FLAT BELT DRIVE

1. Select a flat belt to drive a mill at 250 rpm from a 10 kW, 730 rpm motor, centre distance is to be around 2 m. The mill shaft pulley is of 1 m diameter.

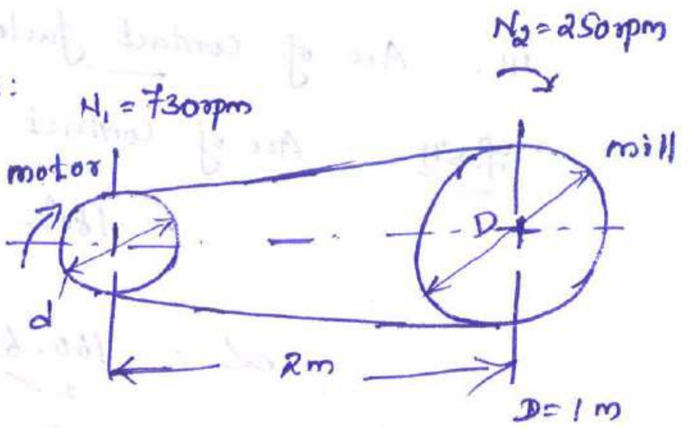
Given: $P = 10\ kW$ $\quad N_2 = 250\ rpm$ $\quad D = 1\ m = 1000\ mm$

$N_1 = 730\ rpm$ $\quad C = 2\ m = 2000\ mm$

Find: Select a flat belt drive

Soln:

1. Calculation of pulley diameters:



Driven pulley dia $D = 1\ m$

$D = 1000\ mm$

$$\text{Velocity ratio} = \frac{N_1}{N_2} = \frac{D}{d}$$

$$\frac{730}{250} = \frac{1000}{d}$$

$d = 342.5\ mm$ for std $\boxed{d = 355\ mm}$ 7.54.

Driver pulley diameter $d = 355\ mm$

2. Calculation of design power in KW.

$$\text{Design KW} = \frac{\text{Rated kW} \times \text{load correction factor } (K_s)}{\text{Arc of contact factor } (K_\alpha) \times \text{Small pulley factor } (K_d)}$$

i. Rated kW = 10 kW

ii. Load correction factor (K_s)

for mill T.53 Intermittent loads

$\boxed{K_s = 1.3}$

iii. Arc of contact factor (K_α)

T.54 Arc of contact $\alpha = 180° - \left(\frac{D-d}{c}\right) \times 60°$

$$= 180° - \left(\frac{1000 - 355}{2000}\right) \times 60°$$

$$\alpha = 160.65°$$

for $\alpha = 160.65°$ $\boxed{K_\alpha = 1.08}$

T.54

iv. Small pulley factor (K_d)

for $d = 355$ mm $\boxed{K_d = 0.8}$

d	K_d
upto 100	0.5
100 - 200	0.6
200 - 300	0.7
300 - 400	0.8
400 - 750	0.9
over 750	1

$$\text{Design KW} = \frac{10 \times 1.3}{1.08 \times 0.8}$$

$\boxed{\text{Design kW} = 15.05 \text{ kW}}$

3. Selection of belting:

for mill → Heavy duty

7.54 FORT 949 g duck belting is selected

Its capacity 0.0289 kw/mm/ply

4. Load rating correction:

7.54 $$\text{Load rating at } V \text{ m/s} = \text{Load rating at } 10 \text{ m/s} \times \frac{V}{10}$$

$$\text{Velocity } V = \frac{\pi d N_1}{60} = \frac{\pi \times 0.355 \times 730}{60}$$

$$V = 13.57 \text{ m/s}$$

$$\therefore \text{Load rating at } 13.57 \text{ m/s} = 0.0289 \times \frac{13.57}{10}$$

$$\boxed{= 0.0392 \text{ kw/mm/ply}}$$

5. Determination of belt width

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

7.52 for $d = 355$ mm & $V = 13.57$ m/s

$$\boxed{\text{No. of plies} = 6}$$

$$\text{width of belt} = \frac{15.05}{0.0392 \times 6} = 63.99 \text{ mm}$$

7.52 for 6 ply belt $\boxed{\text{std belt width} = 100 \text{mm}}$

6. Determination of pulley width :

∴ for belt width upto 125mm

7.54
std Belt width = 100mm

Pulley width = Belt width + 13mm

= 100 + 13

= 113 mm

7.55 $\boxed{\text{std pulley width} = 125\text{ mm}}$

7. Calculation of length of the belt (L)

7.53 for open drive

$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$= (2\times 2000) + \frac{\pi}{2}(1000+355) + \frac{(1000-355)^2}{4\times 2000}$$

$$\boxed{L = 6180.43\text{ mm}}$$

V-BELT DRIVE

2. Design a V-Belt drive to the following specifications:

Power to be transmitted = 75 kW

Speed of driving wheel = 1400 rpm

Speed of driven wheel = 400 rpm

Diameter of driving wheel = 300 mm

Centre distance = 2500 mm

Service = 16 hours/day

Dia of Driving = d

Dia of driven = D

Given: $P = 75\,kW$ $d = 300\,mm$

$N_1 = 1400\,rpm$ $C = 2500\,mm$

$N_2 = 400\,rpm$ Service = 16 hrs/day

find: Design a V-belt drive using manufacturer's data

Soln:

1. Selection of the belt section:

Select 'C' or 'D' section

7.58 for power $P = 75\,kW$

'D' section is selected

2. Selection of pulley diameters (d & D)

Dia of driving wheel $d = 300\,mm$

$$\text{Speed ratio} = \frac{D}{d} = \frac{N_1}{N_2} = 3.5$$

$$\frac{D}{300} = \frac{1400}{400} \qquad \frac{D}{300} = 3.5$$

Dia of driven wheel = $\boxed{D = 1050\,mm}$

3. Selection of centre distance: 'C'

$\boxed{C = 2500 \text{ mm}}$

4. Determination of nominal pitch length 'L'

7.61 $L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$

$= (2 \times 2500) + \left(\frac{\pi}{2} \times (1050+300)\right) + \frac{(1050-300)^2}{4 \times 2500}$

$= 5000 + 2120.58 + 56.25$

$L = 7176.83$ mm

7.60. Std pitch length $\boxed{L = 7648 \text{ mm}}$
for D section

5. Selection of various modification factors:

i. Length correction factor (F_c)

7.60 for D section $\boxed{F_c = 1.05}$

ii. correction factor for arc of contact (F_d)

7.68. Arc of contact $= 180^\circ - \left(\frac{D-d}{C}\right) \times 60^\circ$

$= 180^\circ - \left(\frac{1050-300}{2500}\right) \times 60^\circ$

$= 162^\circ$

for Arc of contact $= 162^\circ$ 7.68 $\boxed{F_d = 0.96}$ V-V

iii. Service factor (F_a)

7.69. for medium duty & 16 hrs continuous duty

$$\boxed{F_a = 1.4}$$

6. Calculation of maximum power capacity:

7.62. for D section

$$kw = \left[3.22\, S^{-0.09} - \frac{506.7}{de} - 4.78 \times 10^{-4} S^2\right] S$$

$$S = \text{Belt speed} = \frac{\pi d N_1}{60} = \frac{\pi \times 0.31 \times 1400}{60}$$

$$S = 21.99 \text{ m/s}$$

de = Equivalent pitch diameter

7.62 $de = dp \times F_b$

pitch diameter of smaller pulley

$$dp = 300 \text{ mm}$$

for speed ratio = 3.5 small dia factor $F_b = 1.14$

7.62

$$de = 300 \times 1.14 = 342 \text{mm}$$

7.62 Maximum value of de = 425 for D' section

∴ calculated de is minimum = 342 mm

$$kw = \left[3.22 \times (21.99)^{-0.09} - \frac{506.7}{342} - 4.78 \times 10^{-4} \times 21.99^2\right] 21.99$$

$$= \left[2.44 - 1.482 - 0.231\right] 21.99$$

$$\boxed{= 15.98 \text{ kw}}$$

7. Determination of number of belts (n_b)

7.70 No. of belts $n_b = \frac{P \times F_a}{kw \times F_c \times F_d}$

$$= \frac{75 \times 1.4}{15.98 \times 1.05 \times 0.96} = 6.52 = 7 \text{ belts}$$

8. Calculation of actual centre distance:

7.61 $C_{actual} = A + \sqrt{A^2 - B}$

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

$$= \frac{7648}{4} - \pi\left[\frac{1050 + 300}{8}\right]$$

$$A = 1381.86$$

$$B = \frac{(D-d)^2}{8} = \frac{(1050 - 300)^2}{8} = 70312.5$$

$$C_{actual} = 1381.86 + \sqrt{(1381.86)^2 - 70312.5}$$

$$= 2738.04 \text{ mm}$$

3. Design a V-belt drive and calculate the actual belt tensions and average stress for the following data. Power to be transmitted = 7.5 KW, Speed of driving wheel = 1000 rpm, Speed of driven wheel = 300 rpm, Diameter of driven pulley = 500 mm, Diameter of driver pulley = 150 mm and center distance = 925 mm.

Given:

$P = 7.5\ KW$ $\quad N_2 = 300\ rpm$ $\quad d = 150 mm$

$N_1 = 1000\ rpm$ $\quad D = 500 mm$ $\quad C = 925 mm$

Find:
i. Design a V-belt drive using manufacturer's data
ii. Actual belt tensions & average stress.

Soln:

I. Design a V-belt drive

1. Selection of the belt section:

For power $P = 7.5$ KW

7.58 'B' section is selected

2. Selection of pulley diameters D & d

$D = 500 mm$ & $d = 150 mm$

3. Selection of center distance $\boxed{C = 925 mm}$

4. Nominal pitch length (L)

7.61
$$L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$$

$$= (2\times925) + \frac{\pi}{2}(500+150) + \frac{(500-150)^2}{4\times925}$$

$$L = 2904.13 mm$$

7.60. for B section

std pitch length $\boxed{L = 3091\ mm}$

5. Selection of various modification factors:

i. Length correction factor (F_c)

7.60. for B section $\boxed{F_c = 1.07}$

ii. Correction factor for arc of contact (F_d)

7.68. $$\text{Arc of contact} = 180^\circ - \left[\frac{(D-d)}{c}\right] \times 60^\circ$$

$$= 180^\circ - \left(\frac{500-150}{925}\right) \times 60^\circ$$

$$= 157.3^\circ$$

7.68 for Arc of contact $= 157.3^\circ$ or 160°

V-V

$$\boxed{F_d = 0.95}$$

iii. Service factor (F_a)

7.69 Assume light duty & 16 hrs continuous service

$$\boxed{F_a = 1.3}$$

6. Maximum power capacity

for B section

7.62 $$Kw = \left[0.79 \cdot S^{-0.09} - \frac{50.8}{d_e} - 1.32 \times 10^{-4} S^2\right] S$$

s = Belt speed = $\frac{\pi d N_1}{60} = \frac{\pi \times 0.15 \times 1000}{60}$

$$\boxed{s = 7.854 \text{ m/s}}$$

d_e = Equivalent pitch dia

7.62 $d_e = d_p \times F_b$

$d_p = d = 150$ mm

speed ratio

$i = \frac{N_1}{N_2} = \frac{D}{d}$

$i = \frac{1000}{300} = 3.33$

Small dia factor $\boxed{F_b = 1.14}$

for $i = 3.33$

$d_e = 150 \times 1.14 = 171$ mm

from 7.62 Maximum $d_e = 175$ mm

$\therefore$ calculated d_e is minimum $d_e = 171$ mm

$$kW = \left[0.79 \times (7.854)^{-0.09} - \frac{50.8}{171} - 1.32 \times 10^{-4} \times (7.854)^2\right] 7.854$$

$$\boxed{= 2.76 \text{ kW}}$$

7. Number of belts (n_b)

7.70 No. of belts $n_b = \frac{P \times F_a}{kW \times F_c \times F_d} = \frac{7.5 \times 1.3}{2.76 \times 1.07 \times 0.95}$

$n_b = 3.48 = 4$ belts

8. Actual center distance:

7.61. $C_{actual} = A + \sqrt{A^2 - B}$

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

$$= \frac{3091}{4} - \pi\left[\frac{500+150}{8}\right]$$

$A = 517.5$

$$B = \frac{(D-d)^2}{8} = \frac{(500-150)^2}{8} = 15312.5$$

$C_{actual} = 517.5 + \sqrt{(517.5)^2 - 15312.5}$

$\boxed{C_{actual} = 1020 \text{ mm}}$

II. Belt tensions & Average stress

a. Belt tensions T_1 & T_2

Power $P = (T_1 - T_2)V$ $\quad v = s = 7.854 \text{ m/s}$

$$\frac{7.5 \times 10^3}{4} = (T_1 - T_2)\, 7.854$$

$\boxed{T_1 - T_2 = 238.73}$ — (1)

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

for B section

7.50 Mass per metre length $\boxed{m = 0.189 \text{ kg/m}}$

7.70 for B section & $d_p = d = 150$ mm

Take

$2\beta = 34^\circ \quad \therefore \boxed{\beta = 17^\circ}$ $\boxed{\mu = 0.3}$

$\alpha = 157.3^\circ \times \frac{\pi}{180} = \boxed{2.745 \text{ rad}}$

$\boxed{V = S = 7.854 \text{ m/s}}$

$$\frac{T_1 - 0.189(7.854)^2}{T_2 - 0.189(7.854)^2} = e^{(0.3 \times 2.745/\sin 17)}$$

$$\frac{T_1 - 11.66}{T_2 - 11.66} = 16.72$$

$$T_1 - 11.66 = 16.72(T_2 - 11.66)$$

$$T_1 - 11.66 = 16.72 T_2 - 194.96$$

$$T_1 - 16.72 T_2 = -194.96 + 11.66$$

$$\boxed{T_1 - 16.72 T_2 = -183.3} \quad \text{--- (2)}$$

solving eqns (1) & (2)

$$T_1 - T_2 = 238.73 \quad \text{--- (1)}$$

$$T_1 - 16.72 T_2 = -183.3 \quad \text{--- (2)}$$

x by 16.72 on eqn (1)

$$16.72 T_1 - 16.72 T_2 = 3991.6$$

$$\underset{(-)}{T_1} \underset{(+)}{-} 16.72 T_2 = \underset{(+)}{-}183.3$$

$$15.72 T_1 = 4174.9$$

$$\boxed{T_1 = 265.6 \text{ N}}$$

$(1) \Rightarrow \quad 265.6 - T_2 = 238.73$

$$\boxed{T_2 = 26.9\ N}$$

Tensions:

$$\boxed{T_1 = 265.6\ N \ \ \& \ \ T_2 = 26.9\ N}$$

Area
A – $80\ mm^2$
B – 140
C – 230
D – 475
E – $695\ mm^2$

b. Stress induced:

$$Stress = \frac{\text{Maximum tension}}{\text{c/s area}} = \frac{265.6}{140}$$

$$\boxed{\sigma = 1.897\ N/mm^2}$$

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4. A V-belt drive is to transmit 45kW in a heavy duty saw mill which works in two shifts of 8 hours each. The speed of motor shaft is 1400 rpm with the approximate speed reduction of 3 in the machine shaft. Design the drive and calculate the average stress induced in the belt.

Given:

$P = 45\ kW$

service = Two shifts of 8 hours each
$\therefore$ 16 hours per day

$N_1 = 1400\ rpm$

speed ratio $i = 3$ $\qquad \frac{N_1}{N_2} = i\ ; \quad \frac{1400}{N_2} = 3$

$$\boxed{N_2 = 466.7\ rpm}$$

Find:

I. Design a V-belt drive using manufacturer's data

II. Average stress induced.

Soln:

I. Design of V-belt drive using manufacturer's data

1. Selection of belt section:

for power $P = 45\ kW$

7.58 'C' section is selected

2 Selection of pulley diameters D & d

for 'C' section

7.58 Minimum pulley dia $\boxed{d = 200\ mm}$

Speed ratio $i = \frac{D}{d} = \frac{N_1}{N_2}$

$$3 = \frac{D}{200}$$

Larger pulley dia $\boxed{D = 600\ mm}$

3. Selection of center distance 'C'

$i = \frac{D}{d} = 3$

for $i = 3$

7.61 $\frac{C}{D} = 1$; $\frac{C}{600} = 1$

Center distance $\boxed{C = 600\ mm}$

Remaining Procedures 4 to 8 & Average stress calculations are same as Problem No. 3

5. Two shafts whose centers are 1m apart are connected by a v- belt drive. The driving pulley is supplied with 100 kW and has an effective diameter of 300 mm. It runs at 1000 rpm, while the driven pulley runs at 375 rpm. The angle of groove on the pulleys is 40°. The permissible tension in 400 mm² cross sectional area of belt is 2.1 Mpa. The density of the belt is 1100 kg/m³. Taking $\mu = 0.28$, estimate the number of belts required. Also calculate the length of the each belt.

Given:

$c = 1m$ $\quad a = 400\ mm^2$ $\quad 2\beta = 40^\circ$

$N_1 = 1000\ rpm$ $\quad P = 100 kW$ $\quad \sigma = 2.1\ Mpa = 2.1\ N/mm^2$

$N_2 = 375\ rpm$ $\quad d = 300\ mm$ $\quad \rho = 1100\ kg/m^3$

$\mu = 0.28$

Find: No. of belts required & length of each belt

Soln: Use Basic Equations.

1. Number of belts required

$$\text{Number of belts} = \frac{\text{Total power transmitted}}{\text{Power transmitted per belt}}$$

$\boxed{\text{Total power transmitted} = 100\ kW = 100 \times 10^3\ W}$

Power transmitted per belt $P = (T_1 - T_2)\, v$

Velocity $v = \dfrac{\pi d N_1}{60} = \dfrac{\pi \times 0.3 \times 1000}{60} = 15.71\ m/s$

If velocity $v > 10\ m/s$ consider T_c

$\boxed{T_{max} = T_1 + T_c}$

$\boxed{\sigma = \dfrac{T_{max}}{A}}$

$\therefore\ T_{max} = \sigma \times A$

$= 2.1 \times 400$

$\boxed{T_{max} = 840\ N}$

$\boxed{\text{mass } m = \rho \times A \times l}$

Centrifugal tension $T_c = m v^2$

$T_c = \rho \times A \times l \times v^2$ $\quad l = 1\ m$ per metre length

$= 1100 \times 400 \times 10^{-6} \times 1 \times (15.71)^2$ $\quad$ Area in m^2

$\boxed{T_c = 108.59\ N}$

$840 = T_1 + 108.59$

$\boxed{T_1 = 731.41\ N}$

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

7.58 Arc of contact angle $\theta = 180° - 60° \left(\frac{D-d}{C}\right)$

Larger pulley dia $D = 800$ mm $\quad d = 300$ mm

$$\boxed{i = \frac{D}{d} = \frac{N_1}{N_2}}$$

$$\frac{D}{300} = \frac{1000}{375}$$

$$\boxed{D = 800 \text{ mm}}$$

$$\theta = 180° - 60°\left(\frac{800-300}{1000}\right)$$

$$\theta = 150°$$

$$\theta = 150 \times \frac{\pi}{180} = 2.62 \text{ rad}$$

$$\boxed{\theta = 2.62 \text{ rad}}$$

$$\boxed{\mu = 0.28}$$

$$2\beta = 40°$$

$$\boxed{\beta = 20°}$$

$$\boxed{T_1 = 731.41 \text{ N}}$$

$$\frac{731.41}{T_2} = e^{(0.28 \times 2.62/\sin 20°)}$$

$$\boxed{T_2 = 85.6 \text{ N}}$$

$$\text{No. of belt} = \frac{100 \times 10^3}{10145.16}$$

Power transmitted per belt $P = (T_1 - T_2)\, v$

$$\boxed{\text{No. of belts required} = 10 \text{ belts}}$$

$$= (731.41 - 85.6) \times 15.71$$

$$\boxed{P = 10145.16 \text{ W}}$$

2. Length of each belts 'L'

7.61 $L = 2C + \frac{\pi}{2}(D+d) + \frac{(D-d)^2}{4C}$

$C = 1000$ mm

$D = 800$ mm

$d = 300$ mm

$$\boxed{L = 3790.4 \text{ mm}}$$

CHAIN DRIVE

6. A truck equipped with a 9.5kW engine uses a roller chain as the final drive to the rear axle. The driving sprocket runs at 900 rpm. and the driven sprocket at 400 rpm with a centre distance of approximately 600mm. Select the roller chain.

Given

Power $N = 9.5kW$

$N_1 = 900$ rpm

$N_2 = 400$ rpm

$a_0 = 600$ mm

find: Select the roller chain.

