

# SCHOOL OF MECHANICAL ENGINEERING DEPARTMENT OF MECHANICAL ENGINEERING

**COURSE CODE: SMEA1601** 

**COURSE NAME: DESIGN OF TRANSMISSION SYSTEMS** 



**Unit 1 Design of Flexible Drives – SMEA1601** 



## **DESIGN OF FLAT BELT DRIVE**

## **STEPS IN FLAT BELT DRIVE**

- 1. Calculation of velocity ratio
- 2. Find the velocity of the belt
- 3. Take load correction factor(Ks) from DDB 7.53 based on given application
- 4. Calculation of Arc of contact from DDB 7.54 and take the value of correction factor(K) for arc of contact from the table DDB 7.54
- 5. Calculation of corrected power
- 6. Calculation of belt rating from DDB 7.54
- 7. Calculation of width of belt
- 8. Calculation of length of belt from DDB 7.53
- 1. Design a flat belt drive to transmit 10kW at 1500 rpm to a line shaft to run at 500 rpm. Approximate centre distance is m. the diameter of the larger pulley is around 750mm.

## Solution

P-10kW N1= 1500rpm N2= 500rpm C=2m D=750 mm  
Velocity ratio 
$$= \frac{N_2}{N_1} = \frac{d}{D}$$
  
 $\frac{500}{1500} = \frac{d}{750} \therefore d = \frac{750 \times 500}{1500} = 225 \text{ mm}$   
Velocity of belt  $= V = \frac{\pi dN_1}{60} = \frac{\pi \times 225 \times 1500}{60 \times 1000} = 17.67 \text{ m/sec}$ 

# Load correction factor $(K_s)$

According to load classification, refer data book, Page No.7.53 and take t value of  $K_S$ .

 $K_s = 1.3....$  (for line shafts)

Correction arc of contact:  $K_{\alpha}$ , refer data book, Page No.7.54.

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Arc of contact = 
$$180 - \frac{D-d}{C} \times 60 = 180 - \frac{750 - 225}{2000} \times 60$$

 $= 164.25^{\circ} = 165^{\circ}$ 

From the databook, page No.7.54 Correction factor for Arc of contact,  $\left\{ (at \ \theta - 165^{\circ}) \right\}$ 

 $K_{\alpha} = 1.06$ 

Interpolation of  $K_{\alpha}$   $170^{\circ} - 104$ for  $5^{\circ} - ? - 0.04 \times \frac{5}{10} = -0.02$   $160^{\circ} - 1.08$  $\frac{5^{\circ} - 0.02}{165 = 1.06}$ 

**Calculation of Corrected Power** 

corrected power =  $\frac{K_s \cdot (\text{Given power in kW})}{K_{\alpha}}$ 

$$=\frac{1.3\times10}{1.06}=12.26\,\mathrm{kW}$$

Refer databook, page No. 7.52,

According to the minimum pulley diameter and the maximum belt speed assume the no. of plies, from table at v = 17.67 m/sec and d = 225 mm;

Take, n = no. of plies = 5

Calculation of Load rating:

Select high speed belt,

The load rating per mm width per ply at 10 m/sec = 0.023 kw/mm/ply

Load rating at belt speed (atV = 17.67 m/sec) =  $\frac{0.023 \times 17.67}{10}$ 

= 0.0406 kW/mm/ply

Calculation of width of the belt

Refer databook, page No.7.54

Millie meter plies of belt = Corrected load (or) Corrected power Load rating /mm/ply

Width  $\times$  no. of plies =  $\frac{12.26}{0.0406}$  = 301.97

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:. Width of the belt =  $\frac{301.97}{5}$  = 60.39 mm

Since n = no. of plies = 5

Refer databook, Page No.7.52,

For 5 ply belt, the standard width of belt = 76 mm

Calculation of length of the belt (L)

Refer databook, page No.7.53

$$L = \frac{\pi}{2} (D+d) + 2C + \frac{(D-d)^2}{4C}$$
$$L = \frac{\pi}{2} (750 + 225) + 2 \times 2000 + \frac{(750 - 225)^2}{4 \times 2000}.$$

= 5565.97 mm

Width of Pulley

Refer databook, Page No. 7.54

up to including 125 mm belt width, pulley than the belt width by 13 mm  $\therefore$  width of pulley = 76 + 13 = 89 mm

Refer databook, page No.7.54

The recommended pulley nominal diameter = 90 mm; with tolerance of nominal diameter as  $\pm 1.2$ 

2. Design a fabric belt to transmit 10 kW at 450 rpm from an engine to a line shaft as 1200 rpm. The diameter of the engine pulley is 6—mm and the distance of the shaft from the engine is 2m. Solution:
P=10 kW N1=450 rpm N2= 1200 rpm

P=10 kW	N1=450 rpm	N2= 1200 rpm
<b>D=600 mm</b>	C=2m	

 $\frac{N_2}{N_1} = \frac{D}{d} \frac{\text{(dia. of Driver pulley)}}{\text{(dia. of Driven pulley)}}$  $\frac{1200}{450} = \frac{600}{d}$  $d = \frac{600 \times 450}{1200} = 225 \text{ mm}$ 

... Dia. of line shaft pulley = d = 225 mm.

• Velocity of the belt =  $v = \frac{\pi D N_1}{60} = \frac{\pi \times 600 \times 450}{60 \times 1000} = 14.137 \text{ m/sec}$ 

Load correction factor  $(K_s)$ 

According to Load classification, refer databook Page No.7.53

Take the value of  $(K_s)$ 

 $K_S = 1.3$  (for line shafts)

Correction factor for Arc of contact  $(K_{\alpha})$ : Refer databook page No. 7.54.

Pref 2

Arc of contact =  $180 - \frac{D-d}{C} \times 60^{\circ}$ 

 $=180 - \frac{600 - 225}{2000} \times 60$ 

= 168.75 take approximately  $\theta = 170^{\circ}$ 

From the table, (refer page No.7.54)

Correction factor (at  $\theta = 170^{\circ}$ ) =  $K_{\alpha} = 1.04$  for ARc of contact

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Calculation of Corrected Power Corrected power =  $\frac{K_S \times (\text{Given power in kW})}{K_u}$ =  $\frac{1.3 \times 10}{1.04}$ = 12.5 kW

Refer Page No. 7.52

According to the minimum pulley diameter and the maximum belt speed, assume the no. of plies

From the table, at v = 14.137 m/sec and d = 225 mm,  $n = n_v = \text{no.of plies} = 5$ 

### Calculation of Load rating

Select high speed belt.

The load rating per mm width per ply at 10 m/sec = 0.023 kW/mm/ply

Load rating at belt spread (at v = 14.137 m/sec) =  $\frac{0.023 \times 14.137}{10}$ 

= 0.0325 kw/mm/ply

# Calculation of width of the belt

Refer page No. 7.54

Millimeter plies of belt = Corrected load or Corrected power Loaded rating /mm/ply

Width  $\times$  no.of plies =  $\frac{12.5}{0.0325}$ 

= 384.43

Since no. of plies n = 5

 $\therefore$  Width of the belt =  $\frac{384.43}{5}$ 

=76.88 mm

Refer P.No. 7.52

For 5 ply belt the standard width of belt = 90 mm

Calculation of Length of the belt (L): Refer Databook Page No. 7.53.

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$$L = \frac{\pi}{2} (D+d) + 2C + \frac{(D-d)^2}{4C}$$
$$= \frac{\pi}{2} (600 + 225) + 2 \times 2000 + \frac{(600 - 225)^2}{4 \times 2000}$$
$$L = 5212.42$$

L = 5313.48 mm

Width of Pulley

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Refer Databook Page No. 7.54

Upto and including 125 mm belt width pulleys to be wider than the belt width by 13 mm

 $\therefore$  Width of the Pulley = 90 + 13 = 103 mm

Refer Databook, Page No. 7.54.

The recommended pulley nominal diameter = 112 mm

With tolerance on nominal diameter as  $\pm 1.2$ .

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## **DESIGN OF V-BELT DRIVE**

### **STEPS IN FLAT BELT DRIVE**

- 1. Select the cross selection of the belt depending on the power to be transmitted.
- 2. Calculate the speed ratio.
- 3. Calculate the length of the belt from DDB 7.53.
- 4. Calculate the design power from DDB 7.64.
- 5. Find number of belts from DDB 7.70
- 6. Calculate new centre distance from DDB 7.61
- 1. A 30 kW 1440 rpm motor is to drive a compressor by means of V-belts. The Diameter of the pulleys are 220 mm and 750 mm. the centre distance between the compressor and the motor is 1440 mm. design a suitable drive.

### Solution:

P=30 kw	N1=1440 rpm	d = 220mm
D=750 mm	C=1440 mm	

### Step 1

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From databook, P. No. 7.58. Select Cross-section of the belt. Select either (C), (D) or (E) Select (C) type Belt (Since min. pulley pitch dia = 220 mm given) Load of drive = P = 22 KW - 150 KWMin. pulley pitch diameter = 200 mmNominal top width = W = 22 mmNominal thickness = T = 14 mm;Weight/meter = 0.343 Kg/m length

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C.S. area of belt  $= \frac{1}{2} (W + b) T$   $= \frac{1}{2} [22 + 11.808] \times 14$   $= 236.66 \text{ mm}^2$ Step 9  $x = T \cdot \tan 20^\circ$   $= 14 \cdot \tan 20^\circ$  = 5.095 mm b = W - 2x = 22 - 2 (5.095)= 11.808 mm

Step 2

Nominal pitch length =  $L = \frac{\pi}{2} (D+d) + \frac{(D-d)^2}{4c} + 2c$ =  $\frac{\pi}{2} (750 + 220) + \frac{(750 - 220)^2}{4 \times 1440} + 2 \times 1440$ = 4452.439 mm

=4.452 m

Take the nearest nominal pitch length from databook.

Refer P.No. 7.60,

Take standard nominal pitch length = 4450 mm; (Nearest) The corresponding nominal inside length = 4394 mm.

The Designation of V-belt C 4394 - 1S2494

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d = 220 mmStep 3  $N_1 = 1440 \text{ rpm}$ Calculation of Design Power Refer Page No. 7.62  $\mathrm{kW} = (1.47S^{-0.09} - \frac{142.7}{de} - 2.34 \times 10^{-4}S^2)\,S$  $S = \frac{\pi dN_1}{60} = \frac{\pi \times 0.22 \times 1440}{60} = 16.587 \text{ m/sec}$  $d_p =$  pitch dia. of smaller pulley de = Equivalent pitch diameter = 220 mm  $= d_p \times F_b$  $\frac{D}{d} = \frac{750}{220}$  $= 220 \times 1.14$  $=3.4 \rightarrow F_b = 1.14$  (P. No. 7.62) = 250.8 m.m  $kW = \left[ 1.47 (16.587)^{-0.09} - \frac{142.7}{250.8} - 2.34 \times 10^{-4} \times 16.587^2 \right] 16.587$ = 8.4312 kW Step 4 To find the no. of belts (n) Refer page No. (7.70) No. of belts  $= n = \frac{P \times F_a}{kW \times F_a \times F_a}$ where P = given power KW  $F_a =$  correction factor (from Page No. 7.69) Let the time period is upto 10 hr.  $\therefore F_a = 1$  (for compressor) kW = Power at the corresponding C.S. (ie at (C) - Cross section)  $F_c = \text{Correction factor for length (P.No.7.60)} = 1.04$  $F_d =$ Correction factor for arc of contact

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$$\begin{aligned} \theta &= 180 - \left(\frac{D-d}{c}\right) 60^{\circ} \\ &= 180 - \left(\frac{750 - 220}{1440}\right) 60 = 157.9^{\circ} \\ &\approx 158^{\circ} \end{aligned}$$

Refer page No. (7.68)  $160^{\circ} - 0.95$ : for  $158^\circ$ ,  $F_d = 0.94$ 0.94157° 30 0.01  $0.01 \times 1$ 1º -3 = 0.0033∴ 158° = 0.94 + 0.0033  $30 \times 1$ n =no. of belts =  $8.4312 \times 1.04 \times 0.94$ = 0.9433=3.63

≈4 belts

 $\therefore$  No. of belts required = n = 4

7.61)

## CALCULATION OF NEW CENTRE DISTANCE

$C = A + \sqrt{A^2 - B}$	(from P.No.
$A = \frac{L}{4} - \pi \left(\frac{D+d}{8}\right)$	L = nominal pitch length
$B = \frac{(D-d)^2}{8}$	=4450 mm
$A = \frac{4450}{4} - \pi \left(\frac{750 + 220}{8}\right) = 731.58$	1.8 - E. P.
$B = \frac{(750 - 220)^2}{8} = 35112.5$	
$C = 731.58 + \sqrt{731.58^2 - 35112.5}$ New centre distance $C = 1438.75$ mm	and a second s

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Calculation of  $T_1$  and  $T_2$ 

$$\frac{T_1 - T_c}{T_2 - T_c} = e^{\mu \ 0 / \sin \beta} \ \dots \ (1)$$

 $2\beta = 40^{\circ}$ 

Semi-groove angle  $\beta = 20^{\circ}$ 

Let  $\mu = 0.25$ 

$$\theta = 158^\circ \times \frac{\pi}{180}$$

= 2.75 radians

 $T_c = mv^2 = 0.343 \times (16.587)^2$ 

= 94.36 N

v = S = 16.587 m/sec

m = 0.343 Kg/m length

Power/belt = 
$$[T_1 - T_c] \left\{ 1 - \frac{1}{e^{\mu\theta/\sin\beta}} \right\} v$$
  
 $8.3412 \times 10^3 = (T_1 - 94.36) \left[ 1 - \frac{1}{e^{0.25 \times 2.75/\sin 20^{\circ}}} \right] \times 16.587$   
 $8.4312 \times 10^3 = (T_1 - 94.36) (14.348)$   
 $T_1 = 681.9819 \text{ N}$   
 $\frac{T_1 - T_c}{T_2 - T_c} = e^{\mu\theta/\sin\beta}$   
 $\frac{681.9819 - 94.36}{T_2 - 94.36} = e^{0.25 \times 2.75/\sin 20^{\circ}}$   
 $\frac{587.8219}{(T_2 - 94.36)} = 7.464$   
 $T_2 = 78.754 + 94.36$ 

$$T_2 = 173.114$$
 N

To Find Stress  $(f_b)$ 

$$\begin{split} T &= (T_1 + T_c) = f_b \times \text{ area of belt} \\ 681.9819 + 94.36 = f_b \times 236.66 \\ \text{Permissible stress in the belt material} \\ f_b &= 3.28 \ \text{N/mm}^2 \end{split}$$

 Design a V belt drive to the following specification Power to be transmitted -75 kW, speed of the driving wheel (N1)=1440 rpm, speed of the driving wheel (N2)=440 rpm, diameter of the driving wheel (d) =300 mm, Centre distance=2500mm, service =16hrs/ day.

