

OBJECTIVES:

- To gain knowledge on the principles and procedure for the design of Mechanical power Transmission components.
- To understand the standard procedure available for Design of Transmission of Mechanical elements
- To learn to use standard data and catalogues (Use of P S G Design Data Book permitted)

UNIT I DESIGN OF FLEXIBLE ELEMENTS 9

Design of Flat belts and pulleys - Selection of V belts and pulleys – Selection of hoisting wire ropes and pulleys – Design of Transmission chains and Sprockets.

UNIT II SPUR GEARS AND PARALLEL AXIS HELICAL GEARS 9

Speed ratios and number of teeth-Force analysis -Tooth stresses - Dynamic effects – Fatigue strength- Factor of safety - Gear materials – Design of straight tooth spur & helical gears based on strength and wear considerations – Pressure angle in the normal and transverse plane- Equivalent number of teeth-forces for helical gears.

UNIT III BEVEL, WORM AND CROSS HELICAL GEARS 9

Straight bevel gear: Tooth terminology, tooth forces and stresses, equivalent number of teeth. Estimating the dimensions of pair of straight bevel gears. Worm Gear: Merits and demerits terminology. Thermal capacity, materials-forces and stresses, efficiency, estimating the size of the worm gear pair. Cross helical: Terminology-helix angles-Estimating the size of the pair of cross helical gears.

UNIT IV GEAR BOXES 9

Geometric progression - Standard step ratio - Ray diagram, kinematics layout -Design of sliding mesh gear box - Design of multi speed gear box for machine tool applications - Constant mesh gear box - Speed reducer unit. – Variable speed gear box, Fluid Couplings, Torque Converters for automotive applications.

UNIT V CAMS, CLUTCHES AND BRAKES 9

Cam Design: Types-pressure angle and under cutting base circle determination-forces and surface stresses. Design of plate clutches –axial clutches-cone clutches-internal expanding rim clutches- Electromagnetic clutches. Band and Block brakes - external shoe brakes – Internal expanding shoe brake.

TOTAL : 45 PERIODS

TEXT BOOKS:

1. Bhandari V, “Design of Machine Elements”, 3rd Edition, Tata McGraw-Hill Book Co, 2010.
2. Joseph Shigley, Charles Mischke, Richard Budynas and Keith Nisbett “Mechanical Engineering Design”, 8th Edition, Tata McGraw-Hill, 2008.

REFERENCES:

1. Sundararajamoorthy T. V, Shanmugam .N, “Machine Design”, Anuradha Publications, Chennai, 2003.
2. Gitin Maitra, L. Prasad “Hand book of Mechanical Design”, 2nd Edition, Tata McGraw-Hill, 2001.
3. Prabhu. T.J., “Design of Transmission Elements”, Mani Offset, Chennai, 2000.
4. C.S.Sharma, Kamlesh Purohit, “Design of Machine Elements”, Prentice Hall of India, Pvt. Ltd., 2003.
5. Bernard Hamrock, Steven Schmid, Bo Jacobson, “Fundamentals of Machine Elements”, 2nd Edition, Tata McGraw-Hill Book Co., 2006.

UNIT – I

DESIGN OF FLEXIBLE ELEMENTS

Introduction

The belts or ropes are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or at different speeds. The amount of power transmitted depends upon the following factors :

- i. The velocity of the belt.
- ii. The tension under which the belt is placed on the pulleys.
- iii. The arc of contact between the belt and the smaller pulley.
- iv. The conditions under which the belt is used.

It may be noted that

- (a) The shafts should be properly in line to insure uniform tension across the belt section.
- (b) The pulleys should not be too close together, in order that the arc of contact on the smaller pulley may be as large as possible.
- (c) The pulleys should not be so far apart as to cause the belt to weigh heavily on the shafts, thus increasing the friction load on the bearings.
- (d) A long belt tends to swing from side to side, causing the belt to run out of the pulleys, which in turn develops crooked spots in the belt.
- (e) The tight side of the belt should be at the bottom, so that whatever sag is present on the loose side will increase the arc of contact at the pulleys.
- (f) In order to obtain good results with flat belts, the maximum distance between the shafts should not exceed 10 meters and the minimum should not be less than 3.5 times the diameter of the larger pulley.

Selection of a belt drive

Following are the various important factors upon which the selection of a belt drive depends:

- i. Speed of the driving and driven shafts,
- ii. Speed reduction ratio,
- iii. Power to be transmitted,
- iv. Centre distance between the shafts,
- v. Positive drive requirements,
- vi. Shafts layout,
- vii. Space available, and
- viii. Service conditions.

Let

S_1 = Percentage slip between the driver and the belt,

S_2 = Percentage slip between the belt and the driven pulley, and

S = Total percentage slip = $S_1 + S_2$

$$\text{Velocity ratio, } \frac{N_2}{N_1} = \frac{D}{d} \left[1 - \frac{S_1 + S_2}{100} \right] = \frac{D}{d} \left[1 - \frac{S}{100} \right]$$

If thickness of the belt (t) is considered, then

$$\text{Velocity ratio, } \frac{N_2}{N_1} = \frac{D+t}{d+t} \left[1 - \frac{S}{100} \right]$$

Effect of Creep of Belt

Let σ_1 and σ_2 = Stresses in the belt on the tight side and slack side respectively, and

E = Young's modulus of the belt material.

$$\text{Velocity ratio, } \frac{N_2}{N_1} = \frac{D}{d} \times \frac{E + \sqrt{\sigma_1}}{E + \sqrt{\sigma_2}}$$

GEOMETRICAL RELATIONSHIPS

For open belt drive: An open belt drive is shown in Fig.

Let D and d = Diameters of the larger and smaller pulleys respectively in meters,

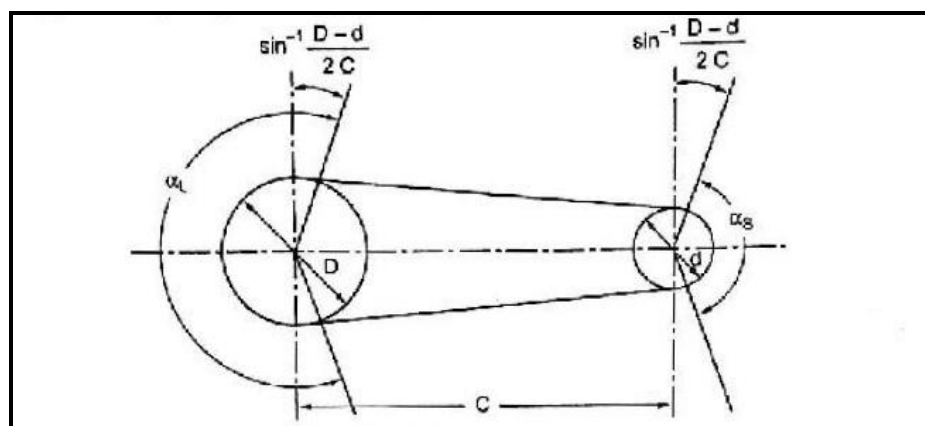
C = Centre distance between the two pulleys in meters,

L = Total length of the belt in meters,

2α = The angle subtended between the straight portions of the belt in degrees,

α_s = Wrap angle (or angle of contact/lap) for small pulley in degrees, and

α_L = Wrap angle for large pulley in degrees.



OPEN BELT DRIVE

$$\sin \alpha = \frac{D-d}{2C}$$

also

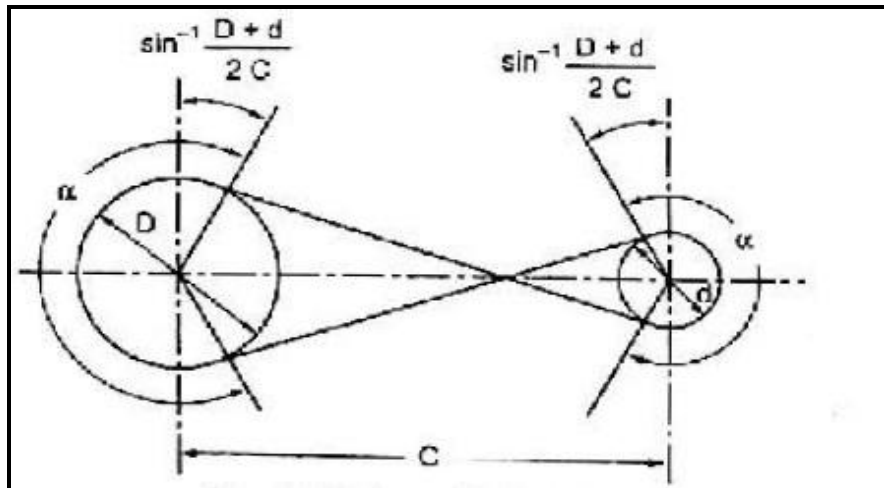
$$\alpha_s = (180 - 2\alpha) \text{ and } \alpha_L = (180 + 2\alpha)$$

Wrap angle for smaller pulley, $\alpha_s = 180 - 2 \sin^{-1} \left(\frac{D-d}{2C} \right)$

Wrap angle for larger pulley, $\alpha_L = 180 + 2 \sin^{-1} \left(\frac{D-d}{2C} \right)$

and Length of the belt, $L = 2C + \left(\frac{\pi}{2} \right) (D + d) + \frac{(D-d)^2}{4C}$

For Crossed Belt Drive:



$$\sin \alpha = \left(\frac{D+d}{2C} \right)$$

also $\alpha_s = \alpha_L = (180 + 2\alpha)$

Therefore, wrap angles for smaller and larger pulleys are same and is given by

$$\alpha_s = \alpha_L = 180 + 2 \sin^{-1} \left(\frac{D+d}{2C} \right)$$

and Length of the belt, $L = 2C + \left(\frac{\pi}{2} \right) (D + d) + \frac{(D+d)^2}{4C}$

STRESSES IN THE BELT

The various stresses acting at various portions of the belt are.

1. Stress due to maximum working tension, T_1 (σ_t):

$$\sigma_t = \frac{\text{Tight side tension}}{\text{Cross-sectional area of the belt}} = \frac{T_1}{b \cdot t}$$

Where b = width of the belt, and

t = Thickness of the belt.

2. Stress due to bending of the belt over the pulley (σ_b):

$$(\sigma_b) = \frac{E \cdot i}{d}$$

Where E = young's modulus of the belt over the pulley (σ_b)

d = diameter of the smaller pulley

3. Stress due to the effect of centrifugal force (σ_c)

$$(\sigma_c) = \frac{\text{centrifugal force}}{\text{cross sectional area of the belt}} = \frac{mv^2}{b \cdot t} = \rho v^2$$

Where $\rho = \text{density of the belt material in Kg/m}^3$

It is noted that the stress will be maximum when the belt moves over the smaller pulley. Therefore the maximum stress in the tight side of the smaller pulley is given by

$$\sigma_{max} = \sigma_t + \sigma_b + \sigma_c$$

Permissible stresses

Leather belt = 2 to 3.45 M Pa

Rubber belt = 1 to 1.7 M Pa

Fabric belt = Less than 1.5 M Pa

DESIGN OF FLAT BELT PULLEYS

Materials Used for Pulleys

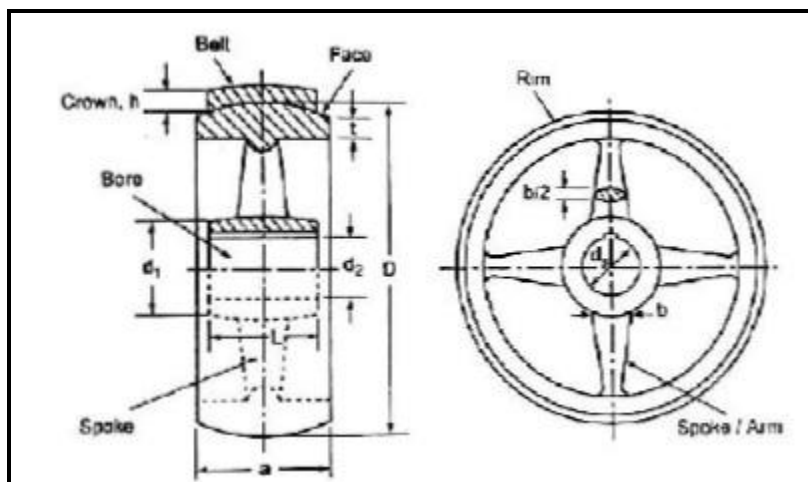
The commonly used pulley materials are:

Cast iron

- Fabricated steel
- Wood or fiber.
- Compressed paper
- Cast Iron pulleys are most widely used in actual practice.

Design Procedure for Cast Iron Pulleys

The cross-section of a cast iron pulley is shown in Fig.1.13. (Refer PSG data book, page no. 7.56).



D = Diameter of the pulley,

a = Width of the pulley,

b = Thickness of the arm,

d_1 = Diameter of the hub,

d_2 = Diameter of the shaft, and

L = Length of the hub,

t = Thickness of the rim,

1. Dimension of pulley:

(i) Diameter of the pulley (D): Obtain the diameter of the pulley either from velocity ration consideration or centrifugal stress consideration. We know that the centrifugal stress induced in the rim of the pulley,

$$\sigma_c = \rho v^2$$

Where ρ = Density of the rim material.

= 7200 kg/m³ for cast iron, and

v = Velocity of the rim = $\frac{\pi D N}{60}$ D being the diameter of pulley and N the speed of the pulley.

(ii) Width of the pulley (a): If the width of the belt is known, then select the width of the pulley referring to tables [from data book, page no. 7.54]

2. Dimension of arms :

(i) Number of arms (n):

Number of arms $\begin{cases} 4 \text{ for diameters upto } 450\text{mm} \\ 6 \text{ for diameters over } 450\text{mm} \end{cases}$ [from data book, page no. 7.56]

(ii) Cross-section of arms (b and b/2):

Major axis of elliptical section near the boss, $b = 2.94 \sqrt[3]{\frac{aD}{4n}}$ for single belt, and

$$= 2.94 \sqrt[3]{\frac{aD}{2n}} \text{ for double belt.}$$

Minor axis of elliptical section near the boss = $\frac{b}{2}$

(iii) Arms taper : The arms are tapered from hub to rim.

Taper = 4mm per 100mm

(iv) Radius of the cross-section of arms : $r = \frac{3}{4}b$

3. Dimensions of hub:

(i) Diameter of hub(d_1):

Diameter of the hub(d_1) = (1.7 to 2.0)* Diameter of the shaft (d_2)

(ii) Length of the hub(l):

Minimum length of bore, $l = \frac{2}{3}a$

Where, a = width of pulley.

4. Crowning of pulley rim:

Selection of crown height (h): Knowing diameter (D) and width (a) of the pulley, select the crown height (h) referring to tables 1.7(a) and (b)

Table 1.7(a). Crown of flat pulleys (40 to 355 mm diameter) **(from data book, page no. 7.55)**

(Crown is unrelated to the width in this diameter range)

Table 1.7(b). Crown of flat pulleys (40 to 2000mm diameter) **(from data book, page no. 7.55)**

(Crown varies with the width in this diameter range)

DESIGN OF FLAT BELT

The two different design procedures used are:

- (i) Using the manufacturer's data, and
- (ii) Using the basic equations.

DESIGN OF FLAT BELT DRIVE BASED ON MANUFACTURER'S DATA

1. Selection of pulley diameters:

Select the pulley diameters and angle of contact (i.e., wrap angle). By using the given belt speed and assuming number of plies, minimum pulley diameter is chosen. Use Table to choose the diameter of the smaller pulley. **(from data book, page no. 7.52)**

2. Calculation of design power in KW:

Calculate the design KW by using the relationship given below

$$\text{Design KW} = \frac{\text{Rated KW} \times \text{Load correction factor (K}_s\text{)}}{\text{Arc of contact factor (K}_a\text{)} \times \text{Small pulley factor (K}_d\text{)}}$$

i. **Load correction factor (k_s):** This factor is used to account for the nature of application and type of load. The value of K_s can be selected from table 1.9. **(From data book, page no. 7.53)**

ii. **Arc of contact factor(k_a):**

$$\text{Arc of contact} = 180 - \left(\frac{D-d}{c} \right) * 60 \quad \dots\dots\dots [\text{from data book pg.no 7.54}]$$

Where D and d are Diameter of larger and smaller pulley resp.

C is the centre distance.

iii. **Small pulley factor (k_d):**

Table. Small pulley factor, K_d (from data book, page no. 7.62)

3. Selection of belting:

Select a belt referring from table below.

Table. Load rating of fabric belts per mm width per ply at 180~arc of contact at 10m/s belt speed **(from data book, page no.7.54)**

4. Load rating correction:

Correct the load rating to the actual speed of the belt by using the relation given below.

$$\text{Load rating at V m/s} = \text{Load rating at 10 m/s} \times \frac{V}{10} \quad \dots [\text{From data book, page no. 7.54}]$$

5. Determination of belt width(b):

Determine the belt width by using the following relation:

$$\text{Width of belt} = \frac{\text{Design power}}{\text{Load rating} \times \text{No. of plies}}$$

Knowing the smaller pulley diameter and velocity of the belt, and consulting table. The number of plies can be found.

The calculated belt width should be rounded off to the standard belt width by consulting Table. Standard widths of transmission belting **(from data book no. 7.52)**

6. Determination of pulley width(B):

Determine the pulley width, by referring the tables 1.6 (a and b) [From data book, page no. 7.54]

Table 1.6(a). Pulley width ... [From data book, page no. 7.54]

Table 1.6(b). Recommended series of width of flat pulley, mm [From data book, page no. 7.55]

7. Calculation of belt length (L):

Calculation the length of the belt by using the equation given below.