Soln:

1. Determination of the transmission ratio (i)

7.74 $$i = \frac{N_1}{N_2} = \frac{900}{400} = 2.25$$

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

7.74 for $i = 2.25$ $Z_1 = 27$ to 25

$$\boxed{Z_1 = 27}$$

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

7.74 $$i = \frac{Z_2}{Z_1} \qquad 2.25 = \frac{Z_2}{27}$$

$$Z_2 = 60.75 = 61$$

$$Z_{2\,max} = 100 \text{ to } 120 \qquad Z_2 < Z_{2\,max}$$

$Z_6 = 61$ is satisfactory

4. Selection of standard pitch (p)

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

Maximum pitch $$p_{max} = \frac{a}{30} = \frac{600}{30} = 20 \text{ mm}$$

Minimum pitch $$p_{min} = \frac{a}{50} = \frac{600}{50} = 12 \text{ mm}$$

Select standard pitch between 12mm & 20mm

7.74 std pitch $p = 15.875$ mm — Std P Nearer to Max. pitch

5. Selection of the chain:

Assume the chain to be duplex — duplex

7.72 for pitch $p = 15.875$ mm & DR

Selected chain number is 10A-2 / DR50.

6. Calculation of total load on the driving side of the chain (ΣP)

7.78 Total load $\Sigma P = P_t + P_c + P_s$ — $P_t + P_c + P_s$

i. Tangential force (P_t)

7.78 $P_t = \frac{102\, N}{V}$ kgf N in KW — $N = 9.5$ KW

$P_t = \frac{1020\, N}{V}$, N.

chain velocity $V = \frac{Z_1 \times p \times N_1}{60 \times 1000} = \frac{27 \times 15.875 \times 900}{60 \times 1000}$

$V = 6.43$ m/s

$P_t = \frac{1020 \times 9.5}{6.43} = \boxed{P_t = 1507\, N}$

ii. Centrifugal tension (P_c)

$P_c = m.v^2$ (or) $P_c = \frac{w\, v^2}{g}$ $\longrightarrow$ $P_c = \frac{17.8 \times 6.43^2}{9.81}$

m = mass per metre.

7.72 for 10A2/DR50 Weight per metre = 1.78 kgf = 17.8 N

(or) $\boxed{P_c = 75.02\, N}$

$= \boxed{\cancel{P_c = 73.59\, N}}$

iii. Tension due to sagging (P_s)

7.78 $P_s = K.\, w.\, a.$

coefficient for sag $k = 6$ (Horizontal)

Weight per metre of chain $W = 1.78$ kgf $= 17.8$ N

Initial centre distance $a = 0.6$ m

$$P_s = 6 \times 17.8 \times 0.6$$

$$\boxed{P_s = 64.08 \text{ N}}$$

Total load $\Sigma P = 1507 + 75.02 + 64.08$

$$\boxed{\Sigma P = 1646.1 \text{ N}}$$

7. Calculation of service factor (k_s)

7.76 $k_s = k_1 \cdot k_2 \cdot k_3 \cdot k_4 \cdot k_5 \cdot k_6$

Load factor $k_1 = 1.25$ (variable load or load with mild shocks)

Distance regulation factor $k_2 = 1$ (Adjustable supports)

factor for centre distance of sprockets
$k_3 = 1$ — $a_p = (30 \text{ to } 50)p$

factor for the position of the sprockets
$k_4 = 1$ (upto 60°) Horizontal drive

Lubrication factor $k_5 = 1$ (drop lubrication)

Rating factor $k_6 = 1.25$ (16 hrs/day)

$$k_s = 1.25 \times 1 \times 1 \times 1 \times 1 \times 1.25$$

$$\boxed{k_s = 1.5625}$$

8. Calculation of design load:

Design load $= \Sigma P \times k_s = 1646.1 \times 1.5625$

$= 2572.03 \text{ N}$

9. Calculation of working factor of safety (F_{SW})

$$F_{SW} = \frac{\text{Breaking load}}{\text{Design load}}$$

7.72 for 10A-2 / DR50

Breaking load = 4440 kgf

= 44400 N

$$F_{SW} = \frac{44400}{2572.03} = \underline{17.26}$$

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

10. Check for factor of safety: Smaller sprocket N_1 & Z_1

7.77 for smaller sprocket speed 900rpm & std pitch 15.875mm

Minimum factor of safety = 11

Working factor of safety 17.26 > Minimum fos 11

Design is safe.

11. Check for the bearing stress in the roller:

$$\sigma_{roller} = \frac{P_t \times K_s}{A} = \frac{1507 \times 1.5625}{140} = 16.8 \, N/mm^2$$

7.72 $A \rightarrow$ Bearing area = $140 mm^2$

$= 1.4 cm^2$

for smaller sprocket speed 900rpm & pitch 15.875mm

Allowable bearing stress = $22.4 N/mm^2$ > Induced stress $16.8 N/mm^2$

$2.24 \, kgf/mm^2$

∴ Design is safe

12. Calculation of length of chain (L)

7.75 Actual length of chain $L = l_p \times p$

Number of links $l_p = 2a_p + \left(\frac{Z_1 + Z_2}{2}\right) + \frac{\left[\frac{(Z_2 - Z_1)}{2\pi}\right]^2}{a_p}$

Approximate centre distance $a_p = \frac{a_0}{p} = \frac{600}{15.875} = \underline{37.795}$

(23)

$$l_p = 2(37.795) + \left(\frac{27+61}{2}\right) + \frac{\left(\frac{61-27}{2\pi}\right)^2}{37.795}$$

$= 120.36$

$l = 122 \times 15.875$

$= 1936.7$ mm

$l_p = 122$ links (rounded to an even number)

13. Calculation of exact centre distance (a)

7.75

$$a = \frac{e + \sqrt{e^2 - 8M}}{4} \times p$$

$$e = l_p - \left(\frac{z_1 + z_2}{2}\right) = 122 - \left(\frac{27+61}{2}\right) = 78$$

$e = 78$

$$M = \left(\frac{z_2 - z_1}{2\pi}\right)^2 = \left(\frac{61-27}{2\pi}\right)^2 = 29.28$$

$$a = \frac{78 + \sqrt{78^2 - 8 \times 29.28}}{4} \times 15.875 = 613.11 \text{ mm}$$

Decrement in centre distance for an initial sag $= 0.01a$

$\Delta a = 0.01a = 0.01 \times 613.11 = 6.1311$ mm

$\therefore$ Exact centre distance $= 613.11 - 6.1311 = 606.978$ mm

14. Calculation of sprocket diameters:

Smaller sprocket:

Pitch circle dia of smaller sprocket (Pcd) $d_1 = \frac{p}{\sin(180/z_1)} = \frac{15.875}{\sin(180/27)} = 136.74$ mm

Sprocket outside dia $d_{o1} = d_1 + 0.8\, d_r$

$d_{o1} = 136.74 + 0.8 \times 10.16 = 144.87$ mm

7.72 d_r = dia of roller $= 10.16$ mm

Larger sprocket

Pcd of larger sprocket $d_2 = \frac{p}{\sin(180/z_2)} = \frac{15.875}{\sin(180/61)} = 308.38$ mm

Sprocket outside dia $d_{o2} = d_2 + 0.8 d_r$

$= 308.38 + 0.8 \times 10.16 = 316.51$ mm

7. Design a chain drive to run a compressor from a 11 kW electric motor running at 970 rpm, the compressor speed being 330 rpm. The compressor operates 16 hrs/day. The centre distance should be approximately 500 mm. The chain tension can be adjusted by shifting the motor on slides.

Given:

Power $N = 11$ kW

$N_1 = 970$ rpm

$N_2 = 330$ rpm

Service = 16 hrs /day

Centre distance $a_0 = 500$ mm

find: Design a chain drive

Soln:

Procedures are same as $\boxed{\text{Problem No. 6}}$

---

8. A 7.5 kW electric motor running at 1400 rpm is used to drive the input shaft of the gear box of a machine. Design a suitable roller chain to connect the motor shaft to the gear box shaft to give an exact speed ratio of 10:1. The center to center distance of the shaft is to be approximately 600mm.

Given:

Power $N = 7.5\ kw$

$N_1 = 1400\ rpm$

Speed ratio $i = 10:1 = 10$

Center distance $a_o = 600\ mm$

Find: Design a roller chain

Soln:

1. Determination of the transmission ratio (i)

$$\boxed{i = 10}$$

$$i = \frac{N_1}{N_2} \qquad 10 = \frac{1400}{N_2}$$

$$\boxed{N_2 = 140\ rpm}$$

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

for $i = 10$ can't take Z_1 from 7.74

$\therefore$ Assume $\boxed{Z_1 = 15}$

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

7.74 $\quad i = \frac{Z_2}{Z_1} \qquad 10 = \frac{Z_2}{15}$

$Z_2 = 150 \qquad$ 7.74 $\boxed{Z_{2max} = 100 \text{ to } 120}$

$\therefore Z_2 = 150$ is not acceptable

$\therefore$ change $Z_1 \Rightarrow$ Assume $\boxed{Z_1 = 11}$

$$10 = \frac{Z_2}{11} \qquad \boxed{Z_2 = 110}$$

Remaining procedures step 4 to 14 are same as Problem No: 6

---

9. Design a chain drive to actuate a compressor from 15kW electric motor running at 1000 rpm, the compressor speed being 350 rpm. The minimum centre distance is 500 mm. The compressor operates 15 hours per day. The chain tension may be adjusted by shifting the motor.

Given:

$N = 15 kW$ Service = 15 hrs/day

$N_1 = 1000$ rpm

$N_2 = 350$ rpm

$a_0 = 500$ mm

Design procedures are same as Problem No: 6

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10. Design a chain drive to actuate a compressor from a 10 kW. electric motor at 960 rpm. The compressor speed is to be 350 rpm. Minimum centre distance should be 0.5m. Compressor is to work for 8 hrs/day.

Given: $N = 10 kW$ $N_2 = 350$ rpm Service = 8 hrs/day

$N_1 = 960$ rpm $a_0 = 0.5 m = 500$ mm

Design procedures are same as Problem No: 6

SPUR GEARS

1. Design a spur gear drive required to transmit 45 KW at a pinion speed of 800 rpm. The velocity ratio is 3.5:1. The teeth are 20° full depth involute with 18 teeth of the pinion. Both the pinion and gear are made of steel with a maximum safe static stress of 180 N/mm². Assume medium shock condition.

Given data

$P = 45\,KW$ $N_1 = 800\,rpm$ $i = 3.5$ $\phi = 20°$ $Z_1 = 18$

$[\sigma_b] = 180\,N/mm^2$

To find Design a spur gear

Solution Since both the pinion and gear are made of the same materials, the pinion is weaker than the gear. So we have to design pinion only.

1. **Selection of Materials**

Given that the pinion and gear are made of steel, Assume steel is hardened to 200 BHN.

2. **Calculation of Z_1 and Z_2**

Number of teeth on pinion $Z_1 = 18$

Number of teeth on gear $Z_2 = i \times Z_1 = 3.5 \times 18 = 63$

$Z_2 = 63$

3. **Calculation of F_t**

Tangential load $F_t = \frac{P}{V} \times K_o$

$V = \frac{\pi d_1 N_1}{60}$

$= \frac{\pi \times m \times 18 \times 800}{60 \times 1000}$

$V = 0.754\,m$

$K_o = 1.25$ For Medium shock condition

$F_t = \frac{45 \times 10^3}{0.754\,m} \times 1.25$

$F_t = \frac{74603}{m}$

4. Calculation of Fd

Initial dynamic load $F_d = \frac{F_t}{C_v}$ (2)

Where

C_v = Velocity factor Assume $V = 12\ m/s$

$= \frac{6}{6+V}$ for accurately hobbed & generated gears with $V < 20\ m/s$

$= \frac{6}{6+12} = 0.333$

$$F_d = \frac{74603}{m} \times \frac{1}{0.333} = \frac{223809}{m}$$

5. Calculation of Fs

Beam strength $F_s = \pi . m . b[\sigma_b] . y$

b = Face width = 10m assume

y = Form Factor

$y = 0.154 - \frac{0.912}{Z_1}$ for 20° full depth system

$= 0.154 - \frac{0.912}{18}$

$y = 0.1033$

$F_s = \pi \times m \times 10m \times 180 \times 0.1033$

$= 584.15 m^2$

6. Calculation of module (m)

We know that, $F_s \geq F_d$

$$584.15 m^2 \geq \frac{223809}{m}$$

module $m \geq 7.26mm$

$\boxed{\text{module } m = 8}$

7. Calculation of b, d and V

Face width (b) $b = 10 \times m = 10 \times 8 = 80mm$

Pitch circle diameter $d_1 = m.z_1 = 8 \times 18 = 144mm$

Pitch line velocity $V = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 144 \times 10^{-3} \times 800}{60} = 6.03 \ m/s$.

8. Recalculation of beam Strength (Fs)

Beam strength $F_s = \pi . m . b [\sigma_b] . y$
$= \pi \times 8 \times 80 \times 180 \times 0.1033$

$F_s = 37385.45 \ N$

9. Calculation of accurate dynamic load (Fd):

Dynamic load $F_d = F_t + \frac{21V(bc+F_t)}{21V + \sqrt{bc+F_t}}$

Where
$F_t = \frac{P}{V} = \frac{45 \times 10^3}{6.03} = 7462.68N$

C – deformation factor
$C = 11860 e$
$= 11860 \times 0.038$
$= 450.68 \ N/mm$

$F_d = 7462.68 + \frac{21 \times 6.03 \times 10^3 (80 \times 450.68 + 7462.68)}{21 \times 6.03 \times 10^3 + \sqrt{80 * 450.68 + 7462.68}}$

$F_d = 50908.19 \ N$

10. Check for beam strength (or tooth breakage) (A)

Since $F_d > F_s$ the design is unsatisfactory. That is dynamic load is greater than the beam strength.

In order to reduce the dynamic load F_d, select the precision gears. Therefore expected error in tooth profiles

$e = 0.019$ precision gears

$\therefore$ deformation factor $C = 11860 \times e = 11860 \times 0.019 = 225.34$

Therefore the dynamic load is recalculated as,

$$F_d = 7462.68 + \frac{21 \times 6.03 \times 10^3 (80 \times 225.34 + 7462.68)}{21 \times 6.03 \times 10^3 + \sqrt{80 \times 225.34 + 7462.68}}$$

$$F_d = 32920.46 \, N$$

$$\boxed{F_d \, (32920.46 N) < F_s (37385.45 N)}$$ Design is safe.

The gear tooth has adequate beam strength and it will not fail by breakage. Therefore the design is satisfactory.

11. Calculation of Maximum wear load (F_w)

Wear load $F_w = d_1 \times b \times Q \times K_w$

$= 144 \times 80 \times 1.555 \times 0.919$

$F_w = 16462.6 N$

Q = Ratio factor

$Q = \frac{2i}{i+1} = \frac{2 \times 3.5}{3.5+1} = 1.555$

K_w - load stress factor

$K_w = 0.919 \, N/mm^2$

12. Check for wear

Since $F_d > F_w$ the design is unsatisfactory. That is dynamic load is greater than the wear load.

In order to increase the wear load (F_w), we have to increase the hardness (BHN) So now for steel hardened to 400 BHN

$K_w = 2.553 \, N/mm^2$

$F_w = 144 \times 80 \times 1.555 \times 2.553 = 45733.42 N$

$$\boxed{\begin{array}{c} F_w\,(45733.42N) > F_d\,(32920.46N) \\ \text{Design is safe} \end{array}}$$

$F_w > F_d$, It means, the gear tooth has adequate wear capacity and it will not wear out. Therefore the design is satisfactory.

13. Basic dimensions of pinion and gear

Module $m = 8mm$

Numbers of teeth $Z_1 = 18$ mm and $Z_2 = 63$

Pitch circle diameter $d_1 = 144mm$ and

$$d_2 = mZ_2 = 8 \times 63 = 504mm$$

Centre distance $a = \frac{m(Z_1+Z_2)}{2} = \frac{8(18+63)}{2} = 324mm$

Face width $b = 80mm$

Height factor $f_0 = 1$ for $20°$ full depth teeth

Bottom clearance $c = 0.25m = 0.25 \times 8 = 2mm$

Tip diameter $d_{a_1} = (Z_1 + 2f_0)m = (18+2\times1)8 = 160mm$

$$d_{a_2} = (Z_2 + 2f_0)m = (63+2\times1)8 = 520mm$$

Root diameter $d_{f_1} = (Z_1 - 2f_0)m - 2c$

$$= (18-2\times1)8 - 2\times2 = 124mm$$

$$d_{f_2} = (Z_2 - 2\times1)8 - 2\times2$$

$$= (63-2\times1)8 - 2\times2 = \underline{\underline{484mm}}$$

2. Design a spur gear drive to transmit 8KW at 720 rpm and the speed ratio is 2. The pinion and wheel are made of the same surface hardened carbon steel with 55RC and core hardness less than 350 BHN. Ultimate strength is 720 N/mm^2 and yield strength is 360 N/mm^2. (6)

Solution.

This problem is very much similar to the problem no 1.

Refer the problem No: 1.

GEAR DESIGN IS BASED ON GEAR LIFE

3. In a spur gear drive for a stone crusher, the gears are made of C40 steel. The pinion is transmitting 30KW at 1200 rpm. The gear ratio is 3. Gear is to work 8 hours per day, six days a weak and for 3 years. Design the drive.

Given Data

Pinion and gear materials : C40 steel

$P = 30KW = 30 \times 10^3 W$ $N_1 = 1200 rpm$ $i = 3$.

To find Design the spur gear drive.

Solution. Since the pinion and gear are made of same materials (i.e C40 steel) therefore we have to do the design of pinion alone.

1) Gear ratio $i = 3$ (given)

2) Material selection

Pinion and gear are made of C40 steel

Assume surface hardness > 350

3. Gear life. Given that the gear is to works 8 hours per day, six days a week and for 3 years. Therefore gear life in terms of hours is give by

$$\text{Gear life} = 8(52 \times 6) \times 3 = 7488 \text{ hours} = 449280 \text{ min}$$

Lifein number of cycles,

$$N = 449280 \times N_1$$
$$= 449280 \times 1200 = 53.9 \times 10^7 \text{ cycles.}$$

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

$$\text{Design Torque } [M_t] = M_t \cdot K \cdot K_d$$

$$\text{Where } M_t = \frac{60 \times P}{2\pi N_1} = \frac{60 \times 30 \times 10^3}{2\pi \times 1200} = 238.73 \text{ Nm}$$

$$K \cdot K_d = 1.3$$
$$[M_t] = 238.73 \times 1.3 = 310.34 \text{ Nm}$$

5. Calculation of Eeq, $[\sigma_b]$ and $[\sigma_c]$

(i) To find Eeq For C40 steel $E_{eq} = 2.15 \times 10^5 \text{ N/mm}^2$

(ii) To find $[\sigma_b]$: The design bending stress $[\sigma_b]$

$$[\sigma_b] = \frac{1.4 K_{bl}}{n \cdot K_\sigma} \sigma_{-1}$$ assuming rotation in one direction only.

$$[\sigma_b] = \frac{1.4 \times 0.7}{2 \times 1.5} \times 340.5$$
$$= 111.23 \text{ N/mm}^2$$

$K_{bl} = 0.7$

$n = 2$

$K_\sigma = 1.5$

$\sigma_{-1} = 0.35 \sigma_u + 120$

$= (0.35 \times 630) + 120$

$\sigma_{-1} = 340.5 \text{ N/mm}^2$

(iii) To find $[\sigma_c]$

$$[\sigma_c] = C_R \cdot HRC \cdot K_{cl}$$

$$= 26.5 \times 55 \times 0.585$$

$$[\sigma_c] = 852.64 \ N/mm^2$$

$C_R = 26.5$

$HRC = 55$

$K_{cl} = 0.585$

6. Calculation of Centre distance (a)

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

where $\psi = 0.3$

$$a \geq (3+1) \sqrt[3]{\left(\frac{0.74}{852.64}\right)^2 \times \frac{2.15 \times 10^5 \times 310.34 \times 10^3}{3 \times 0.3}}$$

$$\geq 152.89 \text{ mm}$$

$$a = 155 \text{mm}$$

7. Selection of Z_1 and Z_2

(i) Assume $Z_1 = 17$ for $20°$ full depth system

(ii) $Z_2 = i \cdot Z_1 = 3 \times 17 = 51$

8. Calculation of module (m)

W.K.T $m = \frac{2a}{Z_1 + Z_2} = \frac{2 \times 155}{17+51} = 4.56 \text{mm}$

The nearest higher standard module

$$\boxed{m = 5 \text{mm}}$$

9. Revision of centre distance

New centre distance $a = \frac{m(Z_1+Z_2)}{2} = \frac{5(17+51)}{2}$

$$a = 170 \text{mm}$$

10. Calculation of b, d_1, V and ψ_p

Face width $b = \psi \cdot a = 0.3 \times 170 = 51\,mm$

Pitch diameter of pinion $\left\{ d_1 = m \cdot Z_1 = 5 \times 17 = 85\,mm \right.$

Pitch line velocity $V = \dfrac{\pi d_1 N_1}{60} = \dfrac{\pi \times 85 \times 10^{-3} \times 1200}{60} = 5.34\,m/s$

$$\psi_p = \frac{b}{d_1} = \frac{51}{85} = 0.6$$

11. Selection of quality of gears

For pitch line Velocity $5.34\,m/s$

IS quality 8 gears are selected.

