DEPARTMENT OF

MECHANICAL ENGINEERING

ME8651 DESIGN OF TRANSMISSION SYSTEMS L T P C
3 0 0 3

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

OUTCOMES:

- Upon completion of this course, the students can able to successfully design transmission components used in Engine and machines

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.
6. Robert C. Juvinall and Kurt M. Marshek, "Fundamentals of Machine Design", 4th Edition, Wiley, 2005
7. Alfred Hall, Halowenko, A and Laughlin, H., "Machine Design", Tata McGraw-Hill BookCo.(Schaum's Outline), 2010
8. Orthwein W, "Machine Component Design", Jaico Publishing Co, 2003.
9. Ansel Ugural, "Mechanical Design – An Integral Approach", 1st Edition, Tata McGraw-Hill Book Co, 2003.
10. Merhyle F. Spotts, Terry E. Shoup and Lee E. Hornberger, "Design of Machine Elements" 8th Edition, Printice Hall, 2003.
11. U.C.Jindal : Machine Design, "Design of Transmission System", Dorling Kindersley, 2010

UNIT I

DESIGN OF FLEXIBLE ELEMENTS

Characteristics of Belt Drives

S.No	Characteristics	Flat belts	V- belts	Toothed or timing belts
1.	Maximum velocity ratio	16	12	11
2.	Maximum belt speed (m/s)	35 to 110	25	80
3.	Slip	1 to 5%	1 to 5%	Nil
4.	Tension	High	Less	Very less
5.	Shock resistance	Good	Good	Fair
6.	Resistance to wear	Good	Fair	Good
7.	Dressing	Required	Not Required	Not Required
8.	Initial cost	Less	Less	Moderate

SELECTION OF A BELT DRIVE

Selection of a belt drive depends upon:

- Power to be transmitted
- Speed of driver and driven shafts
- Shaft relationship
- Service conditions
- Speed reduction ratio
- Centre distance
- Positive drive requirement
- Space available

VELOCITY RATIO OF BELT DRIVE

The ratio between the speeds of the driver and the follower or driven is known as velocity ratio.

D and d = Diameters of the driver and driven respectively,

N_1 and N_2 = Speeds of the driver and driven respectively, and

ω_1 and ω_2 = Angular velocities of the driver and driven respectively.

$$\boxed{\textbf{Velocity ratio, } \frac{N2}{N1} = \frac{\omega 2}{\omega 1} = \frac{D}{d}}$$

Effect of Belt Thickness on Velocity Ratio

When the thickness of belt *(t)* is considered, then velocity ratio is given by

$$\boxed{\frac{N2}{N1} = \frac{D+t}{d+t}}$$

Effect of Slip on Velocity Ratio

Let S1 = Percentage slip between the driver and the belt,

S2 = Percentage slip between the belt and the driven pulley, and

S = Total percentage slip = S1 + S2

$$\text{Velocity ratio, } \frac{N2}{N1} = \frac{D}{d}\left[1 - \frac{S1+S2}{100}\right] = \frac{D}{d}\left[1 - \frac{S}{100}\right]$$

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

$$\text{Velocity ratio, } \frac{N2}{N1} = \frac{D+t}{d+t}\left[1 - \frac{S}{100}\right]$$

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.

$$\text{Velocity ratio, } \frac{N2}{N1} = \frac{D}{d} \times \frac{E+\sqrt{\sigma 1}}{E+\sqrt{\sigma 2}}$$