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#### Solution : P=75kw N1=1440 rpm N2=440 rpm d =300mm C=2500mm service =16 hrs/ day To Find Stress (fb) $T = (T_1 + T_c) = f_b \times \text{ area of belt}$ $681.9819 + 94.36 = f_b \times 236.66$ Permissible stress in the belt material $f_b = 3.28 \text{ N/mm}^2$ Problem 2: Design a V-belt drive to the following specifications Power to be transmitted = 75 kW Speed of the driving wheel = 1440 rpm; $(N_1)$ Speed of the driven wheel = $400 \text{ rpm}; (N_2)$ Diameter of the driving wheel = 300 mm; (d) Centre distance = 2500 mm Service = 16 hours/day Assume any other relevant data if necessary. (Oct. '97) Given Data: $P = 75 \times 10^3$ Watts Driving Wheel Speed $N_1 = 1440$ rpm; Driven Wheel Speed $= N_2 = 400 \text{ rpm}$ Driving wheel diameter = d = 300 mmDriven wheel diameter = D = ?Centre distance = C = 2500 mm; Service = 16 hr/day. b = W - 2xStep 1 = 22 - 10.19Selection of Cross section; From databook, Page No. 7.58 = 11.808 mm Since minimum pulley dia. is 300 mm $x = T \cdot \tan 20^\circ = 14 \times \tan 20^\circ$ Select (C) cross section = 5.095 mmNominal Width = W = 22 mmNominal thickness= T = 14 mm

Mass/kg = m = 0.343 kg/m length

C.S. area = 
$$\frac{1}{2}$$
 (W + b) T  
=  $\frac{1}{2}$  [22 + 11.808] × 14 = 236.656 mm<sup>2</sup>

## Step 2

Nominal Pitch length  $=L=\frac{\pi}{2}(D+d)+\frac{(D-d)^2}{4c}+2c$ 

 $\frac{N_1}{N_2} = \frac{D}{d} = \frac{\pi}{2}(1080 + 300) + \frac{(1080 - 300)^2}{4 \times 2500} + 2 \times 2500$  $D = \frac{N_1}{N_2} \cdot d = 7228.53 \text{ mm}$  $= \frac{1440}{400} \times 300 \qquad L = 7.228 \text{ m}$ = 1080 mm

Take the nearest value of nominal pitch length (from databook) 7.60 L = 6863 mm

\*The Corresponding nominal inside length = 6807 mm

## Step 3

Calculation of Design Power

From page No. 7.62 : for (C) C.S. of the belt

$$kW = \left[ \cdot 1.47S^{-0.09} - \frac{142.7}{de} - 2.34 \times 10^{-4}S^2 \right] S$$
$$S = \frac{\pi dN_1}{60} = \frac{\pi \times 0.3 \times 1440}{60} = 22.61 \text{ m/sec}$$

de = equivalent pitch diameter where

 $d_p$  = pitch dia. of smaller pulley

$$= d_p \times F_b$$
 refer page 7.62  
= 300 × 1.14  
= 342 mm at  $\frac{D}{d}$  = 3.6 the value of  $F_b$  = 1.14

But the max, value of  $(d_e)$  in the formula is 300 mm for (C) cross-section,  $\therefore$  Take  $d_e = 300$  mm

$$kW = \left[ 1.47 (22.61)^{-0.09} - \frac{142.7}{300} - 2.34 \times 10^{-4} (22.61)^2 \right] 22.61$$
  
Max. power transmitted by beft = 11.643 kW

## Step 4

To find the no. of belts

(Refer page No. 7.70)

No. of belts 
$$= n = \frac{P \times F_a}{kW \times F_c \times F_d}$$

P = 75 kW

 $F_a = 16$  hrs/day Page No. 7.6

Select medium duty

[Since specific application is not given]

 $\therefore F_a = 1.2$ 

kW = 11.643 kW

 $F_c = 1.14$  refer Page No. 7.60 at Cross-section C

Refer databook, Page No. 7.68.

$$F_d = 0.95 \qquad \theta = 180^\circ - \left\lfloor \frac{D-d}{c} \right\rfloor \times 60^\circ$$
$$= 180^\circ - \left( \frac{1080 - 300}{2500} \right) 60^\circ$$
$$= 161.28^\circ$$
Let  $\theta = 160^\circ$ 

 $\therefore$  *n* = no. of belts required =  $\frac{75 \times 1.2}{11.643 \times 1.14 \times 0.95}$ 

= 7.137

 $n \approx 8$  belts

8 belts are required.



# **DESIGN OF CHAIN DRIVE**

# **STEPS IN CHAIN DRIVE**

- 1. Calculate transmission ratio
- 2. Select number of teeth on sprocket pinion  $Z_1$  from DDB7.74. calculate the number of teeth on sprocket wheel  $Z_2$ .
- 3. Calculate pitch from DDB 7.74 and select the standard pitch from DDB 7.71-7.73. and select the std pitch with in  $P_{max}$  and  $P_{min}$
- 4. Select chain number from DDB 7.71-7.73.
- 5. Check for breaking load from DDB 7.77
- 6. Check for actual factor of safety from DDB 7.78.
- 7. Find the length of chain from DDB.7.75
- 8. Find exact centre distance from DDB 7.76
- 9. Check for bearing stress from DDB 7.77
- 10. Calculate number of teeth on pinion and wheel  $(d_1 \text{ and } d_2)$  from DDB 7.78

## Problem

Design a chain drive to operate a compressor from a 15kW electric motor at 900 rpm. The compressor is to run at a speed of 300 rpm, the minimum centre distance should be 550mm. Solution:

P=15Kw  $n_1$ =900rpm  $n_2$  = 300 rpm C=550 mm Step 1: Calculation of transmission ratio Transmission ratio =  $\frac{z_2}{z_1} = \frac{n_1}{n_2}$  i = Transmission ratio =  $\frac{n_1}{n_2} = \frac{900}{300} = 3$ Step 2: From Design data book, refer Page No. 7.74. For i = 2 to 3 ;  $z_1$  = 25 to 27 i = 3 to 4 ;  $z_1$  = 23 to 25 Take  $z_1 = 23$  to 27 (select any odd no. of teeth) Select  $z_1 = 27$  teeth (no. of teeth on sprocket pinion)

 $z_2 = i \, z_1 = 3 \times 27 = 81$ 

= S2 (no. of teeth on sprocket wheel)

Step 3: From design data book, P.No.7.74

Optimum centre distance = a = (30 to 50) p

where a = approximate centre distance

$$(pitch)_{max} = p_{max} = \frac{550}{30} = 18.33 \text{ mm}$$

$$(pitch)_{min} = p_{min} = \frac{550}{50} = 11 \text{ mm}$$

Select standard pitch from databook,

Take any standard pitch between 11 to 18.33 mm

 $\therefore$  Select Pitch = p = 15.875 mm

ep 4: Selection of Chain No.

Select roller chain from Page No. 7.72

The available chain No. are 10A and 10B

\*Select 10A-2 Duplex Chain.

Pitch = p = 15.875 mm

Corresponding to chain No. selected, take the values of  $a = \text{Bearing area} = 1.4 \text{ cm}^2$ 

 $w = Weight per m length = 1.78 kg_f$ 

 $Q = Breaking load = 4440 kg_f$ 

5: Calculate Power transmitted based on breaking load: From data book, P. No. 7.77

$$N = \frac{Q \cdot v}{102n \cdot K_s} K_w$$

From the above equation calculate 'Q' breaking load by considering N = given power

N = 15 KW

$$V = \frac{z_1 p n_1}{60 \times 1000} = \frac{27 \times 15.875 \times 900}{60 \times 1000}$$

= 6.429 m/sec

n = Minimum value of factor of safety

 $(K_s = 1; z_1 = 15 to 30)$  from data book, P. No. 7.77,

Select n = 11 for a pitch 15.875 mm and  $n_1 < 1000$  rpm

Since the specific conditions are not given, in the problem, assume  $K_1 = K_2 = K_3 = K_4 = K_5 = K_6 = 1$ 

 $\therefore K_s = 1$   $15 = \frac{Q \times 6.429}{102 \times 11 \times 1}$ 

Breaking load  $Q = 2617.825 \text{ Kg}_{f}$  which is less than the selected chain Breaking load (4440 Kg<sub>f</sub>)

The selection of chain no. is satisfactory based on breaking load.

Step 6

(a) Calculation of Length of chain .

· (b) Final centre distance.

(a) Length of continuous chain in multiples of pitches

$$l_p = 2a_p + \frac{z_1 + z_2}{2} = \frac{\left(\frac{z_2 - z_1}{2\pi}\right)^2}{a_p}$$
  
where  $a_p = \frac{a_0}{p}$   
 $= \frac{550}{15.875} = 34.64$   
 $a_0$  = initially assumed centre distance in mm  
 $= 550$  mm  
 $= 34.64$   
 $p$  = pitch = 15.875 mm

No. of teeth on sprocket wheel  $z_2 = 82$ ;

$$\left(\frac{z_2-z_1}{2\pi}\right)^2 = m$$

No. of teeth on sprocket pinion  $z_1 = 27$ 

Read the value of 'm' directly fro databook at  $(z_2 - z_1) = 81 - 27 = 54;$ 

Refer databook Page No. 7.76,

the value of m = 76.6

$$T_p = 2(34.64) + \frac{27 + 82}{2} + \frac{76.6}{34.64}$$

= 125.99

= 126 (approximated to 126)

Length of chain  $= l = l_p \cdot p$ 

= 126 × 15.875 = 2000.25 mm

Take,

l = 2000 mm(b) Final centre distance

$$a = \frac{e + \sqrt{e^2 - 8m}}{4} \cdot p$$
$$e = l_p \doteq \frac{z_1 + z_2}{4}$$

m = 76.6 (from distance)

$$= 126 - \frac{27 + 82}{2}$$
$$= 126 - \frac{109}{2}$$

2

Final centre distance =  $a = \frac{71.5 + \sqrt{71.5^2 - 8 \times 76.6}}{2}$  $\times 15.875$ a = 549.98a = 550 mmStep 7: Check the actual factor of safety. From databook, Refer P.No. 7.78 Actual factor of safety =  $[n] = \frac{Q}{\Sigma P}$ 

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Q = Breaking load of the chain = 4440 Kg  $\Sigma P = P_t + P_s + P_c$ 

 $P_{l}$  = Tangential force due to power Transmission =  $\frac{102 N}{n}$ 

 $P_{f} = \frac{102 \times 15}{6.429} \qquad N = 15 \text{ kW}$ = 237.98 kg<sub>f</sub> V = 6.429 m/sec

$$P_c = \text{Centrifugal fusion} = \frac{W \cdot v^2}{g}$$
$$= \frac{1.78 \times 6.429^2}{9.81}$$
$$W = 1.78 \text{ kg/m length}$$
$$p = 15.875 \text{ mm}$$
$$z_1 = 27$$

k = Coeff of Sag databook 7.78

and the second second

44

 $z_2 = 82$ 

 $P_s =$  tension due to sagging = kWa metres

 $= 6 \times 1.78 \times \frac{550}{1000} = 6$ (Horizontal)

a = 550 mm = 0.55 m

$$\begin{split} \Sigma p &= 237.98 + 7.499 + 5.874 \\ &= 251.353 \text{ Kg}_{\mathrm{f}} \\ [n] &= \frac{Q}{\Sigma P} = \frac{4440}{251.353} \end{split}$$

= 17.66 > 11

which is greater than allowable factor safety

... The design is safe.

Step 8: Checking of Allowable Bearing Stress

From databook, (P.No. 7.77)

The allowable bearing stress  $= \sigma = 2.24 \text{ Kg/mm}^2$ 

(for a pitch of 15.875 and speed < 1000 rpm)

Refer databook, from Pg. no. 7.77

1

Power transmitted on the basis of allowable bearing stress

$N = \frac{(\sigma) av}{102 K_s}  \mathrm{kW}$	
$15 = \frac{(\sigma) \times 1.4 \times 6.429}{102 \times 1}$	N = 15  kW
$(\sigma) = 169.98 \text{ Kgg/cm}^2$ (or)	$a = 1.4 \text{ cm}^2$
$(\sigma) = 1.69 \text{ Kgg/mm}^2$	v = 6.429  m/sec
$(\sigma) < [\sigma]$	$K_s = 1$

< 2.24 Kgg/mm<sup>2</sup>

Therefore the design is safe.

Therefore the decay. **Step 9:** Pitch dia. of small sprocket  $= d_1 = \frac{p}{\sin \frac{180^\circ}{z_1}}$  $=\frac{15.875}{\sin\frac{180^{\circ}}{27}}$ = 136.74 mm  $=d_2=\frac{p}{\sin\frac{180^\circ}{z_2}}$ Pitch dia. of large sprocket  $=\frac{15.875}{\sin \frac{180^{\circ}}{82}}$ = 363.94 mm Scanned with CamScanner



Unit - 2 Spur Gear and Parallel Axis Helical Gear – SMEA1601



## **DESIGN PROCEDURE FOR SPUR GEARS**

- 1. Selection of materials from DDB 8.4.
- 2. Calculation of life N in cycles
- 3. Calculation of equivalent youngs modulus From DDB 8.14
- 4. Calculation of design bending stress  $[\sigma_b]$  from DDB 8.18
- 5. Calculation of design contact stress  $[\sigma_c]$  from DDB 8.18
- 6. Calculation of design twisting moment [M<sub>t</sub>] from DDB 8.15
- 7. Calculation of approximate centre distance (a) from DDB8.13
- 8. Calculation of Z1 and Z2.
- 9. Calculation of normal module (mn) from DDB 8.22, take nearest value fr DDM8.2
- 10. Re Calculation of centre distance from DDB 8.23
- 11. Calculation of face width b from DDB 8.14, table 10
- 12. Calculation of velocity
- 13. To find the constant K and  $K_d$
- 14. Recalculation of design twisting moment [Mt] from DDB 8.15
- 15. Check for bending stress  $\sigma_b$
- 16. Check for contact stress  $\sigma c$

### 1. Calculation of gear ratio (i):

$$\frac{N_A}{\overline{N}_B} = \frac{Z_B}{Z_A}$$

where,  $N_A$  and  $N_B$ =speed of the drive rand driven respectively, and  $Z_A$  and  $Z_B$ =Number of teeth on driver and driven respectively.