12. Revision of design torque of gears $[M_t]$

Revise k. $k = 1.03$

Revise kd $k_d = 1.4$

Design torque $[M_t] = M_t \cdot k \cdot k_d$

$= 238.73 \times 1.03 \times 1.4$

$= 344.24\,Nm.$

13. Check for bending

Calculation of induced bending stress σ_b:

$$\sigma_b = \frac{i+1}{a \cdot m \cdot b \cdot y} \times [M_t] \qquad y = \text{form factor} = 0.366$$

$$\sigma_b = \frac{3+1}{170 \times 5 \times 51 \times 0.366} \times 344.24 \times 10^3 = 86.78\,N/mm^2$$

$\sigma_b\ (86.78\,N/mm^2) < [\sigma_b]\ (111.23\,N/mm^2)$

The design is safe.

14. Check for wear strength (10)

Calculation of induced contact stress σ_c:

$$\sigma_c = 0.74 \frac{i+1}{a} \sqrt{\frac{i+1}{ib} \times E_{eq}[M_t]}$$

$$= 0.74 \frac{3+1}{170} \sqrt{\frac{3+1}{3 \times 51} \times 2.15 \times 10^5 \times 344.24 \times 10^3}$$

$$\sigma_c = 765.9 \text{ N/mm}^2$$

$$\boxed{\sigma_c = 765.9 < [\sigma_c] = 152.64 \text{ N/mm}^2}$$

The design is safe and satisfactory.

15. Calculation of basic dimensions of pinion & gear

Module $m = 5$ mm

Face width $b = 51$ mm

Height Factor $f_0 = 1$ for full depth ~~system~~ teeth

Bottom clearance $C = 0.25m = 0.25 \times 5 = 1.25$ mm

Tooth depth $h = 2.25m = 2.25 \times 5 = 11.25$ mm

Pitch circle diameter $d_1 = mZ_1 = 5 \times 17 = 85$ mm

$d_2 = mZ_2 = 5 \times 51 = 255$ mm

Helical gear

4. (7) For intermittent duty of an elevator, two cylindrical gears have to transmit 12.5 kW at a pinion speed of 1200 rpm. Design the gear pair for the following specifications. Gear ratio 3.5, pressure angle 20°, involute full depth, helix angle 15°. Gears are expected to work 6 hours a day for 10 years.

Given: $P = 12.5$ kW

$N_1 = 1200$ rpm

$i = 3.5$

$\phi = 20^\circ$

$\beta = 15^\circ$

Soln:

1. Gear ratio $i = 3.5$
2. Selection of material

   For pinion and gear ; alloy steel can be selected.
3. Gear life: 6 hrs a day for 10 years

   $$\text{Gear life} = 6 \times 365 \times 10 = 21,900 \text{ hrs}$$

   $$N = 21,900 \times 1200 \times 60 = 157.7 \times 10^7 \text{ cycles}$$
4. Calculation of initial Design torque $[M_t]$

   P.44

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

   $$M_t = \frac{60 \times P}{2\pi N} = \frac{60 \times 12.5 \times 10^3}{2\pi \times 1200} = 99.47 \text{ N-m}$$

   Assume $K \cdot K_d = 1.3$

   $$[M_t] = 99.47 \times 1.3 = 129.31 \text{ N-m}$$

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

$E_{eq} = 2.15 \times 10^5 \ N/mm^2$ 8.14

$[\sigma_b] = \dfrac{1.4 k_{bl}}{n \cdot k_\sigma} \times \sigma_{-1}$

8.18

$k_{bl} = 0.7$ for $HB > 350$ and $N \geq 25 \times 10^7$

$k_\sigma = 1.5$

$n = 2.5$ steel hardened

$\sigma_{-1} = 0.35 \sigma_u + 120$

8.19

$\sigma_u = 1550 \ N/mm^2$ — 1.C50

$\sigma_{-1} = 0.35 \times 1550 + 120 = 662.5 \ N/mm^2$

$[\sigma_b] = \dfrac{1.4 \times 0.7}{2.5 \times 1.5} \times 662.5 = 173.127 \ N/mm^2$

$[\sigma_c] = C_R \times H_{RC} \times k_{cl}$

8.16

$C_R = 26.5$

$H_{RC} = 40 \text{ to } 55$

$k_{cl} = 0.585$ $HB > 350$ & $N \geq 25 \times 10^7$

$[\sigma_c] = 26.5 \times 55 \times 0.585 = 852.6 \ N/mm^2$

6. Calculation of centre distance (a)

$$a \geq (i+1) \sqrt[3]{\left(\frac{0.7}{[\sigma_c]}\right)^2 \frac{E_{eq}[M_t]}{i \psi}}$$

$\psi = b/a = 0.3$

8.13

$$a \geq (3.5+1) \sqrt[3]{\left(\frac{0.7}{852.6}\right)^2 \times \frac{2.15 \times 10^5 \times 129.31 \times 10^3}{3.5 \times 0.3}}$$

$a \geq 117.6 \text{ mm} \approx 120 \text{ mm}$

7. Assume $Z_1 = 20$

$Z_2 = 3.5 \times 20 = 70$

(18)

8. Calculation of normal module (m_n)

$$m_n = \frac{2a}{z_1+z_2} \times \cos\beta = \frac{2\times 120}{(20+70)} \times \cos 15^\circ = 2.576mm$$

$= 3mm$

9. Revision of centre distance:

$$a = \left(\frac{m_n}{\cos\beta}\right) \times \left(\frac{z_1+z_2}{2}\right)$$

$$= \frac{3}{\cos 15^\circ} \times \left(\frac{20+70}{2}\right) = 139.76mm \Rightarrow 140mm$$

10. Calculation of b, d_1, v & ψ_p.

$$b = \psi \times a = 0.3 \times 139.76 = 41.93 = 42mm$$

$$d_1 = \frac{m_n}{\cos\beta} \times z_1 = \frac{3}{\cos 15} \times 20 = 62.12\ mm$$

$$v = \frac{\pi d_1 n_1}{60} = \frac{\pi \times 62.12 \times 10^{-3} \times 1200}{60} = 3.903\ m/s$$

$$\psi_p = b/d_1 = \frac{42}{62.12} = 0.676$$

11. Selection of quality of gear

for HB > 350 v upto 8 m/s

8.3

IS quality 8 is selected.

12. Revision of design torque $[M_t]$

$[M_t] = M_t \times k \times k_d$

8.15

$k = 1.045$ for $\psi_p = 0.676$

$k_d = 1.2$ for IS quality 8

$[M_t] = 99.432 \times 1.045 \times 1.2 = 124.76\ N\text{-}m$

13. Check for bending:

$$\sigma_b = \frac{0.7\,(i\pm1)\,[M_t]}{a \cdot b \cdot m_n \cdot y_v}$$

8.13A

(14)

8.18 $y_v = 0.402$ for $Z_{v_1} = 22$ $Z_{v_1} = \frac{Z_1}{\cos^3\beta} = \frac{20}{\cos^3 15°} = 22$

$$\sigma_b = \frac{0.7(3.5+1)\ 124.71 \times 10^3}{139.76 \times 42 \times 3 \times 0.402} = 55.5\ N/mm^2$$

$\sigma_b < [\sigma_b]$ safe.

14. Check for wear strength:

$$\sigma_c = 0.7\ \frac{i+1}{a} \sqrt{\frac{i+1}{ib}\ E_{eq}\ [M_t]}$$

P.13 $$= 0.7\left(\frac{3.5+1}{139.76}\right) \sqrt{\frac{3.5+1}{3.5 \times 42} \times 2.15 \times 10^5 \times 124.71 \times 10^3}$$

$= 645.8\ N/mm^2$

$\sigma_c < [\sigma_c]$ safe.

15. Basic dimensions:

Q.22
i. Normal module $m_n = 3$ mm
ii. Centre distance $a = 120$ mm
iii. $Z_1 = 20$ & $Z_2 = 70$
iv. Height factor $f_0 = 1$
v. Bottom clearance $= h = 2.25 \times m_n$
$= 2.25 \times m_n = 6.75$ mm.

vi. Pitch dia $d_1 = \frac{m_n}{\cos\beta} \times Z_1 = 62.12$ mm

$d_2 = \frac{m_n}{\cos\beta} \times Z_2 = \frac{3}{\cos 15} \times 70$

$d_2 = 217.4$ mm

vii. Tip dia $d_{a_1} = \left(\frac{Z_1}{\cos\beta} + 2f_0\right) m_n$

$d_{a_2} = \left(\frac{Z_2}{\cos\beta} + 2f_0\right) m_n$

viii. Root dia $d_{f_1} = \left(\frac{Z_1}{\cos\beta} - 2f_0\right) m_n - 2c$

$d_{f_2} = \left(\frac{Z_2}{\cos\beta} - 2f_0\right) m_n - 2c$

ix. Virtual no. of teeth:

$Z_{v_1} = \frac{Z_1}{\cos^3\beta}$ $Z_{v_2} = \frac{Z_2}{\cos^3\beta}$

5. A pair of helical gear subjected to moderate shock loading is to transmit 40 kW at 1750 rpm of the Pinion. The speed reduction ratio is 4. Design the gear for a life of 10,000 hrs.

Given:

$P = 40 kW$

$N_1 = 1750 rpm$

$Life = 10,000 hrs$

Find:

D/n a helical gear based on gear life.

Soln:

Assume $\beta = 15^\circ$ helix angle & $\phi = 20^\circ$ involute full depth.

Design procedures are same as $\boxed{\text{Problem No: 4}}$ (step 1 to 15)

---

6. Design a helical gear drive to transmit the power of 14.7 kW. Speed ratio 6, Pinion speed 1200 rpm, helix angle is 25°. Select suitable material & design the gear drive.

Given:

$P = 14.7 kW$

$i = 6$ $\quad \frac{N_1}{N_2} = i, \quad \frac{1200}{N_2} = 6 \quad N_2 = 200 rpm$

$N_1 = 1200$

$\beta = 25^\circ$

Find: D/n a helical gear drive based on strength.

Soln:

1. Material Selection:

Select $\boxed{\text{C 45 Steel}}$

Pinion & gear are made of C45 steel

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

2. Calculation of Z_1 & Z_2

Assume $\boxed{Z_1 = 20}$ $\quad \frac{Z_2}{Z_1} = i \quad \frac{Z_2}{20} = 6$

$$\boxed{Z_2 = 120}$$

3. Calculation of Tangential Load (F_t)

8.51 $\quad F_t = \frac{P}{V} \times K_o$

$$V = \frac{\pi d_1 N_1}{60} \qquad d_1 = \frac{m_n}{\cos\beta} Z_1 = \frac{m_n}{\cos 25^\circ} \times 20$$

$$\boxed{d_1 = 22.07\ m_n}\ \text{mm}$$

$$V = \frac{\pi \times 22.07\ m_n \times 1200}{60 \times 1000} = 1.39\ m_n$$

$$\boxed{V = 1.39\ m_n}\ \text{m/s}$$

$\boxed{K_o = 1.5 \text{ for medium shock}}$

$$F_t = \frac{14.7 \times 10^3}{1.39\ m_n} \times 1.5$$

$$\boxed{F_t = \frac{15863.31}{m_n}}$$

4. Initial Dynamic load (F_d)

Initially Assume $\boxed{V = 15\ \text{m/s}}$

8.51 or 8.50 $\quad F_d = F_t \times C_v$

$$C_v = \frac{6+V}{6} = \frac{6+15}{6}$$

$$\boxed{C_v = 3.5}$$

(14)

$$F_d = \frac{15863.31}{m_n} \times 3.5$$

$$\boxed{F_d = \frac{55521.6}{m_n}}$$

5. Beam strength (F_s)

8.51 $F_s = \pi . m_n . b . [\sigma_b] . y'$

8.51 $b = 10 m_n$ $[\sigma_b] = 180 \ N/mm^2$ for C45

$y' = 0.154 - \frac{0.912}{Z_{v_1}}$ for 20° involute / 8.50

$$Z_{v_1} = \frac{Z_1}{\cos^3\beta} = \frac{20}{\cos^3 25} = 27$$

$$y' = 0.154 - \frac{0.912}{27} = 0.1202$$

$$F_s = \pi \times m_n \times 10 m_n \times [180] \times 0.1202$$

$$\boxed{F_s = 679.84 \ m_n^2}$$

6. Normal module (m_n)

Equate F_s & F_d

$$679.84 \ m_n^2 = \frac{55521.6}{m_n}$$

$$\boxed{m_n = 4.33 = 5 \ mm}$$

7. b, d & v

face width $b = 10 \ m_n = 50 mm$

Pcd $d_1 = \frac{m_n}{\cos\beta} Z_1 = \frac{5}{\cos 25} \times 20 = 110.34 \ mm$

$$V_1 = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.11034 \times 1200}{60} = 6.93 \ m/s$$

8. Recalculation of beam strength F_s (18)

$$F_s = 679.84\, m_n^2$$
$$= 679.84 \times 5^2$$
$$\boxed{F_s = 16996\ N}$$

9. Accurate dynamic load (F_d)

8.51 $$F_d = F_t + \left[\frac{0.164\,V\,(Cb \times \cos^2\beta + F_t)\cos\beta}{0.164V + 1.485\sqrt{Cb\cos^2\beta + F_t}}\right]$$

$$F_t = \frac{P}{V}$$ neglect K_o

$$F_t = \frac{14.7 \times 10^3}{6.93} = \boxed{2121.21\ N}$$

Deformation factor

8.53 $C = 11860$ for steel & steel

Expected error $e = 0.0125$ for $m_n = 5$ mm & Precision gear

$$C = 11860 \times 0.0125$$
$$\boxed{C = 148.25\ N/mm}$$

$$F_d = 2121.21 + \left[\frac{(0.164 \times 6.93)(148.25 \times 50 \times \cos^2 25 + 2121.21)\cos 25}{(0.164 \times 6.93) + [1.485 \times \sqrt{148.25 \times 50 \times \cos^2 25 + 2121.21}]}\right]$$

$$= 2121.21 + \frac{8456.4}{135.69}$$

$$\boxed{F_d = 2183.5\ N}$$

10. Check for beam strength

$F_s > F_d$ ∴ Design is safe

16996 N 2183.5 N

11. Maximum wear load (F_w)

8.51 $$F_w = \frac{b\, d_1\, Q\, K}{\cos^2 \beta}$$

Ratio factor $Q = \frac{2i}{i+1} = \frac{2\times 6}{6+1} = 1.714$

$K = 1$

$$F_w = \frac{50 \times 110.34 \times 1.714 \times 1}{\cos^2 25} = 11686.94\ N$$

$$\boxed{F_w = 11686.94\ N}$$

12. Check for wear

$F_w > F_d$

11686.94N   2183.5 N

∴ Design is safe

13. Basic dimensions

8.22 Same a $\boxed{\text{Problem No : 4}}$ $\boxed{\text{Step 15}}$

7. Design a pair of helical gear to transmit 10 kW at 1000 rpm of the pinion. Reduction ratio of 5 is required. Give the details of the drive in a tabular form.

Given:

$P = 10\,kW$

$N_1 = 1000\,rpm$

$i = 5$

$\frac{N_1}{N_2} = i = \frac{1000}{N_2} = 5$

$\boxed{N_2 = 200\,rpm}$

find: Design a helical gear [based on strength]

Assume $\beta = 15°$ or $20°$

Material = C45 $[\sigma_b] = 180\,N/mm^2$

Design procedures are same as [Problem No: 6]

[Step 1 to 13]

(OR)

Design a helical gear [based on gear life]

Assume life = 20,000 hrs

follow [Problem No: 4]

---

8. Design a helical gear drive to connect an electric motor to a reciprocating pump. Gears are over hanging in their shafts. Motor speed = 1440 rpm, Speed reduction ratio = 5, Motor Power = 37 kW, Pressure angle = 20° Helix angle = 25°.

Given:

$N_1 = 1440\,rpm$

$i = 5$

$P = 37\,kW$

$\phi = 20°$

$\beta = 25°$

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

$\frac{1440}{N_2} = 5$

$\boxed{N_2 = 288\,rpm}$

Procedures are same as [Problem No: 6]

(OR)

follow [Problem No: 7]

BEVEL GEAR:

1. Design a bevel gear drive to transmit 7.5 kW at 1440 rpm. Gear ratio 3. Pinion and gear are made of forged C45 steel. Life of gears 10,000 hrs. Assume surface hardened heat treatment and IS quality 6.

Given:

$P = 7.5\,kW$ Material = C45 steel

$N_1 = 1440\,rpm$ Life = 10,000 hrs

$i = 3$

find: Design a bevel gear based on gear life

Soln: Since the materials of pinion and gear are same

Made of C45 steel.

∴ We have to design only the pinion.

1. Gear ratio (i) & pitch angle (δ_1 & δ_2)

$$i = \frac{N_1}{N_2} \;;\; 3 = \frac{1440}{N_2}$$

$$\boxed{N_2 = 480\,rpm}$$

8.39 Pitch angles or Reference angle

$\tan\delta_2 = i$ $\delta_1 = 90° - \delta_2$

$\delta_2 = \tan^{-1}(i)$ $\delta_1 = 90° - 71.6°$

$= \tan^{-1}(3)$ $\boxed{\delta_1 = 18.4°}$

$$\boxed{\delta_2 = 71.6°}$$

2. Material for pinion & gear

Pinion & gear are made of C45 steel

1.9 for C45 steel $\sigma_u = 63$ to 71 kgf/mm^2

$= 630$ to 710 N/mm^2

select $\boxed{\sigma_u = 700\ N/mm^2}$

$\boxed{\sigma_y = 36\ kgf/mm^2 = 360\ N/mm^2}$

3. Gear life:

Gear life in hours $T = 10{,}000$ hrs

8.17 Gear life in number of cycles $N = 60.n.T$ $\boxed{n = N_1 = 1440 rpm}$

$N = 60 \times 1440 \times 10{,}000$

$\boxed{N = 86.4 \times 10^7 \text{ cycles}}$

4. Initial design torque $[M_t]$

8.15 $[M_t] = M_t . k . k_d$

Initially assume $\boxed{k.k_d = 1.3}$

$P = \frac{2\pi N_1 M_t}{60}$; $7.5 \times 10^3 = \frac{2\pi \times 1440 \times M_t}{60}$

$M_t = 49.74 N.m = 49.74 \times 10^3 N.mm$

$\boxed{M_t = 49.74 \times 10^3 \text{ N.mm}}$

Initial design torque $[M_t] = 49.74 \times 10^3 \times 1.3$

$$\boxed{[M_t] = 64.662 \times 10^3 \text{ N.mm}}$$

5. E_{eq}, $[\sigma_b]$ & $[\sigma_c]$

i. Equivalent young's modulus:

8.14 for steel & steel $E_{eq} = 2.15 \times 10^6$ kgf/cm²

Pinion 1 Wheel 2

$$\boxed{E_{eq} = 2.15 \times 10^5 \text{ N/mm}^2}$$

ii. Design bending stress $[\sigma_b]$

8.18 $$[\sigma_b] = \frac{1.4 \; K_{bl}}{n . K_\sigma} \times \sigma_{-1}$$

1.9 for C45 HB = 229 ∴ HB < 350

8.20 life factor for bending $\boxed{K_{bl} = 1}$

for steel, HB < 350 & $N > 10^7$ from Table 22

life in No. of cycles $N = 86.4 \times 10^7$ cycles
∴ $N > 10^7$

stress concentration factor for the fillet (K_σ)

8.19. for surface hardened $0 \leq x \leq 0.1$

$$\boxed{K_\sigma = 1.5}$$

factor of safety 'n'

8.19 for steel, forged & surface hardened

$$\boxed{n = 2.5}$$ table 20

8.19 for forged steel

$$\sigma_{-1} = 0.25\,(\sigma_u + \sigma_y) + 50$$

1.9 for C45 steel $\sigma_u = 700\ N/mm^2$

$\sigma_y = 360\ N/mm^2$

$$\sigma_{-1} = 0.25\,(700 + 360) + 50$$

$$\boxed{\sigma_{-1} = 315\ N/mm^2}$$

Design bending stress $[\sigma_b] = \dfrac{1.4 \times 1}{2.5 \times 1.5} \times 315$

$$\boxed{[\sigma_b] = 117.6\ N/mm^2}$$

iii. Design contact stress $[\sigma_c]$

8.16 $[\sigma_c] = C_R \cdot HRC \cdot K_{cl}$

table 16 co-efficient $\boxed{C_R = 230}$ for C45 & surface hardened

$HRC = 40$ to 55

select $\boxed{HRC = 50}$

life factor for surface (or) contact stress

8.17 for steel, HB < 350 & $N > 10^7$ cycles

table 17

$$\boxed{K_{cl} = 1}$$

Design contact stress $[\sigma_c] = 230 \times 50 \times 1 = 11500 \ kgf/cm^2$

$= 115000 \ N/cm^2 = 115000 \times 10^{-2} \ N/mm^2$

$$\boxed{[\sigma_c] = 1150 \ N/mm^2}$$

6. Cone distance (R)

8.13 table 8

$$R \geq \psi_y \sqrt{i^2+1} \cdot \sqrt[3]{\left[\frac{0.72}{(\psi-0.5)[\sigma_c]}\right]^2 \frac{E_{eq}.[M_t]}{i}}$$

8.39 $\psi_y = \frac{R}{b}$

for $i = 3$ $\boxed{\psi_y = \frac{R}{b} = 3}$ Initially

8.15

table 13

$$R \geq 3\left[\sqrt{3^2+1}\right] \times \sqrt[3]{\left[\frac{0.72}{(3-0.5)[1150]}\right]^2 \times \frac{2.15 \times 10^5 \times [64.662 \times 10^3]}{3}}$$

$R \geq 9.49 \times 6.624$

$$\boxed{R \geq 62.86 = 63 \ mm}$$

7. Number of teeth:

Assume $\boxed{Z_1 = 20}$

$$\frac{Z_2}{Z_1} = i \; ; \; \frac{Z_2}{20} = 3$$

$$\boxed{Z_2 = 60}$$

Virtual no. of teeth $Z_{v_1} = \frac{Z_1}{\cos\delta_1} = \frac{20}{\cos 18.4^\circ} = 21.1 = \underline{\underline{22}}$

8.39

$$Z_{v_2} = \frac{Z_2}{\cos\delta_2} = \frac{60}{\cos 71.6^\circ} = 190.1 = \underline{\underline{191}}.$$

8. Transverse module (m_t)

cone distance $R = 0.5 . m_t \sqrt{Z_1^2 + Z_2^2}$

8.38

$$63 = 0.5 \times m_t \times \sqrt{20^2 + 60^2}$$

$$m_t = 1.99 = \underline{\underline{2\,mm}}$$

8.2 standard module $\boxed{m_t = 2\,mm}$

9. Revision of cone distance (R)

$$R = 0.5 \times 2 \times \sqrt{20^2 + 60^2}$$

$$\boxed{R = 63.25 = 64\,mm}$$

10. Calculation of b, m_{av}, d_{1av}, v & φ_p

face width $b = \frac{R}{\varphi_y}$ $\quad \varphi_y = \frac{R}{b}$

8.15 & 8.39

$\varphi_y = 3$

$b = \frac{64}{3} = 21.33$ mm

$\boxed{b = 21.3 \text{ mm}}$

8.38.