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,

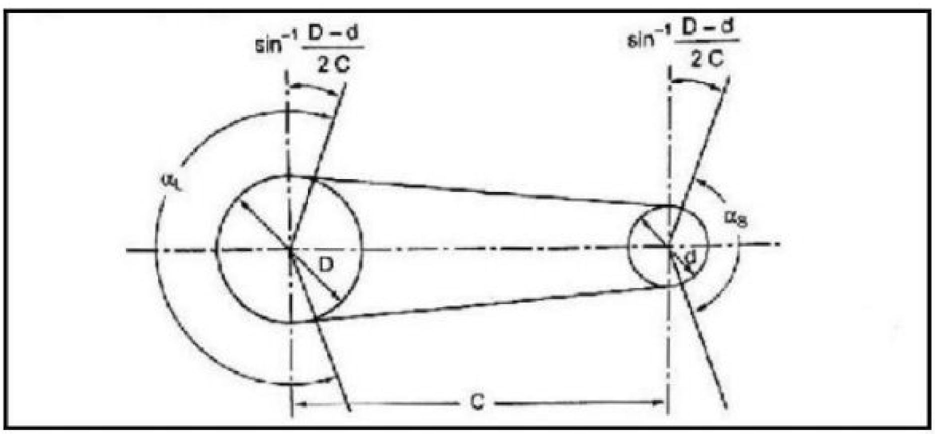
C = Centre distance between the two pulleys in meters,

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

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

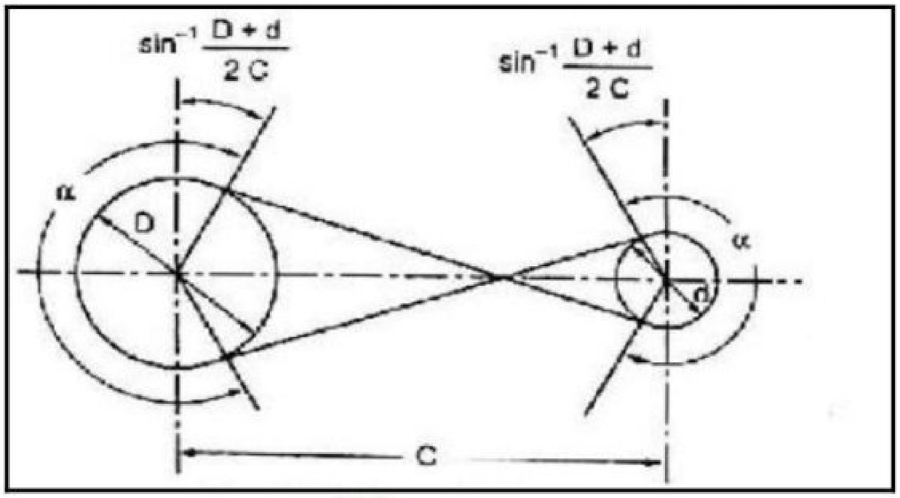
also $\alpha_s = (180 - 2\alpha)$ and $\alpha_L = (180 + 2\alpha)$

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

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

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

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

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 arc.

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

$$\sigma_t = \frac{Tight side tension}{Cross-sectional area of the belt} = \frac{T_t}{b.t}$$

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):

$$(\sigma_b) = \frac{E.t}{d}$$

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)

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

Where ρ = *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 that 1.5 M Pa

DESIGN OF FLAT BELT PULLEYS

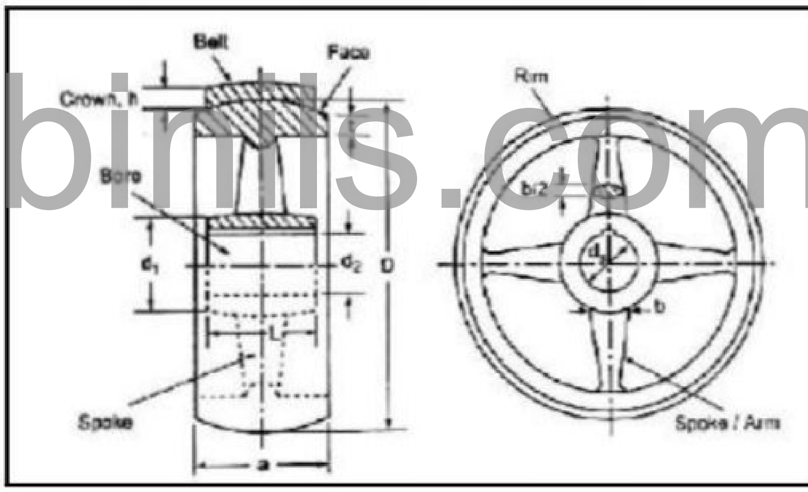
Materials Used for Pulleys

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/m^3 for cast iron, and
v = Velocity of the rim = $\frac{\pi DN}{60}$ 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 $\begin{cases} 4 \text{ for diameters upto } 450mm \\ 6 \text{ for diameters over } 450mm \end{cases}$ ….**[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 raper : 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)