### 2. Selection of material

Consulting Table8.4, knowing the gear ratio i, choose the suitable material.

3. If not given, assume gear life(say20000 hrs)

Life in cycles – life in hrs x 60 x RPM

4. Calculation of initial design torque:

 $[M_t]=M_tK. K_d$ 

where, [M<sub>t</sub>]=transmission torque K =Load factor, Table5.11 K<sub>d</sub> =Dynamic load factor, Table5.12

Assume K. K<sub>d</sub>=1.3 (ifnot given)

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

✓ From table8.14,T.NO.9 Calculate  $E_{eq}$ ✓ Calculate Design bending stress [ $\sigma_b$ ]

✓ Calculate Design contact stress  $[\sigma_c]$ by  $[\sigma_c]=C_B$ . HB.  $K_{cl}$  (or)  $[\sigma_c]=C_R$ . HRC.  $K_{cl}$ 

C<sub>B</sub>C<sub>R</sub>=Coefficient of surface hardness from table8.16,T.No.16

where,

HBHRC =Hardness number

6. Calculation of centre distance (a):

$$a \ge (i+1) \sqrt{\left(\frac{.74}{[c]}\right)^2} \frac{Eed[Mt]}{i\varphi}$$

 $\varphi = b/a$  from table5.21

### 7. Select number of teeth on gear and pinion:

>On pinion,  $Z_1$ =Assume18 >On gear,  $Z_2$ =i x  $Z_1$ 

#### 8. Calculation of module:

$$m = -\frac{2a}{z(1+z2)}$$

Choose standard module from table5.8

#### 9. Revision of centre distance (m):

$$a=\frac{(r-z1-z2)}{2}$$

## 10. Calculate b, $d_1$ , v and $\psi_{p:}$

✓ Calculate face width,  $b = \psi . a$ 



- ✓ Calculatepitch dia,  $d = m.z_1$
- ✓ Calculatepitch linevelocity,  $v = (\pi d_1 N_1)/60$
- ✓ Calculatevalue of  $\psi_p = b/d_1$

#### 11. Selection of quality of gear:

Knowing the pitch linevelocityandconsultingtable5.22, select a suitable quality Of gear.

#### 12. Revision of design torque [M<sub>t</sub>]: Revise K:

Using the calculated value of  $\psi_p$  revise the K value by using table5.11

$$[M_t]=M_tK. K_d$$

#### **13. Check for bending:**

Calculate induced bending stes s,

$$G_{\underline{b}}^{\underline{(i\pm 1)}} bY$$
 []

✓ Compare  $G_{b}$  and  $[G_{b}]$ .

✓ If  $G_b \leq [G_b]$ , then design is safe.

#### 14. Check for wear strength:

Calculate induced contact str.

$$G_{c=}0.74^{\left(\frac{i\pm 1}{a}\right)} \sqrt{\frac{i\pm 1}{ib}} Eeq\left[Mt\right]$$

✓ Compare  $G_c$  and  $[G_c]$ .

✓ If  $G_c \le [G_c]$ , then design is safe.

15. If the design is not satisfactory  $(\mathbf{\sigma}_b > [\mathbf{\sigma}_b]$ and/or  $\mathbf{\sigma}_c > [\mathbf{\sigma}_c]$ ), then increase the module of face width value of the gear material



## PROBLEM

Design a spur gear to transmit 22.5 kW at 900 rpm, speed reduction is 2.5, material for pinion and wheel are C15 and Cast iron grade 30. Take pressure angle of  $20^{\circ}$  and working life of the gears are 10,000 hrs.

Material Given: PINION- C15 - Steel Wheel - Cast iron grade 30 Speed reduction = i = 2.5ie, Gear ratio  $= i = \frac{z_2}{z_1} = 2.5$ Power = P = 22.5 kW Pinion Speed  $= n = N_1 = 900$  rpm; Life = N = 10,000 hrs

# Step 1: Material properties

Refer P.No. 8.16; Table 16 Pinion: C15 (Given) Assume C15 - Case hardened HRC = 55 to 63 assume HRC = 60ie, Surface hardness > 350 BHN  $\therefore$  core hardness < 350 BHN Wheel: Cast Iron - GRADE 30 HB = 200 to 260 Assume surface hardness HB = 250 (Brinell hardness)  $\therefore$  Core hardness < 350

Step 2: Life required = 10,000 hrs = 10000 × 60 × 900 cycles Life required =  $N = 54 \times 10^7$  cycles. Step 3: Calculation of Equivalent Young's Modulus Refer P.No. 8.14; Table 9 Pinion material - Steel -  $E_1 = 2.15 \times 10^6 \text{ Kg}_f/\text{cm}^2$ Gear (or) Wheel material - C.I. Grade 30 -  $E_2 = 1.4 \times 10^6 \text{ Kg}_f/\text{cm}^2$ Therefore, equivalent young's modulus  $= E = 1.7 \times 10^6 \text{ Kg}_f/\text{cm}^2$ Since  $\sigma_u > 28 \text{ Kg/mm}^2$ Kg<sub>f</sub>/cm<sup>2</sup> For CI Grade 30, the ultimate stress  $\sigma_u = 30 \text{ Kg}_f/\text{mm}^2$ (Convert into N/mm<sup>2</sup>)  $E = 1.7 \times 10^5 \text{ N/mm}^2$ Step 4: Calculation of design bending stress  $[\sigma_b]$ (Refer P.No. 8.18) Design bending stress  $[\sigma_b] = \frac{1.4 K_{bl}}{n K_{-1}} \sigma_{-1}$ (assume rotation in one direction)  $K_{bl} = \text{Life factor for bending} = 1$ Table (22); P.No. 8.20 (Consider core hardness) = 1 (for  $N > 10^7$  cycles and BHN < 350) n = Factor of safetyTable (20) P.No. 8.18

=2

(for case hardend)

 $k_{\sigma} =$  fillet stress cont atration factor

Refer Table (21) P.No. 8.19 for case hardened,  $(0 \le x < 0.1)$ = 1.2X = Addendum modification coefficient. $\sigma_{-1}$  = Endurance limit stress in bending for complete reversal of stresses Refer P.No. 8.19; table 19 Assume C15 as forged steel, for forged steel =  $\sigma_{-1} = 0.25 (\sigma_u + \sigma_y) + 500$ From P.No. 1.9 for C15 material.  $\sigma_n = 37 \text{ to } 49 \text{ Kg/mm}^2$  $\sigma_v = 24 \text{ Kg/mm}^2$ where: select  $\sigma_u = 40 \text{ Kg/mm}^2$  $\sigma_y = \text{yield stress Kg/cm}^2$ = 40 × 100 Kg/cm<sup>2</sup>  $\sigma_u =$  ultimate stress Kg<sub>f</sub>/cm<sup>2</sup>  $\sigma_v = 24 \times 100 \text{ Kg/cm}^2$  $\therefore \, \sigma_{\!-1} = 0.25 \, (40 \times 100 + 24 \times 100) + 500$  $= 2100 \text{ Kg}_{f}/\text{cm}^{2}$ 

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... Design bending stress = 
$$[\sigma_b] = \frac{1.4 K_{bl}}{n \cdot K_{\sigma}} \sigma_{-1}$$
  
=  $\frac{1.4 \times 1}{2 \times 1.2} \times 2100 = 1225 \text{ Kg/cm}^2$   
(Convert into N/mm<sup>2</sup>)  
 $[\sigma_b] = 122.5 \text{ N/mm}^2$ 

Step 5: Calculation of Design surface (Contact compressive stress) [σ<sub>c</sub>] P.No. 8.16

Design surface (contact compressive stress)

 $= [\sigma_e] = C_R \cdot HRC \cdot K_{Cl} \cdot K_{cl} \cdot K_{cl} \cdot cm^2$ 

 $C_R$  = Coeff. depending on surface hardness

Table 16 P.No. 8.16

( = 220 (for C15, Case hardened)

HRC = 60 (assumed)

 $K_{cl}$  = Life factor (Consider surface hardness)

Т

Table 17: P.No. 8.17

= 0.585 for N > 107 cycles and surface hardness > 350 BHN

 $[\sigma_c] = 220 \times 60 \times 0.585$ 

 $[\sigma_c] = 7722 \text{ Kg/cm}^2 (\text{Convert into N/mm}^2)$ 

Design surface (Contact compressive) stress =  $[\sigma_c] = 772.2 \text{ N/mm}^2$ 

Step 6: Calculation of design torque  $[M_t]$ 

P. No. 8.15  $[M_t] = M_t \cdot K_d \cdot K$   $M_t = 97420 \frac{\text{kW}}{n}$   $= \frac{97420 \times 22.5}{900}$  $= 2435.5 \text{ Kg}_f - \text{cm}$ 

Initially assume, for symmetry scheme

 $K_d \cdot K = 1.3$ 

(

 $[M_l] = 2435.5 \times 1.3 = 3166.15 \text{ Kg}_{\Gamma}\text{cm}$  (Convert into N-mm)

 $[M_t] = 316615$  N-mm

Step 7: Calculation of Approximate centre distance 'a'. P.No.8.13

Centre distance = 
$$a \ge (i+1)$$
  $\sqrt[3]{\left\{\frac{0.74}{[\sigma_c]}\right\}^2} \frac{E[M_t]}{i\psi}$ 

 $[\sigma_c] = 772.2 \text{ N/mm}^2$  from Step (5)

 $E = 1.7 \times 10^5 \text{ N/mm}^2$  from Step (3)

 $[M_t] = 316615 \text{ N-mm}$  from Step (6)

where  $b = \text{face width -mm}_{a}$ a = centre distance -mm

kW = 22.5 kWn = 900 rpm

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Assume  $z_1 =$  no. of teeth on pinion = 25 teeth.

(Refer P.No. 8.1)

Speed reduction  $i = \frac{z_2}{z_1}$ 

 $z_2 =$  No. of teeth on wheel  $= i \cdot z_1$ 

 $= 2.5 \times 25$ 

= 62.5 teeth

Take  $z_2 = 63$  teeth

Step 9: Calculation of module (P.No. 8.22)

Take, Module  $= m = \frac{2a}{z_1 + z_2}$  $= \frac{2 \times 142.74}{(25 + 63)} = \frac{285.48}{88}$ = 3.24 mm

Take Standard module, from P.No. 8.2 Table (1)

\* module = 4 mm

Step 10: Recalculate centre distance 'a' and rounded to R10 series.

$$a = \frac{m(z_1 + z_2)}{2}$$
$$= \frac{4(25 + 63)}{2}$$

Centre distance =  $a = 176 \text{ mm} \approx \text{Standard R10 series } a = 200 \text{ mm}$ 

Step 11: Calculation of face width (b)

 $\psi$  value already assumed as 0.3 but  $\psi = \frac{b}{a} = 0.3$ 

face width  $= b = 0.3 \times 200 = 60 \text{ mm}$ 

Step 12: Calculation of PCD  $(d_1)$  and PCD  $(d_2)$ 

PCD of pinion  $= (PCD)_1 = d_1 = m \cdot z_1$ 

 $= 4 \times 25 = 100 \text{ mm}$ 

PCD of wheel =  $(PCD)_2 = d_2 = m \cdot z_2$ 

 $= 4 \times 64 = 252 \text{ mm}$ 

Step 13: Cal. of pitch line velocity: (v)

$$v = \frac{\pi d_1 N_1}{60}$$
 m/sec  
=  $\pi \times 0.1 \times \frac{900}{60}$  = 4.712 m/sec

From table (2); P.No. 8.3

at v = upto 8 m/sec

Select the Quality of Gear - IS Quality 8

Step 15: Find load concentration factor (k) for IS Quality 8 Gear,

P.No. 8.15; table (14)

k depends on  $\psi_p = \frac{b}{d_1}$  value

$$=\frac{60}{100}=0.6$$

at  $\psi_p = 0.6$ ; the value of k = 1.03

Step 16: Find Dynamic load factor kd

Refer P.No. 8.16; Table (15)

for pitch line vel up to 8 m/sec, cylindrical gears,

surface hardness > 350 BHN

 $K_{d} = 1.4$ 

 $d_1 = 100 \text{ mm}$  $N_1 = 900 \text{ rpm}$ 

 $b_1 = 52.8 \text{ mm}$  $d_1 = 100 \text{ rpm}$  Accredited "A" Grade by NAAC | 12B Status by UGC | Approved by AlCTE

Step 17: Recalculate, [M1] design twisting moment,

 $[M_t] = M_t k_d K$ 

 $M_t = 2435.5 \text{ Kg}_{\Gamma} \text{cm}$ 

take average value of y

24 
ightarrow 0.414

 $26 \rightarrow 0.427$ 

 $25 \rightarrow 0.4205$ 

Contra an tak

 $= 2435.5 \times 1.4 \times 1.03$ 

 $[M_{f}] = 3511.99 \text{ Kg}_{f} \text{ cm}$ 

[M,] = 351199 N-mm.

Step 18: Check ob induced bending stress

P.No. 8.13; table (8)  $\sigma_b = \frac{i+1}{a \cdot m \cdot b \cdot y} [M_t] \le [\sigma_b]$ (M) = 251100 N

 $[M_i] = 351199 \text{ N-mm}$ 

i = 2.5a = 200 mm

m = 4 mm

b = 60 mm

y = form factor

(P.No. 8.18) table (18)

(y value depends X and z value)

for  $z_1 = 25; X = 0;$ 

= 0.4205

 $\sigma_b = \frac{2.5 + 1}{200 \times 60 \times 0.4205}$  351199 = 60.89 N/mm<sup>2</sup> < [ $\sigma_b$ ] = 1225 N/mm<sup>2</sup>

Step 19: Check oc induced surface (contact compressive stress)

P.No. 8.13, Table (8)

$$σ_c = 0.74 \frac{i+1}{i \cdot b} \sqrt{\frac{i+1}{a}} E[M_i]$$
  
= 0.74  $\frac{2.5+1}{200} \sqrt{\frac{2.5+1}{2.5 \times 60} \times 1.7 \times 10^5 \times 351198}$   
= 483.34 N/mm<sup>2</sup> < [σ<sub>c</sub>] = 772.2 N/mm<sup>2</sup>  
∴ The design of pinion is ssatisfactory.