For open belt drive: $L = 2C + \left(\frac{\pi}{2}\right)(D+d) + \frac{(D-d)^2}{4C}$ [From data book, page no. 7.53]

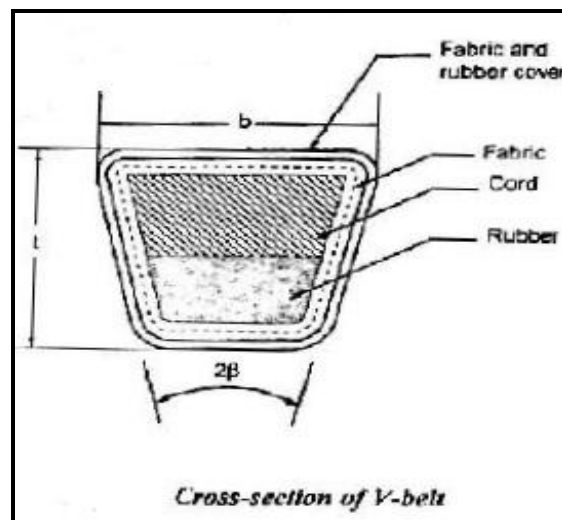
For crossed belt drive: $L = 2C + \left(\frac{\pi}{2}\right)(D+d) + \frac{(D+d)^2}{4C}$ [From data book, page no. 7.53]

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\frac{\mu\alpha}{\sin\beta}} = e^{\mu\alpha \operatorname{cosec}\beta}$$

Where T_1 , T_2 , m , v and α have usual meaning, and

2β is the V- groove angle ($=180^\circ$ for flat belt)

V -Belts and Pulleys



Materials of V-belts

V -belts are made of cotton fabric and cords molded in rubber and covered with fabric and rubber, as shown in Fig

Specification of V-belts

V-belts are designated by its type and nominal inside length. For example, a C2845 belt has a cross-section of type C and has a nominal inside length of 2845 mm.

RATIO OF DRIVING TENSIONS FOR V-BELT

$$\frac{T_1}{T_2} = e^{\frac{\mu\alpha}{\sin\beta}} = e^{\mu\alpha \operatorname{Cosec}\beta}$$

Where T_1 and T_2 = Tensions in the tight and slack sides respectively,

2β = Angle of the groove, and

μ = Coefficient of friction between the belt and sides of the groove.

Note. Number of V-belt = $\frac{\text{Total Power Transmitted}}{\text{Power Transmitted}}$

DESIGN OF SHEAVES (OR V-GROOVED PULLEYS)

1. Materials of V-grooved pulleys:

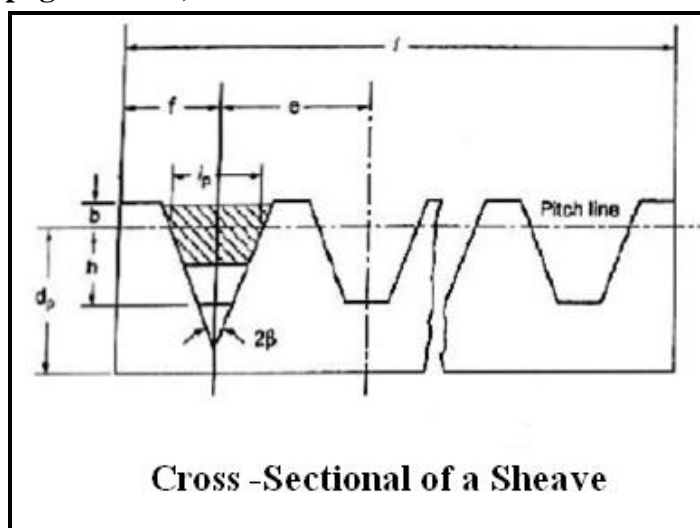
The commonly used sheave materials and their characteristics are summarized In Table 2.1.

Table 2.1

Material of sheaves	Characteristics and /or applications
1. cast iron	It is economical, stable and durable also it has excellent friction characteristics on V-belts.
2.Pressed steel	It is lighter and cheaper but it gives rise to excessive belt slip wear and noise.
3.Formed steel	Primarily used in automotive and agricultural purposes
4.Diecast aluminum	Used for special applications

2. Dimensions of sheaves:

The cross-section of a sheave (i.e., V-grooved pulley) for V-belt drives IS shown In Fig.2.2. (Refer data book, page no. 7.70).



where

I_p = Pitch width,

I = Face width,

f = Edge of pulley to first groove centre,

e = centre to centre distance of grooves

d_p = pulley pitch diameter,

b = Minimum distance down to pitch line,

h = Minimum depth below the pitch line.

Table 2.2 The various dimensions of standard V-grooved pulley in mm(**from data book, page no. 7.70**)

Note Face width, $I = (n - 1) e + 2f$

The two different design procedures used are:

- (i) Using the manufacturer's data, and
- (ii) Using the basic equations.

DESIGN OF V-BELT DRIVE BASED ON MANUFACTURER'S DATA

The design of V-belt is primarily concerned with the selection of belt section, selection of pulley diameters, determination of number of belts and centre distance required for the given transmitted power.

DESIGN PROCEDURE:

1. Selection of belt section:

Consulting Table 2.3, select the cross-section of a belt (i.e., type of belt) depending on the power to be transmitted. ... [From data book, page no. 7.58]

2. Selection of pulley diameters (d and D):

Select small pulley diameter (d) from Table 2.3. Then using the speed ratio, calculate the large pulley diameter (D). These pulley diameters should be rounded off to a standard diameter by using Table 1.5. ... [From data book, page no. 7.58]

3. Determination of nominal pitch length:

Determine the length of the belt L (which is also known as nominal inside length) by using the formula,

$$L = 2C + \left(\frac{\pi}{2}\right)(D + d) + \frac{(D-d)^2}{4C} \dots \text{[From data book, page no. 7.53]}$$

For the calculated nominal inside length and belt section, consulting Table 2.5, select the next standard pitch length.

Tile nominal pitch length is defined as the circumferential length of the belt at the pitch width (i. e., the width at the neutral axis of the belt). The value of the pitch width remains constant for each type of belt irrespective of the groove angle .

For pitch length, add with inside length, 36 mm for A belt, 43 mm for B, 56 mm for C. 79 mm for D and 92 mm for E belt.

4. Selection of various modification factors: In order to calculate the design power the following modification factors have to be determined.

(i) Length correction factor (F_C):

Table. Nominal inside length, nominal pitch length and length correction factor/or standard sizes D/ JI-belts (**from data book, page no. 7.58, 7.59 and 7.60**) (The values for a few cases only given)

(ii) Correction factor for arc of contact (F_d):

- First determine the angle of contact (or arc of contact) of the smaller pulley.
- **Arc of contact** = $180^\circ - \left(\frac{D-d}{C}\right) \times 60^\circ \dots \text{[From data book, page no 7.68]}$
- For the calculated arc of contact, select the correction factor from Table 2.6.
- Arc of contact factor is taken into account because the power transmitted may be limited by slipping of the belt on the smaller pulley.

Table 2.6. Arc of contact factor, F_d ...[From data book, page no. 7.68]

(The values/or a few cases are given below)

(iii) Service factor (F_a):

- Select the service factor (F_a) consulting Table 2.7.

- The service factor takes into account the severity of the load transmitted which depends upon the characteristics of the driving and driven units.

Table 2.7. Service factor for V-Belts, Fa... [From data book, page no. 7.69]

Note. The details of driving units and driven machines under different duties are available in the data book, page no. 7.69.

6. Calculation of maximum power capacity :

Calculate the maximum power capacity (in kW) of a V-belt using the formulas given in Table 2.8.... [From data book, page no. 7.62]

where kW = Maximum power in kW at 180° arc of contact for a belt of average length,

S = Belt speed, m/s,

d_e = Equivalent pitch diameter = $d_p \times F_b$,

d_p = Pitch diameter of the smaller pulley, mm, and

F_2 = Small diameter factor to account for variation of arc of contact, from

Table 2.9.... [From data book, page no. 7.62]

7. Determination of number of belts (n_b):

Determine number of belts the (n_b) from the relation,

$$n_b = \frac{P \times F_a}{KW \times F_c \times F_d} \dots [\text{From data book, page no. 7.70}]$$

Where P = Drive power, in kW,

Fa = Service factor for V-belts,

KW = Rated power (i.e., rating of a single V-belt),

Fc = Length correction factor, and

Fd = Correction factor for arc of contact.

8. Calculation of actual centre distance:

Calculate the actual centre distance from the relation,

$$C_{\text{actual}} = A + \sqrt{A^2 - B} \dots [\text{From data book, page no. 7.61}]$$

$$\text{Where } A = \frac{L}{4} - \pi \left[\frac{D+d}{8} \right]$$

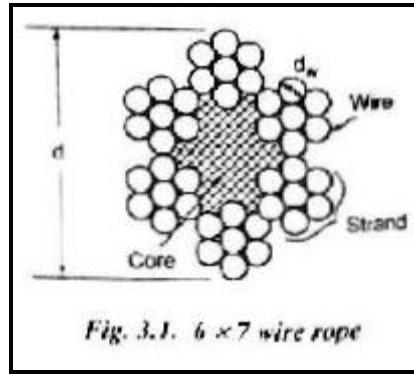
$$B = \frac{(D-d)^2}{8}, \text{ and}$$

L = Nominal pitch length of the belt from table 2.5 (refer step 4)

WIRE ROPES AND PULLEYS

Materials of Wire Ropes

The commonly used materials for wire ropes are \ wrought iron, cast steel, plow steel and alloy steel. For special purposes copper, bronze, aluminum alloys and stainless steel are also used.



SPECIFICATION OF WIRE ROPES

The wire ropes are specified (or designated) by the number of strands and the number of wires in each strand. For example, a 6 x 7 rope means a rope made from six strands with seven wires in each strand.

GUIDELINES FOR THE SELECTION OF WIRE ROPE

The wire rope is selected based on its application. The shows the standard designation of wire ropes and their applications.

STRESSES IN WIRE ROPES

The various types of stresses induced in a wire rope are:

1. Direct stress due to the weight of the load to be lifted and weight of the rope (σ_d):

Let W = Weight of the load to be lifted,

W_r = Weight of the rope, and

A = Area of useful cross-section of the rope.

$$\text{Direct stress } \sigma_d = \frac{W + W_r}{A}$$

2. Bending stress when the rope passes over the sheave or drum (σ_b):

$$\text{Bending stress, } \sigma_b = \frac{d_w E_r}{D}$$

Where E_r = modulus of elasticity of the wire rope

$$= 0.84 \times 10^5 \text{ N/mm}^2, \text{ for steel ropes of ordinary construction,}$$

$$= \frac{3}{8} * E, E = \text{Modulus of elasticity of the wire material}$$

$$d_w = \text{diameter of the wire,}$$

$$D = \text{diameter of sheave.}$$

3. Stress due to acceleration (σ_a):

Due to change in speed, an additional stress is induced. The stress due to acceleration is given by

$$\sigma_a = \left(\frac{W + W_r}{g} \right) \frac{a}{A}$$

Where a = Acceleration of rope and local during hoisting (not at starting or stopping)

$$= \frac{v_2 - v_1}{t}; (v_2 - v_1) \text{ is the change in speed in 't' seconds.}$$

4. Stress during starting and stopping (σ_{st}):

(i) When there is no slack in the rope:

$$\sigma_{st} = 2 \times \sigma_d$$

(ii) When there is slack in the rope before starting or stopping, then there will be a considerable impact load on the rope.

$$\sigma_{st} = \frac{W + W_r}{A} \left[1 + \sqrt{1 + \frac{2 \cdot a_s \cdot h \cdot E_r}{\sigma_d \cdot l \cdot g}} \right]$$

Where a_s = Acceleration during starting or stopping,

h = Slack during starting, and

l = Length of the rope.

Effective stress:

(i) Effective stress in the rope during normal working,

$$\sigma_{en} = \sigma_d + \sigma_b$$

(ii) Effective stress in the rope during starting,

$$\sigma_{est} = \sigma_{st} + \sigma_b$$

(iii) Effective stress in the rope during acceleration of the load,

$$\sigma_{ea} = \sigma_d + \sigma_b + \sigma_a$$

RECOMMENDED FACTOR OF SAFETY FOR WIRE ROPES

The recommended factor of safety for wire ropes based on the ultimate strength are given in Table.

Table. Recommended factor of safety for wire ropes, n' (from data book, page no. 9.1)

DESIGN OF WIRE ROPES

DESIGN PROCEDURE FOR A WIRE ROPE

1. Selection of suitable wire rope:

First select the suitable type of wire rope for the given application, from Table 3.1.

2. Calculation of design load:

Calculate the design load by assuming a larger factor of safety, say 15 (or find the design load by assuming a factor of safety 2 to 2.5 times the factor of safety given in Table 3.2).

Design load = Load to be lifted \times Assumed factor of safety

3. Selection of wire rope diameter (d):

Select the wire rope diameter (d) from Table 3.4, Group 6 x 19 (**from data book, page no. 9.5 and 9.6**) by taking the design load as the breaking strength.

4. Calculation of sheave diameter (D):

Consulting Table 3.5. (**from data book, page no. 9.1**) obtain the diameter of sheave (or drum). Always larger sheave diameter is preferred.

- Ratio for 50 m/min of rope speeds - to be increased by 8% for each 'additional speed of 50m/min

5. Selection of the area of useful cross-section of the rope (A):

Consulting Table 3.6, select the area of useful cross-section of the rope.

Type of construction	Metallic area of rope A, mm ²
6 x 7	0.38 d ²
6 x 19	0.4d ²
6 x 37	0.4d ²

6. Calculation of wire diameter (dw):

Calculate the diameter of wire using the relation

$$d_w = \frac{d}{1.5\sqrt{i}}$$

where

i = Number of wires in the rope

= Number of strands x Number of wires in each strand.

7. Selection of weight of rope (W_r):

Obtain the rope weight (Wr) from Table 3.4.

8. Calculation of various loads:

Calculate the various loads using the relations given below.

(i) Direct load, $W_d = W + W_r$

(ii) Bending load, $W_b = \sigma_b \times A = E_r \cdot \frac{d_w}{D} \times A$

(iii) Acceleration load due to change in the speed of hoisting,

$$W_a = \left[\frac{W + W_r}{g} \right] a$$

Where $a = \frac{v_2 - v_1}{t}$ (When speed of the rope changes from v_1 to v_2 in t seconds)

(iv) Starting or stopping load:

(a) When there is no slack in the rope:

Starting load, $W_{st} = 2 \cdot W_d = 2 (W + W_r)$

(b) When there is slack in the rope:

Starting load, $W_{st} = \sigma_{st} \times A = (W + W_r) \left[1 + \sqrt{1 + \frac{2 \cdot a_s \cdot h \cdot E_r}{\sigma_d \cdot L \cdot g}} \right]$

9. Calculation of effective loads:

(i) Effective load on the rope during normal working, $W_{en} = W_d + W_b$

(ii) Effective load on the rope during acceleration of the load, $W_{ea} = W_d + W_b + W_a$

(iii) Effective load on the rope during starting, $W_{est} = W_b + W_{st}$

10. Calculation of working (or actual) factor of safety (FS_w):

Working factor of safety, $FS_w = \frac{\text{Breaking load from table 3.4 for the selected rope}}{\text{Effective load during acceleration } (W_{e2})}$

11. Check for safe design:

Compare the calculated working factor of safety (FS_w) with the recommended factor of safety (n') given in Table 3.2. If the working factor of safety is greater than the recommended factor of safety (*i. e.*, $FS_w > n'$), then the design is safe and satisfactory.

If $FS_w < n'$, then the design is not satisfactory. Now choose some other rope with greater breaking strength or increase the number of ropes.

12. Calculation of number of ropes:

$$\text{Number of ropes} = \frac{\text{Recommended factor of safety}}{\text{Working factor of safety}} = \frac{n'}{FS_w}$$

FAILURE OF ROPES

The amount of wear that occurs depends upon the pressure between the rope and the sheave and is given

$$P = \frac{2T}{dxD}$$

T = Tension in rope

d = Diameter of rope, and

D = sheave diameter

CHAIN DRIVES

TYPES OF CHAIN DRIVES

The common types of chains are:

1. Link chains (or welded chains),
2. Transmission chains (or roller chains,) and
3. Silent chains (or inverted tooth chains).

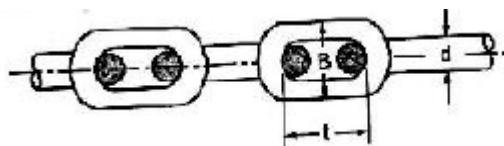
LINK CHAINS

Link chains, also known as *welded load chains*, are widely used

- In low capacity hoisting machines such as hoists, winches and hand operated cranes as the main lifting appliances.
- As slings for suspending the load from the hook or other device.

DIMENSIONS OF A LINK CHAIN

They are pitch (t) equal to the inside length of the link, outside width (B) and diameter (d) of the chain bar.



CLASSIFICATION OF LINK CHAINS

1. Depending on the ratio between the pitch and the diameter of the chain bar:

- (a) Short link chains: If $t \leq 3d$, then the chains are known as short link chains.
- (b) Long link chains: If $t > 3d$, then the chains are known as long link chains.

2. Depending on the manufacturing accuracy:

- (a) Pitched chains: When the permissible deviations from the nominal pitch size is

within $0.03d$ and from the outside width is within $0.05d$, then the chain is called as Pitched chain.

(b) Calibrated chains : When the permissible deviations is within $\pm 0.1d$ of the nominal Size in pitch and outside width, then the chain is known as calibrated chain.