$\boxed{m_t = m_{av} + \frac{b \sin \delta_1}{Z_1}}$

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

8.38

$= 2 - \frac{21.3 \times \sin 18.4^\circ}{20}$

$m_{av} = 1.663$ mm $= 1.7$ mm

Average pitch circle diameter

$d_{1av} = m_{av} \times Z_1$

$= 1.7 \times 20$

$\boxed{d_{1av} = 34}$

$d_{2av} = m_{av} \times Z_2$

$= 1.7 \times 60$

$\boxed{d_{2av} = 102}$

Pitch line velocity $(v) = \frac{\pi \times d_{1av} \times N_1}{60}$

$= \frac{\pi \times 0.034 \times 1440}{60}$

$\boxed{v = 2.564 \text{ m/s}}$

8.15 $\varphi_P = \frac{b}{d_1} = \frac{b}{d_{1av}}$

Table 14 $\therefore \varphi_P = \frac{21.3}{34} = 0.63$

$$\boxed{\varphi_P = 0.63}$$

11. Selection of Quality of gear:

$\boxed{\text{Is Quality 6}}$ gear is given in question

If it is not given 8.3 select quality by using velocity 'v'

12. Revision of design torque $[M_t]$

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

Revise K: load concentration factor

8.15 Table 14 for bevel gear HB < 350 & $\frac{b}{d_{1av}} = \varphi_P = 0.63$

$\varphi_P < 1$

$$\boxed{K = 1.1}$$

Revise K_d: Dynamic load factor

Bevel gear → Straight

8.16 Table 15 for $v = 2.564$ m/s

$\therefore$ v upto 3 m/s

Conical gear

HB < 350

$$\boxed{K_d = 1.35}$$

Revise $[M_t] = 49.74 \times 10^3 \times 1.1 \times 1.35$

Revised $\boxed{[M_t] = 73.864 \times 10^3 \text{ N.mm}}$

13. Check for bending:

8.13A

$$\sigma_b = \frac{R\sqrt{i^2+1}\,[M_t]}{(R-0.5b)^2 \cdot b \cdot m_t \cdot y_{v_1}} \times \frac{1}{\cos\alpha} \le [\sigma_b]$$

form factor $\boxed{y_{v_1} = 0.402}$ $\boxed{\text{for } \alpha = 20^\circ \text{ \& } x = 0 \quad Z_{v_1} = 22}$

8.18

table 18

$$\sigma_b = \frac{64 \times \sqrt{3^2+1} \times [73.864 \times 10^3]}{(64 - 0.5 \times 21.3)^2 \times 21.3 \times 2 \times 0.402} \times \frac{1}{\cos 20^\circ}$$

$$\sigma_b = \frac{14949022.53}{48742.13} \times \frac{1}{\cos 20^\circ} = 326.4 \ N/mm^2$$

$$\sigma_b = 326.4 \ N/mm^2 > [\sigma_b]$$

$\boxed{\therefore \text{Design is unsatisfactory}}$

$\therefore$ Increase the value of transverse module m_t

$\boxed{m_t = 4 mm}$ $\boxed{\text{Repeat from step 9}}$

$$R = 0.5 \times m_t \times \sqrt{z_1^2 + z_2^2} = 0.5 \times 4 \times \sqrt{20^2 + 60^2}$$

$$\boxed{R = 126.5 \ mm}$$

$$b = \frac{R}{\psi_y} = \frac{126.5}{3} = 42.17 \ mm$$

$$\boxed{b = 42.17 \ mm}$$

$$m_{av} = m_t - \frac{b \sin \delta_1}{Z_1} = 4 - \frac{126.5 \times \sin 18.4^\circ}{20}$$

$$\boxed{m_{av} = 2 \text{ mm}}$$

$$d_{1av} = m_{av} \times Z_1 = 2 \times 20 = \underline{40 \text{ mm}}$$

$$d_{2av} = m_{av} \times Z_2 = 2 \times 60 = \underline{120 \text{ mm}}$$

$$V = \frac{\pi \times d_{1av} \times N_1}{60} = \frac{\pi \times 0.04 \times 1440}{60}$$

$$\boxed{V = 3.02 \text{ m/s}}$$

$$\varphi_p = \frac{b}{d_{1av}} = \frac{42.17}{40} = \underline{1.05}$$

HB < 350 $\frac{b}{d_{1av}} = 1.05$ 1 to 1.6

8.15 Load concentration factor $\boxed{k = 1.2}$

8.16 Dynamic load factor $\boxed{k_d = 1.35}$ for $v = 3.02$ m/s $\simeq 3$ m/s, IS quality 6, conical gear, straight bevel

$$[M_t] = 49.74 \times 10^3 \times 1.2 \times 1.35$$

$$\boxed{[M_t] = 80.579 \times 10^3 \text{ N.mm}}$$

$$\boxed{y_v = 0.402}$$

Recalculate σ_b

$$\sigma_b = \frac{126.5 \times \sqrt{3^2+1} \times 80.579 \times 10^3}{(126.5 - 0.5 \times 42.17)^2 \times 42.17 \times 4 \times 0.402} \times \frac{1}{\cos 20^\circ}$$

$$\sigma_b = 45.52 \text{ N/mm}^2 < [\sigma_b] \quad 117.6 \text{ N/mm}^2$$

$\therefore$ Design is Safe

14. Check for wear strength

8.13. $$\sigma_c = \frac{0.72}{(R-0.5b)} \sqrt{\left(\frac{\sqrt{i^2+1}}{ib}\right)^3 \cdot E_{eq}\,[M_t]}$$

$$= \frac{0.72}{(126.5 - 0.5\times 42.17)} \times \sqrt{\left(\frac{\sqrt{3^2+1}}{3\times 42.17}\right)^3 \times 2.15\times10^5 \times 80.579\times10^3}$$

$\sigma_c = 142.13\ N/mm^2 < [\sigma_c]\ 1150\ N/mm^2$

∴ Design is satisfactory

15. Basic dimensions:

8.38 & 8.39 Transverse module $m_t = 4$ mm

No. of teeth $Z_1 = 20$ & $Z_2 = 60$

Cone distance $R = 126.5$ mm

Pitch angles $\delta_1 = 18.4°$ & $\delta_2 = 71.6°$

Tip dia $d_{a1} = m_t (Z_1 + 2\cos\delta_1)$

$= 4\,(20 + 2\times\cos 18.4)$

$d_{a1} = 87.59$ mm

$d_{a2} = m_t (Z_2 + 2\cos\delta_2)$

$d_{a2} = 4(60 + 2\cos 71.6°)$

$d_{a2} = 242.53$ mm

Face width $b = 42.17$ mm

Height factor $f_0 = 1$

Clearance $c = 0.2$

(12)

Addendum angle $\theta_{a1} = \tan\theta_{a1} = \tan\theta_{a2} = \frac{m_t \cdot f_0}{R}$

$$\tan\theta_{a1} = \tan\theta_{a2} = \frac{4\times 1}{126.5}$$

$$\theta_{a1} = \theta_{a2} = \underline{1.8^\circ}$$

Dedendum angle $\tan\theta_{f1} = \tan\theta_{f2} = \frac{m_t(f_0 + c)}{R}$

$$\tan\theta_{f1} = \tan\theta_{f2} = \frac{4\times(1+0.2)}{126.5}$$

$$\theta_{f1} = \theta_{f2} = \underline{2.17^\circ}$$

Tip angle $\delta_{a1} = \delta_1 + \theta_{a1} = 18.4 + 1.8 = \underline{20.2^\circ}$

$$\delta_{a2} = \delta_2 + \theta_{a2} = 71.6 + 1.8 = \underline{73.4^\circ}$$

Root angle $\delta_{f1} = \delta_1 - \theta_{f1} = 18.4 - 2.17 = \underline{16.23^\circ}$

$$\delta_{f2} = \delta_2 - \theta_{f2} = 71.6 - 2.17 = \underline{69.43^\circ}$$

Virtual no. of teeth $Z_{v1} = 22$

$$Z_{v2} = 191$$

2. Design a bevel gear drive to transmit 7 kW at 1600 rpm for the following data:

Gear ratio = 3

Material for pinion & gear = C45 steel

Life = 10,000 hours

Given: $P = 7\,kW$ Material C45 steel for pinion & gear

$N_1 = 1600$ rpm

$i = 3$

Life $T = 10,000$ hrs

Find: Design a bevel gear based on gear life

Assume surface hardened heat treatment forged C45 steel

Design procedures are same as $\boxed{\text{Problem No: 1}}$

---

3. Design a bevel gear drive to transmit 10 kW power at 1440 rpm. Gear ratio is 3 and life of gears 10,000 hrs. Pinion and gear are made of C45 steel and minimum no. of teeth is 20.

Given: $P = 10\,kW$ $i = 3$ $Z_1 = 20$

$N_1 = 1440$ rpm $T = \text{life} = 10,000$ hrs Material = C45 steel

Find: Design a bevel gear based on gear life

Pinion & gear are made of C45 steel

Assume surface hardened & forged C45 steel

Design procedures are same as $\boxed{\text{Problem No: 1}}$

4. Design a pair of straight bevel gears for two shafts whose axis are at right angles. The power transmitted is 25 kW. The speed of pinion is 300 rpm and the gear is 120 rpm.

Given: $P = 25kW$ $N_1 = 300$ rpm $N_2 = 120$ rpm

Find: Design a bevel gear based on strength using lewis and buckinggham's eqn.

Soln:

1. Selection of material:

Select 15Ni 2Cr 1Mo 15 → Alloy steel for case hardening

1.40

Pinion and gear are made of same material

2. Calculation of Z_1 & Z_2

Assume $\boxed{Z_1 = 20}$

$$i = \frac{N_1}{N_2} = \frac{300}{120}$$

$$\boxed{i = 2.5}$$

$$i = \frac{Z_2}{Z_1} \quad ; \quad 2.5 = \frac{Z_2}{20}$$

$$\boxed{Z_2 = 50}$$

3. Calculation of pitch angles and virtual no. of teeth:

Pitch angle: $\tan\delta_2 = i$

8.39

$$\delta_2 = \tan^{-1}(2.5)$$

$$\boxed{\delta_2 = 68.19^\circ}$$

$$\delta_1 = 90 - \delta_2$$

$$= 90 - 68.19$$

$$\boxed{\delta_1 = 21.81^\circ}$$

Virtual no. of teeths:

8.39

$$Zv_1 = \frac{Z_1}{\cos\delta_1} \qquad Zv_2 = \frac{Z_2}{\cos\delta_2}$$

$$= \frac{20}{\cos 21.81^\circ} \qquad = \frac{50}{\cos 68.19^\circ}$$

$$Zv_1 = 21.54 \qquad Zv_2 = 134.6$$

$$\boxed{Zv_1 = 22} \qquad \boxed{Zv_2 = 135}$$

4. Calculation of tangential load (F_t)

$$F_t = \frac{P}{V} \times K_o$$

$$V = \frac{\pi d_1 N_1}{60} \qquad d_1 = m_t \times Z_1$$

$$= m_t \times 20$$

$$\boxed{d_1 = 20\, m_t}$$

$$= \frac{\pi \times 20 m_t \times 300}{60 \times 1000}$$

$$\boxed{V = 0.314\, m_t \quad m/s}$$

$P = 25\,kW$
$= 25 \times 10^3\,W$

$$F_t = \frac{25 \times 10^3}{0.314\, m_t} \times 1.5$$

$K_o = 1$	steady load
$= 1.25$	ligh shock
$= 1.5$	medium "
$= 2$	Heavy "

Assume medium shock condition
$\therefore K_o = 1.5$

$$\boxed{F_t = \frac{119426.75}{m_t}}$$

5. Initial dynamic load (F_d)

8.50

$$F_d = F_t \times C_v \qquad \text{for } v \le 5\,m/s$$

8.52

$$C_v = \frac{3.5 + \sqrt{v}}{3.5} \quad \text{for straight bevel gear}$$

$$\cancel{C_v = \frac{3.5\ \sqrt{0.314 m_t}}{3.5}} \quad \cancel{= \frac{3.5 + 0.56 m_t}{3.5}}$$

Initially assume $v = \underline{5\ m/s}$

$$C_v = \frac{3.5 + \sqrt{5}}{3.5} = \underline{\underline{1.64}}$$

$$F_d = \frac{119426.75}{m_t} \times 1.64$$

$$\boxed{F_d = \frac{195859.87}{m_t}}$$

6. Beam strength (F_s)

$\underline{\underline{8.52}}$ $$F_s = \frac{[\sigma_b]\, b\, Y_v \left(1 - \frac{b}{R}\right)}{P_d}$$

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

Initially assume face width $\boxed{b = 10\ m_t}$

$\underline{\underline{8.50}}$

P_d = Diametral pitch

P_c – circular "

$$P_d = \frac{\pi}{P_c}$$

$$\boxed{P_c = \frac{\pi d}{z} = \pi \times m_t}$$

$$P_d = \frac{\pi}{\frac{\pi d}{z}}$$

$$P_d = \frac{\pi z}{\pi d} = \frac{z}{d}$$

(or)

$\cancel{8.50}$ $$y P_c = \frac{Y_v}{P_d}$$

$$\boxed{\frac{Y_v}{P_d} = y \times \pi \times m_t}$$

$[\sigma_b] = 450\ N/mm^2$

Material is 15 Ni 2 Cr 1 Mo 15 Alloy steel

	$[\sigma_b]$
C.I	56 N/mm^2
Forged steel	112 N/mm^2
Cast "	140 "
Alloy "	450 "

$$y = 0.154 - \frac{0.912}{z_n} \quad \text{for } 20^\circ \text{ involute}$$

$$y = 0.154 - \frac{0.912}{22}$$

form factor $\boxed{y = 0.113}$

$\underline{\underline{8.38}}$ cone distance $R = 0.5 m_t \sqrt{z_1^2 + z_2^2}$

$$= 0.5\ m_t \sqrt{20^2 + 50^2}$$

$$\boxed{R = 26.93\ m_t}$$

Beam strength

$$F_s = 450 \times 10m_t \times 0.113 \times \pi \times m_t \times \left(1 - \frac{10m_t}{26.93m_t}\right)$$

$$= 1597.499\, m_t^2 \times \left(\frac{26.93m_t - 10m_t}{26.93\, m_t}\right)$$

$$= 1597.499\, m_t^2 \times (0.63)$$

$$\boxed{F_s = 1006.424\, m_t^2}$$

7. Transverse module (m_t)

Equate F_s & F_d $\quad F_s \geq F_d$

$$1006.424\, m_t^2 \geq \frac{195859.87}{m_t}$$

$$m_t = 5.795 \text{ mm} \simeq \underline{6 \text{ mm}}$$

Transverse module $\boxed{m_t = 6 \text{ mm}}$ std module $\underline{8.2}$

8. Calculation of b, d_1 & v

face width $b = 10 \times m_t = 10 \times 6 = \underline{60 \text{ mm}}$

Pitch circle dia of pinion $d_1 = m_t \times Z_1 = 6 \times 20 = \underline{120 \text{ mm}}$

Pitch line velocity $V = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.12 \times 300}{60}$

$$v = \underline{1.89 \text{ m/s}}$$

9. Recalculation of beam strength

$$F_s = 1006.424\, m_t^2 = 1006.424 \times 6^2$$

$$\boxed{F_s = 36231.264 \text{ N}}$$

10. Accurate dynamic load (F_d)

8.52

$$F_d = C_v \times N_{sf} \times K_m \times F_t$$

$$F_t = \frac{P}{V} = \frac{25\times10^3}{1.89} = 13227.5\ N$$

$$C_v = \frac{3.5 + \sqrt{1.89}}{3.5} = 1.393$$

Service factor $N_{sf} =$ 1 to 2

take $N_{sf} = 1.5$.

Load distribution factor $K_m = 1.1$ for Both gears straddle mounted

$$F_d = 1.393 \times 1.5 \times 1.1 \times 13227.5$$

$$\boxed{F_d = 30402.75\ N}$$

$$\boxed{F_s = 36231.264\ N}$$

$$F_s > F_d$$

$\therefore$ Design is safe.

11. Maximum wear load (F_w)

$$F_w = \frac{0.75 \times d_1 \times b \times Q' \times k_w}{\cos \delta_1}$$

8.51

$$Q' = \frac{2i'}{i' \pm 1} = \frac{2i'}{i'+1}$$

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

$$i' = \frac{Z_{v_2}}{Z_{v_1}} = \frac{135}{22}$$

$$\boxed{i' = 6.2}$$

$$Q' = \frac{2 \times 6.2}{6.2+1} = 1.722$$

$k_w = 2.553\ N/mm^2$ for 400 BHN & 20° FD

$$F_w = \frac{0.75 \times 120 \times 60 \times 1.722 \times 2.553}{\cos 21.81^\circ}$$

$$\boxed{F_w = 25570.12\,N}$$

$$\boxed{F_d = 30402.75\ N}$$

$$F_w < F_d$$

∴ Design is not safe

∴ Change the module m_t

(or)

Increase $\boxed{m_t = 8\ mm}$

Repeating from step 8 again

$$b = 10 \times m_t = 10 \times 8 = 80\ mm$$

$$d_1 = m_t \times Z_1 = 8 \times 20 = 160\ mm$$

$$V_1 = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.16 \times 300}{60} = 2.513\ m/s$$

Beam strength (F_s)

$F_s = 1006.424\, m t^2 = 1006.424 \times 8^2$

$\boxed{F_s = 64411.14N}$

Dynamic load (F_d)

$F_d = C_v \times N_{SF} \times K_m \times F_t$

$F_t = \frac{P}{V} = \frac{25\times10^3}{2.513} = 9948.3N$

$C_v = \frac{3.5 + \sqrt{2.513}}{3.5} = 1.453$

$N_{SF} = 1.5$ & $K_m = 1.1$

$F_d = 1.453 \times 1.5 \times 1.1 \times 9948.3$

$\boxed{F_d = 23850.55N}$

$\boxed{F_s > F_d}$ ∴ Design is safe

Wear strength (F_w)

$Q' = 1.722$

$K_w = 2.553$

$F_w = \frac{0.75 \times d_1 \times b \times Q' \times K_w}{\cos\delta_1}$

$= \frac{0.75 \times 160 \times 80 \times 1.722 \times 2.553}{\cos 21.81^\circ}$

$\boxed{F_w = 45457.99N}$ $\boxed{F_w > F_d}$ Design is safe

14. Basic dimensions: from 8.38 & 8.39

Methods are same as Problem No: 1, step 15

Worm gear drive:

5. Design a worm gear drive to transmit 20 HP from a worm at 1440 rpm to worm wheel. Assume the bronze is sand chill cast. the speed of the wheel should be 40 ± 2% rpm, initial sliding velocity can be assumed as 3 m/s and efficiency as 80%. Also find the cooling area required, if the temp rise is restricted to 40°C. & heat transfer co-efficient is 10 W/m² °C.