1

Step 20: Check the stresses in the wheel:

(a) Gear Material

Given - cast iron - Grade (30)

Since contact area is same,

 $\sigma_{c_{\rm wheel}} = \sigma_{c_{\rm pinkn}} = 483.34 \ {\rm N/mm}^2$ 

(Already calculated)

Life of Wheel = (N)  $_{\text{Wheel}} = \frac{N_{\text{pinion}}}{2.5} = \frac{54 \times 10^7}{2.5} = 21.6 \times 10^7 \text{ cycles}$ 

From P.No. 8.16,

 $[\sigma_c] = \text{design surface contact compressive stress}$ 

 $= C_B HB \cdot K_{CL}$ 

for CI - Grade (30) HB = 200 to 260 BHN take HB = 260 BHN  $C_B = 23;$  HB = 260

K<sub>CL</sub> = from table (17); P.No. 8.17

$$= \sqrt[6]{\frac{10^7}{N}} = \sqrt[6]{\frac{10^7}{21.6 \times 10^7}} = 0.5992$$

 $[\sigma_c] = 23 \times 260 \times 0.5992 = 3583.216 \text{ Kg}_f/\text{cm}^2$ 

Convert into (N/mm<sup>2</sup>)

$$= \sigma_c > [\sigma_c]$$

The design of wheel is not satisfactory.

Therefore, change the material of wheel and recalculate  $[\sigma_c]$  and  $\sigma_c < [\sigma_c]$ Select another material, $K_{CL} = \text{Surface hardness} > 350$ 

Т

From P.No. 8.16; Table 16

Select either C40 or C45 Steel;

Heat treatment - Surface hardened.

 $=N < 25 \times 10^7$  $= \sqrt{\frac{10^7}{N}}$ = 0.5992 Select C40 - Surface hardened

HRC = 40 to 55

Assume HRC = 50;  $C_R = 230$ ;  $K_{Cl} = 0.5992$ 

Design surface  $[\sigma_c] = C_R HRC K_{Cl}$ 

Contact Compressive stress  $= 230 \times 50 \times 0.5992 = 6890.8 \text{ Kg}_{f}/\text{cm}^{2}$ 

 $[\sigma_c] = 689.08 \text{ N/mm}^2$ 

## $\therefore \sigma_c < [\sigma_c]$

Therefore the selection of material C40, Surface hardened is satisfacto \* Checkup of  $\sigma_b$  induced bending stress;

 $\sigma_{b_1} y_1 = \sigma_{b_2} y_2$   $\sigma_{b_1}$  - (induced bending stress)

 $[\sigma_{b_2}]_2$  - design bending stress of wheel  $\sigma_{b_2}$  (induced bending stress)<sub>wl</sub>

$$=\frac{1.4 K_{bl}}{K_n \cdot n} \sigma_{-1}$$

 $\sigma_{b_2} < [\sigma_b]_{2_{\text{wheel}}}$ 

Find  $[\sigma_b]_{2_{(wheel)}}$ 

Design bending stress

$$= [\sigma_b]_{\text{wheel}} = \frac{1.4 K_{bl}}{K_{\sigma} n} \sigma_{-1}$$

 $k_{bl} = 1$  table (22)

< 350 BHN core hardness;  $N > 10^7$ n = 2.5 table (20)  $y_1 = (\text{form factor})_{\text{pinion}} = 0.42$  $y_2 = (\text{form factor})_{\text{wheel}} = 0.45$ 

at z = 63; x = 0 by interpo

 $\sigma_{-1} = 0.25 (\sigma_u + \sigma_y) + 500$ for C 40 material, P.No. 1  $\sigma_u = 58 \text{ to } 68 = 1054.66 \text{ Kg}_f$ Assume  $\sigma_u = 60 \text{ Kg}_f/\text{mm}^2$  $= 60 \times 10^2 \text{ Kg}_f/$






## **DESIGN PROCEDURE FOR HELICAL GEARS**

- 1. Selection of materials from DDB 8.4.
- 2. Calculation of life N in cycles
- 3. Calculation of equivalent youngs modulus From DDB 8.14
- 4. Calculation of design bending stress  $[\sigma_b]$  from DDB 8.18
- 5. Calculation of design contact stress  $[\sigma_c]$  from DDB 8.18
- 6. Calculation of design twisting moment [M<sub>t</sub>] from DDB 8.15
- 7. Calculation of approximate centre distance (a) from DDB8.13
- 8. Calculation of Z1 and Z2.
- 9. Calculation of normal module (mn) from DDB 8.22, take nearest value fr DDM8.2
- 10. Re Calculation of centre distance from DDB 8.23
- 11. Calculation of face width b from DDB 8.14, table 10
- 12. Calculation of velocity
- 13. To find the constant K and  $K_d$
- 14. Recalculation of design twisting moment [Mt] from DDB 8.15
- 15. Check for bending stress  $\sigma_b$
- 16. Check for contact stress  $\sigma c$

## 1. Calculation of gear ratio(i):

$$\frac{N_A}{\overline{N}_B} = \frac{Z_B}{Z_A}$$

where,  $N_A$  and  $N_B$ =speed of the driver and driven respectively, and  $Z_A$  and  $Z_B$ =Number of teeth on driver and driven respectively.

#### 2. Selection of material

Consulting Table8.3, knowing the gear ratio i, choose the suitable material.

3. If not given, assume gear life(say20000 hrs)

Life in cycles – life in hrs x 60 x RPM

#### 4. Calculation of initial design torque:

 $[M_t]=M_tK. K_d$ 

where, [M<sub>t</sub>]=transmission torque

- K =Load factor, Table
- K<sub>d</sub> =Dynamic load factor, Table

Assume K. K<sub>d</sub>=1.3 ( if not given)

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

✓ From table8.16 Calculate  $E_{eq}$ 

 $\checkmark$  From table8.16 Calculate Design bending stress [ $\sigma_b$ ]

✓ Calculate Design contact stress  $[\sigma_c]$  by

$$[\boldsymbol{\sigma}_{c}]=C_{B}$$
. HB.  $K_{cl}$  (or)

 $[\boldsymbol{\sigma}_{c}]=C_{R}$ . HRC.  $K_{cl}$ 

C<sub>B</sub>C<sub>R</sub>=Coefficient of surface hardness from table8.16

HBHRC =Hardness number

## 6. Calculation of centre distanc e (a):

$$a \ge (i+1) \sqrt{\left(\frac{.74}{[c]}\right)^2 \frac{Eeq}{i\varphi}}$$

 $\varphi = b/a$  from table 5.21

- 7. Select number of teeth on gear and pinion: >On pinion,  $Z_1$ =Assume>17
  - Solution philon,  $Z_1$ -Assume  $Z_1$ Solution,  $Z_2$ =i x  $Z_1$
- 8. Calculation of module:

$$m = \frac{2a}{z(1+z^2)} \times \cos \beta$$

Choose standard module from table5.8

#### 9. Revision of centre distance (m)

$$a=\frac{m(z_{1}+z_{2})}{\cos\beta}$$

#### **10.** Calculate b, $d_1$ , v and $\psi_{p:}$

✓ Calculate face width,  $b = \psi.a$ 



- ✓ Calculate pitch dia,  $d = (m_n.z_1)/\cos\beta$
- ✓ Calculate pitch line velocity, v =( $\pi d_1 N_1$ )/60
- ✓ Calculate value of  $\psi_p = b/d_1$

#### 11. Selection of quality of gear:

Knowing the pitch line velocity and a consulting table, select a suitable quality Of gear.

## 12. Revision of design torque[M<sub>t</sub>]:

$$[M_t]=M_t K. K_d$$

#### **Check for bending:**

Calculate induced beding stress,

$$\mathbf{G}_{\overline{b}}^{0.7\ (i\ \pm 1)}_{\overline{b}}\ Mt]$$

✓ Compare  $G_b$  and  $[G_b]$ . ✓ If  $G_b \le [G_b]$ , then design is safe.

#### 14. Check for wear strength:

Calculate induced contactts se

$$\mathbf{G}_{c=0.7}^{(-i\pm1)}(\sqrt{\frac{i\pm1}{ib}} Eeq [Mt])$$

✓ Compare  $G_c$  and  $[G_c]$ .

✓ If  $G_c \leq [G_c]$ , then design is safe.

15. If the design is not satisfactory( $\mathbf{\sigma}_b > [\mathbf{\sigma}_b]$  and/or $\mathbf{\sigma}_c > [\mathbf{\sigma}_c]$ ), then increase the module of face width value of the gearmaterial.

#### 16. Check for gear:

c. Check for bending: Using  $G_{b1}$ ,  $y_1$  and  $f_{b1}$ ,  $y_1$ .

$$G_{b2} = \frac{b1.y1}{y2}$$



 $\label{eq:compare_star} \begin{array}{l} \checkmark Compare \ \ensuremath{\mathcal{G}_{b2}}\ and \ \ensuremath{[\mbox{$\mathcal{G}_{b2}$}]\ .} \\ \ensuremath{\checkmark}\ \ensuremath{If\mbox{$\mathcal{G}_{b2}$}\ \le \ \ensuremath{[\mbox{$\mathcal{G}_{b2}$}]\ .} \\ \ensuremath{\checkmark}\ \ensuremath{If\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{[\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ .} \\ \ensuremath{\sim}\ \ensuremath{If\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ .} \\ \ensuremath{\sim}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ . \\ \ensuremath{\sim}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{]}\ \ensuremath{}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{}\ \ensuremath{\mbox{$\mathcal{G}_{b2}$}\ \ensuremath{}\ \ensu$ 

## d. Check for wear strength:

Calculate induced contact stress will be same for pinion and gear, So,  $\sigma_{c2} = \sigma_c$ 

✓ Compare  $G_c$  and  $[G_c]$ ✓ If  $G_c \le [G_c]$ , then design is safe

## PROBLEM

Design a pair of helical gears for the following data:

Power- 7.5 kW, speed of pinion -1440 rpm, speed reduction -3, pressure angle- $20^{\circ}$ , Helix angle –  $10^{\circ}$ . Select the materials and heat treatment.

## Solution:

## Step:1

Selection of material and heat treatment fromDDB.8.16, assume same material for both pinion and wheel

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Select Alloy steel with case hardened neat treatment (Refer P. No. 1.14) . 40 Cr 1 MO 28 - Material,  $\sigma_u=70-85~{\rm Kg_f}/{\rm mm^2}$ 

Take  $\sigma_u = 80 \text{ Kg}_f/\text{mm}^2$ 

HRC = 55 to 63 and  $C_R = 280$ 

Take HRC = 60 Surface hardness > 350 BHN Core hardness < 350 BHN

Step 2: Life in cycles.

Assume life of the gear drive = N = 10,000 hrs

 $\therefore N = 10,000 \times 60 \times RPM = 10,000 \times 60 \times 1400$ 

 $= 84 \times 10^{7}$ 

Step 3: Equivalent young's modulus (refer databook P.No. 8.14) Since the material for the pinion and wheel is same,

 $E_{eq} = 2.15 \times 10^{6} \mathrm{Kg_{f}/cm^{2}}$ 

**Step 4:** Calculation of design bending stress  $[\sigma_{\delta}]$ 

(Refer databook P.No. 8.18)

Design bending stress  

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

$$K_{bl} = \text{Life factor for bending (P.No. 8.20)}$$

$$(Consider core hardness) = 1$$

$$(Conster core hardness) = 1$$

$$(Conster core hardness) = 1$$

$$(Const$$

Step 5: Calculation of design surface (contact compressive) stress. (Refer databook, P.No. 8.16)

Design surface (contact compressive stress) =  $[\sigma_c] = C_R HRC K_{Cl}$ 

 $[\sigma_c] = 280 \times 60 \times 0.585 \qquad C_R = 280$  $[\sigma_c] = 9828 \text{ Kgr/cm}^2 \qquad HRC = 60$ KCl = Life factor (P. No. 8.17)

(Consider surface hardness)

1.00

= 0.585

Step 6: Calculation of Design twisting moment [M] (Refer databook, P. No. 8.15)

Design twisting moment  $= [M_t] = M_t \cdot K_d \cdot K$   $= 521.89 \times 1.3$   $M_t = 97420 \frac{kW}{n}$   $[M_t] = 678.46 \text{ Kgr} - \text{cm}$   $= \frac{97420 \times 7.5}{1400}$  = 521.89 Kgr - cm.

Step 7: Calculation of approximate centre distance (a)

(Refer databook, Page No. 8.13)

Centre distance 
$$a \ge (i+1) \sqrt[4]{\left\{\frac{0.7}{|\sigma_c|}\right\}^2 \frac{E[M_t]}{i\psi}}$$
  
 $a \ge (3+1) \sqrt[4]{\left\{\frac{0.7}{9828}\right\}^2 \frac{2.15 \times 10^6 \times 678.46}{3 \times 0.3}}$  assume  $\psi = \frac{b}{a}$   
 $a \ge 8.07$  cm (Open type gearing)  
(Refer P.No. 8.14)

1

i = 3

(Centre distance should be rounded to R10 series)

Refer databook, Page No. 7.20

Take, Centre distance = a = 80 mm

**Step 8:** Calculation of  $z_1$  and  $z_2$ 

Assume,  $z_1 = No.$  of teeth on pinion = 20

No. of teeth on wheel  $z_2 = i \cdot z_2 = 3 \times 20 = 60$  teeth

Step 9: Calculation of Normal module  $(m_n)$ 

(refer page No. 8.22)

$$m_n = \text{Normal module} = \frac{2a \cos \beta}{z_1 + z_2}$$

 $m_n = \frac{2 \times 80 \times \cos 10^\circ}{20 + 60} = 1.9696 \text{ mm}$ 

Select recommended value of module, (from P.No. 8.2) Select, module =  $m_n = 2.5$  mm

(For safe design, always select  $m_h$  slightly greater than required) Step 10: Recalculate the centre distance (a)

 $a = \frac{m_n}{\cos \beta} \frac{(z_1 + z_2)}{2}$  $= \frac{2.5}{\cos 10} \frac{(20 + 60)}{2}$ 

a = 101.54 mm

 $\therefore$  Take, a = centre distance = 100 mm (nearest value) (from R10 series)

tep 11: Calculation of face width 'b'.