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

$$\textbf{Design KW} = \frac{\text{RatedKWxLoadcorrectionfactor}(K_s)}{\text{Arcofcontactfactor}(K_\alpha)\text{xSmallpulleyfactor}(K_d)}$$

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_α):**

Arc of contact= 180 –$(\frac{D-d}{c})$*60 ....**[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.

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

5. Determination of belt width(b):

Determine the belt width by using the following relation:

$$\textbf{Width of belt} = \frac{\text{Designpower}}{\text{LoadratingxNo.ofplies}}$$

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 tables1.6 (a and b)**[From data book, page no. 7.54]**

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

Table1.6(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: $\mathbf{L = 2C + (\frac{\pi}{2})(D+d) + \frac{(D-d)^2}{4C}}$**[From data book, page no. 7.53]**

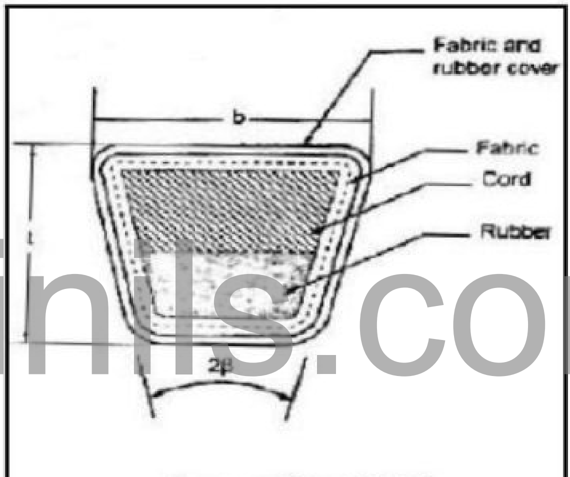
For crossed belt drive: $\mathbf{L = 2C + (\frac{\pi}{2})(D+d) + \frac{(D+d)^2}{4C}}$**[From data book, page no. 7.53]**

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

Where T_1, T2, m, v and α have usual meaning, and

2β is the V- groove angle (=180^0 for flat belt)

V -Belts and Pulleys



Cross-section of V-belt

Materials of V-belts

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

Specification of V-belts

V-belts are designated by its type and nominal inside length. For example, a C2845belt has a cross-section of type C and has a nominal inside length of2845 mm.

RATIO OF DRIVING TENSIONS FOR V-BELT

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

Where TI and T2 = 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 = $\frac{TotalPowerTransmitted}{PowerTransmitted}$

DESIGN OF SHEAVES (OR V-GROOVED PULLEYS)

1. Materials of V-grooved pulleys:

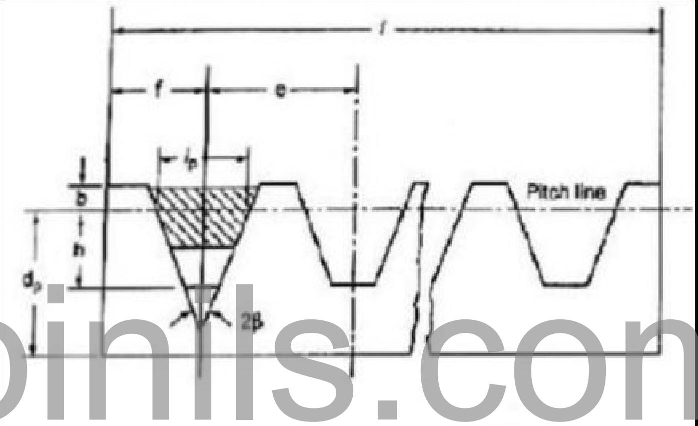
The commonly used sheave materials and their characteristics are summarized In Table 2.1.