SELECTION OF LINK CHAINS

The general formula for selecting link chains in tension is given by

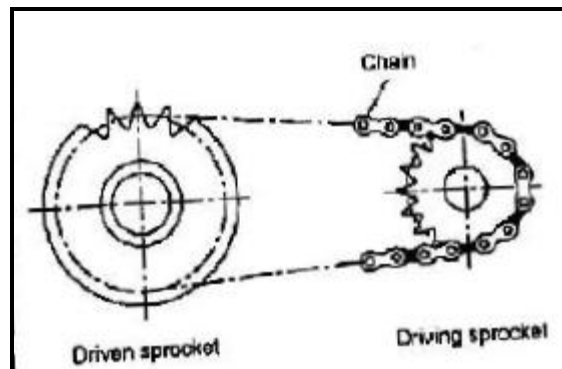
$$P_{safe} = \frac{P_{br}}{n}$$

P_{safe} = Safe load carried by the chain,

P_{br} = Breaking load of the chain, and

n = Factor of safety

TRANSMISSION (OR ROLLER) CHAINS



CHAIN MATERIALS

- Link plates are made of cold-rolled, medium-carbon or alloy steels such as C45, C50 and 40 Cr1.
- Pins, bushings and rollers are made of carburizing steels such as C15, C20, and 30 Ni4 Cr1.

SPECIFICATION OF A ROLLER CHAIN

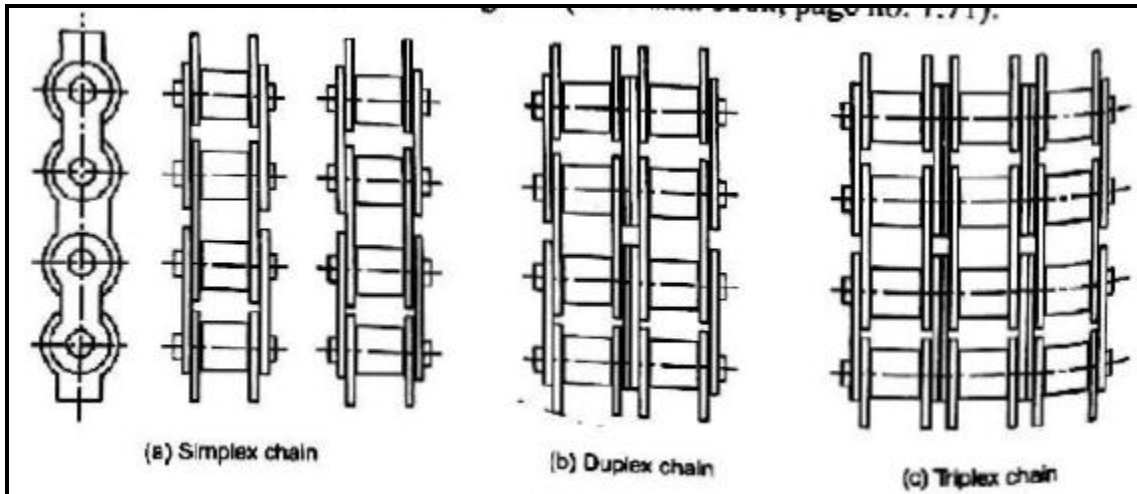
Roller chain is specified by three dimensions - pitch, width and diameter.

Pitch: It is the distance from centre to centre of adjacent pins or rivets.

Width: It is the nominal width of the link or the length of the pin.

Diameter: It refers to the actual outside diameter of the roller.

Roller chains are available in single-row or multi-row construction such as simplex, duplex or triplex strands as shown in Fig. (Refer data book, page no. 7.71).



Shows a sprocket driving a chain in a counter clockwise direction.

Let p = Chain pitch,

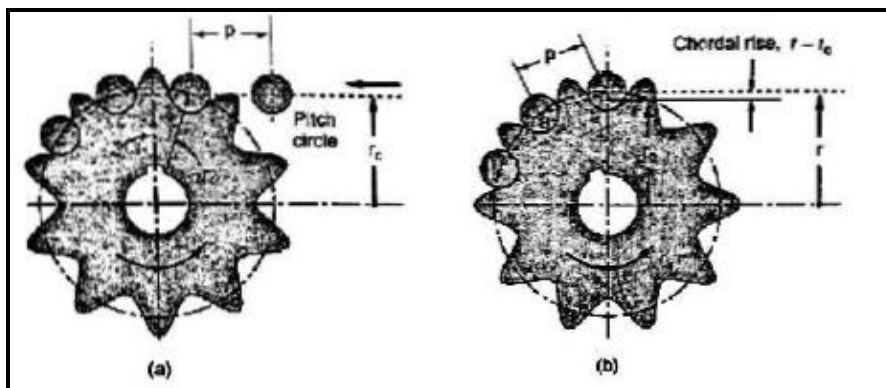
α = Pitch angle,

$\alpha/2$ = Angle of articulation,

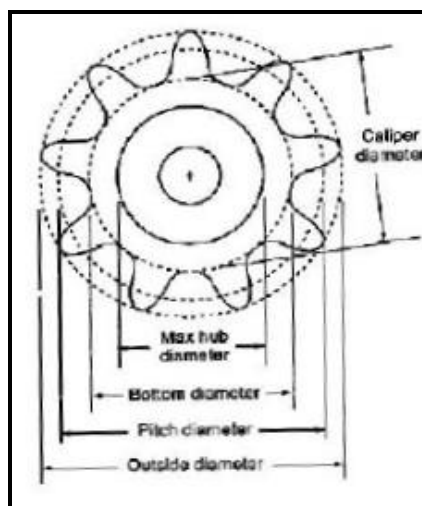
D = Pitch circle diameter of the sprocket, and

z = Number of teeth on the sprocket.

CHORDAL (OR POLYGONAL) ACTION



SPROCKET DIAMETERS



The equations for those diameters are :

(i) Pitch diameter = $\frac{p}{\sin(\frac{180}{z})}$

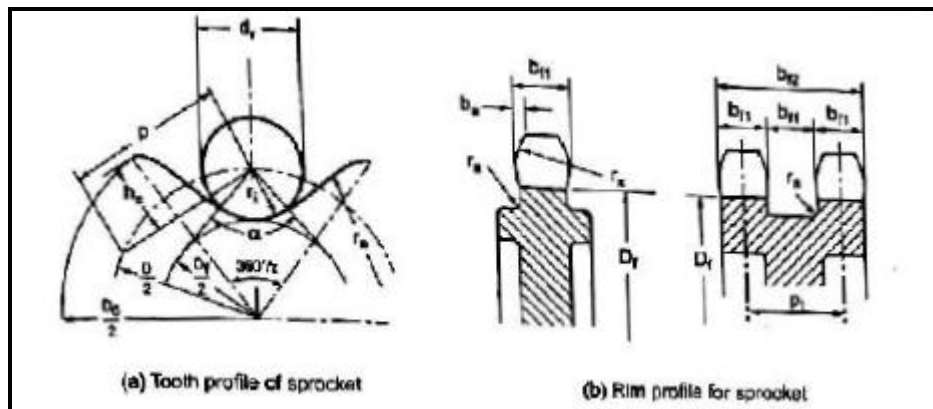
(ii) Outside diameter = $p[0.6 \cot(\frac{180}{z})]$

(iii) Bottom diameter = Pitch diameter – Roller outside diameter

(iv) Caliper diameter = Pitch diameter $\times \cos(\frac{90}{z})$ – Roller outside diameter

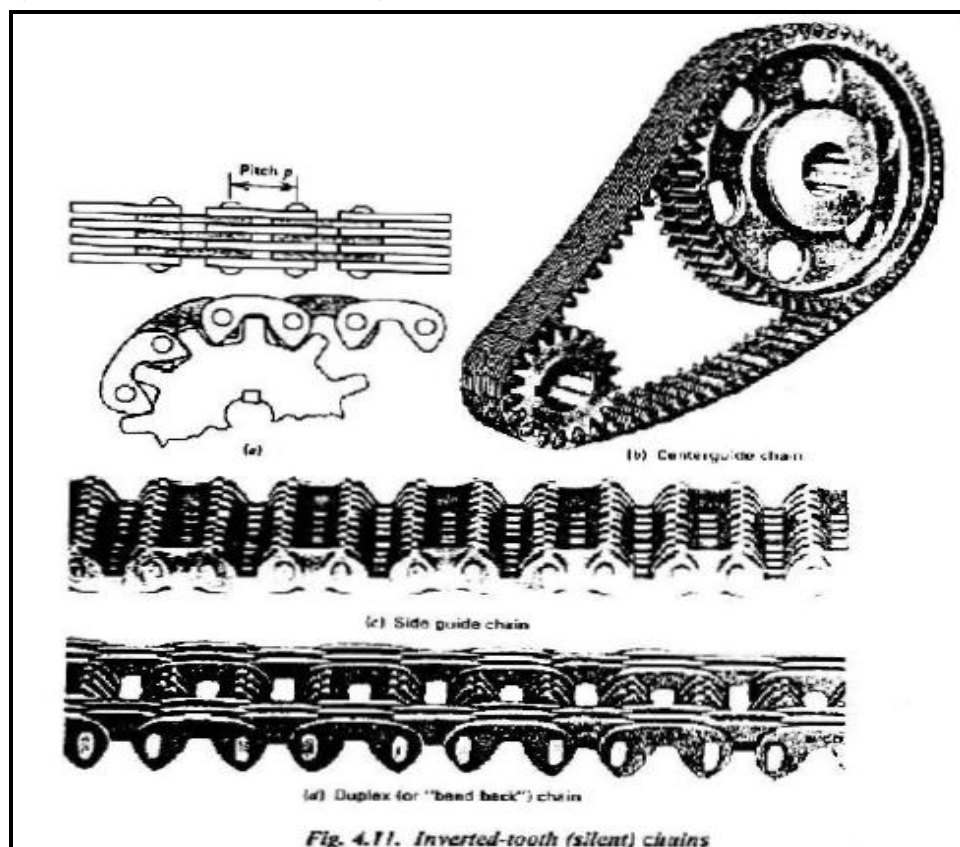
(v) Maximum hub diameter = $p [\cot(\frac{180}{z}) - 1] - 0.03$

TOOTH FORM



DESIGN OF SILENT CHAIN

SILENT (OR INVERTED-TOOTH) CHAIN



TYPES OF SILENT CHAINS

Depending upon the type of joint between links, the silent chains are classified into:

- (i) Reynolds chain: In Reynolds chain, the links are connected by pins resulting in sliding friction.
- (ii) Morse chain: In Morse chain, the rocker pins are used.

DESIGN PROCEDURE OF ROLLER CHAIN

1. Selection of the transmission ratio (i):

Select a preferred transmission ratio, i (from data book, page no. 7.74)

1, 1.12, 1.25, 1.4, 1.6, 1.8, 2, 2.25, 3.15, 4, 4.5, 5, 5.6, 6.3 and 7.1
--

From the pitch range obtained, consulting Table 4.4, select a suitable standard pitch. (From data book, page no. 7.74)

2. Selection of number of teeth on the driver sprocket (Z_1):

Select the number of teeth on the driver sprocket (Z_1)....[From data book, page no. 7.74]

3. Determination of number of teeth on the driven sprocket (Z_2):

$Z_2 = i * Z_1$[From data book, page no. 7.74]

Recommended value of $Z_2: Z_{2max} = 100$ to 120 [From data book, page no. 7.74]

4. Selection of standard pitch (p):

$a = (30 \text{ to } 50)p$

$a = 30p_{max}$ $a = 50p_{min}$

From the pitch range obtained, consulting table 4.4, select a suitable standard pitch.

[From data book, page no. 7.74]

5. Selection of the chain:-

(From data book, page nos. 7.71, 7.72 and 7.73.

This table gives some details for a few Chains.)

Note R - Simplex, DR - Duplex, TR – Triplex

6. Calculation of Total load on The Driving Side of the Chain (P_T):

*{Total load on the
Driving side (P_T)*

$$= \left\{ \begin{array}{l} \text{Tangential force} \\ \text{due to} \\ \text{power transmission} (P_t) \end{array} \right\} + \left\{ \begin{array}{l} \text{Centrifugal tension} (P_c) \\ \text{due to speed} \\ \text{of the chain} \end{array} \right\} + \left\{ \begin{array}{l} \text{Tension} \\ \text{due to chain} \\ \text{sagging} (P_s) \end{array} \right\}$$

$P_T = P_t + P_c + P_s$ [From data book, page no. 7.78]

(i) To find tangential force (P_t):

$$P_t = \frac{1020 N}{v} \dots [\text{From data book, page no. 7.78}]$$

Where N = Transmitted power in KW, and

$$v = \text{Chain velocity in m/s} = \frac{z_1 \times p \times N_1}{60 \times 1000} \text{ or } \frac{z_2 \times p \times N_2}{60 \times 1000}$$

(ii) To find centrifugal tension (P_c):

$$P_c = mv^2 \dots [\text{From data book, page no. 7.78}]$$

Where m = Mass of chain / meter,

V = velocity of chain , m/s

(iii) To find tension due to sagging (P_s):

$$P_s = k \cdot w \cdot a$$

where k = Coefficient of sag taking into account the arrangement of chain drive,

W = Weight of chain / meter = $m \cdot g$, and

a = Centre distance in meter.

Coefficient for sag, k ...[From data book, page no. 7.76]

7. Calculation of service factor (K_s):

The service factor is used to account for variations in the driving and driven sources for roller chains.

Service factor, $K_s = k_1 \cdot k_2 \cdot k_3 \cdot k_4 \cdot k_5 \cdot k_6 \dots$ [From data book, page no. 7.76]

k_1 = Load factor , (from data book, page no.7.76)

k_2 = Factor for distance regulation, (from data book, page no.7.76)

k_3 = Factor for centre distance of sprockets, (from data book, page no.7.76)

k_4 = Factor for position of sprocket (from data book, page no.7.77)

k_5 = Lubrication factor (from data book, page no.7.77)

k_6 = Rating factor (from data book, page no.7.77)

8. Calculation of design load:

Design load = Total load on the driving side of the chain x Service factor

$$\text{Design load} = P_T \times K_s$$

9. Calculation of working factor of safety (FS_w):

$$\text{Factor of safety} = \frac{\text{Breaking load } Q}{\text{Design load}} = \frac{Q}{P_T \times K_s}$$

Where Q is breaking load got from data book, page no.7.72

10. Check for factor of safety:

[Refer From data book, page no. 7.77]

If the working factor of safety (FS_w) is greater than the recommended minimum value of factor of safety (n') then the design is safe and satisfactory.

11. Check for the bearing stress in the rollers:

Calculate the bearing stress in the roller using the formula

$$\sigma = \frac{\text{Tangential load}}{\text{Bearing area}} = \frac{P_t \times K_s}{A}$$

Where A is Bearing Area got From data book, page no. 7.72

Take the values for allowable bearing stress, $[\sigma]$, N/mm² ... [From data book, page no. 7.77]

$\sigma < [\sigma]$. The design is safe and satisfactory.

12. Calculation of actual length of chain (L):

Calculate the number of links (l_p) using the formula

$$L = 2a_p + \frac{Z_1 + Z_2}{2} + \frac{\left[\frac{Z_1 + Z_2}{2\pi}\right]^2}{a_p} \dots [\text{From data book, page no. 7.75.}]$$

$$a_p = \frac{a_0}{p} = \frac{\text{Initial centre distance}}{\text{pitch}}$$

$$L = l_p \times p$$

13. Calculation of exact centre distance:

$$\text{Exact centre distance, } a = \frac{e + \sqrt{e^2 + 8M}}{4} \times p \dots [\text{From data book, page no. 7.75.}]$$

$$\text{Where } e = l_p - \left(\frac{Z_1 + Z_2}{2}\right) \dots [\text{From data book, page no. 7.75.}]$$

$$M = \left(\frac{Z_1 + Z_2}{2\pi}\right)^2 = \text{Constant}$$

14. Calculation of pitch circle diameters (pcd) of sprockets:


$$\text{Pcd of smaller sprocket, } d_1 = \frac{p}{\sin\left(\frac{180}{Z_1}\right)}$$

$$\text{Pcd of Larger sprocket, } d_2 = \frac{p}{\sin\left(\frac{180}{Z_2}\right)} \dots [\text{From data book, page no. 7.78.}]$$

$$\text{Smaller sprocket outside diameter, } d_{01} = d_1 + 0.8d_r$$

$$\text{Larger sprocket outside diameter, } d_{02} = d_2 + 0.8d_r$$

Where d_r is the roller diameter

<p align="center">UNIT – I: DESIGN OF TRANSMISSION SYSTEMS FOR FLEXIBLE ELEMENTS (PART –A)</p>	
<p>1. What are the factors upon which the coefficient of friction between the belt and pulley depends? (May/june2012) (May/June 2014)</p>	
<p>Soln. The coefficient of friction between the belt and the pulley depends upon the following factors:</p> <ol style="list-style-type: none"> 1. The material of belt; 2. The material of pulley; 3. The slip of belt; and 4. The speed of belt. <p>According to C.G. Barth, the coefficient of friction (μ) for oak tanned leather belts on cast iron pulley, at the point of slipping, is given by the following relation, i.e.</p> $\mu = 0.54 - \{42.6/(152.6 + v)\}$ <p align="center">Where, v = Speed of the belt in meters per minute.</p>	
<p>2. Brief the term “crowning of pulley”. (May/June 2014)</p>	
<p>Soln. Pulleys are provided, a -slight conical shapes (or) convex shapes in their rim's r surface in order to Prevent the belt from running off the pulley due centrifugal force. This is known as crowning, of pulley. Usually the crowning height t may be $1/96$ of pulley face width.</p>	
<p>3. What are the materials used for belt-drive? (May/June 2013) (May/June 2016)</p>	
<p>Soln. Leather,, cotton fabrics ,rubber, animal's hair, silk, rayon, woolenetc</p>	
<p>4. What do you mean by Galling of roller chains? (Nov/Dec 2010) (May/june2012) (Nov/Dec 2010)</p>	
<p>Soln. The pin pressure faces have suffered from severe galling where the surfaces have articulated and fused together.</p>	
	
<p>5. When do you prefer chain drive to a belt or rope drive? (May/Jun 2016)</p>	
<p>Soln.</p> <p>Chain drives are preferred for velocity ratio less than 10, chain velocities upto 25 m/s, and for power ratings up to 125KW</p>	

DESIGN OF TRANSMISSION SYSTEMS FOR FLEXIBLE ELEMENTS (PART –A)

6. What are the five parts of roller chain? (April/May 2010)

Soln.