 $\Psi = \frac{b}{a} = 0.3$   $\therefore b = 0.3 \times a = 0.3 \times 100 = 30 \text{ mm}$ Face width = b = 30 mm Step 12: Calculation of pitch line velocity, v in m/sec.

Refer P. No. 8.22

Pitch line velocity  $v = \frac{\pi d_1 n_1}{60} m_{\text{sec}}$ 

 $d_1 =$  pitch circle dia of pinion

= 3.7216 m/sec

 $d_1 = \frac{m_n}{\cos\beta} z_1$  $=\frac{2.5}{\cos 10} \times 20$ 

= 50.77 mm

Selection of Quality of gears: (Refer page No. 8.3)

Select IS Quality - Medium

Preferred Quality - 8

Step 13: To find the constants K and  $K_d$ .

K = Load correction factor

 $= 1.03 \text{ at } \psi = 0.6$ 

 $\psi = \frac{b}{d_1} = \frac{30}{50.77} = 0.59$ 

 $\psi \approx 0.6$ 

 $K_d$  = Dynamic load factor = 1.3(For IS-8 Quality, and

velocity upto 3 m/sec, > 350 BHN surface hardness)

Step 14: Recalculation of  $[M_t]$ 

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

 $= 521.89 \times 1.03 \times 1.3 = 698.81 \text{ Kg}$ -cm

Step 15: Checking of bending stress.

Refer Page No. 8.13  $y_v = \text{form factor}$ Refer P.No. 8.18

$$\sigma_{b} = 0.7 \frac{i \pm 1}{a \cdot b \cdot m_{n} \cdot y_{v}} [M_{t}] \le [\sigma_{b}]$$

$$= 0.7 \frac{(3+1)}{10 \times 3 \times 0.25 \times 0.3955} (698.81)$$

$$= 659.64 \text{ Kgr}/\text{cm}^{2} < [\sigma_{b}]$$

$$= 20.93$$

$$= 21 \text{ teeth}$$

 $\therefore$  Design is satisfactory based on  $y_v = 0.3955$  at  $z_v = 21$  teeth bending stress

# Step 16: Checking of Surface (contact compressive) stress

refer databook, P. No. 8.13

$$\sigma_{c} = 0.7 \, \frac{i \pm 1}{a} \sqrt{\frac{i+1}{ib} E[M_{t}]} \leq [\sigma_{c}]$$
$$= 0.7 \, \frac{3+1}{10} \sqrt{\frac{3+1}{3 \times 3} \times 2.15 \times 10^{6} \times 698.83}$$
$$= 7235.45 \, \text{Kg}_{f}/\text{cm}^{2} < [\sigma_{c}]$$

Therefore, the design is satisfactory based on surface (contact con stress.



# **Unit 3 BEVEL AND WORM GEAR DRIVES – SMEA1601**

Bevel gears are gears where the axes of the two shafts intersect and the toothbearing faces of the gears themselves are conically shaped. Bevel gears are most often mounted on shafts that are 90 degrees apart, but can be designed to work at other angles as well. The pitch surface of bevel gears is a cone.

## TYPES

- Straight bevel gears have conical pitch surface and teeth are straight and tapering towards apex.
- Spiral bevel gears have curved teeth at an angle allowing tooth contact to be gradual and smooth.
- Zerol bevel gears are very similar to a bevel gear, but the teeth are curved: the ends of each tooth are coplanar with the axis, but the middle of each tooth is swept circumferentially around the gear. Zerol bevel gears can be thought of as spiral bevel gears, which also have curved teeth, but with a spiral angle of zero, so the ends of the teeth align with the axis.
- Hypoid bevel gears are similar to spiral bevel, but the pitch surfaces are hyperbolic and not conical. The pinion can be offset above or below the gear center, thus allowing larger pinion diameter, longer life, and smoother mesh. If the beveled surface is made parallel with the axis of rotation, this configuration resembles a worm drive. Hypoid gears were widely used in automobile rear axles

## MITER GEAR

Miter gears are a special case of bevel gears that have equal numbers of teeth. The shafts are positioned at right angles from each other, and the gears have matching pitch surfaces and angles, with a conically-shaped pitch surface. Miter gears are useful for transmitting rotational motion at a 90-degree angle with a 1:1 ratio

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## **MITER GEAR**

## APPLICATIONS

The bevel gear has many diverse applications such as locomotives, marine applications, automobiles, printing presses, cooling towers, power plants, steel plants, railway track inspection machines, etc.

For examples, see the following articles on:

- Bevel gears are used in differential drives, which can transmit power to two axles spinning at different speeds, such as those on a cornering automobile.
- Bevel gears are used as the main mechanism for a hand drill. As the handle of the drill is turned in a vertical direction, the bevel gears change the rotation of the chuck to a horizontal rotation. The bevel gears in a hand drill have the added advantage of increasing the speed of rotation of the chuck and this makes it possible to drill a range of materials.
- The gears in a bevel gear planer permit minor adjustment during assembly and allow for some displacement due to deflection under operating loads without concentrating the load on the end of the tooth.
- Spiral bevel gears are important components on rotorcraft drive systems. These components are required to operate at high speeds, high loads, and for a large number of load cycles. In this application, spiral bevel gears are used to redirect the shaft from the horizontal gas turbine engine to the vertical rotor. Bevel gears are also used as speed reducers



## ADVANTAGES

- This gear makes it possible to change the operating angle.
- Differing of the number of teeth (effectively diameter) on each wheel allows mechanical advantage to be changed. By increasing or decreasing the ratio of teeth between the drive and driven wheels one may change the ratio of rotations between the two, meaning that the rotational drive and torque of the second wheel can be changed in relation to the first, with speed increasing and torque decreasing, or speed decreasing and torque increasing.

## DISADVANTAGES

- One wheel of such gear is designed to work with its complementary wheel and no other.
- Must be precisely mounted.
- The shafts' bearings must be capable of supporting significant forces.

## **DESIGN PROCEDURES**

- 1. Calculation of gear ratio from psg data book pg 8.6
- 2. Selection of materials. from psg data book pg 1.40
- 3. Selection of gear life based on given data.
- 4. Calculation of initial design torque from psg data book pg 8.15. table no13
- 5. Determination of equivalent young's modulus from psg data book pg8.14. table no.9
- 6. Calculation of design contact stress from psg data book pg 8.16. table no.15
- 7. Calculation of design of bending stress from psg data book pg 8.18.
- 8. Calculation of cone distance from psg data book pg8.13. table no.8
- 9. Selection of Z1 and Z2 assume Z1 initially as  $\geq 17$ .



10.Calculation for transverse module psg data book pg8.38 table no31.

- 11.Revision of cone distance psg data book pg8.38 table no.31.
- 12.Calculation of face width from psg data book pg8.15 table no. 13
- 13.Calculation of reference diameter from psg data book pg8.38 table no. 31.
- 14.Selection of quality of gear from psg data book pg8.3 table no. 2
- 15.Revision of design torque from psg data book pg8.15
- 16.Check for contact stress from psg data book pg8.13
- 17.Check for bending stress from psg data book pg8.13 A.



#### Problem:

A pair of 20° full depth involute teeth bevel gear connects two shafts at right angles having a velocity ratio of 3.2: 1. The gear is made of cast steel with an allowable static stress as72 N/mm<sup>2</sup>, and the pinion is made of steel having a static of 100 N/mm<sup>2</sup>. The pinion transmits 40kW and at 840 rpm. Find the module, face width, and pitch diameter from the stand point of the beam strength, and check the design from the stand point of wear.

#### Given data:

 $\Theta = 90^{\circ}$ ,  $\dot{\alpha} = 20^{\circ}$ , i = 3.2,  $[\sigma b2] = 72 \text{ N/mm}^2$ ,  $[\sigma b1] = 100 \text{ N/mm}^2$ , P = 40 kW,

N1= 840 rpm

To find:

Module, face width, and pitch diameter of the gears.

#### Solution:

Step: 1: Selection of materials Material for gear : Cast Iron Material for pinion: Steel

Step: 2: Assume  $Z_1 = 20$ , then  $Z_2 = i \times Z1$   $Z_2 = 3.2 \times 20 = 64$  $Z_2 = 64$ 

Step : 3: Calculation of pitch angle [ (ie)  $\delta 1$  and  $\delta 2$  ] and the virtual number of teeth (ie) Zp<sub>1</sub> and Zp<sub>2</sub> using the following relations ,

$$\tan \delta_2 = i = 3.2 \text{ or } \delta_2 = \tan^{-1} (3.2) = 72.64^{\circ}$$
  

$$\tan \delta_1 = 90^{\circ} - \delta_2$$
  

$$\delta_1 = 90^{\circ} - 72.64^{\circ} = 17.36^{\circ}$$
  
The virtual number of the teeth on the gears is given by  

$$Zv1 = Z1 = 20$$

$$Zv2 = \begin{array}{c} Cos \, \delta 1 & cos \, 17.36^{\circ} \\ Zv1 = 21 \\ Cos \, \delta 2 & cos \, 72.64^{\circ} \\ Zv2 = 215 \end{array}$$

Then form factors based on virtual number of teeth are given by

Ý1 = 0.154 - 0.912  
Zv1  
Ý1 = 0.1106  
Ý2 = 0.154 - 0.912  
Zv2  
Ý2 = 0.1497  
For pinion: 
$$[\sigma b1] \times ý1 = 100 \times .0116 = 11.06 \text{ N/mm}^2$$
  
For gear:  $[\sigma b2] \times ý2 = 72 \times 1497 = 10.78 \text{ N/mm}^2$   
e the value of gear is less than pinion. Thus we have to design for ge

Hence the value of gear is less than pinion. Thus we have to design for gear only.

Step : 4: Calculating the tangential load using the relation We know that ,

where  $v = \prod d1 N1 = \prod x N1 \times (mt x Z1)$  60 = 60 1000  $= \prod x 840 \times (mt x 20) = .879 mt$  60 1000Ko = 1.25, assuming medium shock

$$Ft = \frac{40 \times 10^{3}}{0.879 \text{ mt}} \times 1.25 = \frac{56841}{\text{mt}}$$

Step :5:

$$Fd = \frac{Ft}{Cv} \qquad \text{where } Cv = \frac{5.6}{5.6 + \sqrt{V}}$$

where V = 5 m/s

$$Cv = 0.715$$
Fd = 56841 x 1
mt 0.715
Fd = 79497.9
mt

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Step: 6: Calculating the preliminary value of dynamic load using the relation

 $Fs = \prod x mtx bx [Ob] x y x [R-b]$ R b = 10 mt ; Ý2 = 0.1497 Where  $[\sigma b2] = 72 \text{ N/mm}^2$ ; R= 0.5 mt x  $\sqrt{(Z1^2 + Z2^2)} = 33.53$  mt Fs = T x mt x 10 mt x 72x 0.1497x [33.53 mt - 10 mt] 33.53 mt  $Fs = 237.62 \text{ mt}^2$ Step :7: Calculation of transverse module mt We know that  $Fs \ge Fd$ 237.62 mt<sup>2</sup> ≥ <u>79497.9</u> mt mt ≥ 6.94 mm Step:8: Calculate the values of b, d1 and v Face width b = 10 mt = 10 x 7 = 70 mmPitch circle diameter d1 = mt x Z1 = 7 x 20 = 140 mm Pitch line velocity  $V2 = V1 = \Pi dN = 6.61 m/s$ 60 Step:9: Recalculation of the beam strength Fs = 237.62 x mt<sup>2</sup> = 11643.38 N Step: 10: Calculation of the dynamic load, using Buckingham's equation, Fd = Ft +21 V (bc + Ft) 21 V +V( bc +Ft) Ft = = 40 x 103 = 6493.55 N P 6.16 v c = 11860 x 0.017 = 201.62 N/mm  $Fd = 6493.5 + 21 \times 6.16 \times 10^3 \times (70 \times 201.62 + 6493.5)$  $21 \times 6.16 \times 10^3 + \sqrt{(70 \times 201.62 + 6493.5)}$ Fd = 27077.55 N Step :11: Check for the beam strength or tooth breakage, but Fd >> Fs Taking module as 14 b = 10 mt = 10 x 14 = 140 mm Face width Pitch circle diameter d1 = mt x Z1 = 14 x 20 = 280 mm Pitch line velocity  $V2 = V1 = \prod dN = 12.315 m/s$ 

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Fs = 237.62 x mt<sup>2</sup> = 237.62 x 14<sup>2</sup> = 46573.52 N Ft =  $\frac{P}{v}$  =  $\frac{40 \times 10^3}{12.315}$  = 3248 N c = 11860 x .025 = 296.62 N/mm Ed = 3248 + 21 x 12 315 x 103 x (140 x 296 5 + 3248)

$$\frac{140 \times 296.5 + 3248}{21 \times 12.315 \times 10^{3} + \sqrt{(140 \times 296.5 + 3248)}}$$
  
Fd = 47969.4 N

We find Fs > Fd , now the design is safe and satisfactory against the tooth failure.

#### Step: 12:

Calculation of wear load (Fw)

$$Fs = \underbrace{0.75 \text{ x } d1 \text{ x } b \text{ x } Q \text{ x } \text{Kw}}_{\text{Cos } \delta1}$$

$$Q = \text{Ratio Factor} = \underbrace{2 \text{ x } 2\text{v2}}_{2\text{ x } 1 \pm 2\text{v2}} = \underbrace{2 \text{ x } 215}_{21 \pm 215} = 1.822, \text{ and}$$

$$Zv1 \pm Zv2 = 21 \pm 215$$

$$Kw = 0.919 \text{ N/mm}^2 \text{ , for steel gears hardened to 250 BHN ,}$$

$$Fs = \underbrace{0.75 \text{ x } 280 \text{ x } 140 \text{ x } 1,822 \text{ x } 0.919}_{\text{Cos } 17.36^\circ} = 51578.25 \text{ N}$$

Step : 13:

Checking for wear, we found that Fw > Fd, it means the gear tooth has adequate wear capacity and will not wear out. Thus the design is safe against wear failure also. Step: 14

Module mt = 14 mmFace width b = 10 x mt = 140 mmPitch diameter  $d1 = mt \times Z1 = 14 \times 20 = 280 \text{ mm}$  $d2 = mt \times Z2 = 14 \times 64 = 896 \text{ mm}$ 

CO3 11.30



#### Problem on bevel gear:

Design a cast iron bevel gear drive for a pillar drilling machine to transmit 1875 W, at 800 Rpm, to a spindle at 400 rpm. The gear is to work for 40 hours per week for 3 years. Pressure angle is 20°.