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).**



Cross -Sectional of a Sheave

where

Ip= 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, l= (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 thepower 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,

   $$L = 2C + \left(\frac{\pi}{2}\right)(D + d) + \frac{(D-d)^2}{4C}$$ **... [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 lengthis 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 0 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 bedetermined.

(i) Length correction factor (F_C):

Table.Nominal inside length, nominal pitch length and length correction/actor/or standard sizes 0/ Jl-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 = $180^\circ - \left(\frac{D-d}{C}\right) \times 60^\circ$... [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 0/ contact/actor, Fd…[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 (Fa) 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 180^0 arc of contact for a belt of average length,

S = Belt speed, m/s,

d_e= Equivalent pitch diameter = dpx Fb,

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,

$$n_b = \frac{P * F_a}{KW * F_c * F_d}$$ ….**[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

F_d= Correction factor for arc of contact.

8. Calculation of actual centre distance:

Calculate the actual centre distance from the relation,

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

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

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

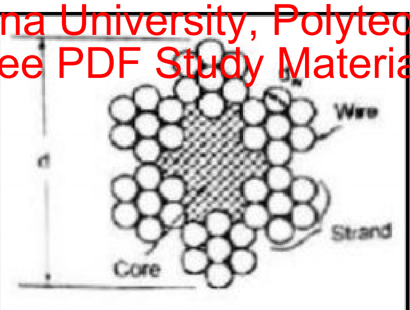


Fig. 3.1. 6 × 7 wire rope

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 = 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**

= $0.84*10^5$ N/mm^2, for steel ropes of ordinary construction,

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

d_W= diameter of the wire,

D = 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) 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 slopping, then there will be a considerable impact load on the rope.

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

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 x 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, mm²
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,):

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, $W_b - \sigma_b \times A = E_r . \frac{d_w}{D} \times A$

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

$$Wa = [\frac{W + W_r}{g}]\, a$$

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, $WS1 = 2 \cdot W_d = 2\,(W + Wr)$

(b) When there is slack in the rope:

Starting load, $W_{st} = \sigma st \times A = (W + Wr)\,[1 + \sqrt{1 + \frac{2 . a_s . h . E_r}{\sigma_d . l . g}}\,]$

9. Calculation of effective loads:

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

(ii) Effective load on the rope during acceleration of the load, $Wea = W_d + W_b + W_a$

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

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

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

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

$$P = \frac{2T}{d \times D}$$

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).

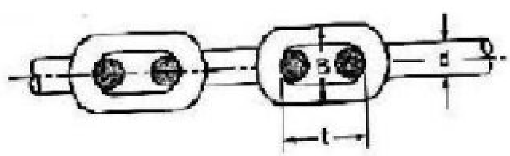
LINK CHAINS

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 $t \leq 3$ d, then the chains are known as short link chains.

(b) Long link chains: If $l > 3$ d, 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 within60.05 d and from the outside width is within 6 0.05 d, then the chain is called as Pitched chain.

(b) Calibrated chains :When the permissible deviations is within i 0.1 d 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

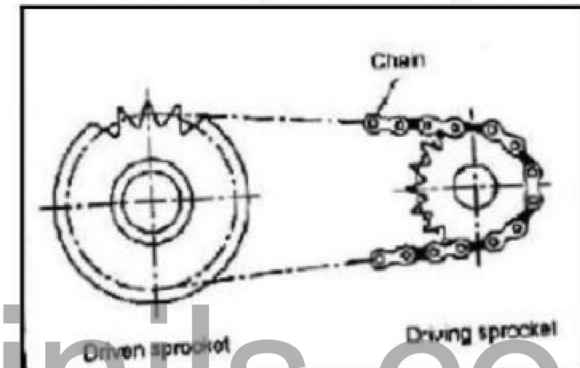
$$P_{safe} = \frac{P_{br}}{n}$$

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 asC45, C50 and 40 Crl.
- Pins, bushings and rollers are made of carburizing steels such as CIS, C20, and 30 Ni4 Crl.