The five parts of roller chain are

1. Pin link 2. Pin 3. Bushing 4. Roller and 5. plates

7. Sketch and name the different types of compound wire ropes. (April/May 2010)

Soln. They are two type namely

a) Fiber ropes

b) Wire ropes. (a) Based on number of strands and wires..

i) 6x7 ii) 6 x 19 iii) 6 x 37 iv) 8 x 19 ropes.

(b) Based on the direction of wire I a with respect to strands in twisting.

i) Cross -lay ropes.

ii) Parallel-lay ropes.

iii) Compound laid ropes.

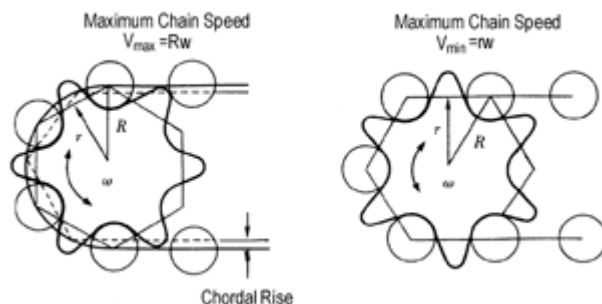
8. How are the ends of flat-belt joined?

(April/May 2011)

Soln. (i) Cemented joints (ii) Laced joints (iii) Crest joints. (iv) Hinged joints.

9. What is chordal action (Polygonal action) in chain drives? (Nov/Dec 2012)

Soln. When chain passes over the sprocket, it moves as a series of chords instead of a continuous arc as in the case of a belt drive. It results in varying speed of the chain drive. This phenomenon is known as chordal action



10. Give an expression for ratio of tensions in a flat belt drive. (Apr/May2011)

Soln. Tension ratio, $\frac{T_1}{T_2} = e^{\mu\alpha}$...[neglecting centrifugal tension]

$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\mu\alpha} \dots[\text{considering centrifugal tension}]$$

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 $v < 3 \text{ m / s}$

(ii) Medium velocity gears $v = 3 \text{ m / s}$

(iii) High velocity gears $v > 15 \text{ m / s}$

4. Classification based on the position of teeth on the wheel:

(i) Straight gears (ii) Helical gears

(iii) Herringbone gears

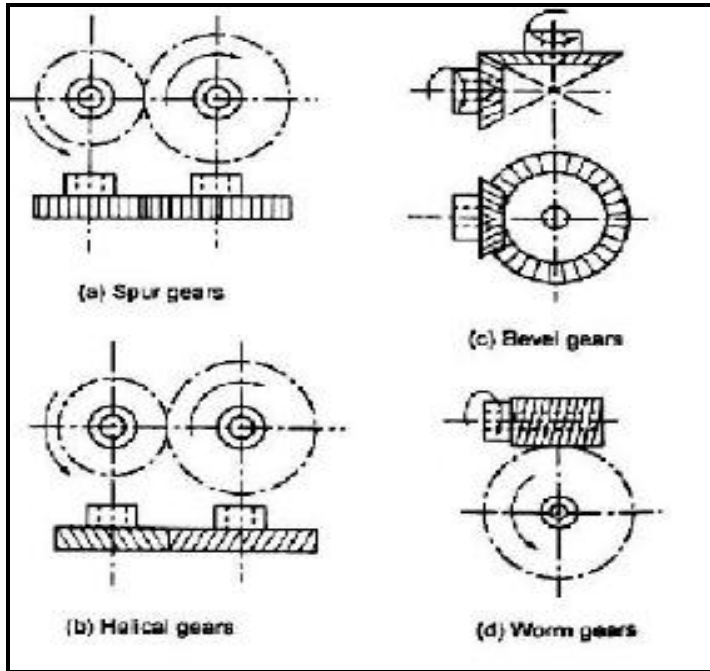
(iv) Curved teeth gears

5. Classification based on the type of gearing:

(i) External gearing

(ii) Internal gearing

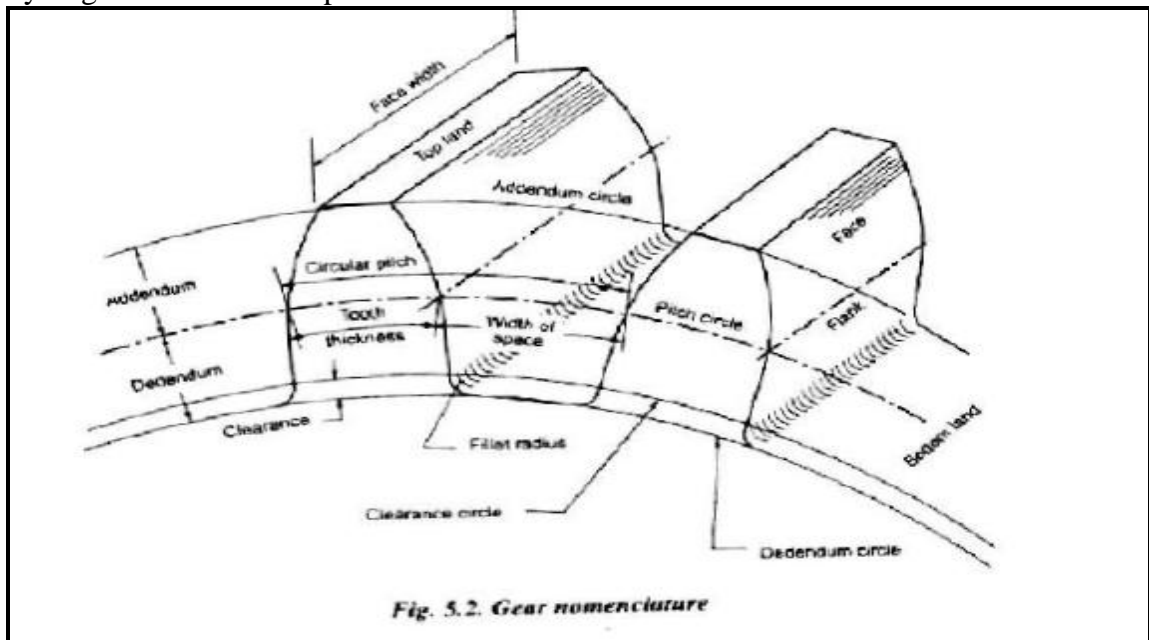
(iii) Rack and pinion



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



(a) Circular pitch (P_c):

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.

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

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.

$$\text{Diametral pitch, } P_d = \frac{Z}{D} = \frac{\pi}{P_c}$$

(c) Module pitch (m):

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

$$\text{Module, } m = \frac{D}{Z}$$

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

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

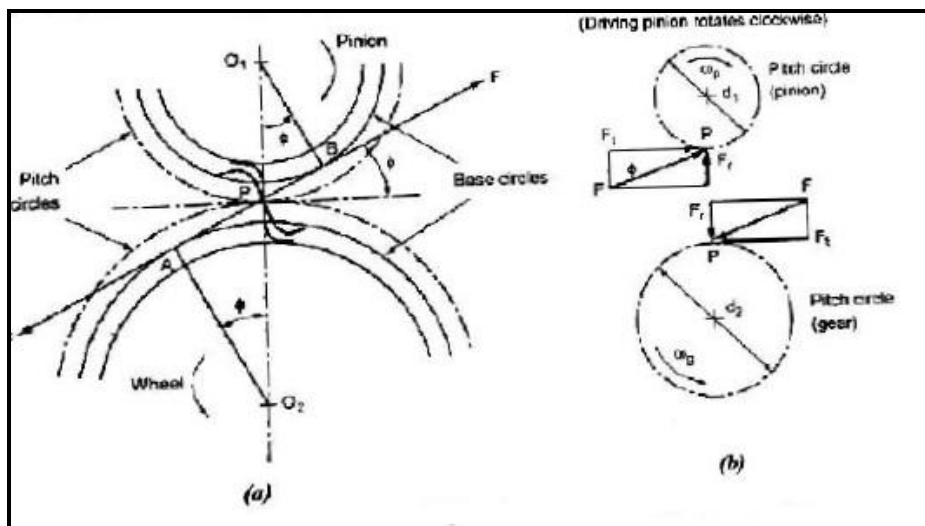
Where N_A and N_B = Speeds of driver and driven respectively, and,
 Z_A and Z_B = Number of teeth on driver and driven respectively,

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

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

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

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

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. $Z_2 = i \cdot z_1$. Where i = gear ratio.

3. Calculation of tangential load on tooth (F_t):

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

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 (F_d):

$$1. F_d = \left(\frac{F_t}{C_v} \right) \dots \dots \dots \text{(From data book page no. 8.50)}$$

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

5. Calculation of beam strength (Fs):

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

m = Module in mm.

F_s = 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):

$$1. F_s \geq F_d$$

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 **d₁ = z.m**

3. Pitch line velocity $v = \left(\frac{\pi d_1 N_1}{60} \right)$

8. Recalculation of beam strength (F_s):

$$1. F_s = \pi \cdot m \cdot b \cdot [\sigma_b] \cdot y$$

9. Calculation of accurate dynamic load (F_d):

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

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) :

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

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

F_w = Maximum wear load.

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

$$f = (2.8 * \text{BHN} - 70) \text{ N/mm}^2$$

d₁ = 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:**

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

3. Gear Life:

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

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

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

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

$$K \times K_d = 1.3$$

$$M_t = \text{Transmitted torque} = \frac{60 \times P}{2\pi N}$$

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

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

b. $[\sigma_b] = \left(\frac{1.4 K_{bl}}{n K_\sigma} \right) \sigma_{-1} \dots \dots [\text{PSG data book page no:8.18}]$

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. $[\sigma_c] = C_B \cdot H_B \cdot K_{cl} \dots \dots [\text{PSG data book page no: 8.16}]$

$C_B = \frac{C_E}{10} \dots \dots [\text{PSG data book page no: 8.16}]$

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):

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

Assume $\phi = .3$

7. Selection of number of teeth:

Assume $Z_1 = 17$

$$Z_2 = i \times Z_1$$

8. Calculation of module:

$$m = \frac{2a}{(z_1 + z_2)} \dots \dots \dots [\text{PSG data book page no: 8.22}]$$

Standard transverse module in PSG data book pg no: 8.2

9. Revision of center distance:

$$a = \frac{m(Z_1 + Z_2)}{2} \dots \dots \dots [\text{PSG data book page no: 8.22}]$$

10. Calculation of b, d, v and ϕ :

Face width, $b = \phi \cdot a$

pitch circle dia, $d_1 = m \cdot Z_1$

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

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

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 $[M_t]$:

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

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:

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

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:

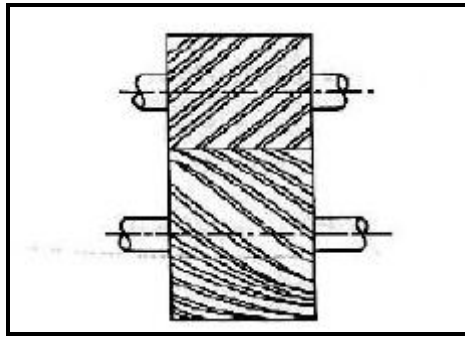
$$\sigma_c = 0.74 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{i b} E_{sq} [M_t]} \dots \dots \dots [\text{PSG data book page no: 8.13}]$$

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

15. Basic Dimensions:

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

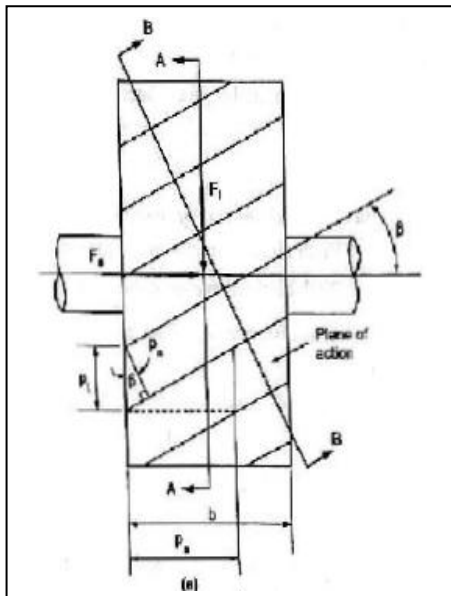
Helical Gears



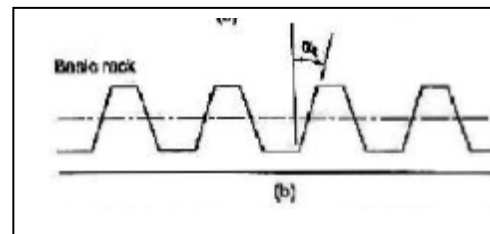
KINEMATICS AND NOMENCLATURE OF HELICAL GEARS

Let

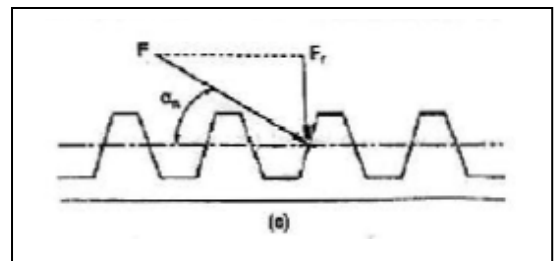
- β = Helix angle,
- p_t = Transverse circular pitch,
- p_n = Normal circular pitch,
- p_a = Axial pitch,
- p_d = Diametral pitch,
- α_t and α_n = Transverse and normal pressure angles respectively,
- m_t and m_n = Transverse and normal modulus respectively,
- z_1 and z_2 = Number of teeth on pinion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

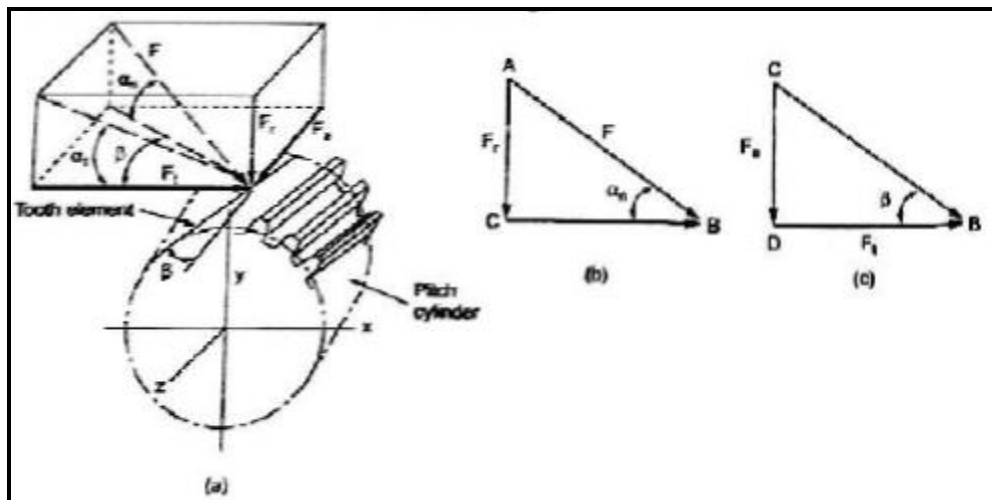
- Normal Pressure angle (α_n) = 15° to 25°

- 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

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

FORCE ANALYSIS ON HELICAL GEARS



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. $z_2 = i \cdot z_1$. Where i = gear ratio.

3. **Calculation of tangential load on tooth (F_t):**

$$3. F_t = \left(\frac{P \cdot K_0}{V} \right)$$

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 (F_d):**

$$4. F_d = \left(\frac{F_t}{C_v} \right) \dots \dots \dots \text{(From data book page no. 8.50)}$$

$$5. C_v = \frac{6}{6 + V} \text{ (Assume } V=15) \dots \text{ (From data book page no. 8.51)}$$

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

$$F_s = \pi \cdot m_n \cdot b \cdot [\sigma_b] \cdot y' \dots \text{(From data book page no. 8.51)}$$

m_n = Normal module in mm.

F_s = Strength of gear tooth.

$[\sigma_b]$ = Allowable static stress.

b = Face width = $10m_n$.

y' = Form factor = $\left(0.154 - \frac{0.912}{z_{eq}} \right)$ for 20° involute.

$$Z_{eq} = \frac{Z_1}{\cos^3 \beta}$$

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 :

$$1. \text{ Face width } b = 10m_n.$$

$$2. \text{ Pitch circle diameter } d_1 = \left(\frac{m_n}{\cos \beta} \times Z_1 \right) \dots \text{ (From pg no. 8.22)}$$

$$3. \text{ Pitch line velocity } v = \left(\frac{\pi d_1 N_1}{60} \right)$$

8. Recalculation of beam strength (F_s):

$$1. F_s = \pi \cdot m \cdot b \cdot [\sigma_b] \cdot y'$$

9. Calculation of accurate dynamic load (F_d):

$$F_d = F_t + \frac{21v(bc \cos^2 \beta + F_t) \cos \beta}{21v + \sqrt{bc \cos^2 \beta + F_t}} \dots \text{ (From data book page no. 8.51)}$$

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 :

- Compare F_d and F_s
- If $F_s \geq F_d$, Design is safe and satisfactory.

11. Calculation of maximum wear load (F_w) :

$$F_w = \left(\frac{d_1 \cdot b \cdot Q \cdot K_w}{\cos^2 \beta} \right) \dots \dots \text{ (From data book page no. 8.51)}$$

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

F_w = Maximum wear load.

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

$$f = (2.8 * \text{BHN} - 70) \text{ N/mm}^2$$

d_1 = Pitch circle diameter.

B = Face width.