Given: $P = 20\ HP$

$N_1 = 1440\ rpm$

$N_2 = 40 \pm 2\ \%\ rpm$

$V = 3\ m/s$

$\eta = 80\%$

$i = \frac{N_1}{N_2} = \frac{1440}{40} = 36 \pm 2\%$

$t_o - t_a = 40°c$

$k_t = 10\ W/m^2 °c$

Find: I. Design a worm gear

II. Cooling area required

Soln: I. Design of worm gear.

1. Selection of material:

Worm – steel

Worm Wheel – Bronze (sand chill cast) is given

8.54

2. Initial design torque $[M_t]$

8.44

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

$K = 1$ & $K_d = 1$ $\boxed{k \cdot K_d = 1}$

(or) use 8.44 $\boxed{M_t = 71620 \frac{Hp\ i\eta}{n}}$

$n = N_1 = 1440 rpm$

$i = 36$

$\eta = 0.8$

$\boxed{N = N_2}$

$P = \frac{2\pi N M_t}{60}$

$15 \times 10^3 = \frac{2 \times \pi \times 40 \times M_t}{60}$

$M_t = 3580.99\ N.m$

$\boxed{M_t = 3580.99 \times 10^3\ N.mm}$

Power $P = 20\ HP$

$1\ HP = 0.7457\ KW$

$\therefore 20\ Hp = 0.7457 \times 20$

$P = 14.91 = 15\ KW$

(22)

Initial design torque

$[M_t] = 3580.99 \times 10^3 \times 1$

$\boxed{[M_t] = 3580.99 \times 10^3 \text{ N.mm}}$

3. Selection of Z_1 & Z_2

8.46 for $\eta = 80\% = 0.8$

$\boxed{Z_1 = 2}$ $i = \frac{Z_2}{Z_1}$; $36 = \frac{Z_2}{2}$

$\boxed{Z_2 = 72}$

4. Selection of $[\sigma_b]$ & $[\sigma_c]$

1.40 for Bronze → chill cast → tensile strength $= 240 \text{ N/mm}^2$

8.45 " " $\sigma_u < 39 \text{ kgf/mm}^2$

$\boxed{\sigma_u < 390 \text{ N/mm}^2}$ $\sigma_u = 240 \text{ N/mm}^2$

table 33 for chill cast & $\sigma_u < 39 \text{ kgf/mm}^2$

$[\sigma_b] = 550 \text{ kgf/cm}^2 = 5500 \text{ N/cm}^2$

$\boxed{[\sigma_b] = 55 \text{ N/mm}^2}$

8.45 table 32 for $V = 3 \text{ m/s}$ sliding velocity v is given $V = 3 \text{ m/s}$

Bronze

$[\sigma_c] = 1590 \text{ kgf/cm}^2$

$= 15900 \text{ N/cm}^2$

$\boxed{[\sigma_c] = 159 \text{ N/mm}^2}$

5. centre distance (a)

8.44 $$a=\left(\frac{Z_2}{q}+1\right)\sqrt[3]{\left[\frac{540}{\left(\frac{Z_2}{q}\right)[\sigma_c]}\right]^2\frac{[M_t]}{10}}$$

$Z = Z_2$ no.of teeth on wheel
$Z = Z_1$ no.of teeth on worm

Assume $q = 11$ Diameter factor

$Z_2 = 72 \quad [\sigma_c] = 159\,N/mm^2$

$[M_t] = 3580.99 \times 10^3\ N.mm$

$$a=\left(\frac{72}{11}+1\right)\sqrt[3]{\left[\frac{540}{\left(72/11\right)\times 159}\right]^2\times\frac{3580.99\times 10^3}{10}}$$

$$\boxed{a = 345.99\ mm = 346\ mm}$$

6. Axial module (m_x)

8.43 $$m_x=\frac{2a}{(q+Z_2)} = \frac{2\times 346}{11+72}$$

$m_x = 8.4\,mm$

8.2 std $\Rightarrow \boxed{m_x = 10\ mm}$

7. Revise (a)

8.43 $$a = 0.5 m_x (q+Z_2)$$

$$= 0.5\times 10\,(11+72)$$

$$\boxed{a = 415\ mm}$$

8. Calculation of $d, v, \gamma, \& V_s$

$d_1 = q \times m_x = 11 \times 10 = \underline{110\ mm}$ $\boxed{q = \frac{d_1}{m_x}}$

8.43 $d_2 = Z_2 \times m_x = 72 \times 10 = \underline{720 mm}$

$Z = Z_2$

$Z = Z_1$

$V_1 = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.11 \times 1440}{60} = \underline{8.29\ m/s}$

$\boxed{V_2 = \frac{\pi d_2 N_2}{60} = \frac{\pi \times 0.72 \times 40}{60} \quad V_2 = 1.51 m/s}$

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

$\gamma = \tan^{-1}\left(\frac{2}{11}\right)$

$\boxed{\gamma = 10.3^\circ}$

8.44 Sliding Velocity $V_s = \frac{V_1}{\cos\gamma} = \frac{8.29}{\cos 10.3^\circ}$

$\boxed{V_s = 8.43 m/s}$

8.43 m/s not in table 32

∴ Take $[\sigma_c]$ for 8 m/s

9. Recalculation of $[\sigma_c]$

8.45 for $V_s = 8.43 m/s$ $[\sigma_c] = 1490\ kgf/cm^2$

$\boxed{[\sigma_c] = 149\ N/mm^2}$

10. Revise $[M_t]$

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

8.44 $M_t = 3580.99 \times 10^3\ N.mm$

$K = 1$ for $V_2 = 1.51 m/s$

$K_d = 1$

$[M_t] = 3580.99 \times 10^3 \times 1$

Revised $\boxed{[M_t] = 3580.99 \times 10^3}$

11. check for bending:

8.44 $$\sigma_b = \frac{1.9[M_t]}{m_x^3 \cdot q \cdot Z_2 \cdot y_{v_2}} \le [\sigma_b]$$

$$Z_{v_2} = \frac{Z_2}{\cos^3 \gamma} = \frac{72}{\cos^3 10.3^\circ} = 75.5$$

$$\boxed{Z_{v_2} = 76}$$

for $Z_{v_2} = 76$ or 75 $\boxed{y_{v_2} = 0.496}$

8.53 20° F.D

$$\sigma_b = \frac{1.9 \times 3580.99 \times 10^3}{10^3 \times 11 \times 72 \times 0.496}$$

$$\sigma_b = 17.32 \ N/mm^2 < [\sigma_b] \ 55 \ N/mm^2$$

∴ Design is safe

12. check for wear:

8.44 $$\sigma_c = \frac{540}{(Z_2/q)} \sqrt{\left[\frac{(Z_2/q)+1}{a}\right]^3 \times \frac{[M_t]}{10}} \le [\sigma_c]$$

$$= \frac{540}{(72/11)} \sqrt{\left[\frac{(72/11)+1}{415}\right]^3 \times \frac{3580.99 \times 10^3}{10}}$$

$$\sigma_c = 121.04 \ N/mm^2 < [\sigma_c] \ 149 \ N/mm^2$$

Design is safe

13. check for efficiency

8.49 $$\eta = \frac{\tan \gamma}{\tan(\gamma + \rho)} \times 0.95$$

$\rho = \tan^{-1}(\mu) = \tan^{-1}(0.02)$ for $v_s = 8.43$

Take $\boxed{\mu = 0.02}$

8.49 $\boxed{\mu = \tan \rho}$ $\boxed{\rho = 1.72^\circ}$

$$\eta_{actual} = 0.95 \times \frac{\tan 10.3^\circ}{\tan(10.3 + 1.72)}$$

$\eta_{actual} = 0.8108 = 81.08\% >$ Desired η 80%.

$\therefore$ Design is safe

14. Basic dimensions: i. Axial module $m_x = 10$ mm

ii. centre distance $a = 415$ mm

8.43 table 39

for $z_1 = 2$ iii. Length of the worm $L = (11 + 0.06\, z_2)\, 10$ 8.48

$L = (11 + (0.06 \times 72))\, 10 = 153.2$ mm

iv. $z_1 = 2$ & $z_2 = 72$

v. Height factor $f_0 = 1$

vi. face width $b = 0.75 d_1 \to \boxed{\text{for } z_1 = 2}$

8.48 table 38

$= 0.75 \times 110 = 82.5$ mm

vii. Bottom clearance $c = 0.25\, m_x = 2.5$ mm

viii. Pitch circle dia $d_1 = 110$ mm & $d_2 = 720$ mm

8.43

ix. Tip dia $d_{a1} = d_1 + 2f_0 \cdot m_x$

$$= 110 + (2 \times 1 \times 10)$$

$$= \underline{130\ mm}$$

$$d_{a2} = (Z_2 + 2f_0)\, m_x$$

$$= (72 + (2 \times 1))\, 10$$

$$= \underline{740\ mm}$$

x. Root dia $d_{f1} = d_1 - 2f_0\, m_x - 2C$

$$= 110 - (2 \times 1 \times 10) - (2 \times 2.5)$$

$$d_{f1} = \underline{85\ mm}$$

$$d_{f2} = (Z_2 - 2f_0)\, m_x - 2C$$

$$= (72 - 2 \times 1)\, 10 - (2 \times 2.5)$$

$$d_{f2} = \underline{695\ mm}$$

II. Cooling area required 'A'

8.52

$$(1 - \eta) \times \text{Input power} = k_t \cdot A\,(t_o - t_a)$$

temp rise $t_o - t_a = 40°c$

calculated η or $\eta_{actual} = 0.8108$

20 Hp ⇒ Input power $= 15\ kw = 15 \times 10^3\ W$

$\boxed{1\ HP = 0.7457\ kw}$

$k_t = 10\ W/m^2\,°c$

$$(1 - 0.8108) \times 15 \times 10^3 = 10 \times A\ (40)$$

$$A = \underline{7.095\ m^2}$$

6. A steel worm running at 240 rpm receives 1.5kW from its shaft. The speed reduction is 10:1. Design the drive so as to have an efficiency of 80.1. Also determine the cooling area required, if the temperature rise is restricted to 45°c. Take overall heat transfer co-efficient as 10 W/m² °c.

Given: $N_1 = 240$ rpm　　$\eta = 80.1 = 0.801$

$P = 1.5$ kW　　$t_0 - t_a = 45°c$　　$\frac{N_1}{N_2} = i$

$i = 10:1 = 10$　　$K_t = 10$ W/m² °c.　　$\frac{240}{N_2} = 10$

$\boxed{N_2 = 24 \text{ rpm}}$

Find:

I. Design a Worm gear drive

II. Cooling area required

Soln:

I. Design of worm gear:

1. Selection of material

Worm – Steel

P.B. Worm Wheel – Bronze (Sand cast)

Remaining Procedures [Step (2 to 15)] are same as

$\boxed{\text{Problem No: 5}}$

7. Design a worm gear drive to transmit 22.5 kW at a worm speed of 1440 rpm. Velocity ratio is 24:1. An efficiency of atleast 85% is desired. The temperature raise should be restricted to 40°c. Determine the required cooling area.

Given:

$P = 22.5\ kW$

$N_1 = 1440\ rpm$

$t_o - t_a = 40^\circ c$

$i = 24:1 = 24$

$\eta = 85\% = 0.85$

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

$\frac{1440}{N_2} = 24$

$\boxed{N_2 = 60\ rpm}$

$\boxed{\text{Take } K_t = 10\ W/m^2 {}^\circ C}$ not given

find:

I. Design a worm gear drive

II. Cooling area required.

Design procedures are same as $\boxed{\text{Problem No : 5}}$

GEAR BOX

1. Design a 9 speed gear box for a milling machine with speeds ranging from 56 to 900 rpm. The output speed is 720 rpm. Make a neat sketch of the gear box. Indicate the number of teeth on all the gears and their speeds. Assuming the gears and shafts are made of C45, calculate module, centre distance and diameter of the spindle.

Given:

$n = 9$ Material = C45 steel

$N_{max} = 900$ rpm

$N_{min} = 56$ rpm

$N_{output} = 720$ rpm

Find: Design a 9 speed gear box

Soln: 1. Selection of spindle speeds

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

$$\frac{900}{56} = \phi^{9-1} \quad ; \quad 16.0714 = \phi^{8}$$

$$\boxed{\phi = 1.415}$$

$\phi = 1.415$ is not a standard step ratio

$\phi = 1.12 \times 1.12 \times 1.12 = 1.41$

7.20

$\boxed{\phi = 1.12}$ (skip 2 speeds)

R_{20} series $\phi = 1.12$

$\underline{\underline{7.20}}$

Select Nine speeds

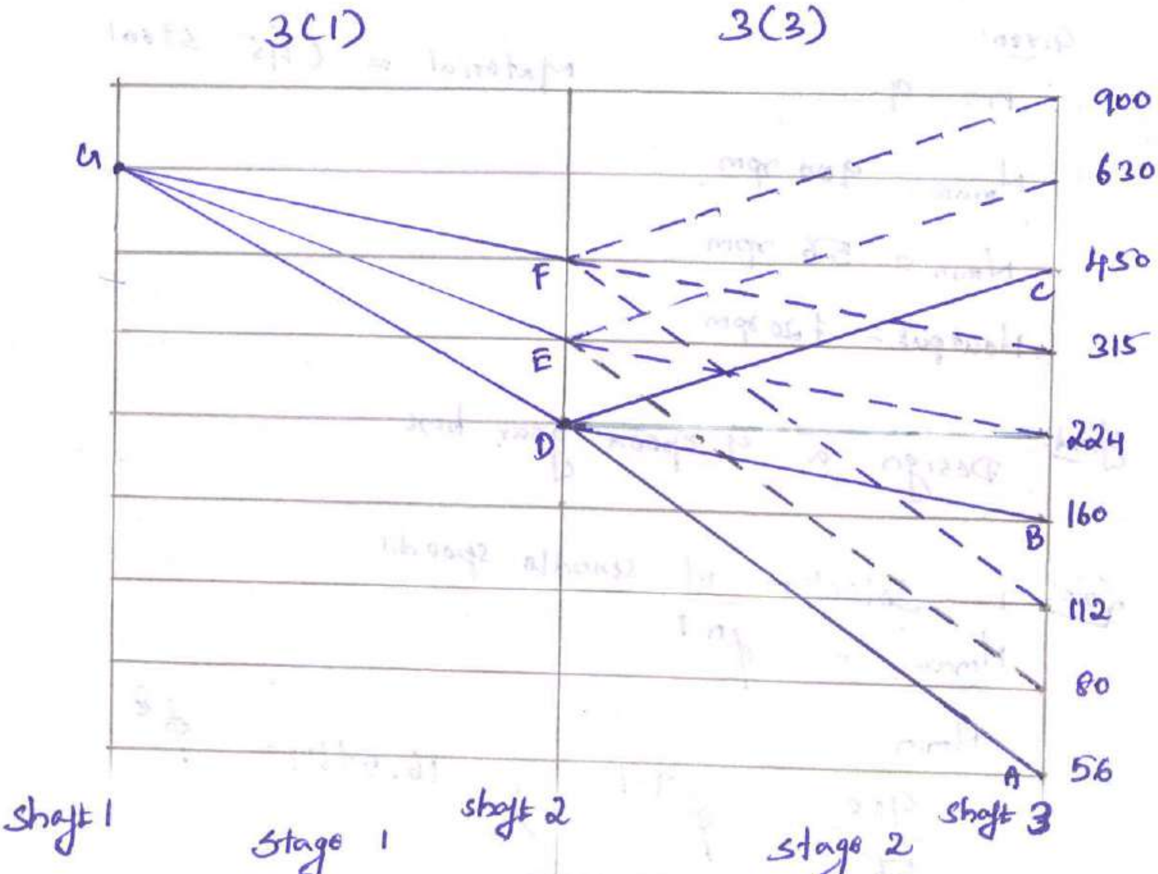
56 rpm, 80, 112, 160, 224, 315, 450, 630, 900 rpm

$\uparrow$ N_{min} ... $\uparrow$ N_{max}

2. Construct the ray diagram:

Structural formula

for 9 speeds 3(1) · 3(3)



$$\frac{N_{min}}{N_{input}} \geq \frac{1}{4} \text{ (or) } 0.25$$

$$\frac{N_{max}}{N_{input}} \leq 2$$

Stage II

224 → 450 max, 160, 56 min

$$\frac{N_{min}}{N_{input}} \geq 0.25 \; ; \; \frac{56}{224} = 0.25 = 0.25$$

$$\frac{160}{224} = 0.714 > 0.25$$

$$\frac{N_{max}}{N_{input}} \leq 2 \quad \frac{450}{224} = 2$$

Stage I

630 → 450 max, 315, 224 min

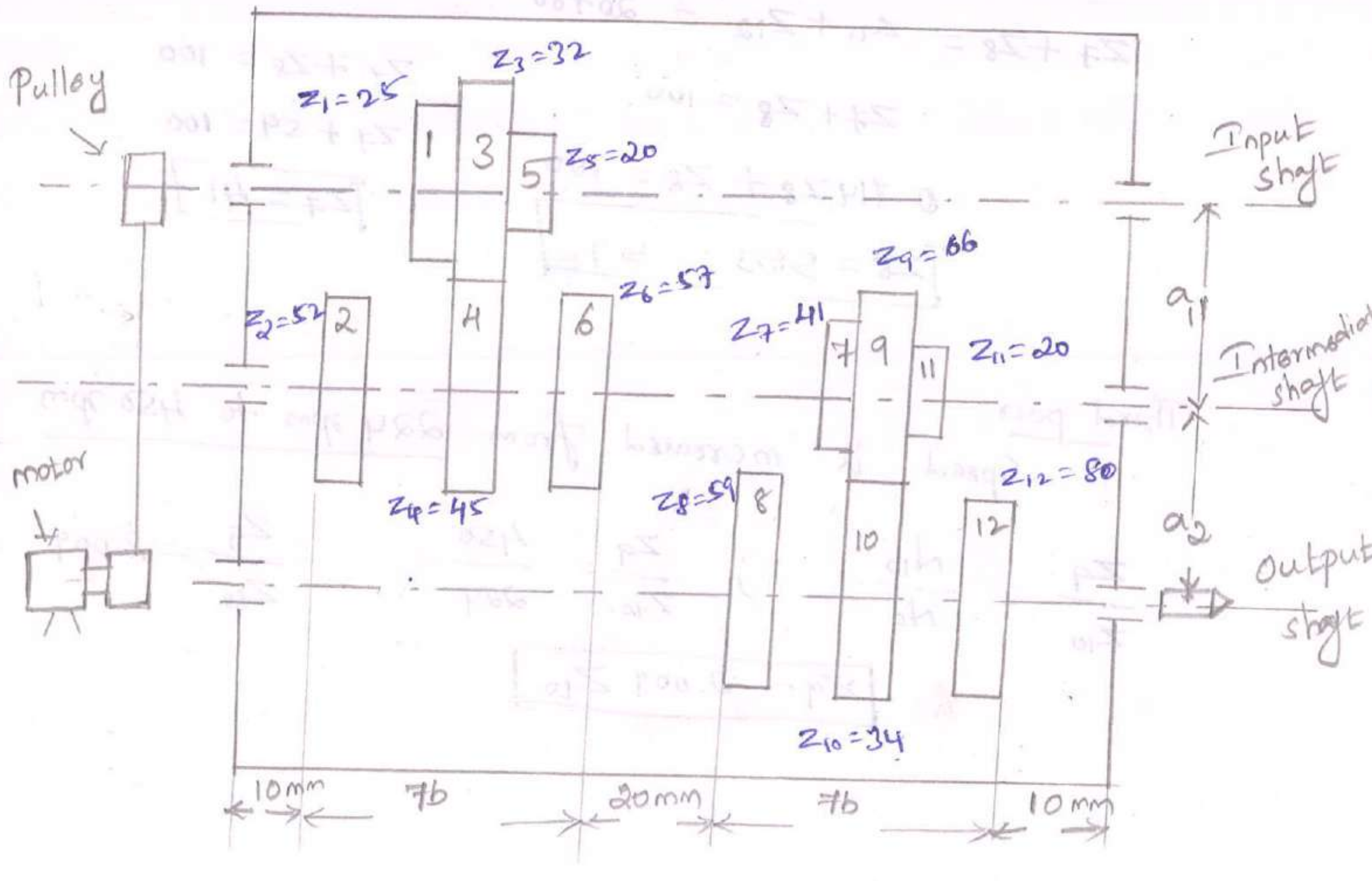
$$\frac{N_{min}}{N_{input}} \geq 0.25 \quad \frac{224}{630} = 0.36 > 0.25$$

$$\frac{315}{630} = 0.5 > 0.25$$

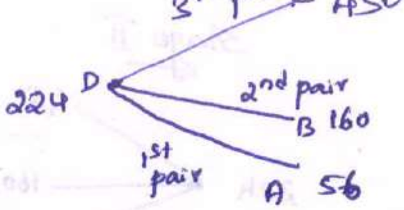
$$\frac{N_{max}}{N_{input}} \leq 2 \quad \frac{450}{630} = 0.714 < 2$$

∴ Ratio requirements are satisfied.