SPECIFICATION OF A ROLLER CHAIN

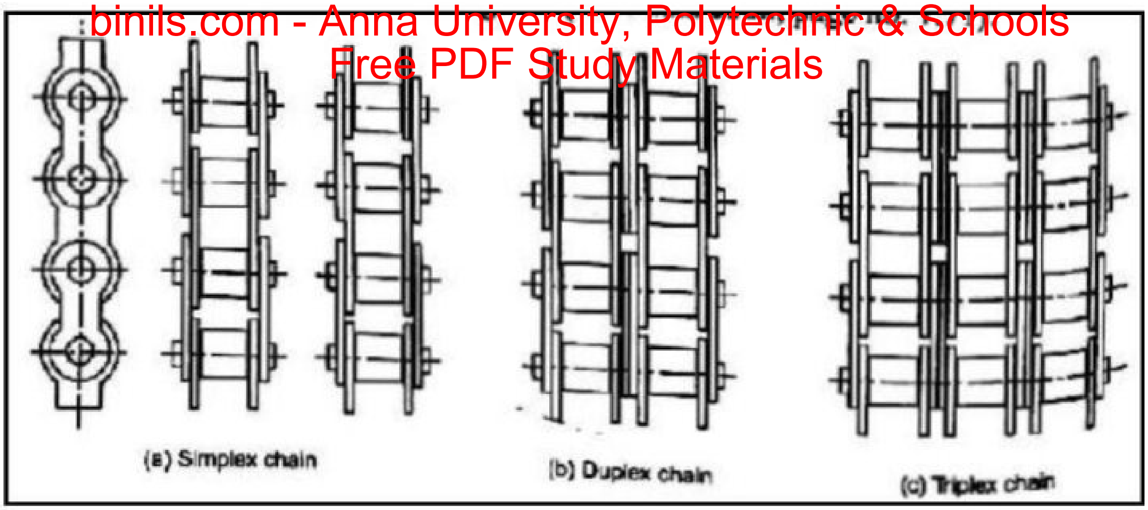
Roller chain is specified by three dimensions - pitch, width and diameter.

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,

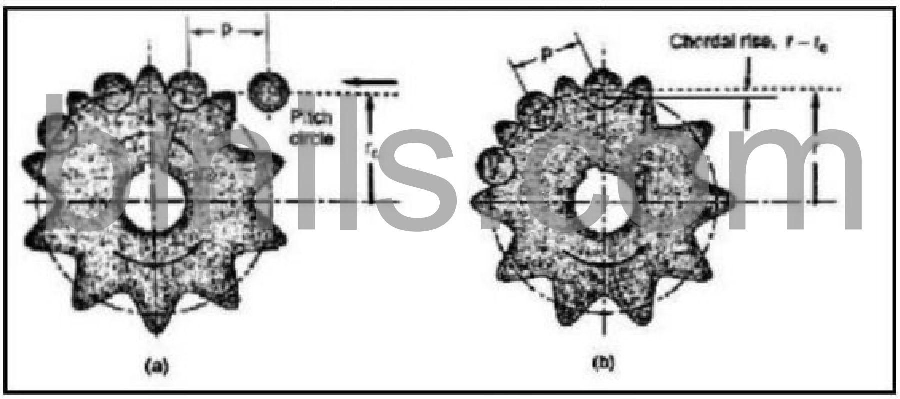
a = Pitch angle,

$\alpha/2$ = Angle of articulation,

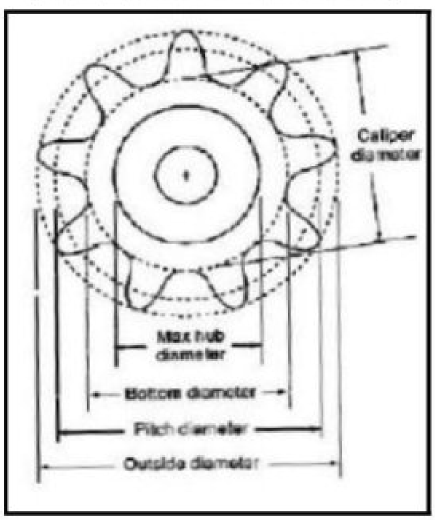
D = Pitch circle diameter of the sprocket, and

z = Number of teeth on the sprocket.