12. Check for wear:

- Compare F_d and F_w
- 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:**

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

3. **Gear Life:**

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

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

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

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

$$K \times K_d = 1.3$$

$$M_t = \text{Transmitted torque} = \frac{60 \times P}{2\pi N}$$

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

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

- e. $[\sigma_b] = \left(\frac{1.4 K_{bl}}{n K_\sigma} \right) \sigma_{-1} \dots [\text{PSG data book page no:8.18}]$

$$K_{bl} = \text{Life factor for bending} \dots [\text{PSG data book page no:8.20}]$$

$$K_\sigma = \text{Stress concentration factor} \dots [\text{PSG data book page no:8.19}]$$

$$n = \text{Factor of safety} \dots [\text{PSG data book page no:8.19}]$$

$$\sigma_{-1} = \text{Endurance limit stress} \dots [\text{PSG data book page no: 8.19}]$$

- f. $[\sigma_c] = C_B \cdot H_B \cdot K_{cl} \dots [\text{PSG data book page no: 8.16}]$

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

$$H_B = \text{Brinell hardness number} \dots [\text{PSG data book page no: 8.16}]$$

$$K_{cl} = \text{Life factor for surface strength} \dots [\text{PSG data book page no: 8.17}]$$

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

$$a \geq (i+1) \sqrt[3]{\left[\frac{0.7}{[\sigma_c]} \right]^2 \times \frac{E_{eq}[M_t]}{i \cdot \phi}} \dots [\text{PSG data book page no: 8.13}]$$

$$\text{Assume } \phi = \frac{b}{a} = 0.3$$

7. Selection of number of teeth:

Assume $Z_1 = 17$

$$Z_2 = i \times Z_1$$

8. Calculation of module:

$$m_n = \frac{2a}{(z_1 + z_2)} \times \cos \beta \dots \dots \dots [\text{PSG data book page no: 8.22}]$$

where β = helix angle.

Standard transverse module in PSG data book pg no: 8.2

9. Calculation of center distance:

$$a = \frac{m_n (Z_1 + Z_2)}{2 \cos \beta} \dots \dots \dots [\text{PSG data book page no: 8.22}]$$

10. Calculation of b, d , v and ϕ :

Face width, $b = \phi \cdot a$

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

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

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

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_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:

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

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:

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

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

15. Basic Dimensions:

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

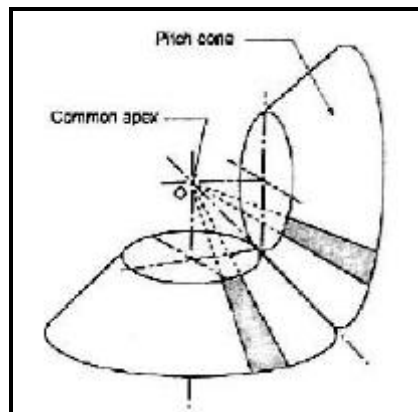
UNIT– II: SPUR GEARS AND PARALLEL AXIS HELICAL GEARS (PART - A)
1. What is pressure angle? What is the effect of increase in pressure angle? (May/june 2014)
<p>Soln.</p> <p>It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. The standard pressure angle is $14\frac{1}{2}^{\circ}$ and 20°</p>
2. What condition must be satisfied in order that a pair of spur gears may have a constant velocity ratio? (May/june 2014)
<p>Soln.</p> <p>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.</p>
3. What are the profiles of spur gear (May/june 2016)
<p>Soln.</p> <ol style="list-style-type: none"> 1. Involute tooth profile 2. Cycloidal tooth profile
4. What are the main types of gear tooth failure? (May/june2013) (May/june2012)
<p>Soln. The two modes of gear tooth failures are:</p> <ol style="list-style-type: none"> 1. Tooth breakage (due to static and dynamic loads),and 2. Tooth wear (or) surface deterioration <ol style="list-style-type: none"> (a). abrasion, (b). pitting, and (c). scoring or seizure
5. Define the various pitch in a helical gear. (May/June 2012)
<p>Soln.</p> <ol style="list-style-type: none"> 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)
<p>Soln.</p> <p>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.</p> <p>Transmitted load, $= W_t = F_t$</p> <p>Radial component (F_r): 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</p>
9. Compare the contact between mating teeth of spur and helical gears. (April/May 2010)
<p>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.</p> <p>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.</p>
10. why is a gear tooth subjected to dynamic loading (April/May 2015)
<p>Soln.</p> <p>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:</p> <p>(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.</p>

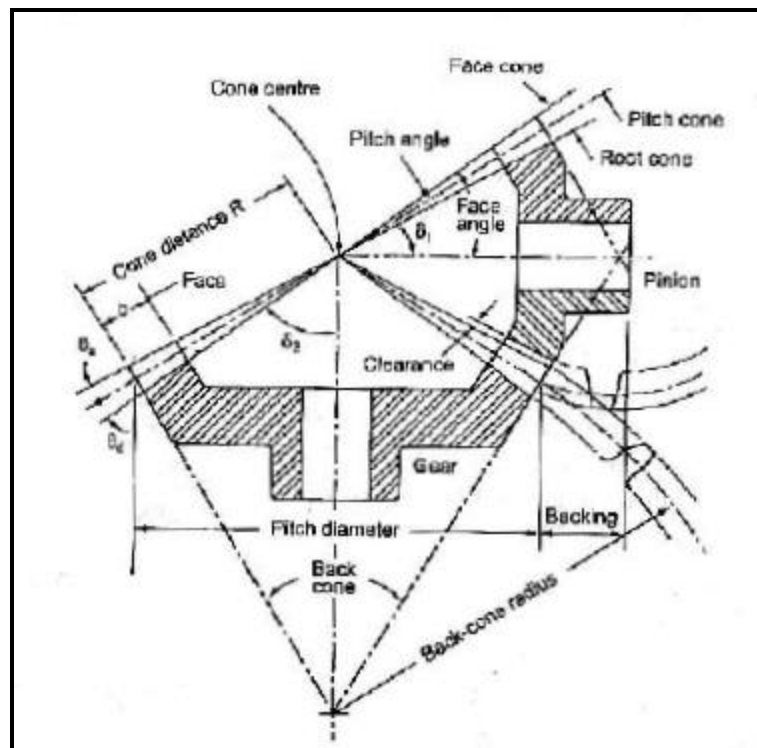
UNIT-III

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. $Z_2 = i \cdot z_1$. 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: $\tan \delta_2 = i$ and $\delta_1 = 90^\circ - \delta_2, \dots$ (From data book page no. 8.39)

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

4. Calculation of tangential load on tooth (F_t):

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

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 (F_d):

$$1. F_d = \left(\frac{F_t}{C_v} \right), \dots \text{ (From data book page no. 8.50)}$$

$$2. C_v = \frac{5.6}{5.6 + \sqrt{V}} \text{ (Assume } V=5)$$

6. Calculation of beam strength (F_s):

$$F_s = \pi \cdot m_t \cdot b \cdot [\sigma_b] \cdot y' \cdot \left(\frac{R-b}{R} \right), \dots \text{ (From data book page no. 8.52)}$$

m_t = Transverse module in mm.

F_s = Strength of gear tooth.

$[\sigma_b]$ = Allowable static stress.

b = Face width.

y' = Form factor = $\left(0.154 - \frac{0.912}{z_{v1}} \right)$ for 20° involute.... (Pg no. 8.50)

$$R = \sqrt{\left(\frac{d_1}{2} \right)^2 + \left(\frac{d_2}{2} \right)^2}$$

7. Calculation of module (m_t):

$$1. F_s \geq F_d$$

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 :

$$1. \text{ Face width } b = 10m_t.$$

$$2. \text{ Pitch circle diameter } d_1 = m_t \cdot z_1$$

$$3. \text{ Pitch line velocity } v = \left(\frac{\pi d_1 N_1}{60} \right)$$

9. Recalculation of beam strength (F_s):

$$1. F_s = \pi \cdot m_t \cdot b \cdot [\sigma_b] \cdot y' \cdot \left(\frac{R-b}{R} \right)$$

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) :

$$F_w = \left(\frac{0.75.d_1.b.Q'.K_w}{\cos \delta_1} \right)$$

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

F_w = Maximum wear load.

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

$$f = (2.8 * \text{BHN} - 70) \text{ N/mm}^2$$

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:

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

$$i = \tan \delta_2$$

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

3. Gear Life:

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

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

4. Calculation of Initial design Torque: [M_t]

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

$$K \times K_d = 1.3$$

$$M_t = \text{Transmitted torque} = \frac{60 \times P}{2\pi N}$$

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

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

h. $[\sigma_b] = \left(\frac{1.4 K_{bl}}{n K_\sigma}\right) \sigma_{-1}$ [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]

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

$C_B = \frac{C_B}{10}$ [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 cone radius:

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

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

7. Selection of number of teeth:

Assume $Z_1 = 17$

$$Z_2 = i \times Z_1$$

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

8. Transverse module:

$$m_t = \frac{R}{0.5 \sqrt{Z_1^2 + Z_2^2}} \dots\dots\dots[\text{PSG data book page no: 8.38}]$$

Standard transverse module in PSG data book pg no: 8.2

9. Revision of cone distance:

$$R = 0.5 m_t \sqrt{Z_1^2 + Z_2^2} \dots\dots[\text{PSG data book page no: 8.38}]$$

10. Calculation of b , m_{av} , d_{1av} , v , φ_y :

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

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

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

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

$$\phi_y = \frac{b}{d_{av}}$$

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:

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

Y_v from **PSG data book page no: 8.18** for value of Z_v

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

14. Check for wear strength:

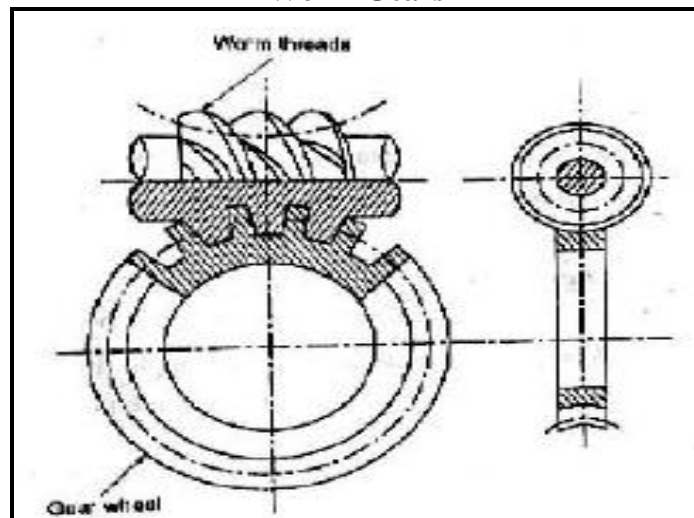
$$\sigma_c = \frac{0.72}{(R - 0.5b)} \sqrt{\left[\frac{\sqrt{(i^2 + 1)^3}}{i \times b} \times E_{sq} [M_t] \right]} \dots\dots\dots[\text{PSG data book page no:8.13}]$$

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

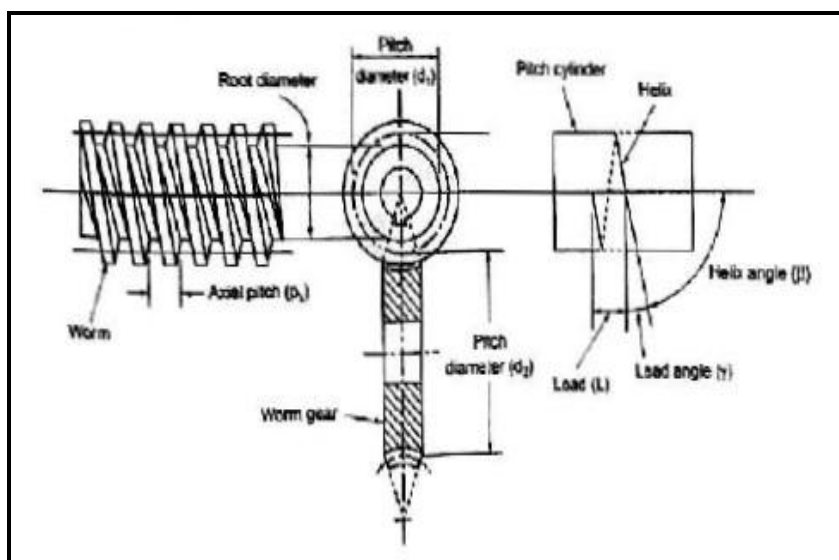
15. Basic Dimensions:

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

$H_g = (1 - \eta) \times \text{Input power}$,

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

K_t = Heat transfer coefficient of housing walls ($\text{W/m}^2\text{8C}$),

A = Effective surface area of the housing (m^2),

t_o = Temperature of lubricating oil ($^{\circ}\text{C}$), and

t_a = Temperature of the atmospheric air ($^{\circ}\text{C}$).

Therefore $(1 - \eta) \times \text{Input power} = K_t \times A \times (t_o - t_a)$

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. $Z_2 = i \times Z_1$. Where i = gear ratio.

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

Diameter factor, $q = \frac{d_1}{m_x}$. If not assume $q = 11$

Lead angle, $\gamma = \tan^{-1}\left(\frac{z_1}{q}\right)$

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

$$1. F_t = \left(\frac{P \times K_0}{v} \right)$$

P= Transmitter power in watts.
V= Pitch line velocity in m/s.
K₀ = Service factor (Assume 1.25).

5. Calculation of initial dynamic load (F_d):

1. $F_d = \left(\frac{F_t}{C_v} \right) \dots \dots \dots$ (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 (F_s):

$$F_s = \pi \cdot m_x \cdot b \cdot [\sigma_b] \cdot y' \dots \dots \text{(From data book page no. 8.52)}$$

m_x= Axial module in mm.
F_s = Strength of gear tooth.
[σ_b] = Allowable static stress.

b = Face width = **8.25m_x**

y' = Form factor = $\left(\frac{y}{\pi} \right) \dots \dots \text{(From data book page no. 8.52)}$

7. Calculation of axial module (m_x):

1. $F_s \geq F_d$

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 **d₁ = m_x * z₂**
3. Pitch line velocity $v = \left(\frac{\pi d_1 N_1}{60} \right)$

9. Recalculation of beam strength (F_s):

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

10. Calculation of accurate dynamic load (F_d):

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

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) :

$$F_w = d_2 * b * K_w \dots \dots \dots \text{(From data book page no. 8.52)}$$

F_w= Maximum wear load.

$$K_w = \left(\frac{\text{Pressure angle}}{10} \right) \dots \dots \dots \text{(From data book page no. 8.54)}$$

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. $\eta = 0.96 \left(\frac{\tan \gamma}{\tan(\gamma + \rho)} \right) \dots \dots \dots$ (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)
- $(1 - \eta) * P = K_t * (\Delta t) * A$
- $(\Delta t) = \text{Oil temp} - \text{Air temp}$
- $K_t = \text{Heat transfer coeff.}$

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: $[M_t]$

$$[M_t] = M_t \times K \times K_d \dots \dots \dots [\text{PSG data book page no: 8.13}]$$

Assume $K \times K_d = 1$

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

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):

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

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:

$$a = 0.5 m_x (q + Z_2) \dots \dots \dots [\text{PSG data book page no: 8.43}]$$

8. Calculation of b, d, v, γ, v_3 :

$$\text{Pitch diameter: } d_1 = q \times m_x$$

$$d_2 = Z_2 \times m_x$$

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

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

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

Assume $q = 11$

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

9. Recalculation 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:

$$\sigma_b = \frac{1.9 [M_t]}{m_x^3 \times q \times Z_2 \times y_v} \dots \dots \dots [\text{PSG data book page no: 8.44}]$$

Y_v 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:

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

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

13. Check for efficiency:

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

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

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]

UNIT– III: BEVEL, WORM AND CROSS HELICAL GEARS (PART - A)
1. Define the following terms: (a) cone distance (b) face angle. (May/ June 2014)
<p>Soln. cone distance (R): it is the length of the pitch cone element mathematically, cone distance (R)</p> $R = \frac{\text{pitch radius}}{\sin \delta}$ <p>Tip or face angle: it is the angle subtended by the face of the tooth at the cone centre mathematically,</p> <p>Tip angle = pitch angle + addendum angle</p>
2. What is virtual number of teeth in bevel gears? (May/ June 2014)
<p>Soln. Soln. on order to simplify the design calculation and analysis, bevel gears are replaced equivalent spur gear. An imaginary spur gear considered in a plane perpendicular to the tooth at the larger end, is known as virtual or formative or equivalent spur gear.</p>
3. Where do we use worm gears? (May/June 2013)
<p>Soln.</p> <p>When we require to transmit power between nonparallel and non-intersecting shafts and very high</p> <p>Velocity ratio, of about 100, worm gears, can be employed. Also worm-gears provide self-locking Facility</p>
4. What is helical angle of worm? (May/ Jun 2016)
<p>Soln. Helical angle is the angle between any helix and an axial line on its right, circular cylinder or cone. Common applications are <u>screws</u>, <u>helical gears</u>, and <u>worm gears</u>. The helical angle is measured in degrees.</p>
5. What are the main losses in the worm gear drive? (May/June 2012)
<p>Soln.</p> <p>Merits</p> <ol style="list-style-type: none"> 1) Used for very high velocity ratio of about 100 2) Smooth and noiseless operation. 3) Self-locking facility is available. <p>Demerits</p> <ol style="list-style-type: none"> 1) Low efficiency. 2) More heat will be produced and hence this drive can be operated inside an oil reservoir or extra <p>Cooling fan is required in order to dissipate the heat from the drive.</p> <ol style="list-style-type: none"> 3) Low power transmission.