#### Given Data:

P= 1875 W, N1= 800 rpm, N2= 400 rpm, α= 20°

#### To find:

Design a bevel gear dive

#### Solution:

Since the materials of pinion and gear are same we have to design only the pinion

#### 1. Gear ratio:

$$Mt = \left(\frac{60 \times 1875}{2 \pi \times 800}\right) = 22.38$$
 N-m and

K.K<sub>d</sub> = 1.3 ( as per assumption)

[Mt] = 22.38 x 1.3 = 29.095 N-m

Calculation of Eeq , [6b], [6c]:

To find Eeq: Eeq =  $1.4 \times 10^5 \text{ N/mm}^2$  for cast iron,  $6u > 280 \text{ N/mm}^2 \text{ PSG 8.14}$ 

To find [6b] = 
$$\frac{1.4 K_{bl}}{n. K_{\sigma}} \times \sigma_{-1}$$
, for rotation in one direction PSG 8.20

$$K_{bl} = 9\sqrt{\frac{10^7}{N}} = 0.8852$$
, for CI

Кб = 1.2;	PSG 8.19
n = 2,	PSG 8.19
б-1 = 0.45 би	PSG 8.19

i = N1 / N2 = 800 / 400 = 2

pitch angle : for right angle bevel gear, tan  $\delta 2 = I = 2$ or  $\delta 2 = tan^{-1} (2) = 63.43^{\circ}$ and  $\delta 1 = 90 - \delta 2 = 26.57^{\circ}$ 

#### 2. Material for pinion and gear:

Cast iron grade 35 heat treated

Би = 350 N/ mm<sup>2</sup>, from PSG 1.40



Then [6b] = ( 1.4 x 0.8852 x 157.2 / (2x 1.2) ) = 81.33 N/ mm<sup>2</sup>

#### To find [6c] :

$$[6c] = Cb \times HB \times Kcl$$
  

$$Cb = 2.3 \qquad PSG 8.16$$
  

$$Hb = 200 \text{ to } 260 \qquad PSG 8.16$$
  

$$Kcl = 6\sqrt{\frac{10^7}{N}} = 6\sqrt{\frac{10^7}{29.952 \times 10^7}}$$
  
= 0.833, for C l

[ 6c] = 2.3 x 260 x 0.833 = 498.08 N/ mm<sup>2</sup>

#### 6. Calculation of cone distance (R):

We know that 
$$R \ge \psi y \left(\sqrt{(i^2+1)}\right) \left\{ \sqrt[3]{\left[\frac{0.72}{\psi y - 0.5[6c]}\right]^2 x Eeq \frac{[Mt]}{i}} \right\}$$
  
$$\psi y = \frac{R}{b} = 3$$

R ≥ 50.2

R= 51 mm

7. Assume Z1= 20, Then Z2 = I x Z1= 2 x 20 = 40

Virtual number of teeth  $Zv_1 = Z_1 / \cos \delta_1 = 20 / (\cos 26.57^\circ) = 23$ 

And  $Zv_2 = Z_2 / \cos \delta_2 = 40 / (\cos 63.43) = 90$ 

#### 8. Calculating the transverse module (mt):

$$Mt = \frac{R}{(0.5\sqrt{Z1^2 + Z2^2})} = 2.28 \text{ mm take as } 2.5 \text{ PSG 8.2}$$



9. Revision of cone distance R:

we know that ,

 $R = (0.5 Mt \sqrt{Z1^2 + Z2^2}) = 0.5 x 2.5 \sqrt{(20^2 + 40^2)} = 55.9 \text{ mm}$ 

10. Calculation of b, Mav, d 1av, v, and ψy:

Face width (b);  $b = R / \psi y = 55.9 / 3 = 18.63 \text{ mm}$ 

Average module (Mav) :  $m_t - (b \sin \delta_1/Z1) = 2.0863 \text{ mm}$ 

Average pcd of pinion ( $d_{1av}$ ) = d1av = Mav x z1 = 2.083 x 20 = 41.66 m

Pitch line velocity v :  $\frac{\pi x d_{1av x N1}}{_{60}}$  = 1.745 m /s  $\Psi y = b / d_{1av} = 18.63 / 41.66 = 0.477$ 

#### 14. Check for wear strength :We know that the induced contact stress,

$$\mathbf{b}c = \left(\frac{0.72}{R - 0.5b}\right) \left( \left(\frac{\sqrt{(i^2 + 1)^3}}{(i \, x \, b)}\right) x \, Eeq \, [Mt] \right]^{\frac{1}{2}}$$

K= 1.1 for  $b / d_{1av} \le 1$ , PSG 8.15 Kd = 1.35 P SG 8.16

[Mt] = 22.38 x 1.1 x 1.35 = 33.24 N-m

13. Check for bending stress

We know that the induced bending stress

$$\mathbf{b} \ b \ = \left\{ \frac{R \sqrt{(i^2 + 1)}[Mt]}{((R - 0.5b)^2 x \ b \ x \ mt \ x \ Y_{v1})} \right\}$$

Where Yv1= 0.408 for Zv1 = 23 .,,

PSG 8.18

Бb = 100.75 N/ mm<sup>2</sup>

Which is not satisfactory Recalculate with various b,  $d_{av}$ , v,  $\psi_{y_{s}}m_{av}$ ,



= 439.33 N / mm<sup>2</sup>

We find that 6c < [6c], thus the design is safety

15. Calculation of basic dimensions of pinion and gear : Transverse module : mt = 3 mm Number of teeth : Z1 = 20, Z2 = 40 Pitch circle diameter : d1= mt x Z1 = 3 x20 = 60 mm and D2 = Mt x Z2 = 3 x 40 = 120 mm Cone distance R = 67.08 mm Face width b = 22.36 mm Pitch angle =  $\delta_1$  = 26.57°, and  $\delta_2$  = 63.43° Height factor : fo = 1 Clearance : c = 0.2 Virtual number of teeth : Zv\_1= 23, and Zv\_2= 90



## **1. DESIGN OF WORM AND WORMWHEEL**

When the shafts are non-parallel and non-intersecting worm and worm wheel drive is used. It can be treated as screw and nut pair. Since the sliding occurs the materials used should have low coefficient of friction.

A worm gear is a species of helical gear, but its helix angle is usually somewhat large (close to 90 degrees) and its body is usually fairly long in the axial direction. These attributes give it screw like qualities. The distinction between a worm and a helical gear is that at least one tooth persists for a full rotation around the helix.

**Worm :** It is made of steel. The threads are grounded and polished to reduce the surface hardness as low as possible.

Worm wheel: They are made of bronze and cast iron.

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#### **DESIGN PROCEDURE OF WORM GEARS**

STEP-1: Selection of Material	PSG. 8.5
STEP-2: Calculation of Initial Design Torque	
[Mt]=Mt x K x Kd.	
Initially, Assume K x Kd = 1.	
Mt=60 x p / 2ΠΝ.	
STEP-3: Selection of Z1&Z2.	
Select Z1 FOR VARIOUS EFFICIENCIES	PSG. 8.46
$Z_2 = I \times Z_1.$	
STEP-4: Selection of [ob] & [oc]	PSG.8.45
STEP-5: Calculation of Centre Distance	PSG.8.44
a = $[(z/q) +1] \times 3\sqrt{540/(Z/q) \times [\sigma c]} \times [Mt] / 10$	
STEP-6: Calculation of Axial Module	PSG.8.43
m= 2a / (q+z)	
STEP-7: Calculation of Revised Centre Distance	PSG.8.43
a=0.5m (q+Z <sub>2</sub> )	
STEP-8: Calculation of d, v, γ, Vs.	
d = q x m	
$v = \pi dn / 60$	
$\gamma = \tan^{-1} \{Z/q\}$	
$Vs = v/\cos \gamma$	
STEP-9: Recalculation of Design Contact Stress Using Vs.	PSG.8.45
STEP-10: Revise K, d, Mt Values.	
STEP-11: CHECK FOR BENDING STRESS	PSG.8.44
[σb] = 1.9[Mt] / m3 x q x z x y	
STEP-12: Check for Wear oc	PSG.8.44

STEP-13: Check for Efficiency η=0.95 x tan γ/tan (γ+ρ) ρ=TAN-1(μ)

STEP-14: Calculation of Cooling Area Required (1-η) x INPUT POWER = Kt x A (to-ta)

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STEP-15: Calculation of Basic Dimensions

#### PROBLEM ON WORM GEARS

A steel worm running at 240 rp , receives 1.5 kw from its shaft. The speed reduction is 10:1, design the drive so as to have an efficiency of 80 %, also determine the cooling area required, if the temperature rise is restricted to 45° C, and take overall heat transfer co efficient as 10 W /  $m^2$  ° C.

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#### Given data:

N1= 240 rpm, P= 1.5 kW, I= 10, η desired = 80%, to - ta = 45°C, Kt = 10 W/ m<sup>2</sup> °C.

#### To find:

1. Design the worm gear drive

2. The cooling area required

#### Solution:

STEP-1: Selection of Material Worm – Steel Wheel – Bronze ( sand cast), selected from Table

PSG. 8.5

STEP-2: Calculation of Initial Design Torque

[Mt]=Mt x K x Kd. Initially, Assume K x Kd = 1. Mt=60 x p / 2ΠΝ. Mt= ( 60 x 1.5 x 10<sup>3</sup> / 2πN2) = 596.83 N-m

#### K.Kd = 1

[Mt]= 596.83 N-m

STEP-3: Selection of Z1&Z2.

Select Z1, η desired = 80%,, Z1 = 3 PSG. 8.46 Z<sub>2</sub> = i x Z<sub>1</sub>.= 10 x 3 = 30 PSG.8.43

12B Status by UGC | Approved by AICTE NAAC I Accredited "A" Grade by STEP-4: Selection of [ob] & [oc] PSG.8.45 For bronze wheel 6u < 390 N / mm<sup>2</sup>, [6b]= 50 N/ mm<sup>2</sup> is selected in one rotation in one direction [6c]= 159 N / mm<sup>2</sup> is selected STEP-5: Calculation of Centre Distance PSG.8.44  $a = [(z/q)|+1] \times 3\sqrt{540/(Z/q)} \times [\sigma c]]2 \times [Mt] / 10$ a = [(30/11)|+1] x 3V[540/(30/11) x [oc]]2 x [596.83x 10<sup>3</sup>] / 10 a = 168.6 mm STEP-6: Calculation of Axial Module PSG.8.43 m = 2a / (q+z)m= 2168.6/ (11+30) = 8.22 mm STEP-7: Calculation of Revised Centre Distance PSG.8.43  $a=0.5m(q+Z_2)$ a=0.5x 10 (11+30) = 205 mm STEP-8: Calculation of d, v, y, Vs.  $d = q \times m$ d 1= q x m= 11 x 10 = 110 mm d 2= Z2 x m= 30 x 10 = 300 mm  $v_1 = \pi dn_1 / 60 = 1.382 m / s$  $v_2 = \pi dn_2/60 = 0.377 \text{ m}/\text{s}$  $y = \tan^{-1} \{Z/q\} = 15.25^{\circ}$ Vs = v/cos y = 1.432 m/sSTEP-9: Recalculation of Design Contact Stress Using Vs. For vs = 1.432 m /s , [6c] = 172 N/mm<sup>2</sup> PSG.8.45 STEP-10: Revise K, d, Mt Values. [Mt]=Mt x K x Kd. = 596.83 x 1 x1 = 596.83 N-m



STEP-11: CHECK FOR BENDING STRESS

PSG.8.44

[ob] = 1.9[Mt] / m3 x q x z x y

 $[\sigma b] = 1.9 \times 596.863 \times 10^3 / 10^3 \times 11 \times 30 \times 0.432$ 

= 7.6 N / mm<sup>2</sup>

STEP-12: Check for Wear oc

 $Bc = 540 / (Z2/q) V((Z2/q) + 1) / a)^3 x (Mt / 10)$ 

= 118.59 N / mm<sup>2</sup>

**STEP-13:** Check for Efficiency η=0.95 x tan γ/tan (γ+ρ) ρ=TAN-1(μ) = 2.862 ° η=0.95 x tan γ/tan (γ+ρ) = 80%

STEP-14: Calculation of Cooling Area Required (1-η) x INPUT POWER = Kt x A (to-ta)

 $(1 - 0.8) \times 1.5 \times 10^3 = 10 \times A \times 45^\circ$ = 0.666 m<sup>2</sup>

STEP-15: Calculation of Basic DimensionsPSG.8.43Axial module = Mx = 10 mmNumber of starts = Z1 = 3Number of teeth = Z2 = 30Length of worm 152 mm (  $L \ge (12.5 + 0.09 \times Z2 ) mx = 152 mm$ Center distance = a = 205 mmHeight factor = 1

## **ASSIGNMENT PROBLEMS**

- 1. Design a pair of straight bevel gears to transmit 10 kW at 1000 rpm between two perpendicular shafts. A speed ratio desire is 3.5; Select suitablematerials.
- 2. Design a suitable worm and wonn wheel drive to transmit 10 kW at 1000 rpm of worm. Select suitable materials. The speed of wheel is to be around 40rpm.
- 3. Design a bevel gear drive to transmit 7.4 kW at 1440 rpm for the following data: Gear ratio=3 ,Material for pinion and gear :C45 steel ,Life=10,000hrs.7



# Unit 4 GEAR BOXES – SMEA1601



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gearbox -Constant mesh gear box. - Design of multi speed gear box.

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Gear boxes are employed to convert input from a high speed power sources to low speed (Eg. Lift, Cranes and Crushing Machine) or into a many of speeds (Lathe, Milling Machine and Automobiles). A gearbox that converts a high speed input into a single output it is called a single stage gearbox. It usually has two gears and shafts.

A gearbox that converts a high speed input into a number of different speed output it is called a multi-speed gear box. Multi speed gear box has more than two gears and shafts.

