gear tooth.
$[\sigma_b]$ = Allowable static stress.
b = Face width = **10m**
y = Form factor = $(0.154 - \dfrac{0.912}{z_1})$ for **20° involute.**

6. **Calculation of module (m):**
   1. $Fs \geq F_d$

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

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

8. **Recalculation of beam strength (F_s):**
   1. $Fs = \pi.m.b.[\sigma_b].y$

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

$$F_d = F_t + \dfrac{21v(bc + F_t)}{21v + \sqrt{bc + F_t}} \ldots\ldots \text{(From data book page no. 8.51)}$$

F_d = Total dynamic load on gear tooth.

F_t = Transmitted load.

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

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

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

$$F_w = d_1.b.Q.K_w \ldots\ldots \text{ (From data book page no. 8.51)}$$

Q = Ratio factor = $\dfrac{2*i}{i+1}$

F_w = Maximum wear load.

$$K_w = \dfrac{f^2 \sin\phi}{1.4}\left(\dfrac{1}{E_p} + \dfrac{1}{E_g}\right) \ldots \text{ (From data book page no. 8.51)}$$

f = (2.8 * BHN – 70) N/mm^2

d_1 = Pitch circle diameter.
B = Face width.

12. Check for wear:
  i. Compare F_d and F_w
  ii. If $F_w \geq F_d$,Design is safe and satisfactory.

13. Calculation Basic Dimensions of:
  Basic dimensions of spur gear ……... **(From data book page no. 8.22)**

DESIGN PROCEDURE FOR SPUR GEAR WITH GEAR LIFE

INDIAN STANDARD

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

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

3. **Gear Life:**

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

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

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

$$[M_t] = M_t \times K \times K_d \dots [\textbf{ PSG data book page no:8.15}]$$

$$K \times K_d = 1.3$$

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

5. **Calculation of $[E_{eq}]$, $[\sigma_b]$, $[\sigma_c]$:**
  a. $[E_{eq}]$ = Equivalent young's modulus….[**PSG data book page no:8.14**]
  b. $[\sigma_b] = \left(\frac{1.4\,K_{bl}}{n K_\sigma}\right)\sigma_{-1}$ …….[**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**]

  c. $[\sigma_c] = C_B . H_B . K_{cl}$…………. [**PSG data book page no: 8.16**]

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

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

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

6. **Calculation of center distance (a):**

$$a \geq (i+1)^3 \sqrt{\left[\frac{0.74}{[\sigma_c]}\right]^2 \times \frac{E_{eq}[M_t]}{i.\varphi}} \quad ....[\text{ PSG data book page no: 8.13}]$$

Assume φ=.3

7. Selection of number of teeth:

Assume $Z_1 = 17$

$$Z_2 = i \times Z_1$$

8. Calculation of module:

$$m = \frac{2a}{(z1+z2)} \quad ....[\text{PSG data book page no: 8.22}]$$

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

9. Revision of center distance:

$$a = \frac{m(Z1+Z2)}{2} \quad .... [\text{PSG data book page no: 8.22}]$$

10. Calculation of b, $d, v \text{ and } \varphi$:

Face width, $b = \varphi.a$

pitch circle dia, $d_1 = m.Z1$

Pitch line velocity, $v = \frac{\pi \times d_1 \times N_1}{60}$

$$\varphi_p = \frac{b}{d_1}$$

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_d]:

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

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:

$$\sigma_b = \frac{i \pm 1}{a.m.b.Y}[M_t] \quad ....[\text{PSG data book page no:8.13A}]$$

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

$\sigma_b <$ [σ_b] – Design is safe and satisfactory.
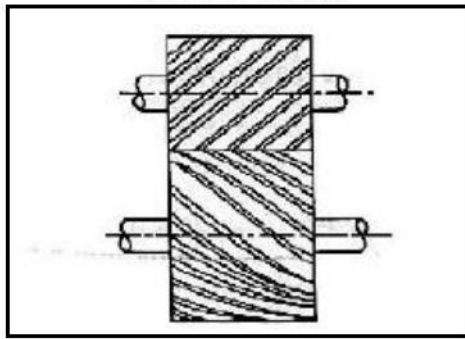
14. Check for wear strength:

$$\sigma_c = 0.74 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{ib} E_{eq}[M_t]} \quad ....[\text{PSG data book page no:8.13}]$$

