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

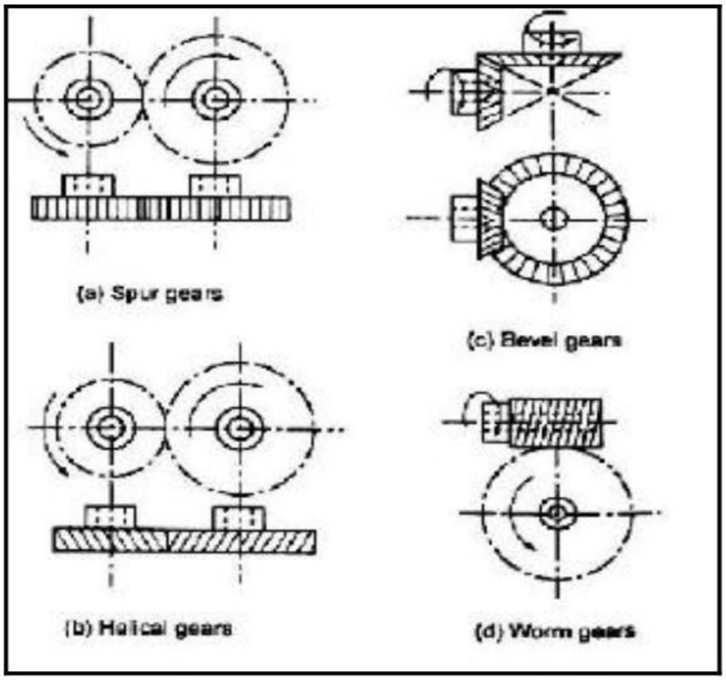
(iv) Curved teeth gears

5. Classification based on the type of gearing:

(i) External gearing

(ii) Internal gearing

(iii) Rack and pinion



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

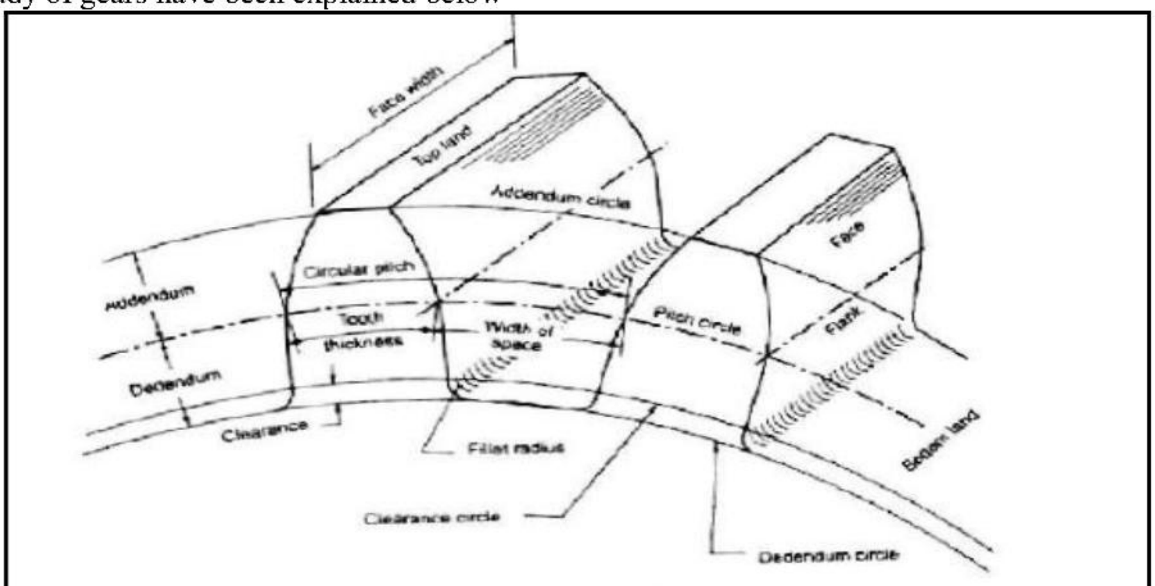


Fig. 5.2. Gear nomenclature

(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.