3. Kinematic arrangement:



4. Number of teeth on all gears:



Stage 2:

first pair:

Speed is reduced from 224 rpm to 56 rpm

Assume $\boxed{Z_{11} = 20}$

$$\frac{Z_{11}}{Z_{12}} = \frac{N_{12}}{N_{11}} \qquad \frac{20}{Z_{12}} = \frac{56}{224}$$

$$\boxed{Z_{12} = 80}$$

Second pair:

speed is reduced from 224 rpm to 160 rpm

$$\frac{Z_7}{Z_8} = \frac{N_8}{N_7} \quad ; \quad \frac{Z_7}{Z_8} = \frac{160}{224}$$

$$\frac{Z_7}{Z_8} = 0.714 \qquad \boxed{Z_7 = 0.714 Z_8}$$

$$Z_7 + Z_8 = Z_{11} + Z_{12} = 20 + 80$$

$$Z_7 + Z_8 = 100$$

$$0.714 Z_8 + Z_8 = 100$$

$$\boxed{Z_8 = 58.3 = 59}$$

$$Z_7 + Z_8 = 100$$

$$Z_7 + 59 = 100$$

$$\boxed{Z_7 = 41}$$

Third pair:

Speed is increased from 224 rpm to 450 rpm

$$\frac{Z_9}{Z_{10}} = \frac{N_{10}}{N_9} \quad ; \quad \frac{Z_9}{Z_{10}} = \frac{450}{224} \qquad \frac{Z_9}{Z_{10}} = 2.009$$

$$\boxed{Z_9 = 2.009 Z_{10}}$$

$$Z_9 + Z_{10} = Z_{11} + Z_{12} = Z_7 + Z_8$$

$$2.009\, Z_{10} + Z_{10} = 100$$

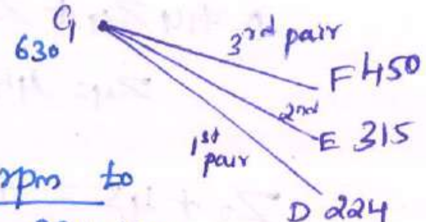
$$\boxed{Z_{10} = 33.23 \simeq 34}$$

$$Z_9 + Z_{10} = 100 \quad , \quad Z_9 + 34 = 100$$

$$\boxed{Z_9 = 66}$$

Stage 1:

first pair:



speed is reduced from 630 rpm to 224 rpm

Assume $\boxed{Z_5 = 20}$

$$\frac{Z_5}{Z_6} = \frac{N_6}{N_5} \qquad \frac{20}{Z_6} = \frac{224}{630} \qquad Z_6 = 56.3$$

$$\boxed{Z_6 = 57}$$

$$\boxed{Z_5 + Z_6 = 20 + 57 = 77}$$

Second pair:

Speed is reduced from 630 rpm to 315 rpm

$$\frac{Z_1}{Z_2} = \frac{N_2}{N_1} \; ; \quad \frac{Z_1}{Z_2} = \frac{315}{630} \qquad \frac{Z_1}{Z_2} = 0.5$$

$$Z_1 = 0.5\, Z_2$$

$$\boxed{Z_1 + Z_2 = Z_5 + Z_6} \qquad 0.5 Z_2 + Z_2 = 77$$

$$Z_2 = 51.33 \qquad Z_1 + Z_2 = 77$$

$$\boxed{Z_2 = 52} \qquad Z_1 + 52 = 77$$

$$\boxed{Z_1 = 25}$$

Third pair:

Speed is reduced from 630 rpm to 450 rpm

$$\frac{Z_3}{Z_4} = \frac{N_4}{N_3} \qquad \frac{Z_3}{Z_4} = \frac{450}{630} = 0.714$$

$$Z_3 = 0.714 Z_4$$

$$\boxed{Z_3 + Z_4 = Z_1 + Z_2 = Z_5 + Z_6}$$

$$Z_3 + Z_4 = 77$$

$$0.714 Z_4 + Z_4 = 77$$

$$Z_4 = 44.9 \qquad \boxed{Z_4 = 45}$$

$$Z_3 + 45 = 77$$

$$\boxed{Z_3 = 32}$$

5. Material Selection: C45 steel is given

6. Calculation of module:

$$\boxed{F_t \text{ or } P_c}$$

$$\text{module } m = \sqrt{F_t / \psi_m \times M}$$

8.32

$$F_t = \frac{T}{r} = \frac{T_{12}}{r_{12}} \qquad \text{Power } P = \frac{2\pi N T}{60}$$

N - lowest speed

$N = 56$ rpm

in 12th gear

$$\boxed{\therefore T = T_{12}}$$

Assume Power $\boxed{P = 2 \text{ kW}}$

$$2 \times 10^3 = \frac{2\pi \times 56 \times T_{12}}{60}$$

$$T_{12} = 341.05 \text{ N.m} = 341.05 \times 10^3 \text{ N.mm}$$

$$F_t = \frac{341.05 \times 10^3 \times 2}{m \times 80}$$

$$d_{12} = m \cdot Z_{12}$$

$$\boxed{r_{12} = \frac{m \cdot Z_{12}}{2}}$$

$$Z_{12} = 80$$

$$\boxed{F_t = F_{t(2} = \frac{8526.16}{m}}$$

$\varphi_m = b/m = 10$

Material constant $M = 30$ for C45

	Material constant M
C45	30
15Ni 2Cr 1Mo15	80
40Ni 2Cr Mo28	100

module $m = \sqrt{\dfrac{\frac{8526.16}{m}}{10 \times 30}}$ $m = 5.3$ mm

$\boxed{m = 6 \text{ mm}}$ ... std m from 8.2

7. Calculation of centre distance:

$a_1 = \left[\frac{z_1 + z_2}{2}\right] m = \left[\frac{25+52}{2}\right] \times 6$

$\boxed{a_1 = 231 \text{ mm}}$

$a_2 = \left[\frac{z_7 + z_8}{2}\right] m = \left[\frac{41+59}{2}\right] \times 6$

$\boxed{a_2 = 300 \text{ mm}}$

8. Calculation of face width

$b = \varphi \times m = 10 \times 6 = \underline{60 \text{ mm}}$

9. Calculation of length of shaft:

$L = 25 + 10 + 76 + 20 + 76 + 10 + 25$

$b = 60$ mm

$\boxed{L = 930 \text{ mm}}$

(8)

10. Design of shafts:

i. Design of spindle (or) Output shaft

$$T_{eq} = \frac{\pi}{16} \times \tau \times d_s^3$$

$\boxed{\tau = 30\ N/mm^2 \text{ for C45}}$

$$T_{eq} = \sqrt{M^2 + T_{12}^2}$$

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

$$F_n = \frac{F_t}{\cos\alpha}\ ;\ F_t = \frac{8526.16}{m} = \frac{8526.16}{6} = 1421.03\ N$$

$\boxed{\alpha = 20^\circ}$ Assume

$$F_n = \frac{1421.03}{\cos 20^\circ}$$

$$\boxed{F_n = 1512.2\ N}$$

$$M = \frac{1512.2 \times 930}{4} = 351592.3\ N.mm$$

$$T = T_{12} = 341.05 \times 10^3\ N.mm$$

$$T_{eq} = \sqrt{(351592.3)^2 + (341.05 \times 10^3)^2}$$

$$\boxed{T_{eq} = 497226.61\ N.mm}$$

$$497226.61 = \frac{\pi}{16} \times 30 \times d_s^3$$

$$\boxed{d_s = 43.9\ mm = 44\ mm} \text{ or } 45\ mm$$

ii: Design of other shafts:

a. Diameter of shaft 1 Input speed = 630 rpm

$$P = \frac{2\pi N T}{60} \qquad 2\times10^3 = \frac{2\pi\times 630\times T}{60}$$

$$T = 30.32\ N.m = \underline{\underline{30.32\times10^3\ N.mm}}$$

$$T = 0.2\times d_{s_1}^3\times \tau$$

$$30.32\times10^3 = 0.2\times d_{s_1}^3\times 30$$

$$\boxed{d_{s_1} = 17.16mm = 18mm}$$

b. Diameter of shaft 2: $P = \frac{2\pi NT}{60}$ Input speed = 224 rpm

$$2\times10^3 = \frac{2\pi\times 224\times T}{60}$$

$$T = \underline{\underline{85.26\times10^3\ N.mm}}$$

$$T = 0.2\times d_{s_2}^3\times\tau$$

$$85.26\times10^3 = 0.2\times d_{s_2}^3\times 30$$

$$\boxed{d_{s_2} = 24.2\ mm = 25mm}$$

2. A nine speed gear box, used as a head stock gear box of a turret lathe, is to provide a speed range of 180 rpm to 1800 rpm. Using standard step ratio, draw the speed diagram and the kinematic layout. Also find and fix the number of teeth on all gears.

Given: $n = 9$

$N_{min} = 180$ rpm

$N_{max} = 1800$ rpm

find: 1. Draw the speed diagram & kinematic layout

2. Number of teeth on all gears.

Soln: 1. Speed diagram & kinematic layout:

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \qquad \frac{1800}{180} = \phi^{9-1}$$

$$\boxed{\phi = 1.334}$$

$\phi = 1.334$ is not a standard step ratio

$\phi = 1.06 \times 1.06 \times 1.06 \times 1.06 \times 1.06 = 1.338$

7.20

$\therefore \boxed{\phi = 1.06}$ is a standard step ratio

(Skip 4 speeds) R40 Series

Spindle speeds:

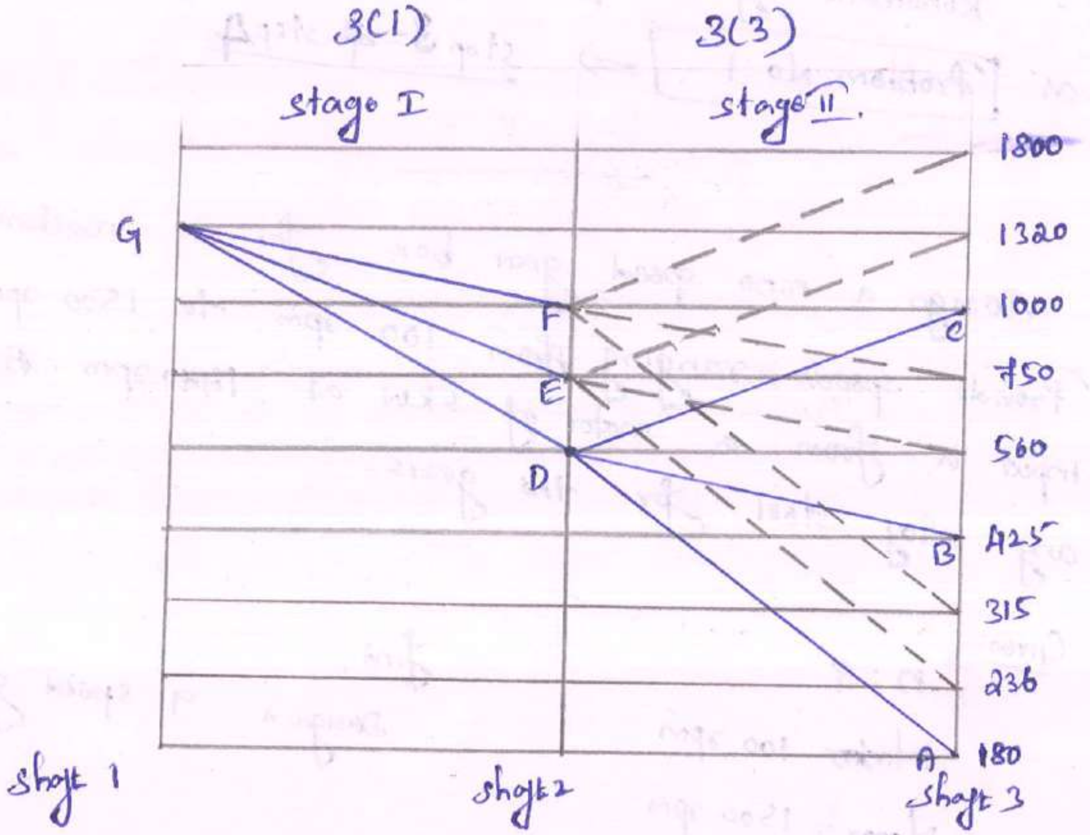
180 rpm, 236, 315, 425, 560, 750, 1000, 1320, 1800 rpm

$\uparrow N_{min}$ $\qquad \uparrow N_{max}$

(11)

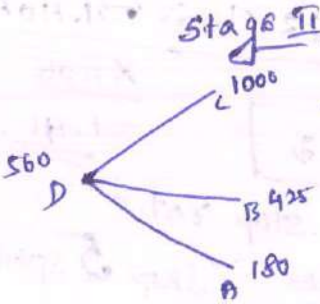
Speed (or) Ray diagram:

Structural formula:
3(1). 3(3).



$$\frac{N_{min}}{N_{input}} \geq 0.25 \qquad \frac{N_{max}}{N_{input}} \leq 2$$

Stage II

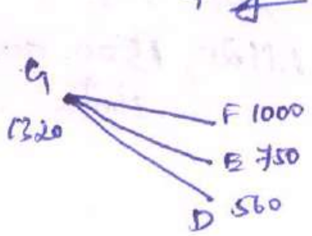


$$\frac{N_{min}}{N_{input}} = \frac{180}{560} = 0.32 > 0.25$$

$$= \frac{425}{560} = 0.75 > 0.25$$

$$\frac{N_{max}}{N_{input}} = \frac{1000}{560} = 1.79 < 2$$

Stage I



$$\frac{N_{min}}{N_{input}} = \frac{560}{1320} = 0.42 > 0.25$$

$$= \frac{750}{1320} = 0.6 > 0.25$$

$$\frac{N_{max}}{N_{input}} = \frac{1000}{1320} = 0.78 < 2$$

∴ Ratio requirements are satisfied

Kinematic layout:

2. Number of teeth on all gears

Procedure for kinematic layout & No. of teeth calculations are same as Problem No: 1 ⟹ step 3 & step 4

---

3. Design a nine speed gear box for a machine to provide speeds ranging from 100 rpm to 1500 rpm. The input is from a motor of 5kW at 1440 rpm. Assume any alloy steel for the gears.

Given:

$n = 9$

$N_{min} = 100$ rpm

$N_{max} = 1500$ rpm

$P = 5$ kW

N_{input} $N = 1440$ rpm

Find:

Design a 9 speed gear box

Soln:

1. Selection of spindle speeds

$$\frac{N_{max}}{N_{input}} = \phi^{n-1}$$

$$\frac{1500}{100} = \phi^{9-1}$$

$$\boxed{\phi = 1.403}$$

is not a std step ratio

7.20

$\phi = 1.06 \times 1.06 \times 1.06 \times 1.06 \times 1.06 \times 1.06 = 1.41$

$\therefore \boxed{\phi = 1.06}$

std step ratio

R40 series (Skip 5 speed)

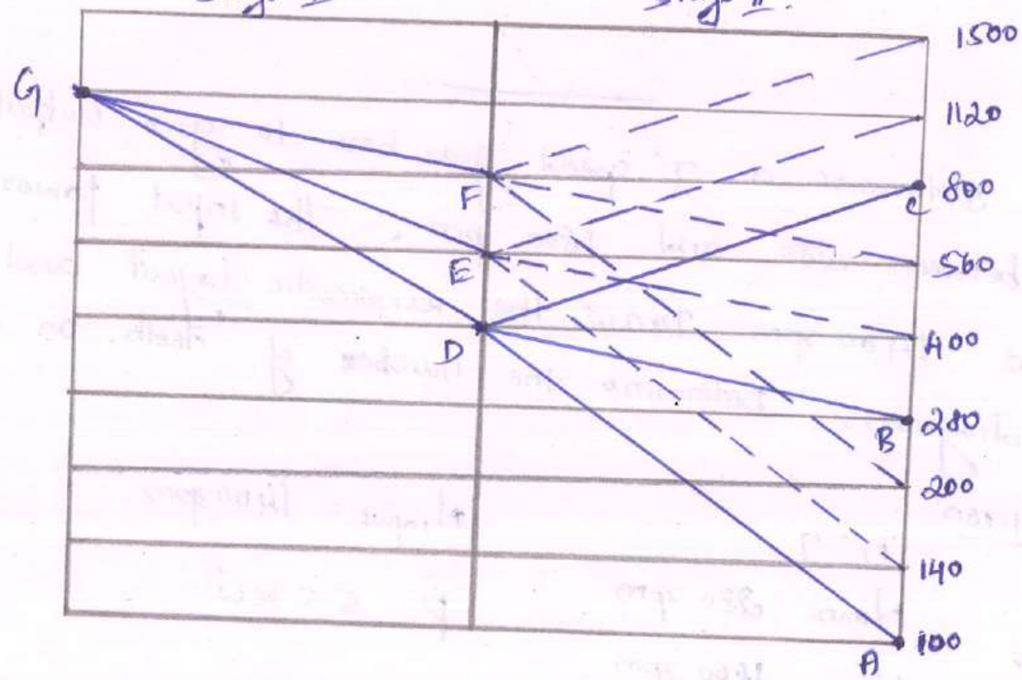
100 rpm, 140, 200, 280, 400, 560, 800, 1120, 1500 rpm

N_{min} ... N_{max}

2. construct the ray diagram:

Structural formula $= 3(1) \cdot 3(3)$.

3(1) Stage I 3(3) Stage II.



$$\frac{N_{min}}{N_{input}} \geq 0.25 \qquad \frac{N_{max}}{N_{input}} \leq 2$$

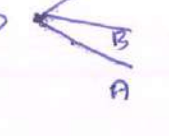
Stage II

$$\frac{N_{min}}{N_{input}} = \frac{100}{400} = 0.25 \qquad \frac{N_{max}}{N_{input}} = \frac{800}{400} = 2$$

$$= \frac{280}{400} = 0.7 > 0.25$$

Stage I

$$\frac{N_{min}}{N_{input}} = \frac{400}{1120} = 0.36 > 0.25 \qquad \frac{N_{max}}{N_{input}} = \frac{800}{1120} = 0.7 < 2$$



$$= \frac{560}{1120} = 0.5 > 0.25$$

$\therefore$ Ratio requirements are satisfied

Remaining Procedures (step 2 to 10) are same as

Problem No: 1

Assume Material = C45

Power p = 5kw is given

---

4. Determine a 9 speed gear box to give output speeds between 280 and 1800 rpm. The input power is 5.5kw at 1400 rpm. Draw the kinematic layout and speed diagram. Determine the number of teeth on all gears.

Given:

$n = 9$

$N_{min} = 280$ rpm

$N_{max} = 1800$ rpm

$N_{input} = 1400$ rpm

$p = 5.5$ kw.