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

15. Basic Dimensions:

For Basic Dimensions of spur gear ..[**PSG data book page no:8.22**]

Helical Gears

## KINEMATICS AND NOMENCLATURE OF HELICAL GEARS

Let
β = Helix angle,
p_t = Transverse cirular pitch,
p_n = Normal circular pitch,
p_a = Axial pitch,
p_d = Diametral pitch,
α_t and α_n = Transvers and normal pressure angles respectively,
m_t and m_n = Transverse and normal modulus respectively,
z_1 amd z_2 = Number of teeth on pinion and gear respectively,
d_1 and d_2 = pitch circle diameters of pinion and gear respectively,
N_1 and N_2 = Speeds of pinion and gear respectively, and
a = Centre to centre distance between pinion and gear.
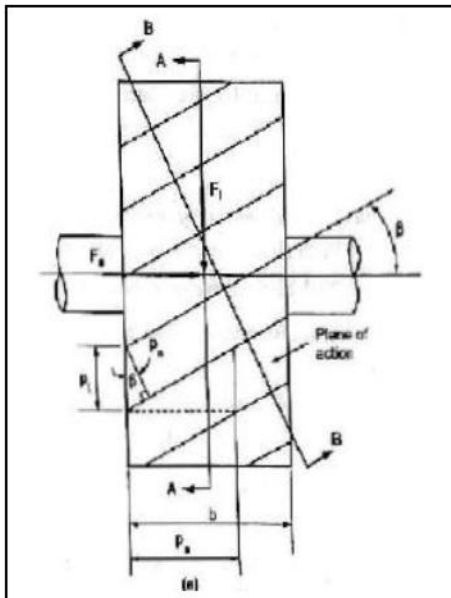
(a) Gear
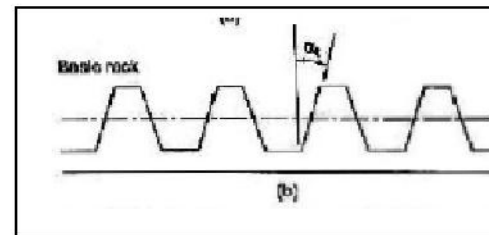
(b) Section AA (transverse plane),
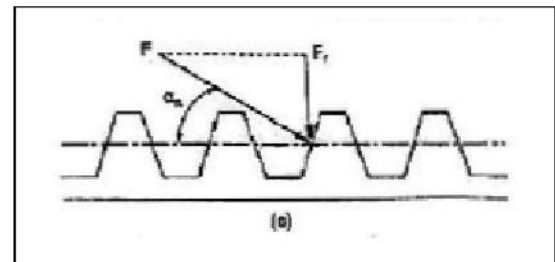
(c) Section BB (normal plane).

The various terms used in the study of helical gears have been explained below.

TOOTH PROPORTIONS FOR HELICAL GEARS

There are no standard proportions for helical gears. The proportions recommended by American Gear Manufacturer's Association (AGMA) are as follows:

➢ Normal Pressure angle **(α_n) = 158 to 258**

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- Helix angle (β) = **88 to 258, for helical** = **258 to 408 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