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

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

(b) Diametral pitch (Pd):

It is the ratio of number of teeth to the pitch circle diameter.

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

(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.

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

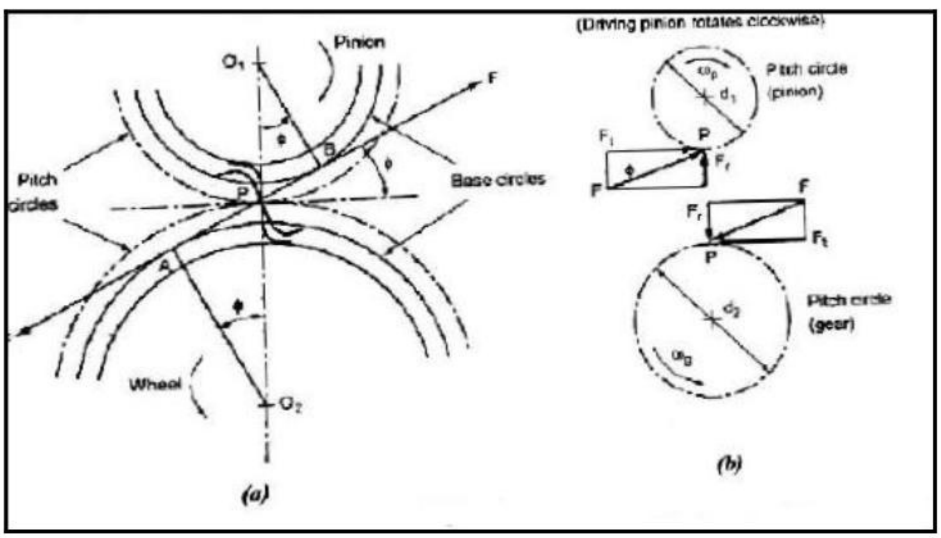
Where N_A and N_B= Speeds of driver and driven respectively, and,
Z_A and Z_B = Number of teeth on driver and driven respectively,

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
ϕ = 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

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

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

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

Or

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

Pitch line velocity (ν) is given by

$$\nu = \frac{\pi d N}{60} \ m/s$$

Then the transmitted power is calculated as

$$P = F_t \times \nu$$

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)**

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

5. Calculation of beam strength (Fs):

Fs= π.m.b.[σ_b].y……. (From data book page no. 8.50)

m= Module in mm.
Fs = Strength of gear tooth.
[σ_b] = Allowable static stress.
b = Face width = **10m**
y = Form factor = **$(0.154 - \frac{0.912}{z_1})$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 = $\left(\frac{\pi d_1 N_1}{60}\right)$**

8. Recalculation of beam strength (F_s):

1. Fs= π.m.b.[σ_b].y

9. Calculation of accurate dynamic load (F_d):

$F_d = F_t + \frac{21v(bc+F_t)}{21v+\sqrt{bc+F_t}}$……**(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) :

$F_w = d_1.b.Q.K_w$……. (From data book page no. 8.51)

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

F_w= Maximum wear load.

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

f = (2.8 * BHN – 70) N/mm²

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:**

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

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

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

$[M_t] = M_t$ x K x K_d….[**PSG data book page no:8.15**]

K x K_d= 1.3

M_t =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. $[\sigma_b] = (\frac{1.4\, K_{bl}}{n K_\sigma})\sigma_{-1}$ …….[**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]**

c. $[\sigma_c] = C_B. HB. K_{cl}$…………. **[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….. [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 center distance (a):**

$$a \geq (i+1)\sqrt[3]{\left[\frac{0.74}{[\sigma_c]}\right]^2 \times \frac{E_{eq}[M_t]}{i.\varphi}}$$ ….[**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:**

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

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

9. **Revision of center distance:**

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

10. **Calculation of b,** d, v *and* φ:

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. **Selectionofqualityofgears**:

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:**

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

Yv 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:**

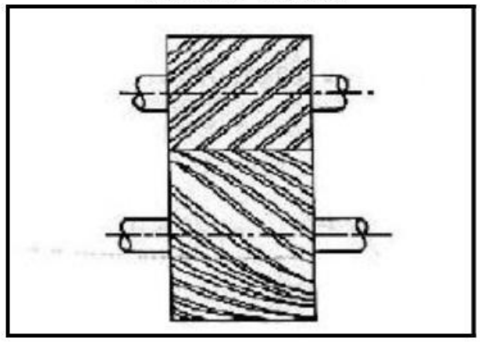
$$\sigma_c = 0.74\frac{i \pm 1}{a}\sqrt{\frac{i \pm 1}{ib}E_{eq}[M_t]}$$ ………..[**PSG data book page no:8.13]**

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

15. **Basic Dimensions:**

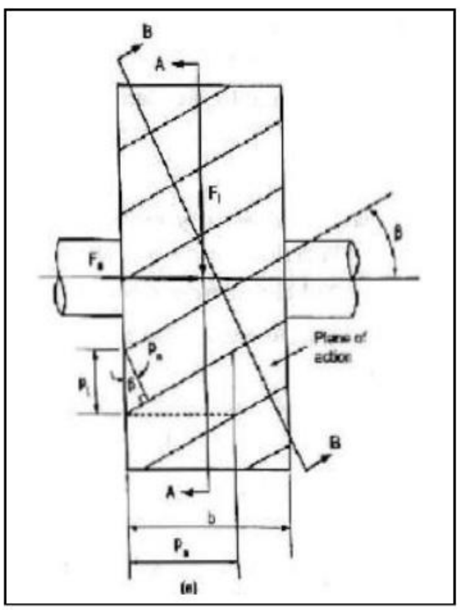
For Basic Dimensions of spur gear ..[**PSG data book page no:8.22]**

Helical Gears

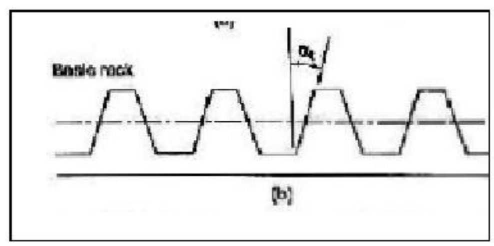


KINEMATICS AND NOMENCLATURE OF HELICAL GEARS

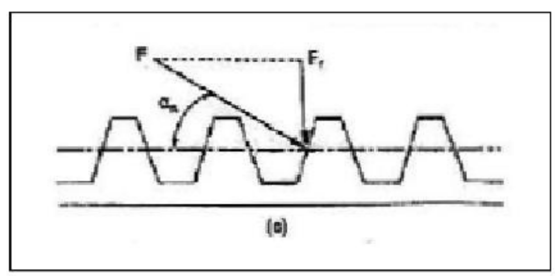
Let β = Helix angle,
p_t = Transverse cirular pitch,
p_n = Normal circular pitch,
p_a = Axial pitch,
p_d = Diametral pitch,
α_t and α_n = Transvers and normal pressure angles respectively,
m_t and m_n = Transverse and normal modulus respectively,
z_1 amd z_2 = Number of teeth on pinion and gear respectively,
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