#### **Requirements of Gear Box:**

- Provides the designed series of spindle speeds
- Transmits the required amount of power to the spindle
- Provides smooth silent operation of the transmission
- Simple in construction
- Mechanism of speed gear boxes should be easily accessible so that it is easier to carry out preventive maintenance

## Law of speed in Multispeed Gear box

- ARITHMATIC PROGRESSION
- GEOMETRIC PROGRESSION
- HARMONIC PROGRESSION

ARITHMATIC PROGRESSION: The difference between and two successive spindle speed is constant.

GEOMETRIC PROGRESSION: The ratio of any successive spindle speed is constant.

HARMONIC PROGRESSION: The difference between reciprocal of any two successive speed is constant.

## **Advantages of Geometric Progression**

- In order to get a series of output speeds from a gearbox.
- The speed is reduced uniformly in different stages.
- The speed loss is minimum i.e., Speed loss = Desired optimum speed Available speed
- The number of gears to be employed is minimum
- G.P. provides a more even range of spindle speeds at each step
- The layout is comparatively very compact.
- Productivity of a machining operation, i.e., surface area of the metal removed in unit time, is constant in the whole speed range.
- G.P. machine tool spindle speeds can be selected easily from preferred numbers. Because **PREFERRED NUMBERS** are in geometric progression.
- Geometric progression, also known as a geometric sequence



## **Standard progression**

When the spindle speed are arranged in geometric progression then the ratio between thetwo adjacentspeeds as step ratio or progression ratio

 $\frac{\frac{\lambda^2}{\lambda 1}}{\frac{\lambda^2}{\lambda 2}} = \frac{\frac{\lambda^4}{\lambda 3}}{\frac{\lambda^3}{\lambda 3}} = \dots = \frac{\frac{\lambda n}{\lambda (\tau - 1)}}{\frac{\lambda}{\tau}} = \text{constant} = \phi$ 

## **Step ratio**

• The ratio between the two adjacent speeds is known as step ratio or progression ratio. It is denoted by  $\phi$ .

Basic	Step ratio
series	(φ)
R5	1.58
R10	1.26
R20	1.12
R40	1.06
R80	1.03

## Preferred basic series (or) Preferred Numbers:

- Preferred numbers are the conventionally rounded off values derived from geometric series.
- There are five basic series, denoted as R 5, R 10, R 20, R 40 and R 80 series.
- Each series has its own step ratio i.e., series factor.
- The series of preferred numbers 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

Basic series	Preferred number
R5 (φ=1.6)	1.00, 1.60, 2.50, 4.00, 6.30, 10.00
R10 (φ=1.26)	1.00, 1.25, 1.60, 2.00, 2.50, 3.15, 4.00,
	5.00, 6.30, 8.00, 10.00
R20 (φ=1.12)	1.00, 1.06, 1.25, 1.18, 1.60, 1.25, 2.00,
	2.24, 2.50, 2.80, 3.15, 3.55, 4.00, 4.50,
	5.00, 5.60, 6.30, 7.10, 8.00, 9.00,
	10.00
R40 (q=1.06)	1.00, 1.06, 1.18, 1.25, 1.32, 1.18,
	1.40,1.60, 1.70, 1.25,1.80, 1.90, 2.00,
	2.10, 2.24, 2.36, 2.50, 2.65, 2.80, 3.00,
	3.15, 3.35, 3.55, 4.00, 4.25, 4.50, 4.75,
	5.00, 5.30, 5.60, 6.00, 6.30, 6.70, 7.10,
	7.50, 8.00, 8.50, 9.00, 9.50, 10.00



## **Design of gear box**

Selection of SpindleSpeed:
 Find thestandardstep by using therelations,= Φn-1

## 2. Structuralformula

It can be selected based on the number of speed:

	-
Number of speed	Structural formula
	3(1) 2(3)
6	2(1) 3(2)
	2(1) 2(2) 2(4)
8	4(1) 2(4)
	3(1) 3(3)
9	
	3(1) 2(3)2(6)
12	2(1) 3(2)2(6)
	2(1) 2(2)3(4)
	3(1) 3(3)2(5)
14	4(1) 2(4)2(6)
15	3(1) 3(3) 2(6)
	4(1) 2(4)2(8)
16	2(1) 4(2)2(8)
	2(1) 2(2)4(4)
	3(1) 3(3)2(9)
18	3(1) 2(3)3(6)
	2(1) 3(2)3(6)

## 3. Ray Diagram

The ray diagram is the graphical representation of the drive arrangement in general from. In other words, The ray diagram is the graphical representation of the structural formula. The basic rules to be followed while designing the gear box as

✓ Transmissi n ration (i):

$$\frac{1}{4} \leq i \leq 2$$
(or)  $\frac{1}{4}$ 



 $N_{min} / N_{input} \ge$ 

 $i_{max} = N_{max} / N_{input} \le 2$ 

For stable operation the speed ratio at any stage should not be greaterthan 8.

 $N_{max} / N_{min} \le 8$ 



## 4. KinematicLayout:

The kinematic arrangement shows the arrangement of gears in a gear box.

Formula for kinematic arrangement,

$$n = p_1(X_1) \cdot p_2(X_2)$$



## 5. Calculation of number of teeth.

In each stage first pair,

Assume, driver  $Z_{min} \ge 17$ ,



#### Assume Z = 20 (driver)

#### **SOLVEDPROBLEMS:**

1. Design a 12 speed gear box with a minimum and maximum speed of 100 rpm and 355 rpm. Draw the ray and kinematic diagram and calculate number of teeth in each gear.

#### **Given Data:**

Z = 12, N1 = 100 rpm, N2 = 355 rpm

**To Find:** Design a Gear Box

#### Solution:

.

**Step 1:** Calculation of Step Ratio ( $\phi$ ) :

$$\frac{N_{max}}{N_{min}} = \phi^{z-1}$$

$$\frac{355}{100} = \phi''$$

$$\phi = 1.12.$$

**STEP 2:** Selection of Intermediate speed from DB: 7.2

Based on the Step ratio 1.12 select the values

$$N_1 = 100 \text{ rpm}.$$
 $N_2 = 200 \text{ rpm}$ 
 $N_2 = 112 \text{ rpm}.$ 
 $N_8 = 224 \text{ rpm}$ 
 $N_8 = 125 \text{ rpm}.$ 
 $N_8 = 250 \text{ rpm}$ 
 $N_4 = 140 \text{ rpm}.$ 
 $N_{10} = 280 \text{ rpm}$ 
 $N_5 = 160 \text{ rpm}.$ 
 $N_{11} = 315. \text{ rpm}$ 
 $N_6 = 180 \text{ rpm}.$ 
 $N_{12} = 355 \text{ rpm}.$ 

Step 3: Formation Of Structural Formula

For 12 speed 3 x 2 x 2

	0	
	Speed	Space.
2= 3x2x2	P1 = 3	$X_1 = I$
1 A.	P2 = 8	x2= A=3
	Pg=2	$X_3 = P_1 X P_2 = 0$

No. of Speed,  $Z = P1 (X1) \cdot P2 (X2) \cdot P3(X3)$ 

$$Z = 3 (1)$$
 . 2 (3) . 2 (6)  
I II III STAGE

#### **Step 4:** FORMATION OF RAY DIAGRAM

U - No. of Stages + 1 = 3 + 1



4 vertical lines to be drawn and 12 Horizontal lines to be drawn



Location of C

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Nmax = 200. Nz/p = 100, 112, 125, 140, 160, 180 mm.  $\frac{N_{max}}{N_{2/P}} = \frac{200}{100} = 2.$ 200 = 1.785 112 200 = 1.6 . . . 125 200 = 1.428 200 -= 1.25. 160 200 = 1-11 Avg. = 1.52. The Avay be closer to the 125 tpm. Optimum input. Speed. = 125 rpm. Location of E: Nmmx 42 NE/P Nimax = 180 mm. NI/P = 125, 140, 160 Mpm. Nmax = 180 = 1.44 N±1p 1245 180 = 1.28 140 180 = 1-125 160 Ang. = 1.28 The areage is closer to 14 orpm. optimum to speed is 1401pm.

#### **STEP 5: CONSTRUCTION OF KINEMATIC DIAGRAM**



**STEP 6:**Calculation of No. of Teeth:
(Note: In every Stage Driver Speed should be constant)

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IIIrd Stage:

Driver Speed (C) = 125 rpm

N2 = N4 = 125 rpm

Driven Speed (A, B) : N1 (A) = 100 rpm, N2 (B) = 200 rpm

Cased):  $[1^{s+}$  Pair 'Ac']  $\frac{N_{1}}{N_{2}} = \frac{Z_{2}}{Z_{1}}$ NOTE:  $\frac{N_{1}}{N_{2}} = \frac{Z_{2}}{Z_{1}}$ NOTE:  $\frac{Always}{driver Gear = 20}$   $\frac{10L}{125}$   $Z_{1} = 25.$ Case (ii):  $L^{2rd}$  Pair 'BL'  $\frac{N_{8}}{N_{4}} = \frac{Z_{4}}{Z_{3}}$   $\frac{200}{125} = \frac{Z_{4}}{Z_{3}}$   $\frac{200}{125} = \frac{Z_{4}}{Z_{3}}$   $\frac{200}{Z_{4}} = 1.6Z_{3} - (15)$ 

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We know that,  $Z_1 + Z_2 = Z_3 + Z_4.$  $20 + 25 = Z_3 + 1.6 Z_3$ 

> $2 \cdot bz_3 = 45$  $z_3 = 17.$

 $z_4 = .1.6 \times 17. \Rightarrow 28.$ 

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$$Z_{3} = l \overline{q}.$$

$$Z_{4} = .l \cdot b \times l \overline{q}. \Rightarrow 28.$$

$$2nd \text{ stage, } LED_{*}EC \text{ Pair}]$$

$$cased): \text{ Reducing pair.}$$

$$* \text{ Driver Speed (E)}.$$

$$N_{6} = N_{8} = 140 \text{ rpm.}$$

$$* \text{ Driven Speed (C, D)}$$

$$N_{5} = 126 \text{ rpm.}.$$

$$N_{9} = 180 \text{ rpm.}.$$

$$* \frac{N_{5}}{N_{5}} = \frac{Z_{5}}{Z_{5}}.$$

$$Assume Z_{5} = 20.$$

$$\frac{125}{140} = \frac{20}{Z_{5}}.$$

$$Z_{5} = .$$

$$Case(i) \text{ In creasing pair (cD)}$$

$$\frac{N_{7}}{N_{6}} = \frac{Z_{8}}{Z_{7}}.$$

$$\frac{180}{N_{6}} = \frac{Z_{8}}{Z_{7}}.$$

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(13)

We know that,  $Z_{5} + Z_{6} = Z_{7} + Z_{8}$ .  $43 = 2.28Z_{7}$ .  $Z_{7} = 19$ .  $Z_{8} = 1.27 \times 19$ .  $Z_{8} = 24$ .

1st stage,

. .

\* HE . HF . HG Pair.

\* Driver Speed (+)

N10 = N12 = N14 = 355 mm.

.

\* Driven Speed (E,F,G)

Na (E) = 140 pm. N11 (F) = 160 pm. N13 (G) = 180 pm.

Case (i): Reducing pour (EH).

 $\frac{Nq}{N_{10}} = \frac{Z_{10}}{Zq}$   $\frac{140}{355} = \frac{Z_{10}}{Zq} \Rightarrow \frac{20}{Zq}$ 

Zq =. 51.

case jis : Intermediate pair.

 $\frac{N_{11}}{N_{12}} = \frac{742}{Z_{11}}$   $\frac{160}{355} = \frac{Z_{12}}{Z_{11}}$ 

Z12 = 0.4521 We know that, Zq+Z10 = Z11 + Z12. . 71 = Z1 + 0.45Z1 + Zu = 49. -+ Z12= 22. case (iii) : Increasing pair (HG).  $\frac{N_{18}}{N_{14}} = \frac{Z_{14}}{Z_{18}}$ 180 = ZH 355 ZIS Z14 = 0.507 Z13 --- (3). we know that, Z13+Z4 = Z1+Z12 71 = Z13+0.507Z13 - Zi3 = 47. - Z14 = 24.

Unit V – CLUTHES AND BRAKES – SMEA1601

# **CLUTCHES:**

A clutch is a device used to engage and disengage the driving shaft to the driver shaft according to the requirement. In case of automobiles, the clutch is required during changing of gears. By operating a lever, the clutch engages and disengages the driver and driven shafts. While changing gears, the driven shaft should be disengaged from the driving shaft.

## **Classification of Clutch**

Clutches are classified in two ways, based on

- 1. Method of energy transfer (actuation).
- 2. Method of engagement.

According to the method of energy transfer, the

- clutches are categorised into
- 1. Mechanically actuated clutches.
- 2. Hydraulically actuated clutches.
- 3. Pneumatically actuated clutches.
- 4. Electrically actuated clutches.
- According to the method of engagement, clutches are

categorised into

- 1. Friction clutches.
- 2. Positive contact clutches.
- 3. Over running clutches.
- 4. Magnetic clutches.

In this book, mechanically operated clutches,

especially friction clutches are discussed.

The actuating force or the force required to engage

the clutches is supplied mainly by springs.

## Working principle of friction clutch

The principle of transmission of power is by contacting friction surfaces, to develop frictional force and this frictional force is used to start the driven shaft from rest and gradually increases the speed without slipping. In case of automobiles, the friction clutch is used to engage the ngine shaft to the driven shaft.

## Material for friction surface

The materials used for lining of friction surfaces should have the following properties.

- 1. The material should have high coefficient of friction.
- 2. The material should withstand high temperatures caused during operation.
- 3. The material should have high heat conductivity, high resistance to wear and should not be effected by moisture or oil.

## **Types of friction clutches**

- 1. Disc or plate clutches.
- (a) Single disc clutch
- (b) Multiple disc clutch
- 2. Cone clutches.
- 3. Centrifugal clutches.

## Advantages and Disadvantages of friction clutch.

## Advantages of friction clutches.

- 1. They have a very little shock during engagement as they can slip relative to each other.
- 2. It can be used for high speed engagement applications.

## **Disadvantages of friction clutches**

- 1. They are not suitable for application that require positive transmission because they do slip.
- 2. Replacement of friction material is often required as they wear out. Cams, Clutches and Brakes

## Design considerations for friction clutches

- 1. Sufficient torque transmitting capacity.
- 2. Light weight.
- 3. Smooth engagement without shock and fast disengagement with drag.
- 4. Proper friction material (high co-efficient of friction) that should not be affected by moisture and oil, etc.
- 5. Provision for taking up wear of the contact surfaces.
- 6. Provision for transmission of heat which is generated during operation.