UNIT– III: BEVEL, WORM AND CROSS HELICAL GEARS (PART - A)
6. How Bevel gears are manufactured? (May/Jun 2016)
<p>Soln. Bevel gears are manufactured by following methods. They are</p> <ul style="list-style-type: none"> a) Gear Milling b) Gear Hobbing c) Gear Shaping d) Bevel Gear Cutting
7. A pair of worm gears is designated as R2/54/10/5. Find the gear ratio (Nov/Dec 2012)
<p>Soln. for example, R2/54/10/5 worm drive means, a right hand worm of star 2, meshes with a worm wheel of 54 teeth and of diameter quotient 10, and with module 5mm</p>
8. Why is phosphor bronze widely used for worm gear? (Apr/May 2015)
<p>Soln. The phosphor bronze is widely used for worms drive in order to reduce wear of the worms which will be excessive with cast iron or steel.</p>
9. What are the forces acting on bevel gears?
<p>Soln. the components of the resultant forces are:</p> <ul style="list-style-type: none"> 1. tangential or useful component (F_t), and 2. Separating force (F_s): it is resolved into two components. They are <ul style="list-style-type: none"> (i). Axial force (F_a), and (ii). Radial force (F_r).
10. List out the main types of failure in worm gears. (Apr/May 2011)
<p>Soln. the different worm gear tooth failures are:</p> <ul style="list-style-type: none"> (i). Seizure (ii). Pitting and rupture

UNIT-IV:

GEAR BOXES

Procedure for typical gear box design.

Step 1: Selection of spindle speeds:

1. Determine the progression ratio = $\phi^{n-1} = \frac{N_{max}}{N_{min}}$

(where n = number of speed)

2. Find step ratio series and speed of gears (PSG Data book pg no. 7.20)

3. Write the structural formulae

$$Z_1 = p_1(x_2) * p_2(x_2) * p_3(x_3) * p_4(x_4)$$

$$X_1 = 1, X_2 = p_1, X_3 = p_1 p_2, X_4 = p_1 p_2 p_3$$

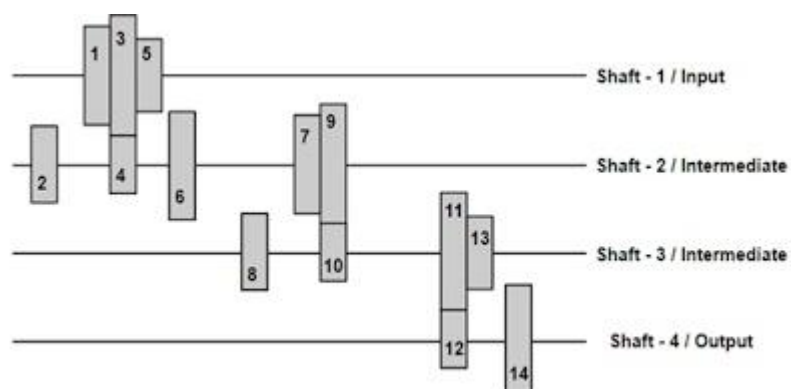
<u>Preferred Structural Formulas</u>	
6 speeds	2 x 3 (or) 3 x 2
8 speeds	2 x 4 (or) 4 x 2 (or) 2 x 2 x 2
9 speeds	3 x 3
12 speeds	3 x 2 x 2 (or) 2 x 2 x 3 (or) 2 x 3 x 2
16 speeds	4 x 2 x 2 (or) 2 x 4 x 2 (or) 2 x 2 x 4

Step 2:

Draw the ray diagram *or* speed diagram

Step 3:

Draw the kinematic diagram.



Step 4:

Calculate the number of teeth.

Step 5: Select the suitable materials

Materials constant PSG.1.15

Materials	Materials constant (M)
C45	30
15 Ni 2Cr 1 Mo 15	80
40 Ni 2Cr 1 Mo 28	100

Permissible shear stress(τ) N/mm²

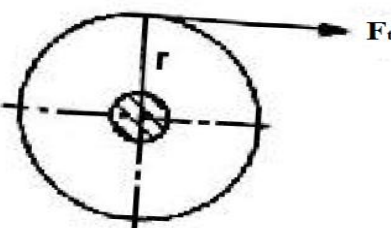
S.No	Shaft materials	(τ) N/mm ²
1.	C14 (as supplied)	25
2.	C45 (case hardened)	30
3.	Low carbon alloyed steel (case hardened)	40
4.	40 Ni 2Cr 1 Mo 28 (hardened and tempered)	55

Step 6: calculation of module

Calculate the torque for the gear which has the lowest speed using the relation,

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

Calculate the tangential force on the gear in terms of module using the relation



Gear

$$F_t = \frac{T}{r} = \frac{2T}{zm}$$

$$[T = F_t \times r \text{ and } r = \frac{zm}{2}]$$

Now calculate the module using the relation

$$m = \sqrt{(F_t / \phi_m) * M}$$

ϕ_m = Ratio between the face width and module = $\frac{b}{m} = 10$

M = Material constant

Standard module in PSG data book pg no: 8.2

Step 7: Calculation of centre distance in all stages:

Calculate the centre distance in each stage by using the relation

$$a = \left[\frac{Z_x + Z_y}{2} \right] m$$

Z_x and Z_y = Number of teeth on the gear pair in engagement in each stage

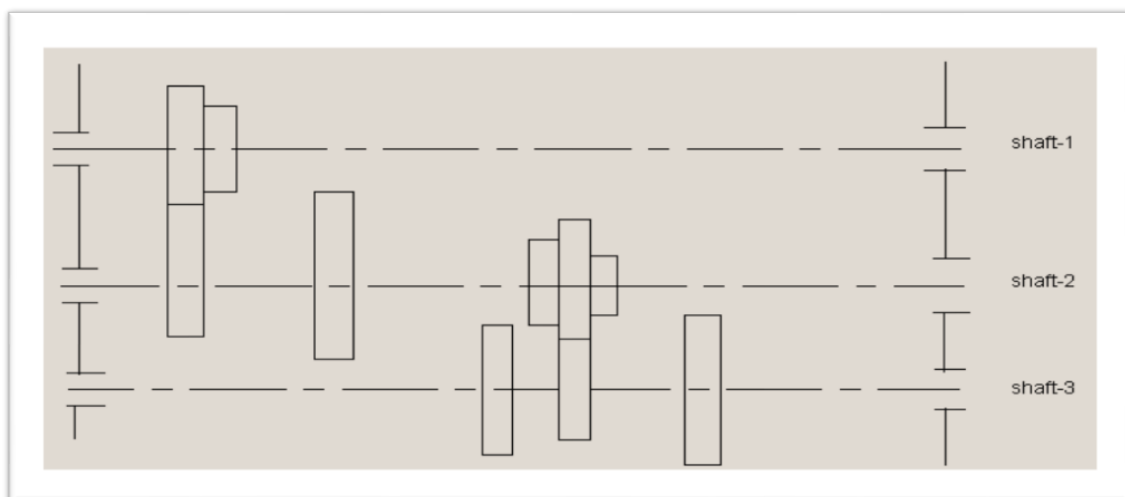
Step 8: calculation of face width: $b=10*m$

Step 9: calculation of distance between the bearings ie., length of shafts:

Calculate the distance between the bearings by using the following assumptions

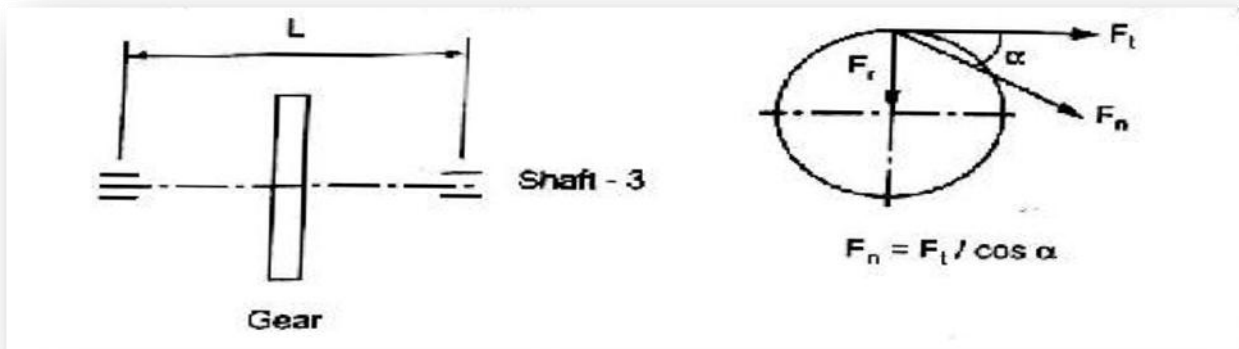
ASSUMPTIONS:
Give 10mm clearance between the gear and the bearings on both sides.
Take the distance between the adjacent groups of gears as 20 mm.
Take the total length for two pairs gear group as 4b and for three pairs gear group as 7b as shown in figure.
Assume the width of the bearings as 25mm

Distance between the bearings is given by $L=25+10+4b$ (or $7b$) + 20 (or $4b$) + 10 + 25



Step 10: Design of shafts:

(i) **Design of spindle i.e., output shaft:** Design the output shaft for maximum bending moment by considering the shaft as simply supported on bearings.



Step 11 Calculate the maximum bending moment due to normal load (F_n) using the relation

$$M = \frac{F_n \times L}{4}$$

Where F_n = Normal load on gear = $\frac{F_t}{\cos \alpha}$

Step 12 Calculate the equivalent torque using the relation

$$T_{eq} = \sqrt{M^2 + T^2}$$

Where T = Torque on the spindle = $\frac{P \times 60}{2\pi N_{low}}$

Step 13 Calculate the diameter of the spindle using the relation

$$d_s = \left[\frac{16 \times T_{eq}}{\pi [\tau]} \right]^{\frac{1}{3}}$$

Where $[\tau]$ = Permissible shear stress,

T = Torque on the spindle = $\frac{P \times 60}{2\pi N}$

Step 14 Design of other shafts: Determine the diameter of the input and intermediate shafts using the relation.

$$T = 0.2 d_s^3 [\tau]$$

UNIT– IV: GEAR BOXES (PART - A)

1. What is step ratio? Name the series in which speeds of multispeed gear box are arranged. (May/June 2014)

Soln. Step ratio is the ratio of one speed of the shaft to its previous lower speed. Since the spindle speeds are arranged in geometric progression, the ratios adjacent speeds (i.e., step ratios) are constant.

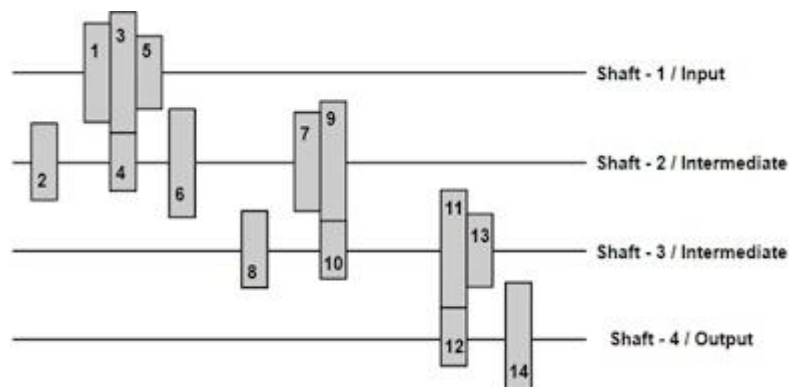
If N_r is the maximum speed and N is the minimum speed, then,

$$\frac{N_r}{N} = (Step\ ratio)^r = r^r - 1$$

Basic series	Step ratio(ϕ)
R5	$\sqrt[5]{10} = 1.58$
R10	$\sqrt[10]{10} = 1.26$
R 20	$\sqrt[20]{10} = 1.12$
R 40	$\sqrt[40]{10} = 1.06$
R 80	$\sqrt[80]{10} = 1.03$

2. Sketch the kinematic layout of gears for 3 speeds between two shafts. (May/June 2014)

Soln.



3. What are preferred numbers? (May/June 2013)

Soln. The series of preferred number is obtained by multiplying a step ratio with the first number to get the second number. The third number is obtained by multiplying a step ratio with the second number. Similarly the procedure is continued until the series is completed. (from data book page no.7.20)

4. List four applications where constant mesh gear box is used. (Nov/Dec 2012)

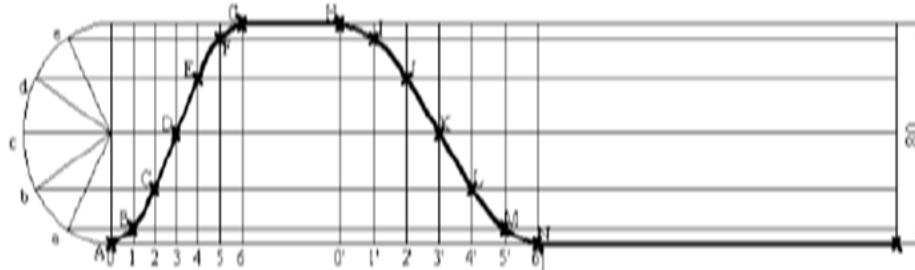
Soln.

1. Automobile
2. Rolling mill
3. Machine tools
4. Crane

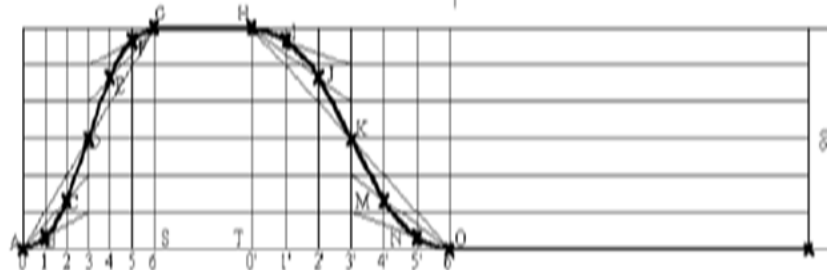
UNIT– IV: GEAR BOXES (PART - A)		
5. Which type of gear is used in constant mesh gear box? Justify. (Nov/Dec 2012)		
Soln. Helical gears are used in constant mesh gear boxes to provide quieter and smooth operation		
6. What are the possible arrangements to achieve 12 speeds from a gear box? (April/May2015)		
Soln.		
S.No	Number Of Speeds	Preferred Structural Formula
1.	12speed	<i>(i). 3(1) 2(3) 2(6)</i> <i>(ii). 2(1) 3(2) 2(6)</i> <i>(iii). 2(1) 2(2) 3(4)</i>
7. Define the term progression ratio? (April /May2015)		
Soln. When the spindle speeds are arranged in geometric progression, the ratio between the two adjacent speeds is known as step ratio or progression ratio.		
8. What are the points to be considered while designing a sliding mesh type of multi speed gear box? (April /May2010)		
Soln. i) The transmission ratio in a gear box is limited by $\frac{1}{4} < i < 2$ ii) Speed ratio of any stage should not be greater than 8.		
9. Where is multi-speed gear boxes employed? (Apr/May2011)		
Soln. (i) Automobiles (ii) Machine tools and (iii)Aeronautical.		
10. Where multispeed gearbox is used? (May/Jun 2016)		
Soln. Multi speed gearbox is used for speed adjustment at constant power level. Heavy duty Industrial Multi-speed gearboxes are designed for continuous operation.		

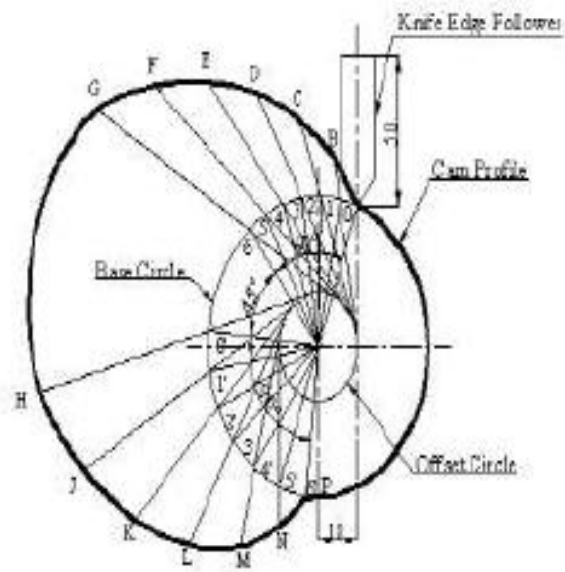
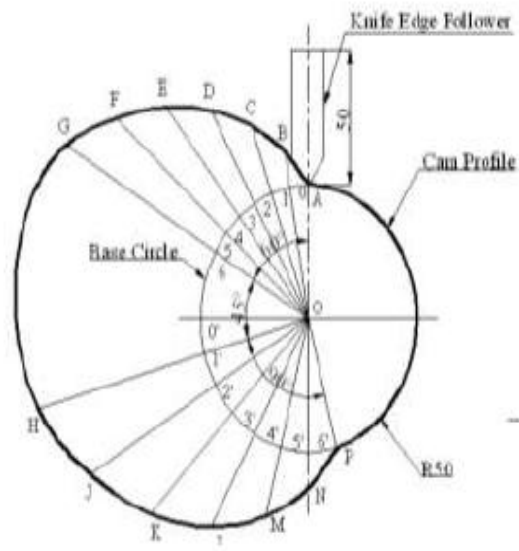
UNIT V
CAMS, CLUTCHES AND BRAKES
CAM TYPES

SIMPLE HARMONIC MOTIONS:



UNIFORM ACCELERATION & RETARDATION:





CLUTCHES

Design of a single plate clutch

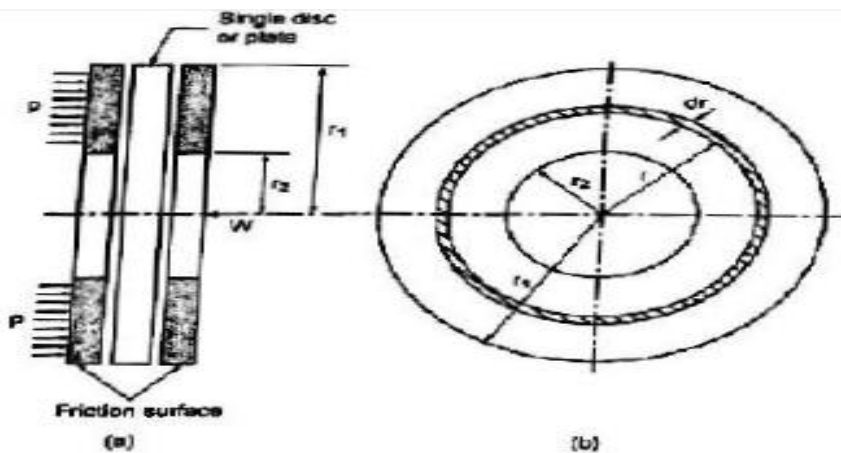
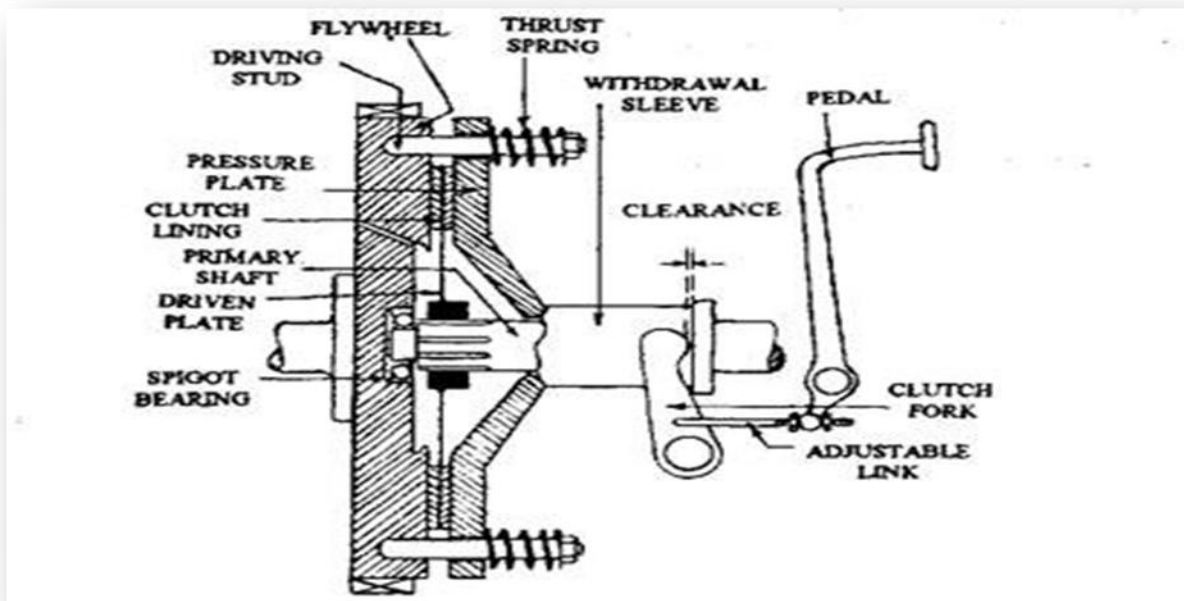


Fig. 10.3. Forces on a single disc or plate clutch

T = Torque transmitted by the clutch,

P = Intensity of axial pressure acting on contact surfaces,

r_1 = External radius of friction surface,

r_2 = Internal radius of friction surface.

Area of the elemental ring = $2\pi r.dr$

Normal or axial force on the ring, $\delta W r.dr = p \times 2\pi r.dr$

Friction torque acting on the ring, $t_r = 2\pi \mu p r^2.dr$

(i) Considering uniform pressure:

$$P = \frac{W}{A}$$

$$A = \pi(r_1^2 - r_2^2)$$

$$T = \mu WR$$

$$R = \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right]$$

(ii) Considering uniform wear

$$P = \frac{c}{r}$$

$$P_1 \cdot r_1 = P_2 \cdot r_2 = c$$

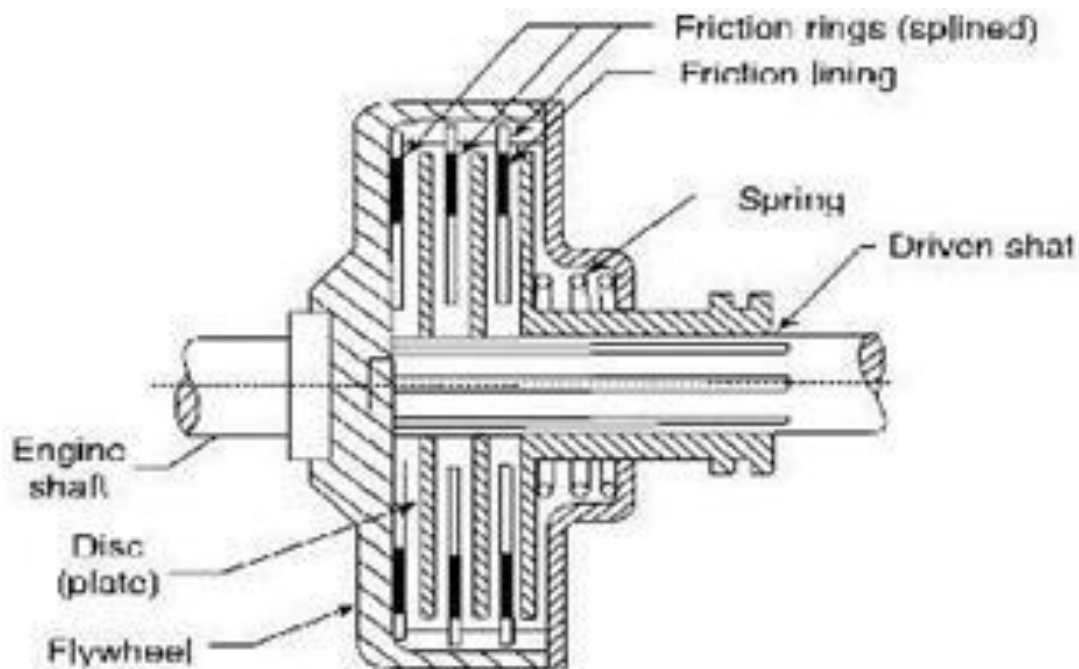
$$C = \frac{W}{2\pi(r_1 - r_2)}$$

$$T = \mu WR$$

$$R = \left[\frac{r_1 + r_2}{2} \right]$$

Design of a Multiplate clutch

(Torque transmitted on multiplate clutch)



n_1 = Number of discs on the driving shaft, and

n_2 = Number of discs on the driven shaft.

Number of pair of contact surfaces,

$$n = n_1 + n_2 - 1$$

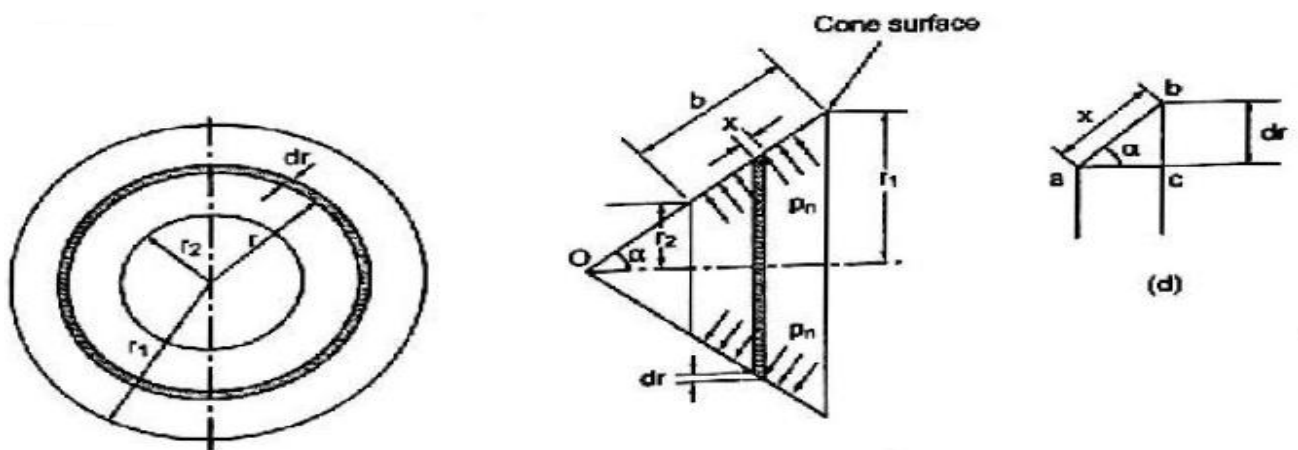
Total frictional torque on the clutch is given by

$$T = n\mu WR$$

$$R = \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] \text{ [For uniform pressure]}$$

$$R = \left[\frac{r_1 + r_2}{2} \right] \text{ [For uniform wear]}$$

Design of a cone clutch



Torque transmitted on the cone clutch is given by

$$T = \mu WR \operatorname{cosec} \alpha$$

$$R = \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right] \text{ [For uniform pressure]}$$

$$R = \left[\frac{r_1 + r_2}{2} \right] \text{ [For uniform wear]}$$

Axial force required at the engagement of clutch is given by

$$W_e = Wn (\sin \alpha + \mu \cos \alpha)$$

And axial force required at the disengagement of clutch is given by

$$W_e = Wn (\sin \alpha - \mu \cos \alpha)$$

BRAKE

SINGLE BLOCK OR SHOE BRAKE

The friction between the block and the brake drum causes the retarding of the drum. This type of Brake is commonly used on railway trains and tram cars.

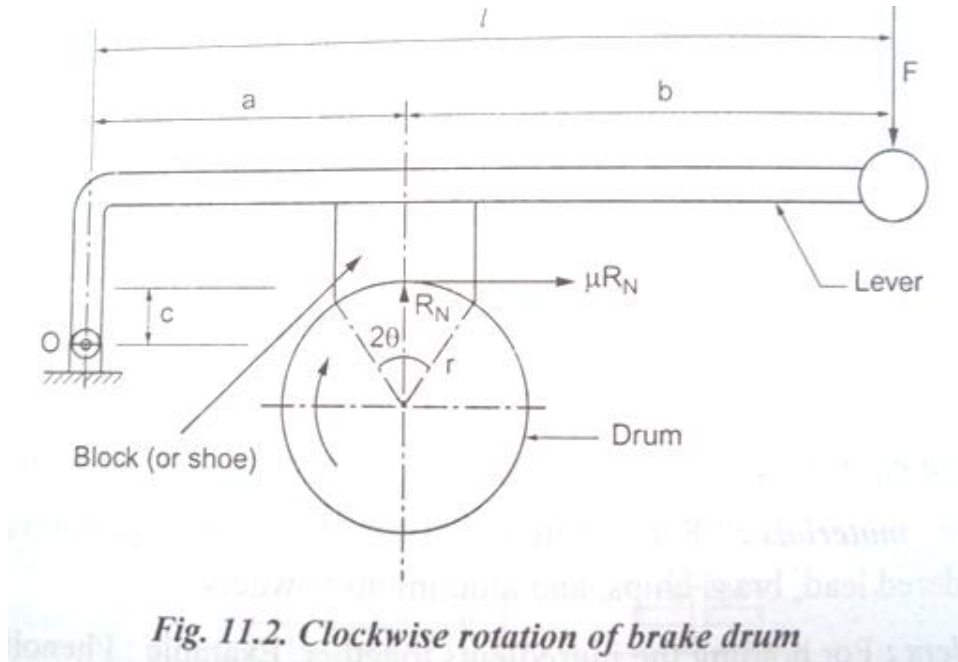


Fig. 11.2. Clockwise rotation of brake drum

The block is pressed against the drum by a force applied on one end of the lever. The other end of the lever is pivoted on a fixed fulcrum O.

Let r = Radius of the drum

$$R_N = \text{Normal reaction of the block} = \frac{F \cdot l}{a - \mu c} \text{ (clockwise)}$$

$$= \frac{F \cdot l}{a + \mu c} \text{ (anti-clockwise)}$$

F = Force applied at the lever end

μ = Coefficient of friction

μR_N = Frictional force

$$T_B = \text{Braking torque} = \mu R_N \cdot r = \mu \cdot \frac{F \cdot l \cdot r}{a - \mu c} \text{ (clockwise)}$$

$$= \mu \cdot \frac{F \cdot l \cdot r}{a + \mu c} \text{ (anti clockwise)}$$

DOUBLE BLOCK OR DOUBLE SHOE BRAKE

If only one block is used for braking, then there will be side thrust on the bearing of wheel shaft. This drawback can be removed by providing two blocks on the two sides of the drum, as shown in fig 11.7. This also doubles the braking torque. The double shoes on the drum reduce the unbalanced force on the shaft. The blocks or shoes are held on the drum by means of spring force.

Let S = Spring force required to set the blocks on the drum.

R = Radius of drum.

R_{N1} and μR_{N1} = Normal Reaction and the braking force on the left hand side shoe, and

R_{N2} and μR_{N2} = Normal Reaction and the braking force on the right hand side shoe.

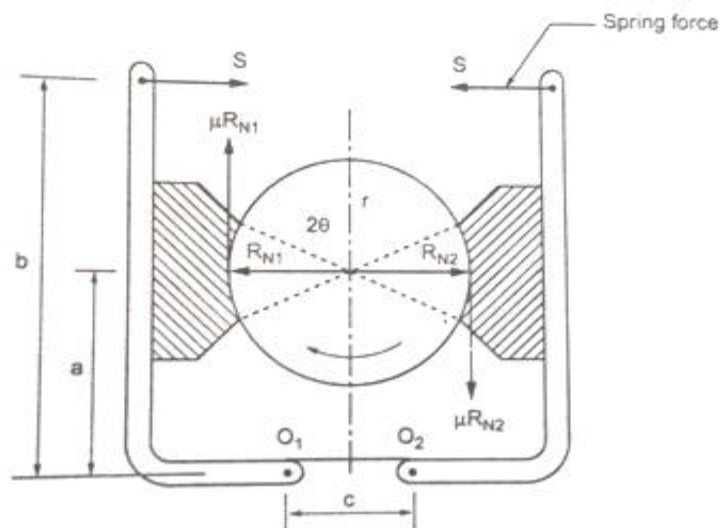


Fig. 11.7. Double shoe brake

DESIGN PROCEDURE FOR BLOCK BRAKE

Step 1: calculate the total energy absorbed by the brake

$$E_T = \frac{1}{2} m v^2 + \frac{1}{2} I \omega^2 + W X$$

Step 2: calculate the torque capacity by using the relation

$$T_B = \frac{60 E_T}{\pi N_i t}$$

N_i = Initial speed of brake drum and

t = Time of application of brake

Step 3: calculate the initial braking power by using the relation

$$P = \frac{2\pi N_f T_b}{60}$$

Step 4: Select the brake drum diameter

Step 5: Select the suitable brake drum and shoe materials. For the chosen materials, consulting Table 11.1, the coefficient of friction is obtained.

Table 11.1 properties of brake lining materials

Material	μ	Allowable pressure (p_{\max}) Mpa	Max.Temp. (°C)
Wood on metal	0.25	0.48	65
Metal on metal	0.25	1.4	315
Leather on metal	0.35	0.17	65
Asbestos on metal in oil	0.40	0.34	260
Powdered metal lining on C.I in oil.	0.15	2.8	260

Step 6: Consulting table 11.2 , calculate the induced bearing pressure (p)

Table 11.2 Limiting values of p_v (from PSG 7.130)

Operating conditions	P_v (mpa) (m/s)
Continuous service, poor heat dissipation	1.05
Intermittent service, poor heat dissipation	2.1
Continuous service, good heat dissipation as in oil bath	3.0

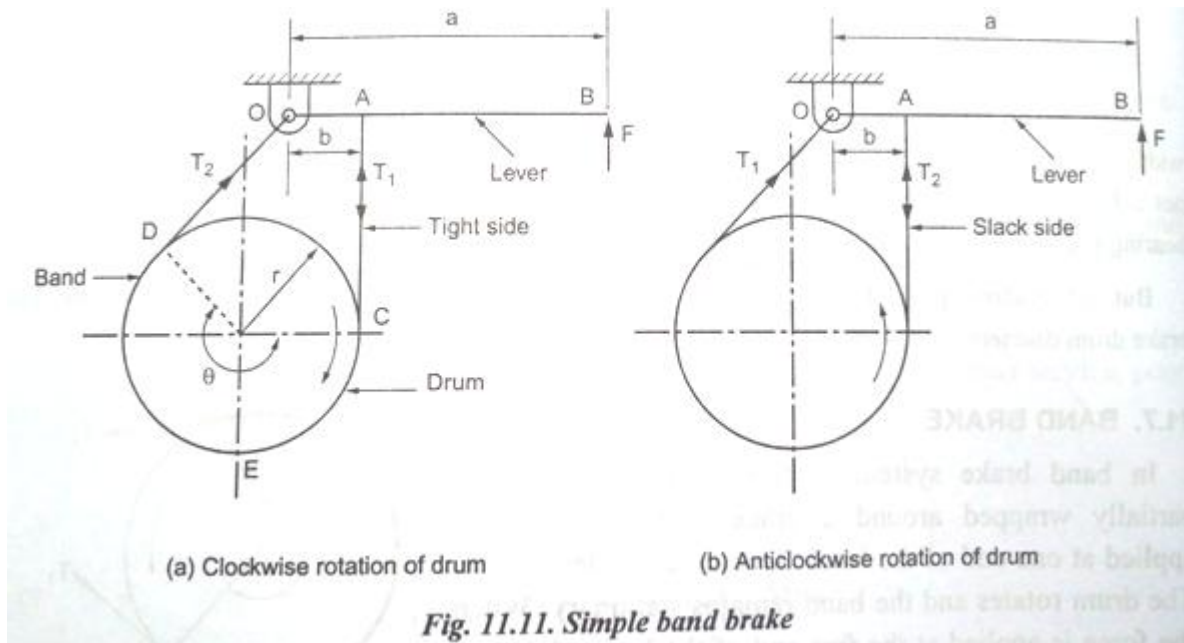
Step 7: calculate the projected area of the shoe by using the relation, $A = \frac{R_N}{P}$

Step 8: Finally calculate the breath and width of the shoe by using the relation projected area of the shoe,

$$A = \text{Breadth} \times \text{Width}$$

SIMPLE BAND BRAKES

The band or rope is wrapped round the cylindrical drum. When a force F is applied to the lever at B , the level turns about the fulcrum pin O and tightens the band on the drum and hence the brakes are applied. The friction between the band and the drum provides the braking torque.