Find:

1. Draw the speed diagram & kinematic layout.
2. No. of teeth on all gears.

Soln: 1. Speed diagram: (first select spindle speeds)

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \qquad \frac{1800}{280} = \phi^{9-1}$$

$\boxed{\phi = 1.262}$ is not a std step ratio

$\phi = 1.06 \times 1.06 \times 1.06 \times 1.06 = 1.262$

7.20 $\boxed{\phi = 1.06}$ Std step ratio

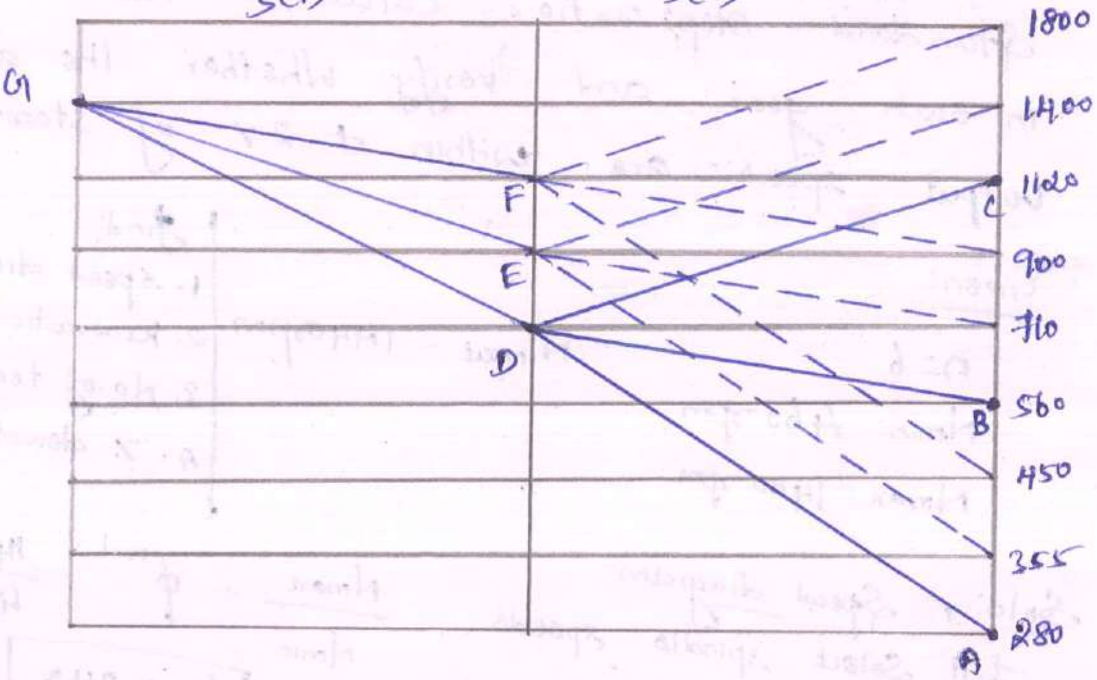
R40 series 280 rpm, 355, 450, 560, 710, 900, 1120, 1400, 1800 rpm

Spindle speeds $\rightarrow$ $\uparrow N_{min}$ $\qquad \uparrow N_{max}$

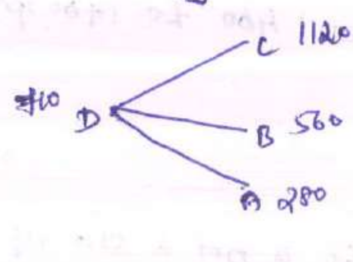
Speed (or) Ray diagram:

Structural formula $= 3(1).\ 3(3)$.

Stage I $3(1)$ Stage II $3(3)$



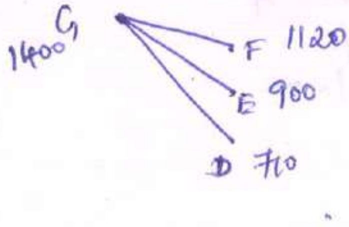
Stage II



$$\frac{N_{min}}{N_{input}} = \frac{280}{710} = 0.39 > 0.25 \qquad \frac{N_{max}}{N_{input}} = \frac{1120}{710} = 1.57 < 2$$

$$= \frac{560}{710} = 0.8 > 0.25$$

Stage I



$$\frac{N_{min}}{N_{input}} = \frac{710}{1400} = 0.51 > 0.25 \qquad \frac{N_{max}}{N_{input}} = \frac{1120}{1400} = 0.8 < 2$$

$$= \frac{900}{1400} = 0.64 > 0.25$$

$\therefore$ Ratio requirements are Satisfied

Procedure for kinematic layout and 2. Number of teeth calculations are same as

$\boxed{\text{Problem No: 1}} \Rightarrow$ Step 3 & step 4

5. Sketch the arrangements of a six speed gear box. The minimum and maximum speeds required are around 460 and 1400 rpm. Drive speed is 1440 rpm. Construct speed diagram of the gear box and obtain various reduction ratios. Use standard output speeds and standard step ratio. Calculate number of teeth in each gear and verify whether the actual output speeds are within ± 2% of standard speeds.

Given:

$n = 6$ $\quad N_{input} = 1440 rpm$

$N_{min} = 460$ rpm

$N_{max} = 1400$ rpm

Find:
1. Speed diagram
2. Kinematic arrangement
3. No. of teeths
4. % deviation

Soln: 1. Speed diagram

first select spindle speeds

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \qquad \frac{1400}{460} = \phi^{6-1}$$

$$\boxed{\phi = 1.249} = 1.25$$

$\phi = 1.25$ R10 series can't take 460 to 1400 rpm

$\phi = 1.12 \times 1.12 = 1.254$

$\boxed{\phi = 1.12}$ R20 Series

460 is not a std speed in R20

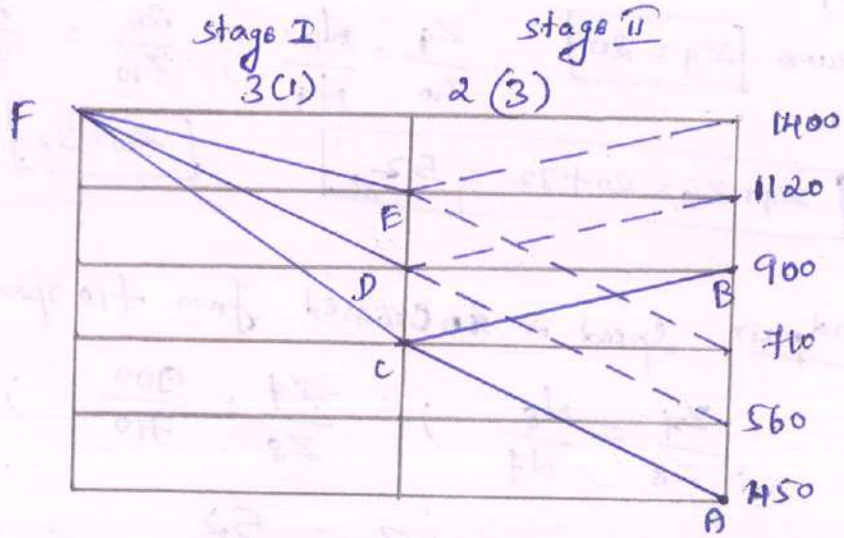
450 rpm, 560, 710, 900, 1120, 1400 rpm

$\uparrow$ N_{min} $\qquad$ $\uparrow$ N_{max}

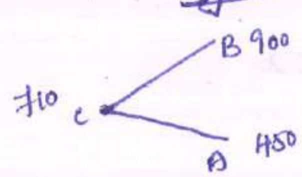
3×2

Speed or ray diagram:

Structural formula $3(1).\ 2(3)$



Stage II



$\frac{N_{min}}{N_{input}} = \frac{450}{710} = 0.63 > 0.25$ $\qquad \frac{N_{max}}{N_{input}} = \frac{900}{710} = 1.27 < 2$

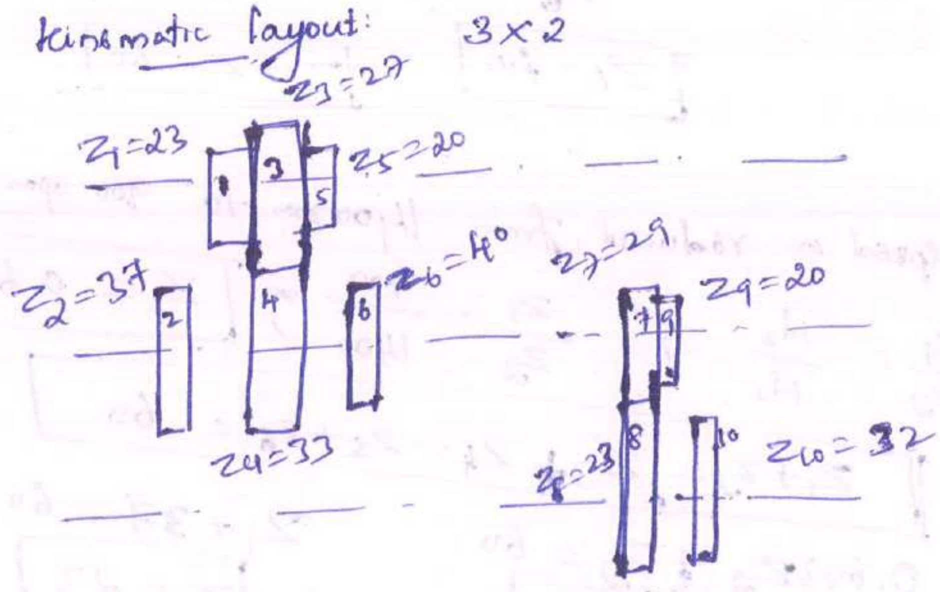
Stage I.

$\frac{N_{min}}{N_{input}} = \frac{710}{1400} = 0.51 > 0.25$ $\qquad \frac{N_{max}}{N_{input}} = \frac{1120}{1400} = 0.8 < 2$

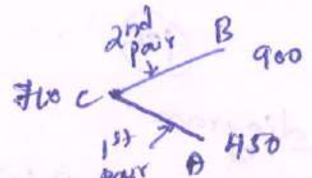
$= \frac{900}{1400} = 0.64 > 0.25$

$\therefore$ Ratio requirements are satisfied

2. Kinematic layout: 3×2



3. Number of teeth:



Stage II — 1st pair

Speed is reduced from 710 rpm to 450 rpm

Assume $\boxed{Z_9 = 20}$ $\quad \frac{Z_9}{Z_{10}} = \frac{N_{10}}{N_9}$; $\frac{20}{Z_{10}} = \frac{450}{710}$

$\boxed{Z_{10} = 32}$

$\boxed{Z_9 + Z_{10} = 20 + 32 = 52}$

2nd pair — Speed is increased from 710 rpm to 900 rpm

$\frac{Z_7}{Z_8} = \frac{N_8}{N_7}$; $\frac{Z_7}{Z_8} = \frac{900}{710}$; $\boxed{Z_7 = 1.27\, Z_8}$

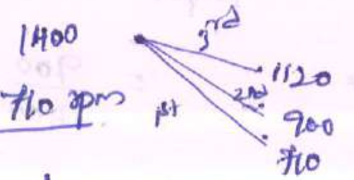
$Z_7 + Z_8 = Z_9 + Z_{10} = 52$

$1.27 Z_8 + Z_8 = 52$ $\quad \boxed{Z_8 = 23}$

$Z_7 + Z_8 = 52 \Rightarrow Z_7 + 23 = 52$

$\boxed{Z_7 = 29}$

Stage I — 1st pair



Speed is reduced from 1400 rpm to 710 rpm

Assume $\boxed{Z_5 = 20}$ $\quad \frac{Z_5}{Z_6} = \frac{N_6}{N_5}$, $\frac{20}{Z_6} = \frac{710}{1400}$

$\boxed{Z_6 = 40}$ $\quad \boxed{Z_5 + Z_6 = 60}$

2nd pair:

Speed is reduced from 1400 rpm to 900 rpm

$\frac{Z_1}{Z_2} = \frac{N_2}{N_1}$; $\frac{Z_1}{Z_2} = \frac{900}{1400}$; $\boxed{Z_1 = 0.643\, Z_2}$

$\boxed{Z_1 + Z_2 = Z_3 + Z_4 = Z_5 + Z_6 = 60}$

$0.643 Z_2 + Z_2 = 60$ $\quad Z_1 + 37 = 60$

$\boxed{Z_2 = 37}$ $\quad \boxed{Z_1 = 23}$

3rd pair: speed is reduced from 1400 rpm to 1120 rpm

$$\frac{Z_3}{Z_4} = \frac{N_4}{N_3}, \quad \frac{Z_3}{Z_4} = \frac{1120}{1400} \quad \boxed{Z_3 = 0.8 Z_4}$$

$$Z_3 + Z_4 = 60 \qquad 0.8Z_4 + Z_4 = 60$$

$$\boxed{Z_4 = 33} \qquad Z_3 + 33 = 60$$

$$\boxed{Z_3 = 27}$$

4. Percentage deviation

$$\%\ \text{deviation} = \frac{N_{obtain} - N_{Table}}{N_{obtain}}$$

Output speeds:

$$N_{01} = N_I \times \frac{Z_1}{Z_2} \times \frac{Z_7}{Z_8}; \quad 1440 \times \left(\frac{23}{37}\right) \times \left(\frac{29}{23}\right) = 1128.6 \text{ rpm}$$

$$N_{02} = N_I \times \frac{Z_1}{Z_2} \times \frac{Z_9}{Z_{10}}, \quad 1440 \times \left(\frac{23}{37}\right) \times \left(\frac{20}{32}\right) = \underline{559.5} \text{ rpm}$$

$$N_{03} = N_I \times \frac{Z_3}{Z_4} \times \frac{Z_7}{Z_8}; \quad 1440 \times \left(\frac{27}{33}\right) \times \left(\frac{29}{23}\right) = \underline{1485.5} \text{ rpm}$$

$$N_{04} = N_I \times \frac{Z_3}{Z_4} \times \frac{Z_9}{Z_{10}}; \quad 1440 \times \left(\frac{27}{33}\right) \times \left(\frac{20}{32}\right) = \underline{736.4} \text{ rpm}$$

$$N_{05} = N_I \times \frac{Z_5}{Z_6} \times \frac{Z_7}{Z_8}; \quad 1440 \times \left(\frac{20}{40}\right) \times \left(\frac{29}{23}\right) = \underline{907.8} \text{ rpm}$$

$$N_{06} = N_I \times \frac{Z_5}{Z_6} \times \frac{Z_9}{Z_{10}}; \quad 1440 \times \left(\frac{20}{40}\right) \times \left(\frac{20}{32}\right) = \underline{450} \text{ rpm}$$

Sl. No.	Obtainable speed N_{obt} rpm	Table (or) Calculated speed N_{cal} or N_{Table} rpm	% deviation
1	450	450	0
2	559.5	560	-0.09
3	736.4	710	3.6
4	907.8	900	0.86
5	1128.6	1120	0.76
6	1485.5	1400	5 %

4 actual speeds are within 2 % of standard speeds.

2 " " are not within the limit.

(21)

6. Design a 12 speed gear box for an all geared headstock of a lathe. Maximum and minimum speeds are 600 rpm and 25 rpm respectively. The drive is from an electric motor giving 2.25 kW at 1440 rpm.

Given:

$n = 12$

$N_{max} = 600$ rpm

$N_{min} = 25$ rpm

$P = 2.25$ kW

$N_{input} = 1440$ rpm

Soln:

1. Selection of spindle speeds

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

$$\frac{600}{25} = \phi^{12-1}$$

$\underline{\underline{\phi = 1.335}}$ is not a std step ratio

$1.06 \times \underline{(1.06 \times 1.06 \times 1.06 \times 1.06)} = 1.338$ (Skip 4 speeds)

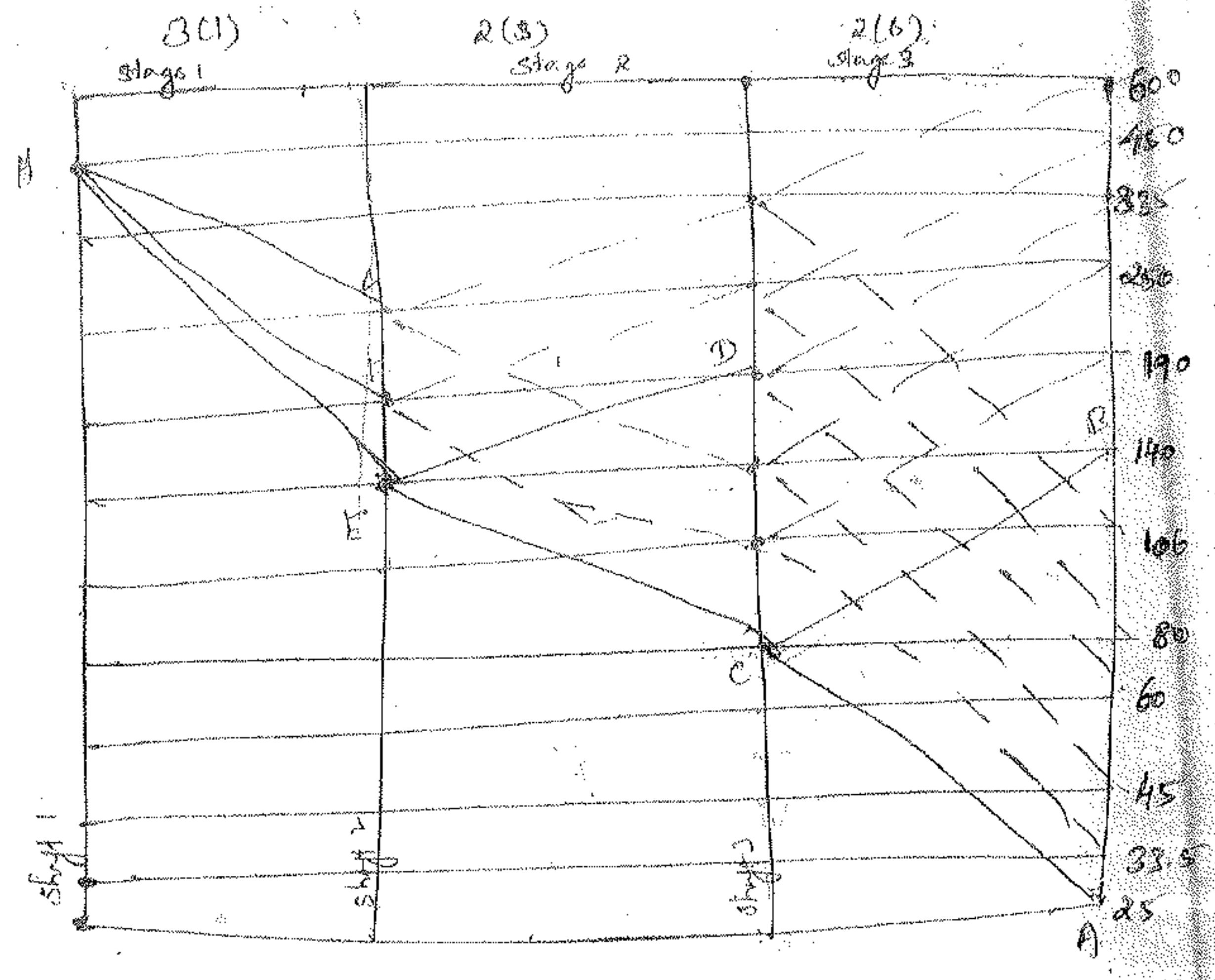
$$\boxed{\phi = 1.06}$$

25, 33.5, 45, 60, 80, 106, 140, 190, 250, 335, 450, 600 rpm

2. Ray diagram:

Structural formula:

3(1) . 2(3) . 2(6)



Stage 3: $\frac{N_{min}}{N_{input}} = \frac{25}{80} = 0.31 > \frac{1}{4}$

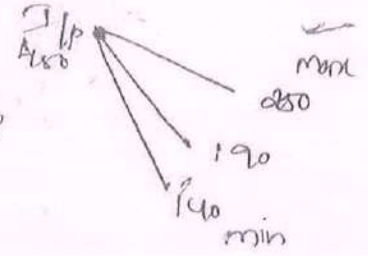
$\frac{N_{max}}{N_{input}} = \frac{140}{80} = 1.75 < 2$

Stage 2: $\frac{N_{min}}{N_{input}} = \frac{80}{140} = 0.57 > \frac{1}{4}$

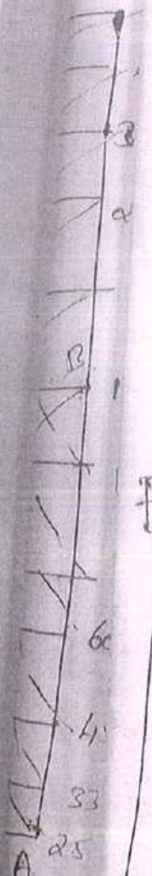
$\frac{N_{max}}{N_{input}} = \frac{190}{140} = 1.36 < 2$