## **PROBLEM 1:**

A single dry plate clutch is to be designed to transmit 10 HP. at 900 rpm. Find (a) diameter of the shaft, (b) mean radius and face width of the friction lining assuming the ratio of the mean radius to the face width as 4, (c) outerand inner radii of the clutch plate. Take  $\sigma y$  for the shaft material as 38 kg/mm2

Power = 10 HP = 7.36 kW

Speed = N = 900 rpm

Refer P.S.G Data book, page No. 7.89

(a) Diameter of shaft 
$$= d = \sqrt[3]{\frac{495000 \times kW \times k_w}{N[\tau]}}$$

where kW = rated power = 7.36 kW N = rpm of the drive N = 900 rpm  $k_w =$  factor based on working conditions  $= k_1 + k_2 + k_3 + k_4$  $[\tau] =$  Allowed shear stress

Since working conditions are not given,

Assume  $k_1, k_2, k_3$ , and  $k_4$  values

From P.S.G Data book, Page No. 7.90 take

```
k_1 = 0.5 ; k_2 = 1.25 ; k_3 = 0.32 and k_4 = 0.75 k_w = k_1 + k_2 + k_3 + k_4 k_w = 0.5 + 1.25 + 0.32 + 0.75 = 2.82
```

Assume, material of shaft as C50;

 $\sigma_v = 38 \text{ kg}_f/\text{mm}^2$ ; FOS = 2

 $[\tau] =$  Allowable shear stress

$$= \frac{\sigma_y}{2} \times \frac{1}{FOS}$$
$$= \frac{38}{2} \times \frac{1}{2} = 9.5 \text{ kg}_{\text{f}}/\text{mm}^2$$
$$[\tau] = 950 \text{ kg}_{\text{f}}/\text{cm}^2$$

Diameter of clutch shaft

$$= d = \sqrt[3]{\frac{495000 \times 7.36 \times 2.82}{900 \times 950}} = 2.29 \text{ cm or } 2.3 \text{ cm}$$

d = 23 mm

(b) Let b = face width

 $r_m =$  mean radius

$$b = \frac{r_m}{4}$$
 (given)

Refer PSG data book Page No 7.89

$$i_m = \frac{[M_t]}{2\pi P_b b \mu r_m^2}$$

W.k.t For single plate clutch  $i_m = 2$ 

where,  $i_m =$  Number of friction surfaces

From PSG data book, **Page No. 7.89**, allowable pressure =  $P_a = k . \, P_b$ 

where k = speed factor = 1(assume)

and  $P_b = {\rm basic \ pressure} = 2.5 \ {\rm to} \ 3 \ {\rm kg_f/cm}^2$  ( for dry running)

Take  $P_b=3~{\rm kg_f/cm^2}$ 

 $\mu = \text{coefficient of friction} = 0.25 (Assume)$ 

From PSG Databook, Page No. 7.8

$$M_t = \frac{71620 \text{ HP}}{N} = \frac{71620 \times 10}{900} = 795.77 \text{ kg}_f - \text{cm}$$

Design torque  $= [M_t] = M_t \times k_{\overline{w}}$ 795.77 × 2.82 = 2244.07

$$= 2244 \text{ kg}_{f} - \text{cm}$$

$$i_{m} = \frac{\left[ M_{t} \right]}{2\pi P_{b} b \mu r_{m}^{2}}$$

$$2 = \frac{2244}{2 \times \pi \times 3 \times \frac{r_{m}}{4} \times 0.25 \times r_{m}^{2}}$$

$$r_{m}^{3} = \frac{2244 \times 4}{2 \pi \times 3 \times 0.25 \times 2} \Rightarrow r_{m} = \sqrt[3]{952.38117}$$

$$r_{m} = 9.8 \text{ cm or } 98 \text{ mm}$$

Mean radius  $= r_m = 98 \text{ mm}$ 

(c) To find  $r_i$  = inner radius and  $r_o$  = outer radius

b = face width of the plate

$$= r_o - r_i = \frac{r_m}{4}$$
  
 $r_o - r_i = \frac{98}{4} = 24.5$   
 $b = 24.5$  mm .... (1)

But 
$$r_m = \frac{r_o + r_i}{2} = 98$$
  
 $\therefore r_o + r_i = 98 \times 2 = 196$  ... (2)  
By solving eq. 1 and eq. 2 we get  $r_o = 110.25$  mm and

 $r_i = 85.75 \text{ mm}$ 

Inner radius of the clutch plate  $r_i = 85.75~\mathrm{mm}$ 

Outer radius of the clutch plate  $\,=r_o=110.25~{\rm mm}$ 

# **PROBLEM 2:**

A multiplate clutch is to be designed to transmit a power of 50 kW at 500 rpm. Assuming suitable materials and data, determine the main dimensions of friction lining used in that clutch.

## Given data

Design power = Rated power × service factor i.e.,  $P_d = P \times \text{Service factor}$ 

 $= 50 \times 10^{3} \times 1.25$ = 62500 W

Now calculate design torque that is to be transmitted.

$$T = \frac{60 \times P_d}{2 \pi N} = \frac{60 \times 62500}{2\pi \times 500}$$
$$= 1193.66 \text{ N} - \text{m}$$
$$= 1193.66 \times 10^3 \text{ N} - \text{mm}$$

Let,

 $\tau =$  Permissible shear stress for shaft material.

 $\tau = 60 \text{ N/mm}^2 \text{ (Assume)}$ 

and d = diameter of clutch shaft.

Then

$$d > \left[\frac{16T}{\pi\tau}\right]^{1/3}$$
$$d > \left[\frac{16 \times 1933660}{\pi \times 60}\right]^{1/3}$$
$$d > \left[\frac{30938560}{188.49}\right]^{1/3}$$
$$d > [164134.15]^{1/3}$$
$$d > 54.75$$
$$\therefore d = 55 \text{ mm}$$

÷

Let

$$\begin{split} r_i &= \text{Inner radius }; \ r_o &= \text{ Outer radius } \\ r_i &= 2d = 2 \times 55 \\ &= 110 \text{ mm} \\ r_o &= 1.5 \ r_i = 1.5 \times 110 \\ &= 165 \text{ mm} \end{split}$$

 $n_1 =$  Number of driving plates = 5 (Assume)

 $n_2 =$  Number of driven plates = 6 (Assume)

W.k.t.

 $\therefore n = n_1 + n_2 - 1 = 5 + 6 - 1 = 10$ 

Let the friction material be asbestos based material.

Assume,  $\mu = 0.35$ 

Allowable pressure = 0.21 to 0.56 N/mm<sup>2</sup>

Now

$$T = n \ \mu W r_m$$
  

$$\therefore r_m = \frac{r_i + r_o}{2} = \frac{110 + 165}{2} = 137.5 \ \text{mm}$$
  

$$\therefore W = \frac{T}{n \ \mu r_m} = \frac{1193660}{10 \times 0.35 \times 137.5}$$
  

$$W = 2480.33 \ \text{N}$$

Now

$$p_{\max} = \frac{W}{2 \pi r_i (r_o - r_i)}$$
$$= \frac{2480.33}{2 \times \pi \times 110 (165 - 110)}$$
$$= 0.06 \text{ N/mm}^2 < \text{ allowable pr. } (0.21 \text{ N/mm}^2)$$

Since the pressure is less than allowable pressure, our design is safe.

Now find face width

$$b = r_o - r_i = 165 - 110$$
  
 $b = 55 \text{ mm}$ 

Thickness

t = 5 mm (Assumed)

Hence, the main dimensions of friction lining are

- 1. Inner radius = 110 mm
- 2. Outerradius = 165 mm
- 3. Face width = 55 mm
- 4. Thickness = 5 mm

# CONE CLUTCH

# **PROBLEM:**

An engine is developing 50 kW at 1200 rpm by using a cone clutch. The cone angle is 30°. The mean radius is 300 mm; The coefficient of friction is 0.25; The normal pressure should not exceed 0.2 N/mm2; Determine (1) face width of the cone (2) axial force required to engage the clutch.

P = 50 kW; N = 1200 rpm;  $2\alpha = 30^{\circ}$ ;  $\alpha = 15^{\circ}$ 

 $p_n=0.2~\mathrm{N/mm}^2$  ;  $\mu=0.25$  ;  $r=300~\mathrm{mm}$  Let;  $b=\mathrm{face}$  width

Assume uniform wear, Refer PSG data book Page.No. 7.89

$$M_t = \text{Transmitted torque} = \frac{9400 \text{ kW}}{N}$$
$$= \frac{97400 \times 50}{1200} = 4058.33 \text{ kg}_f - \text{cm}$$

(or) 
$$M_t = 405833$$
 N - mm

Frictional torque =  $F_t = 2\pi \mu p_n \cdot b \cdot r^2$ 

$$405833 = 2\pi \times 0.25 \times 0.2 \times b \times 300^2$$

Face width = b = 14.35 mm

Let  $(W_a)_{\tt engage} =$  Axial force required to engage the clutch

 $W_n =$ Normal force on the contact surface

$$\begin{split} &= p_n \cdot 2\pi r \cdot b \\ &= 0.2 \times 2 \times \pi \times 300 \times 14.35 \\ &= 5409.82 \text{ N} \\ &\therefore \text{ Axial force required} \\ &\text{ to engage the clutch} \\ \end{split} \\ &= (W_a)_{\text{engage}} = W_n (\sin \alpha + \mu \cos \alpha) \end{split}$$

= 5409.82 [ sin  $15^{\circ} + 0.25 \cos 15^{\circ}$ ] ]

$$(F_a)_{engage} = 2707.17 \text{ N}$$

## BRAKES

## Introduction:

Brake is a device which is used to stop the motion of a machine or retard its motion. It is due to the friction between contact surfaces. The factors required in the design are:

- 1. Coefficient of friction between the braking surfaces.
- 2. Unit pressure between braking surfaces.
- 3. The projected area of the braking surfaces.
- 4. Heat dissipating capacity.

# **PROBLEM:**

A single block brake shown in the diagram has the diameter 300 mm. The angle of contact is 120°. The coefficient of friction is 0.3; If the torque transmitted by the brake is 80 N – m, find the force P required. Assume length of the lever (l) = 300 mm and distance between centre of wheel to fulcrum (x) = 150 mm.



Taking moments about O

$$\begin{split} R_N \, x &= P \times l \\ R_N = \frac{Pl}{x} = \frac{P \times 300}{150} = 2P \\ R_N &= 2 \ (P) \end{split}$$

Since angle of contact is greater than (or) equal to  $60^{\,\circ}$ 

$$\therefore \mu' = \frac{4 \mu \sin \theta}{2 \theta + \sin 2 \theta}$$
$$= \frac{4 \times 0.3 \times \sin 60^{\circ}}{120^{\circ} + \sin 120^{\circ}}$$
$$= \frac{4 \times 0.3 \times \sin 60^{\circ}}{\frac{\pi}{1.5} + \sin 120^{\circ}}$$
$$= \frac{1.03923}{2.96042}$$
$$\mu' = 0.351$$

Braking torque 
$$T_B = \mu' R_N \cdot r$$
  
 $80 \times 10^3 = 0.351 \times [2 \ (P)] \times 150$   
 $P = \frac{80 \times 10^3}{105.3} = 759.7 \text{ N}$ 

## **PROBLEM**:

Design a differential band brake for a crane lifting a load of 50 kN through a rope wound round a barrel of 550 mm diameter, as shown in Fig. The brake drum keyedto the same shaft be 650 mm in diameter and the angle of lap of the brake band over the drum is 240°. Operating arms of the brake are 45 mm and 210 mm. Operating lever is 1.5 m long. Take  $\mu = 0.25$ .



Barrel radius = 225 mm Brake drum diameter = 650 mm Brake drum radius = 325 mm Angle of lap  $(\theta) = 240^{\circ}$ 

$$= 240^\circ \times \frac{\pi}{180}$$
 rad

= 4.188 rad

Length of lever = 1.5 m

 $\mu = 0.25$ 

Designing differential band brake by following steps.

#### Step 1

Find braking torque

$$T_B = \text{Load} \times \text{Barrel radius}$$
  
 $T_B = 50 \times 10^3 \text{ N} \times 225 \text{ mm}$   
 $T_B = 11250 \times 10^3 \text{ N-mm}$   
 $T_B = 11250 \text{ N.m}$ 

#### Step 2

Find Tension  $T_1 \& T_2$ Tension ratio  $\frac{T_1}{T_2} = e^{\mu \theta}$   $\begin{cases} T_1 = \text{ Tight side tension} \\ T_2 = \text{Slack side tension} \end{cases}$   $\therefore \frac{T_1}{T_2} = e^{0.25 \times 4.188} = 2.849$ or  $T_1 = 2.849 T_2$ ...(1)

Now,

 $T_{B} = (T_{1} - T_{2}) \times r$ 

11250 Nm =  $(T_1 - T_2) \times 0.325$  m

$$T_1 - T_2 = 34,615.4$$
 N

Comparing and solving both equation 1 and 2, we get

...(2)

 $T_1 = 53336.5$  N and  $T_2 = 18721.14$  N

Step 3

Thickness of band brake

$$t = 0.005 \times D$$
  
= 0.005 × 650  
 $t = 3.25$  mm

Now find band width (w)

Assume Induced tensile stress  $\sigma_t = 60 \; \mathrm{N/m} \, \mathrm{m}^2$ 

$$\sigma_t = \frac{T_1}{w \times t}$$
$$\therefore \frac{53,336.5}{w \times 3.25} \le 60 \text{ N/mm}^2$$

Band Width w = 273.5 mm

$$w = 273.5 \text{ mm}$$

#### Step 4

Calculate the bearing pressure

$$P_{\max} = \frac{T_1}{w.r}$$
$$= \frac{53,336.5}{273.5 \times 325}$$

$$P_{\rm max} = 0.599 \, {\rm N/mm}^2$$

[P] for Metal drum 
$$= 1.5 \text{ N/mm}^2$$

Since  $P_{\max} < [P]$ ,

 $\therefore$  The design is satisfactory.

### Step 5

Force applied at the end of lever. From the figure. Taking moments about O, we get  $F \times 1500 + T_1 \times 45 = T_2 \times 210$   $F \times 1500 + 53336.5 \times 45 = 18721.14 \times 210$ F = 1020.85 N