Let F = Force applied on the lever,

R = Radius of the drum,

T = Thickness of the band,

R_f = Effective radius of the band = $r + \frac{t}{2}$

a = Length of lever = OB

b = Distance between the fulcrum O and point A .

For clockwise rotation of drum

$$F \cdot a = T_1 \cdot b$$

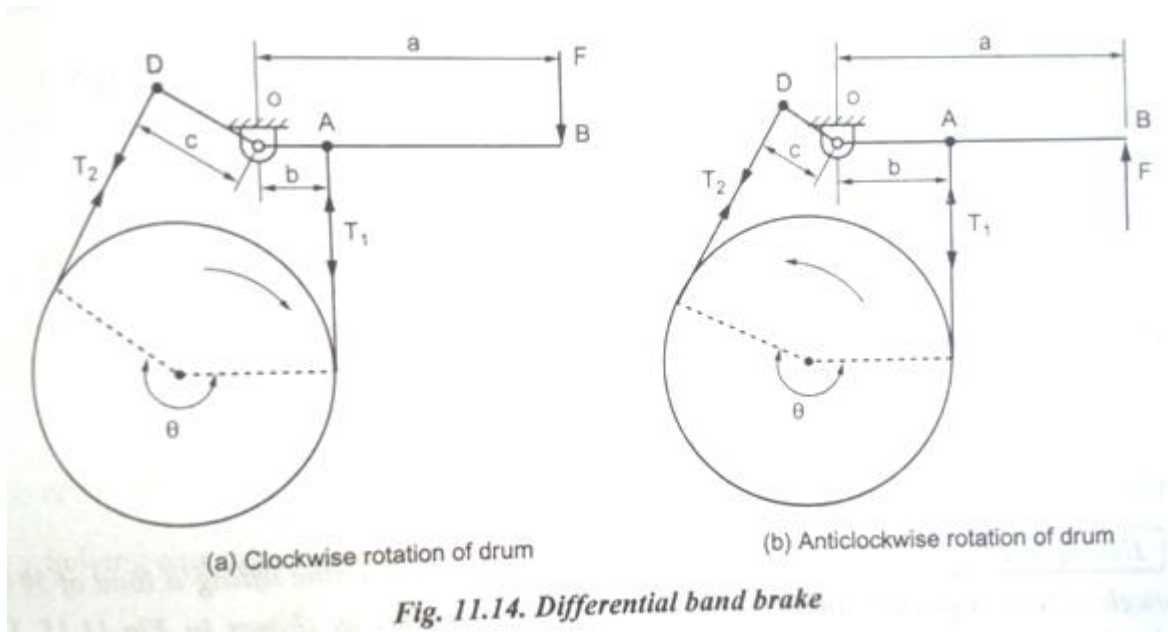
For anticlockwise rotation of drum

$$F \cdot a = T_2 \cdot b$$

$$\text{Braking torque } T_B = (T_1 - T_2) \cdot r = \left[T_1 - \frac{T_1}{e^{\mu \theta}} \right] r = F \times \frac{a}{b} \left[1 - \frac{1}{e^{\mu \theta}} \right] r$$

DIFFERENTIAL BAND BRAKE

In a differential band brake the ends of the band are joined to the lever at points D and A . Point D is the fulcrum. It may be noted that for the band to tighten, the length OD must be greater than the length OA .



- (i) Downward force on lever for clockwise rotation of drum:

This type of arrangement is shown in fig. 11.14(a). Taking moments about O, we get

$$F \cdot a = T_2 \cdot c - T_1 \cdot b$$

$$\text{Thus } T_2 \cdot c > T_1 \cdot b \quad (\text{or}) \quad \frac{c}{b} > \frac{T_1}{T_2}$$

Thus $c > b$ for the system to work satisfactorily.

If $\frac{c}{b} = \frac{T_1}{T_2}$, the external applied force $F = 0$, which is the condition for self-locking.

- (ii) Downward force on lever for anticlockwise rotation of drum:

Taking moments about O, we get

$$F \cdot a = T_1 \cdot b - T_2 \cdot c$$

$$\text{Thus } T_1 \cdot b > T_2 \cdot c \quad (\text{or}) \quad \frac{T_1}{T_2} > \frac{c}{b}$$

Condition for self-locking: If $T_1 \cdot b = T_2 \cdot c$, then external applied force $F = 0$

$$\frac{T_1}{T_2} = \frac{c}{b}$$

- (iii) Upward force on lever for anticlockwise rotation of drum:

This type of arrangement is shown in fig. 11.14(b). Taking moments about O, we get

$$F \cdot a = T_1 \cdot b - T_2 \cdot c$$

$$\text{Thus } T_1 \cdot b > T_2 \cdot c$$

$$\frac{T_1}{T_2} > \frac{c}{b}$$

- (iv) Upward force on lever for clockwise rotation of drum:

Taking Moment about O, we get,

$$F \cdot a = T_2 \cdot c - T_1 \cdot b$$

$$\text{Thus } T_2 \cdot c > T_1 \cdot b \text{ (or) } \frac{c}{b} > \frac{T_1}{T_2}$$

Condition for self-locking:

$$\text{If } \frac{T_1}{T_2} = \frac{c}{b} ; \text{ then } F = 0$$

In this case, c must be less than b for proper braking.

DESIGN PROCEDURE FOR BAND BRAKES

Step 1: Calculate the braking torque required from the data given.

Step 2: If not given, select the suitable diameter (D) of the brake drum, consulting table 11.3

Table 11.3 Dimensions of brake drum (from PSG 7.98)

Power of the motor, KW	Brake drum diameter, mm	Brake drum width, mm
7.36	160	50
11.04	200	65
14.72	250	80
25.76	320	100
44.16	400	125
73.6	500	160
110.4	630	200
184	800	250

Step 3: Determine the tight and slack side tensions.

Step 4: Calculate the thickness (t) of band: Take thickness of band as 0.005XDiameter of brake drum.

Step 5: calculate the band width (w)

$$\text{Induced tensile stress, } \sigma_t = \frac{T_1}{w \cdot t}$$

T_1 = Tight side tension in the band,

W = width of the band

t = Thickness of the band = 0.005 D

$$\sigma_t = \text{permissible tensile stress} = 50 \text{ to } 80 \text{ N/mm}^2$$

Step 6: check for bearing pressure

$$p_{\max} = \frac{T_1}{w \cdot r}$$

r = Radius of the drum

Step 7: calculate the force to be applied at the end of the lever

Table 11.4 Safe bearing pressure in band brakes (from PSG.7.98)

<i>Types of brake</i>	<i>Materials of the rubbing, surfaces</i>			
	<i>Steel band on C.I or steel drum</i>	<i>Asbestos brake band on steel or C.I drum</i>	<i>Rolled, pressed and shaped friction material on metal drum</i>	<i>Wood on C.I drum</i>
<i>Holding</i>	<i>1.5</i>	<i>0.6</i>	<i>0.8</i>	<i>0.6</i>
<i>Lowering</i>	<i>1.0</i>	<i>0.3</i>	<i>0.4</i>	<i>0.4</i>

BAND AND BLOCK BRAKE

This arrangement is a combination of both the band and the block brakes, as shown. The band is lined with a number of wooden blocks, each of which is in contact with the rim of the brake drum. When the brake is applied, the blocks are pressed against the drum. The advantage of using wooden blocks is that they provide higher coefficient of friction and they can be easily and economically replaced after being worn out.

- Let T_n = Tension in the band on tight side,
 T_o = Tension in the band on slack side,
 T_1 = Tension in band between the first and second block,
 T_2 = Tension in band between second and third block,
 T_3 = Tension in band between third and fourth blocks.
 n = Number of wooden blocks,
 μ = Coefficient of friction between block and drum.
 2θ = Angle subtended by each block at the drum centre.
 R_N = Normal reaction on the block.

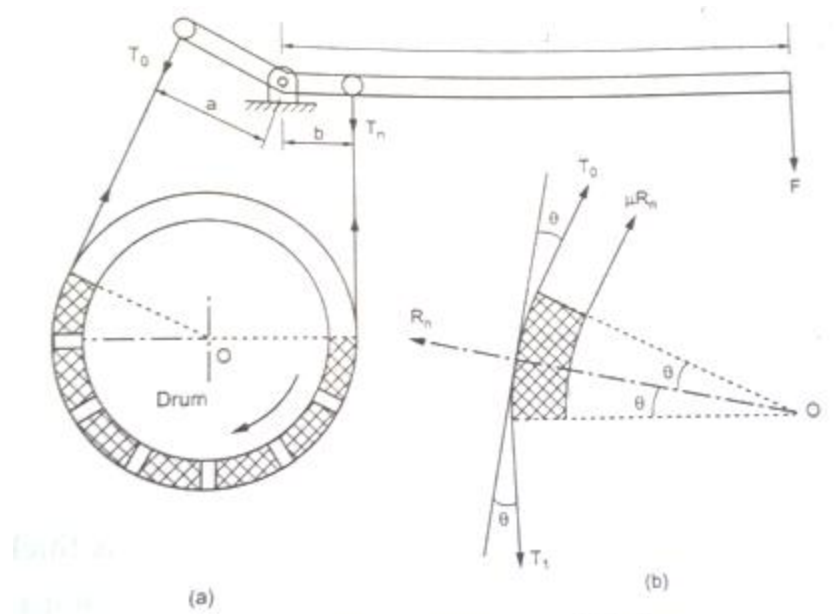


Fig. 11.19. Band and block brake

Resolving the forces radially, we get

$$(T_1 + T_0) \sin\theta = R_n$$

Resolving the forces tangentially, we get

$$(T_1 - T_0) \cos\theta = \mu R_n$$

Dividing equation (i) by (ii), we get

$$\mu \tan\theta = \frac{T_1 - T_0}{T_1 + T_0}$$

$$\frac{1 + \mu \tan\theta}{1 - \mu \tan\theta} = \frac{1 + \left(\frac{T_1 - T_0}{T_1 + T_0}\right)}{1 - \left(\frac{T_1 - T_0}{T_1 + T_0}\right)} = \frac{2 T_1}{2 T_0} = \frac{T_1}{T_0}$$

or

$$\frac{T_1}{T_0} = \frac{1 + \mu \tan\theta}{1 - \mu \tan\theta}$$

Similarly, it can be proved for each of the blocks that

$$\frac{T_2}{T_1} = \frac{1 + \mu \tan\theta}{1 - \mu \tan\theta} \quad \text{and} \quad \frac{T_3}{T_2} = \frac{1 + \mu \tan\theta}{1 - \mu \tan\theta}$$

Therefore,
$$\frac{T_1}{T_0} = \frac{T_2}{T_1} = \frac{T_3}{T_2} = \dots = \frac{T_n}{T_{n-1}} = \frac{1 + \mu \tan\theta}{1 - \mu \tan\theta}$$

So the ratio of tensions for all 'n' block is given by

$$\frac{T_n}{T_0} = \frac{T_1}{T_0} \times \frac{T_2}{T_1} \times \frac{T_3}{T_2} \times \dots \times \frac{T_n}{T_{n-1}} = \left[\frac{1 + \mu \tan \theta}{1 - \mu \tan \theta} \right]^n$$

Braking torque on the drum is given by

$$T_B = (T_1 - T_2) r$$

$$T_B = (T_1 - T_2) \left(\frac{d+2t}{2} \right)$$

DESIGN OF INTERNAL EXPANDING SHOE BRAKE

$$\text{Moment of normal force, } M_N = \frac{1}{2} p l . b . r . \cos \theta_1 [(\theta_1 - \theta_2) + \frac{1}{2} (\sin 2\theta_1 - \sin 2\theta_2)]$$

$$\text{Moment of frictional force, } M_F = \mu p l . b . r . [r (\cos \theta_1 - \cos \theta_2) + \frac{r \cos \theta_1}{4} (\cos 2\theta_2 - \cos 2\theta_1)]$$

Braking torque in Block or shoe brake is given by

$$T_B = \frac{\mu F l r}{a - \mu c} \text{ [when the rotation of drum is clockwise]}$$

$$T_B = \frac{\mu F l r}{a + \mu c} \text{ [when the rotation of drum is anticlockwise]}$$

where T_B - Braking torque,

r = Radius of drum,

F = Force applied at lever end,

μ = Coefficient of friction, and

a, c & l = Dimensions of lever.

Equivalent coefficient of friction (μ^1) used when $2\theta > 40^\circ$ is given by

$$\mu^1 = \frac{4\mu \sin \theta}{2\theta + \sin 2\theta}$$

where 2θ = Angle of contact

Braking torque in Double block or double shoe brake is given by

$$T_B = \mu r (R_{N1} + R_{N2})$$

where r = Radius of drum,

R_{N1} & R_{N2} = Normal reaction on the left & right hand side shoes

In **Band brake system**,

Tension ratio, $\frac{T_1}{T_2} = e^{\mu\theta}$ and

Braking torque, $T_B = (T_1 - T_2)r$

T_1 and T_2 = Tension in the band on tight and slack sides respectively

θ = Angle of lap

r = Radius of drum,

Force applied on the lever in Simple band brake is given by

(i) $F = T_1 \left(\frac{b}{a} \right)$. [For clockwise rotation of the drum]

$F = T_2 \left(\frac{b}{a} \right)$ [For anticlockwise rotation of the drum]

Tension ratio in Band and block brake is given by

$$\frac{T_n}{T_o} = \left[\frac{1 + \mu \tan \theta}{1 - \mu \tan \theta} \right]^n$$

where T_n = Tension in the band on tight side (maximum tension),

T_o = Tension in the band on slack side (minimum tension),

2θ - Angle subtended by each block at the drum centre, and

n = Number of wooden blocks.

Actuating force on leading (or left hand) shoe. $F_1 = \frac{M_N - M_F}{l}$

Actuating force on Trailing (or right hand) shoe. $F_2 = \frac{M_N + M_F}{l}$

Energy considerations:

(i) Total energy absorbed by brake: $E_r = \frac{1}{2}mv^2 + \frac{1}{2}I\omega^2 + Wx$

(ii) Heat generated in brakes: $H_g = \mu \times R_N \times V = \mu p \cdot A \cdot V$

(iii) Heat dissipated in brakes: $H_d = C \times A \times \Delta t = C \times A \times (t_s - t_a)$

Temperature rise: $\Delta t = \frac{E}{c.m}$

UNIT-V: CAM TYPES (PART - A)
1. What is meant by self energizing brake? (May/June 2014) (April/May 2015)
Soln. When the frictional force helps in applying the brake, then the brake is said to be self-energised brake .
2. What are the effects of temperature rise in clutches and Explain the function of clutch ? (May/June 2016)
Soln. i) Excessive surface temperature results in premature clutch failure. ii) May causes the individual plates to be welded together in metal clutches. iii) May cause excessive wear in non-metal clutches. Functions The clutch is a mechanical device which is used to connect or disconnect the source of power at the operator's will.
3. In cone clutches, semi cone angle should be greater than 12°, why? (May/June 2012)
Soln. We know that the torque capacity is inversely proportional to $\sin \alpha$. The value of α is less than the angle of friction (ϕ), the clutch has a tendency to grab, resulting in self-engagement. The self- engagement is not desirable because the clutch should engage or disengage only at the operations will.
4. List the characteristics of material used for brake lining? (Nov/Dec, 2010)
Soln. (i). A high and uniform coefficient of friction. (ii). The ability to withstand high temperatures, together with high heat dissipation capacity. (iii). Adequate mechanical and thermal strengths. (iv). High resistance to wear. (v). resistance against environmental conditions, such as moisture and oil.
5. Define base circle and pitch circle with respect to cam? (Nov/Dec, 2010)
Soln. Base circle: The smallest circle from the cam center through the cam profile curve Pitch circle: A circle from the cam center through the pitch point. The pitch circle radius is used to calculate a cam of minimum size for a given <i>pressure angle</i> .
6. What is the advantage of block brake with one short shoe? What is the remedy?(April/May 2010)
Soln. 1. Self- locking brake 2. Self-energizing brake

UNIT–V: CAM TYPES (PART - A)
7. List the advantages and applications multi plate clutches? (April/May 2015)
Soln. a multiplate clutch is used when large amount of torque is to be transmitted. In a multiplate clutch, the number of friction linings and the metal plates are increased which increases the capacity of the clutch to transmit torque, the multiplate clutch works in the same way as the single plate clutch by operating the clutch pedal. They are extensively used in motor car, machine tools, etc.
8. Under what condition of a clutch, uniform rate of wear assumption is more valid? (May/June 2009)
Soln In clutches, the value of normal pressure, axial load for the given clutch is limited by the rate of wear that can be tolerated in the brake linings. Moreover, the assumption of uniform rate wear gives a lower calculated clutch capacity than the assumption of uniform pressure. Hence clutches are usually designed on the basis of uniform wear.
9. Name four profiles normally used in cams. (Nov/Dec 2012)
Soln. Classify cam based on a shape? (i) wedge cam (ii) radial cams (iii) spiral cams (iv) drum cams (v) spherical cams
10. What is a self-locking brake? (Apr/May2011)
Soln. When the frictional force is sufficient enough to apply the brake with no external force, then the brake is said to be self-locking brake