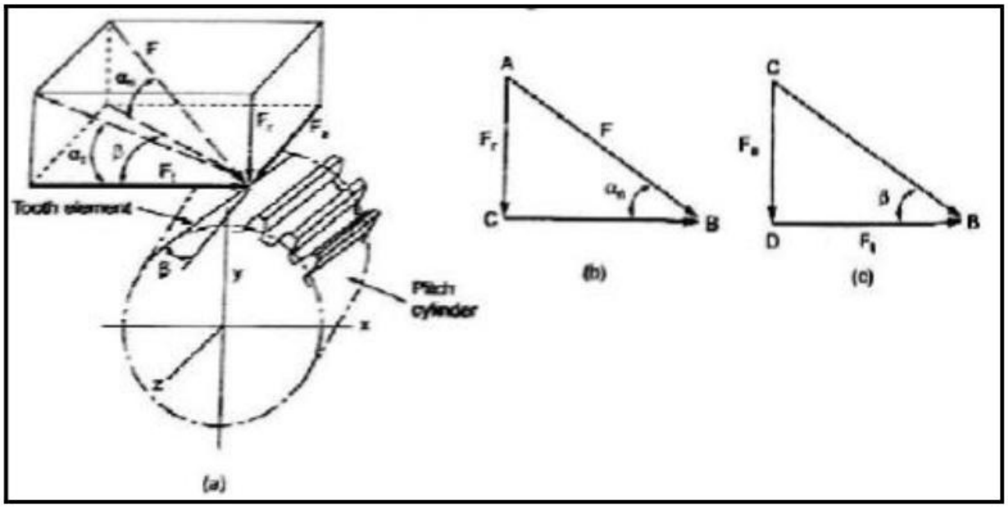
There are no standard proportions for helical gears. The proportions recommended by American Gear Manufacturer's Association (AGMA) are as follows:

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

- Helix angle **(β)** = **88 to 258, for helical** = **25**8 **to 40**8 **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. $\mathbf{F_t} = \left(\frac{P * K_0}{V}\right)$

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):

4. $\mathbf{F_d} = \left(\frac{Ft}{C_v}\right)$……… **(From data book page no. 8.50)**
5. $\mathbf{C_v} = \frac{6}{6+V}$ (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 = **$(0.154 - \frac{0.912}{z_{eq}})$for 20° involute.**

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

6. **Calculation of module (m_n):**
   1. **Fs ≥ 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 $d_1 = \left(\frac{m_n}{cos\beta} \; x \; z_1\right)$ **... (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. **Fs= π.m.b.[σ_b].y'**

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

   $F_d = F_t + \frac{21v(bc.cos^2\beta + F_t).cos\beta}{21v + \sqrt{bc.cos^2\beta + F_t}}$ **..(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) :**

    $F_w = \left(\frac{d1.b.Q.Kw.}{cos^2\beta}\right)$……. **(From data book page no. 8.51)**

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

    F_w= Maximum wear load.

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

    f = (2.8 * BHN – 70) N/mm^2

    d_1 = Pitch circle diameter.
    B = Face width.

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)**

DESIGN PROCEDURE FOR HELICAL 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:**

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

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

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

$[M_t] = M_t \times K \times K_d$….[**PSG data book page no:8.15**]

$K \times K_d = 1.3$

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

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

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

   e. $[\sigma_b] = (\frac{1.4\, K_{bl}}{n K_\sigma})\sigma_{-1}$ …….[**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]**

   f. $[\sigma_c] = C_B . H_B . K_{cl}$………… **[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….. [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 center distance (a):**

$$a \geq (i+1)\sqrt[3]{[\frac{0.7}{[\sigma_c]}]^2 \times \frac{E_{eq}[M_t]}{i.\varphi}}$$ ….[**PSG data book page no: 8.13]**

Assume $\varphi = \frac{b}{a} = 0.3$

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

Assume $Z_1 = 17$

$Z_2 = i \times Z_1$

8. **Calculation of module:**

$$m_n = \frac{2a}{(z1+z2)} \text{ x } \cos\beta$$ ……….[**PSG data book page no: 8.22]**

where β = helix angle.

Standard transverse module in **PSG data book pg no**: **8. 2**

9. **Calculation of center distance:**

$$a = \frac{m_n(Z1+Z2)}{2 * \cos\beta}$$ ………. [PSG **data book page no: 8.22]**

10. **Calculation of b,** d, v *and* φ:

Face width, $b = \varphi.a$

pitch circle dia, $d_1 = \frac{m_n}{\cos\beta} * Z_1$……[PSG **data book page no: 8.21]**

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

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

11. **Selectionofqualityofgears**:

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:**

$$\sigma_b = 0.7 \frac{i \pm 1}{a.m.b.y_v}[M_t]$$ ....[**PSG data book page no:8.13A]**