CHORDAL (OR POLYGONAL) ACTION



SPROCKET DIAMETERS



The equations for those diameters are :

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

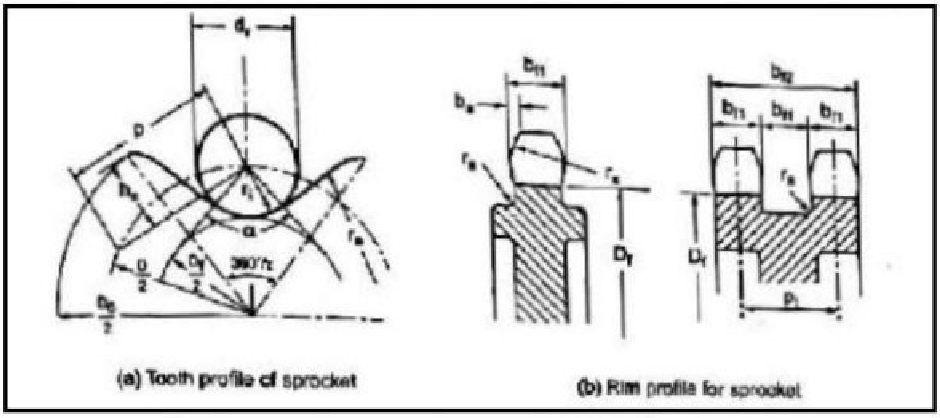
(ii) Outside diameter = **p[0.6 cot ($\frac{180}{z}$)]**

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

(iv) Caliper diameter = Pitch diameter x 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