All the basic dimension of helical and herringbone gears are listed in table 6.1(from data book,page no. 8.22)
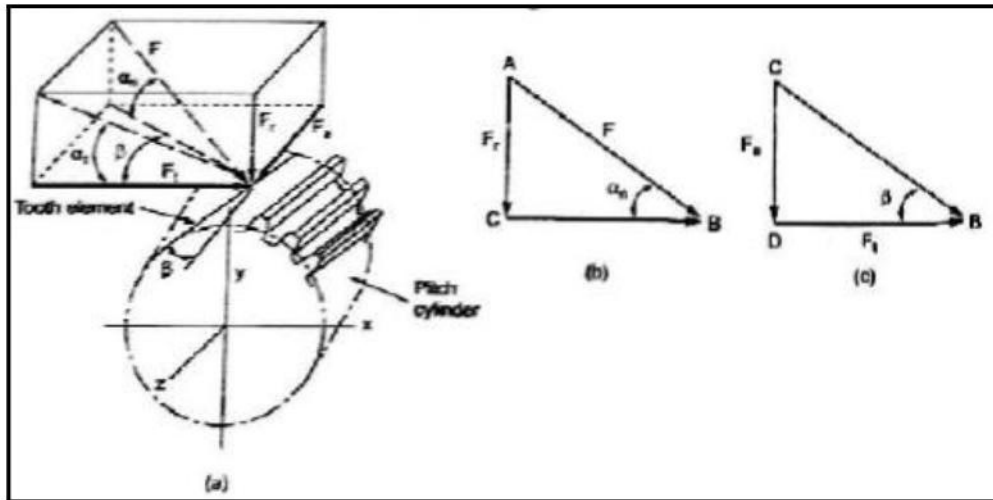
Tooth forces acting on helical gear

DESIGN PROCEDURE FOR HELICAL 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. Calculation of tangential load on tooth (Ft):

3. $\mathbf{F_t} = \left(\dfrac{P*K_0}{V}\right)$

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

4. Calculation of initial dynamic load (Fd):

4. $\mathbf{F_d} = \left(\dfrac{Ft}{C_v}\right)$ .... **(From data book page no. 8.50)**

5. $\mathbf{C_v} = \dfrac{6}{6+V}$ (Assume V=15)... **(From data book page no. 8.51)**

5. Calculation of beam strength (Fs):

$Fs = \pi.m_n.b.[\sigma_b].y'$.... **(From data book page no. 8.51)**

m_n= Normal module in mm.
Fs = Strength of gear tooth.
$[\sigma_b]$ = Allowable static stress.
b = Face width = **$10m_n$.**
y' = Form factor = **$(0.154 - \dfrac{0.912}{z_{eq}})$ for 20° involute.**

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$$Z_{eq} = \frac{z1}{cos^3\beta}$$

6. **Calculation of module (m_n):**
      1. $F_s \geq F_d$

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

7. **Calculation of b,d and v :**
      1. Face width **b = 10m_n.**
      2. Pitch circle diameter **$d_1 = \left(\frac{m_n}{cos\beta} \times Z_1\right)$ ... (From pg no. 8.22)**
      3. Pitch line velocity **v $= \left(\frac{\pi d_1 N_1}{60}\right)$**

8. **Recalculation of beam strength (F_s):**
      1. $F_s = \pi.m.b.[\sigma_b].y'$

9. **Calculation of accurate dynamic load (F_d):**
$$F_d = F_t + \frac{21v\left(bc.cos^2\beta + F_t\right).cos\beta}{21v + \sqrt{bc.cos^2\beta + F_t}} ..\text{(From data book page no. 8.51)}$$
F_d= Total dynamic load on gear tooth.

F_t = Transmitted load.

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

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

11. **Calculation of maximum wear load (F_w) :**
$$F_w = \left(\frac{d1.b.Q.Kw.}{cos^2\beta}\right) .... \text{(From data book page no. 8.51)}$$
$$Q = \text{Ratio factor} = \left(\frac{2*i}{i+1}\right) ..... \text{(From data book page no. 8.51)}$$
F_w= Maximum wear load.
$$K_w = \frac{f^2 sin\varnothing}{1.4}\left(\frac{1}{E_p} + \frac{1}{E_g}\right) .....\text{(From data book page no. 8.51)}$$
f = (2.8 * BHN – 70) N/mm^2

d_1 = Pitch circle diameter.
B = Face width.

12. **Check for wear:**
      i. Compare F_d and F_w
      ii. If $F_w \geq F_d$ Design is safe and satisfactory.

13. **Calculation Basic Dimensions of:**
      Basic dimensions of Helical gear ……... **(From data book page no. 8.22)**