Yv 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:**

$$\sigma_c = 0.7 \left(\frac{i \pm 1}{a}\right)\sqrt{\frac{i \pm 1}{ib} E_{eq} [M_t]}$$ ………..[**PSG data book page no:8.13]**

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

15. **Basic Dimensions:**

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

UNIT– II: SPUR GEARS AND PARALLEL AXIS HELICAL GEARS (PART - A)
1. What is pressure angle? What is the effect of increase in pressure angle? (May/june 2014)
Soln.   It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. The standard pressure angle is $14\frac{1}{2}^0$ and 20^0
2. What condition must be satisfied in order that a pair of spur gears may have a constant velocity ratio? (May/june 2014)
Soln.   Normally spur gear are the replaced by other gears like helical, double helical gears, bevel gears etc. Spur gear is normally used in lower speed due its ability of generating zero axial thrust. Now in order to maintain constant gear ratio or speed ratio, their centre of pitch circle must be from fixed and the pitch circle of two mating gears should meet at a point and the line of action should meet at pitch point in order to satisfy the law of gearing.
3. What are the profiles of spur gear (May/june 2016)
Soln.   *1. Involute tooth profile*   *2. Cycloidal tooth profile*
4. What are the main types of gear tooth failure? (May/june2013) (May/june2012)
Soln. *The two modes of gear tooth failures are:*   *1. Tooth breakage (due to static and dynamic loads),and*   *2. Tooth wear (or) surface deterioration)*   *(a). abrasion,*   *(b). pitting, and*   *(c). scoring or seizure*
5. Define the various pitch in a helical gear. (May/June 2012)
Soln.   *1. Transverse circular pitch (p_t):the distance between corresponding points on adjacent teeth measured in a plane perpendicular to the shaft axis is known as Transverse circular pitch*   *2. normal circular pitch (p_n): the distance between corresponding points on adjacent teeth measured in a plane perpendicular to helix is known as normal circular pitch*   *3. Axial pitch (p_a): the distance between corresponding points on adjacent teeth measured in a plane parallel to the shaft axis is known as axial pitch.*

UNIT– II: SPUR GEARS AND PARALLEL AXIS HELICAL GEARS (PART - A)

6 .What is herringbone gear (April/May 2016)

Soln. Herringbone gears, also called double helical gears, are gear sets designed to transmit power through parallel or, less commonly, perpendicular axes. It do not have any grooves in between the gears

7. State the law of gearing or conditions of correct gearing (Nov/Dec 2010)

Soln. It states that for obtaining a constant velocity ratio, at any instant of teeth the common normal at each of contact should always pass through a pitch point, situated on the line joining the centre of rotation of the pair of mating parts.

8. What is tangential component of gear tooth force called useful component? (April/May 2010)

Soln.

Tangential component (F_t) *: the tangential* F_t *is a useful component. Because it transmits power. Using the value of* F_t *the magnitudes of torque transmitted power can be determined.*

Transmitted load, $= W_t = F_t$

Radial component (F_t)*: the radial component* F_r *is a separating force which is always directed towards the centre of the gear.* F_r *does no work. So it is not really a useful component. This force* F_r *causes bending of the shaft. The force* F_r*is also called as transverse force or bending force*

9. Compare the contact between mating teeth of spur and helical gears. (April/May 2010)

Soln. i) In spur gears the line of contact is parallel to the axis of rotation. The total length of contact line is equal to the face width.

ii) In helical gears the line of contact is diagonal across the face of the tooth. The total length of contact line is greater than the face width. This lowers the unit loading & increases load carrying capacity.

10. why is a gear tooth subjected to dynamic loading (April/May 2015)

Soln.

In addition to the static load due to power transmission, there are dynamic loads between the meshing teeth. The dynamic loads are due to the following reasons:

(a). Inaccuracies of tooth spacing, b). Irregularities in tooth profiles, (c). Elasticity of parts, (d). Misalignment between bearings, (e). Deflection of teeth under load, and (f). Dynamic unbalance of rotating masses.