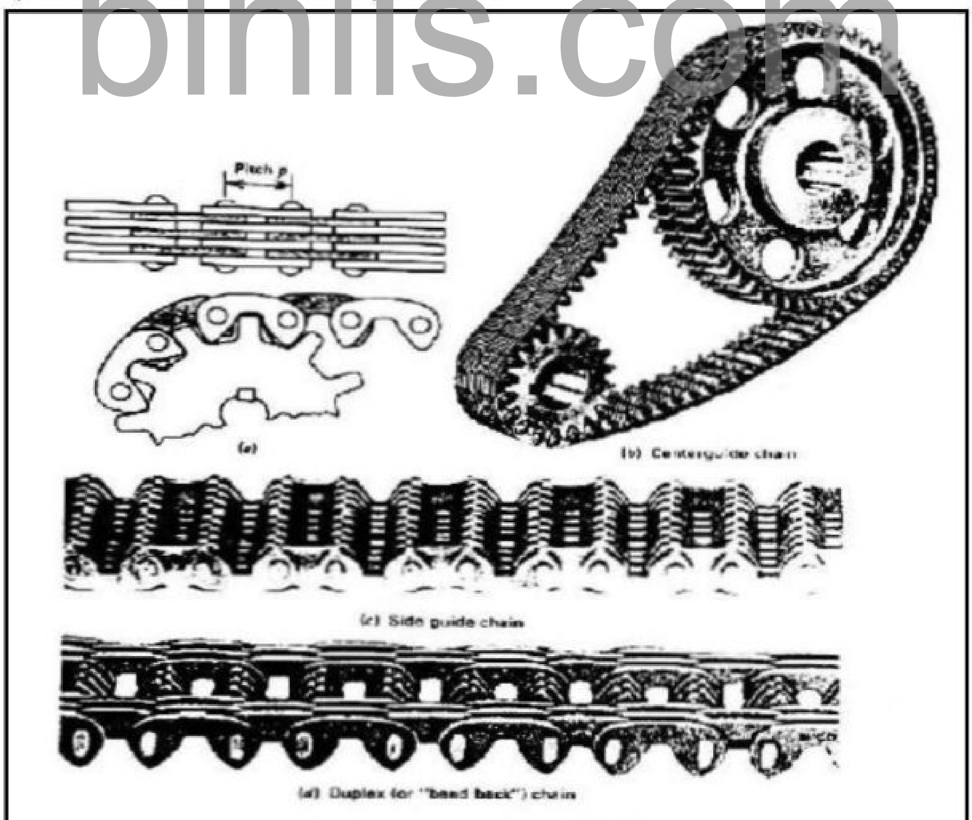


Fig. 4.11. Inverted-tooth (silent) chains

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 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):

$$\begin{Bmatrix} \textit{Total load on the} \\ \textit{Driving side } (P_T) \end{Bmatrix} = \begin{Bmatrix} \textit{Tangential force} \\ \textit{due to} \\ \textit{power transmission}(P_t) \end{Bmatrix} + \begin{Bmatrix} \textit{Centrifugal tension}(P_c) \\ \textit{due to speed} \\ \textit{of the chain} \end{Bmatrix} + \begin{Bmatrix} \textit{Tension} \\ \textit{due to chain} \\ \textit{sagging}(P_s) \end{Bmatrix}$$

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

(i) To find tangential force (P_t):

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

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):

$P_c = mv^2$…[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):

$P_s = k . w. a$

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

W = Weight of chain / meter = m- 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, $K_s = k_1 . k_2 . k_3 . k_4 . k_5 . k_6$…**[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

Design load = **P_T x 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

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

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

Take the values for allowable bearing stress,$[\sigma]$, N/mm^2 …[**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

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

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

$L = l_p \times p$

13. Calculation of exact centre distance:

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

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

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

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

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

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

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

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

Where d_r **is the roller diameter**

UNIT – I: DESIGN OF TRANSMISSION SYSTEMS FOR FLEXIBLE ELEMENTS (PART –A)
1. What are the factors upon which the coefficient of friction between the belt and pulley depends? (May/june2012) (May/June 2014)
Soln. *The coefficient of friction between the belt and the pulley depends upon the following factors:* 1. *The material of belt;* 2. *The material of pulley;* 3. *The slip of belt; and* 4. *The speed of belt.* *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.* $\mu = 0.54 - \{42.6/(152.6 + v)\}$ *Where, v = Speed of the belt in meters per minute.*
2. Brief the term "crowning of pulley". (May/June 2014)
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.*
3. What are the materials used for belt-drive? (May/June 2013) (May/June 2016)
Soln. Leather,, cotton fabrics ,rubber, animal's hair, silk, rayon, woolenetc
4. What do you mean by Galling of roller chains? (Nov/Dec 2010) (May/june2012) (Nov/Dec 2010)
Soln. *The pin pressure faces have suffered from severe galling where the surfaces have articulated and fused together.* 
5. When do you prefer chain drive to a belt or rope drive? (May/Jun 2016
Soln. *Chain drives are preferred for velocity ratio less than 10, chain velocities upto 25 m/s, and for power ratings up to 125KW*

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 nup4ber 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.

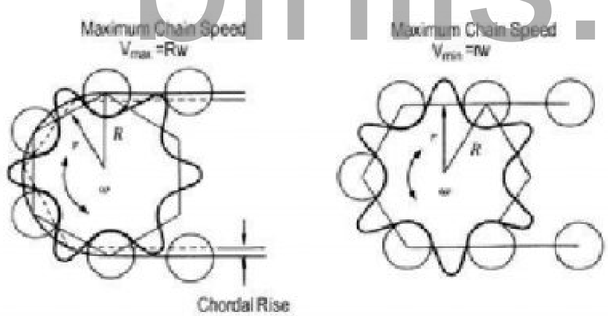
iii) Compound laid ropes.

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]

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