Stage 1. $\frac{N_{min}}{N_{input}} = \frac{140}{450} = 0.311 > \frac{1}{4}$

$\frac{N_{max}}{N_{input}} = \frac{250}{450} = 0.56 < 2$



Ratio requirements are satisfied



3. Kinematic arrangement

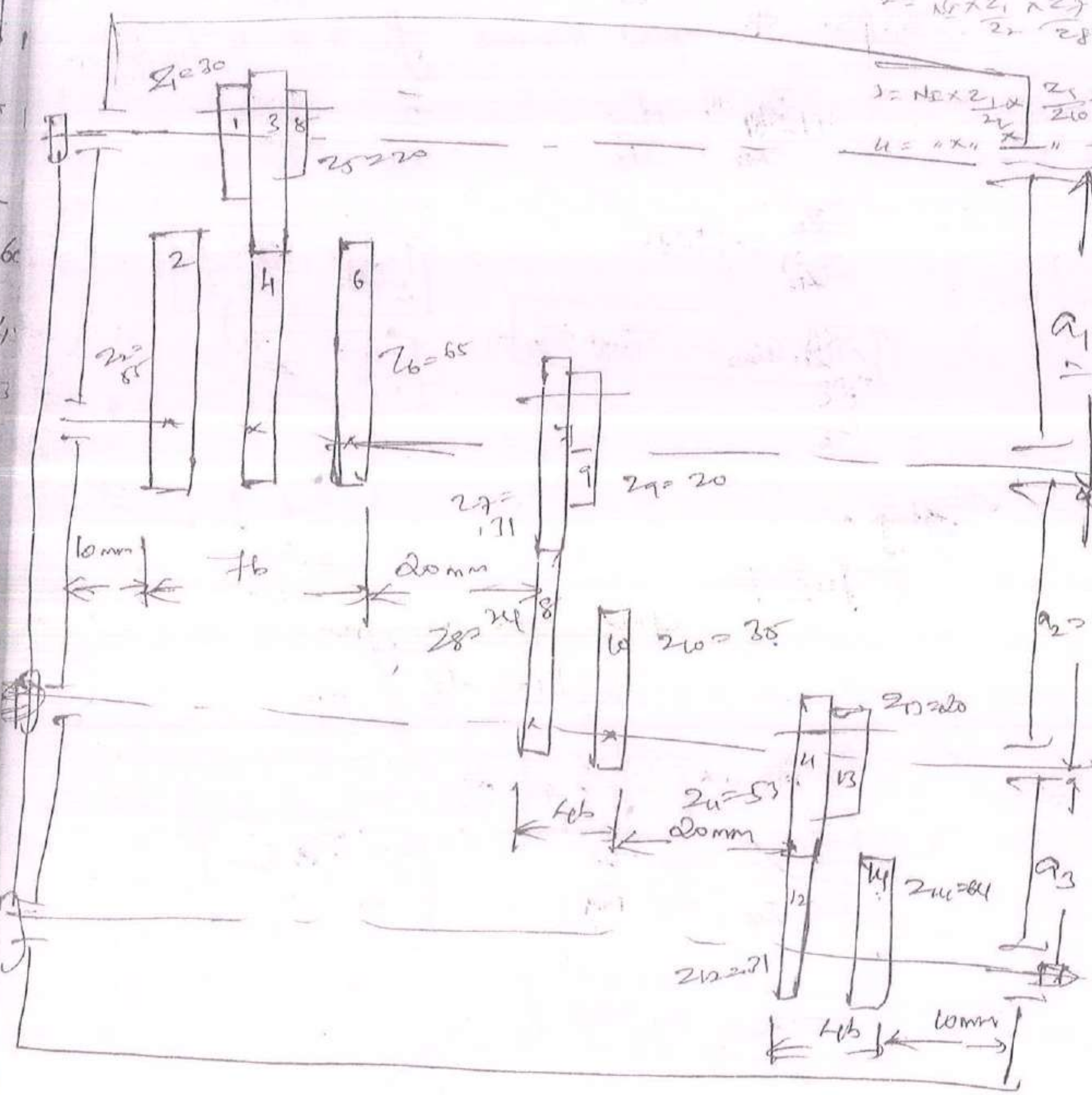
12 speeds

$1 = N_I \times \frac{Z_1}{Z_2} \times \frac{Z_7}{Z_8} \times \frac{Z_{11}}{Z_{12}}$

$2 = N_I \times \frac{Z_1}{Z_2} \times \frac{Z_7}{Z_8} \times \frac{Z_{13}}{Z_{14}}$

$3 = N_I \times \frac{Z_1}{Z_2} \times \frac{Z_9}{Z_{10}} \times \frac{Z_{11}}{Z_{12}}$

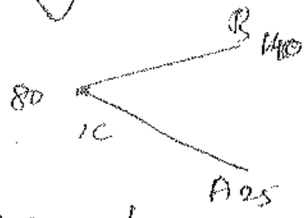
$4 = " \times " \times " \times \frac{Z_{13}}{Z_{14}}$



(24)

4. Calculation of number teeth on all gears:

Stage 3:



First pair:

Speed reduced from 80 rpm to 25 rpm

Assume $\boxed{Z_{13} = 20}$

$$\frac{Z_{13}}{Z_{14}} = \frac{N_{14}}{N_{13}} \qquad \frac{20}{Z_{14}} = \frac{25}{80} \qquad \boxed{Z_{14} = 64}$$

Second pair

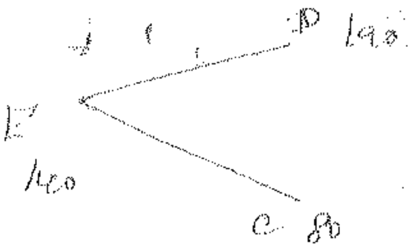
Speed increased from 80 to 140 rpm

$$\frac{Z_{11}}{Z_{12}} = \frac{N_{12}}{N_{11}} \qquad \frac{Z_{11}}{Z_{12}} = \frac{140}{80}$$

$$\frac{Z_{11}}{Z_{12}} = 1.75$$

$$\boxed{Z_{11} + Z_{12} = Z_{13} + Z_{14}} \qquad \boxed{Z_{11} = 53} \qquad \boxed{Z_{12} = 31}$$

Stage 2:



First pair

Speed reduced from 140

140 to 80 rpm

Assume $\boxed{Z_9 = 20}$

$$\frac{Z_9}{Z_{10}} = \frac{N_{10}}{N_9} \qquad \boxed{Z_{10} = 35}$$

$$\boxed{Z_9 + Z_{10} = Z_7 + Z_8}$$

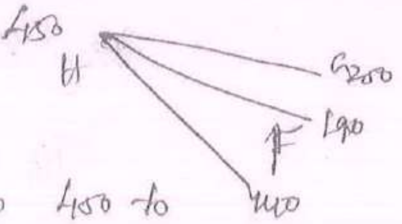
Second pair Speed is increased from 140 to 190

$$\frac{Z_7}{Z_8} = \frac{N_8}{N_7} = \frac{190}{140}$$

$$\boxed{Z_7 = 31} \quad \boxed{Z_8 = 27}$$

Stage I:

first pair



Speed is reduced from 450 to 140 rpm

Assume $\boxed{Z_5 = 20}$

$$\frac{Z_5}{Z_6} = \frac{N_6}{N_5} = \frac{140}{450} \qquad \boxed{Z_6 = 65}$$

$$Z_1 + Z_2 = Z_3 + Z_4 = Z_5 + Z_6$$

Second pair 450 to 190 rpm

$$\frac{Z_1}{Z_2} = \frac{N_2}{N_1} = \frac{190}{450}$$

$$\boxed{Z_1 = 25} \quad \boxed{Z_2 = 60}$$

Third pair 450 to 250 rpm

$$\frac{Z_3}{Z_4} = \frac{N_4}{N_3} = \frac{250}{450}$$

$$\boxed{Z_3 = 30} \quad \boxed{Z_4 = 55}$$

P8

5. Material selection : C45

6. Calculation of module.

To find torque

Torque for the gear : which has the low speed :

$$T_{14} = \frac{P \times 60}{2\pi N_{low}} = \frac{2.25 \times 10^3 \times 60}{2\pi \times 25} = 85$$

To find tangential force:

$$\boxed{F_{t\,14} = \frac{T}{r}} = \frac{2 \times T_{14}}{Z_{14} \times m}$$

$$= \frac{2 \times 859.44 \times 10^3}{64 \times m} = \frac{26857.5}{m}$$

module, $\boxed{m = \sqrt{F_t / \psi_m \times M}}$ P.32

$\psi_m = b/m = 10$

$M = 30$ for C45 steel

$$m = \sqrt{\frac{\frac{26857.5}{m}}{10 \times 30}}$$

$$= \sqrt{\frac{89.525}{m}}$$

$$m^2 = \frac{89.525}{m}$$

$\boxed{m = 5mm}$ ← $\boxed{m = 4.47\ mm}$ = 5m

7. calculation of centre distance:

$$a_1 = \left(\frac{Z_1 + Z_2}{2}\right) m$$

$$= \left(\frac{30+55}{2}\right) \times 5 = 212.5 \text{ mm}, \quad a_1 = 212.5 \text{ mm}$$

$$a_2 = \left(\frac{Z_7 + Z_8}{2}\right) m = \frac{(31+24)}{2} \times 5 = 137.5 \text{ mm}$$

$$a_3 = \left(\frac{Z_{11} + Z_{12}}{2}\right) m = \left(\frac{53+31}{2}\right) 5 = 210 \text{ mm}$$

8. calculation of face width

$$b = \varphi \times m = 10 \times 5 = 50 \text{ mm}$$

9. calculation of length of shaft:

$$L = 25 + 10 + 7b + 20 + 4b + 20 + 4b + 10 + 25$$

$$= 110 + 15b$$

$$= 110 + (15 \times 50) = 860 \text{ mm}$$

10. Design of shaft

i) D/n of Spindle:

To find Max. bending moment (M)

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

$$\boxed{F_n = \frac{F_t}{\cos\alpha}} = \frac{\frac{26857.5}{5}}{\cos 20^\circ} = \underline{5716.23N}$$

$$M = \frac{5716.23 \times 860}{4} \Rightarrow \underline{12.29 \times 10^5} \text{ N.mm}$$

To find equivalent torque (T_{eq})

$$T_{eq} = \sqrt{M^2 + T_4^2}$$

$$= \sqrt{(12.29 \times 10^5)^2 + (859.44 \times 10^3)^2}$$

$$= \underline{1.5 \times 10^6} \text{ N.mm}$$

Diameter of the Spindle

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

$\tau = 30$ N/mm² for C45

$d_s = \underline{63.38}$ mm

for R40 series $d_s = \underline{67}$ mm

ii) Design of other shafts

a) Dia of shaft 1:

input speed = 450 rpm

$T = 0.2\, d_s^3\, [\tau]$ $\qquad P = \frac{2\pi N T}{60}$

$T = \frac{P \times 60}{2\pi N} = \frac{2.25 \times 10^3 \times 60}{2\pi \times 450} = 47.746\ Nm$

$47.746 \times 10^3 = 0.2 \times d_{s_1}^3 \times 30$

$d_{s_1} = 19.9 = 20\ mm$ for R40

b) Dia of shaft 2:

Minimum speed = 140 rpm

$T = \frac{P \times 60}{2\pi N} = \frac{2.25 \times 10^3 \times 60}{2\pi \times 140} = 153.47\ N\cdot m$

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

$153.47 \times 10^3 = 0.2 \times d_{s2}^3 \times 30$

$d_{s2} = 29.46\ mm = 30\ mm$

c) Dia of shaft 3: Min speed = 80 rpm

$T = \frac{2.25 \times 10^3 \times 60}{2\pi \times 80} = 268.57\ Nm$

$d_{s3} = 35.5\ mm$

(7). Design a 12 speed gear box for a lathe. The minimum and maximum speeds are 100 and 1200 rpm. Power is 5 kw from 1440 rpm induction motor.

Given: $n = 12$ $\quad P = 5kw$

$N_{min} = 100$ rpm $\quad N_{input} = 1440$ rpm

$N_{max} = 1200$ rpm

Find: Design a 12 speed gear box.

Soln:

1. Selection of spindle speeds:

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \qquad \frac{1200}{100} = \phi^{12-1}$$

$$\boxed{\phi = 1.253}$$

f.20 $\boxed{\phi = 1.25}$ R10 series

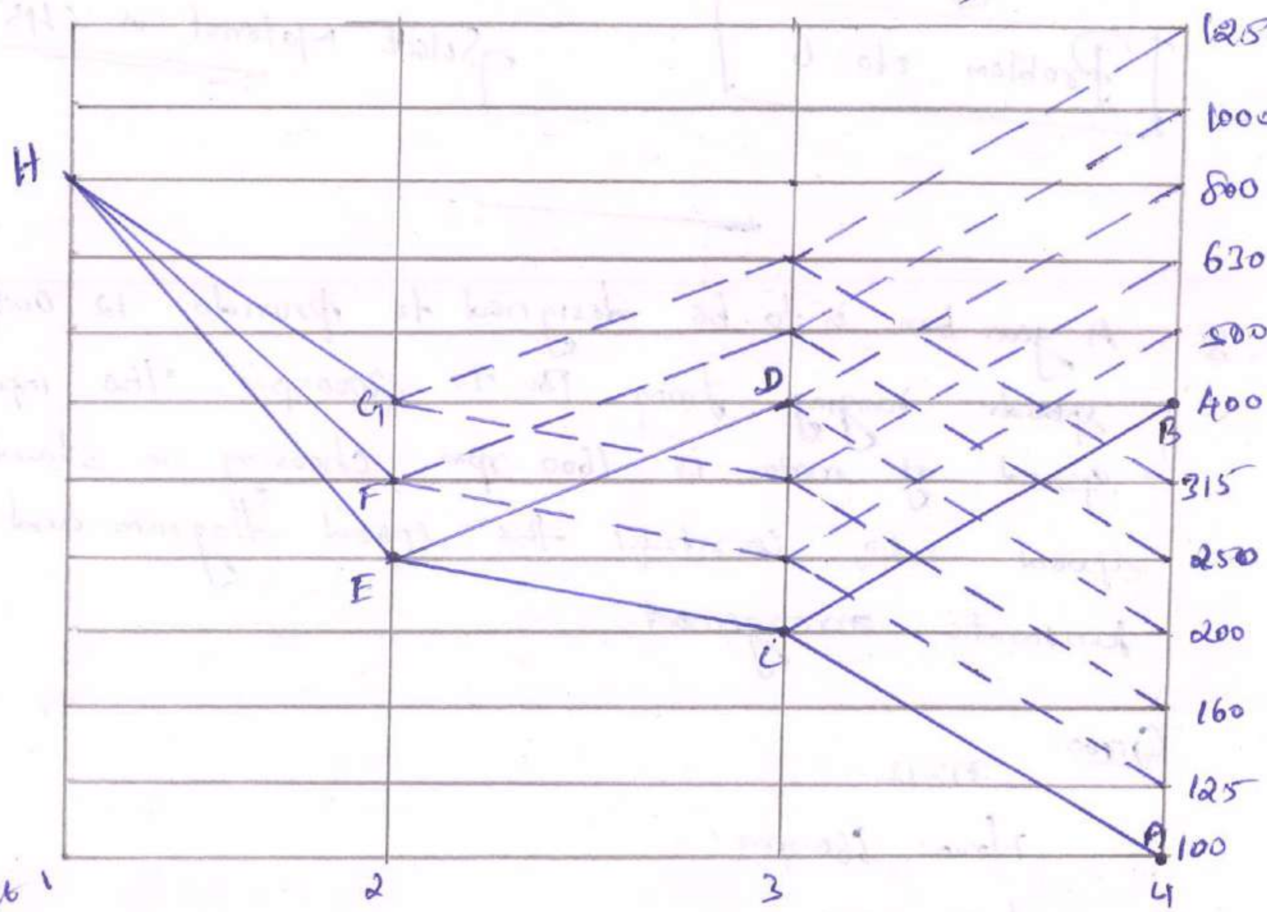
100 rpm, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250 rpm

2. speed or Ray diagram:

Structural formula:

$3(1).\ 2(3).\ 2(6)$

Stage I 3(1) II 2(3) III 2(6)



Stage III $\frac{N_{min}}{N_{input}} = \frac{100}{200} = 0.5 > 0.25$ $\frac{N_{max}}{N_{input}} = \frac{400}{200} = 2$

200 C → B 400, A 100

Stage II $\frac{N_{min}}{N_{input}} = \frac{200}{250} = 0.8 > 0.25$ $\frac{N_{max}}{N_{input}} = \frac{400}{250} = 1.6 < 2$

250 E → D 400, C 200

Stage I $\frac{N_{min}}{N_{input}} = \frac{250}{800} = 0.3 > 0.25$ $\frac{N_{max}}{N_{input}} = \frac{400}{800} = 0.5 < 2$

$= \frac{315}{800} = 0.39 > 0.25$

800 H → G 400, F 315, E 250

∴ Ratio requirements are satisfied

Remaining Procedures (step 3 to step 10) are same as

$\boxed{\text{Problem No: 6}}$ Select Material is C45

8. A gear box is to be designed to provide 12 output speeds ranging from 160 to 2000 rpm. The input speed of motor is 1600 rpm. Choosing a standard speed ratio, construct the speed diagram and the kinematic arrangement.

Given:

$n = 12$

$N_{min} = 160$ rpm

$N_{max} = 2000$ rpm

Find: Construct

i. Speed diagram

ii. Kinematic arrangement

Soln: Selection of spindle speeds:

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \qquad \frac{2000}{160} = \phi^{12-1} \qquad \boxed{\phi = 1.258}$$

is not a std step ratio

$$1.12 \times 1.12 = 1.254 \qquad \boxed{\phi = 1.12}$$ is satisfy the requirements

F.20 R20 series

160 rpm, 200, 250, 315, 400, 500, 630, 800, 1000, 1250,

N_{min}

1600, 2000 rpm

N_{max}

Construction procedure for speed diagram and kinematic layout are same as [Problem No: 6] [Step 2 & 3]

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9. Design a 12 speed gear box. The required speed range is 100 to 355 rpm. Draw the ray diagram, kinematic arrangement and find the number of teeth on each gear. Check for the interference.

Given:

$n = 12$

$N_{min} = 100$ rpm

$N_{max} = 355$ rpm

Find: Design a 12 speed gear box

Soln:

1. Selection of spindle speeds:

$$\frac{N_{max}}{N_{min}} = \phi^{n-1} \quad \frac{355}{100} = \phi^{12-1}$$

$\boxed{\phi = 1.12}$ is a std step ratio

T.20

R 20 series

N_{min} 100 rpm, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355 rpm N_{max}

Remaining procedures are same as [Problem No: 6] [Step 2, 3 & 4]

UNIT 5 CAMS, CLUTCHES AND BRAKES

CLUTCHES

1. An automobile single plate clutch consists of two pairs of contacting surfaces. The Inner and outer radii of friction plate are 120mm and 250mm respectively. The co-efficient of friction is 0.25 and the total axial force is 15 KN. Calculate the power transmitting capacity of the clutch plate at 500 rpm. using (i) uniform wear theory and (ii) Uniform pressure theory.

Given data.

$n = 2 \quad r_1 = 250\,mm = 0.25\,m \qquad \mu = 0.25$

$r_2 = 120\,mm = 0.12\,m$,

$W = 15\,KN = 15\times10^3\,N$

$N = 500\,rpm$

Solution.

(i) Using uniform wear theory

Torque $T = n.\mu.W\left(\frac{r_1+r_2}{2}\right)$

$= 2\times0.25\times15\times10^3\left(\frac{0.25+0.12}{2}\right)$

$T = 1387.5\,Nm$

$\therefore$ Power transmitted

$$P = \frac{2\pi NT}{60} = \frac{2\pi\times500\times1387.5}{60}$$

$P = 72.65\,KW$

(ii) Using uniform pressure theory

$$\text{Torque, } T = n \cdot \mu \cdot W \; \frac{2}{3} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2} \right]$$

$$= 2 \times 0.25 \times 15 \times 10^3 \times \frac{2}{3} \left[\frac{(0.25)^3 - (0.12)^3}{(0.25)^2 - (0.12)^2} \right]$$

$$T = 1444.6 \text{ Nm}$$

Power transmitted,

$$P = \frac{2\pi NT}{60} = \frac{2\pi \times 500 \times 1444.6}{60} = 75.64 \text{ kW}$$

2. A plate clutch with maximum diameter 6cm has maximum Lining pressure of 350 KPa. The power to be transmitted at 400 rpm is 135 kW and $\mu = 0.3$. Find inside diameter and spring force required to engage the clutch. Springs with spring index 6 and material of the spring is steel with safe shear stress 600 MPa are used. Find the diameters if 6 springs are used.

Given data

Power $P = 135 \text{ kW} = 135 \times 10^3 \text{ W}$

$N = 400 \text{ rpm}$

$\mu = 0.3$

$d_1 = 6 \text{ cm} = 600 \text{ mm}$

Max Pressure $= 350 \text{ KPa}$

To find

Diameter $d_2 = ?$

Solution

Calculate the torque to be transmitted,

W.K.T

$$P = \frac{2\pi NT}{60}$$

$$T = \frac{P\times 60}{2\pi N} = \frac{135\times10^3\times 60}{2\pi\times 400}$$

$$T = 3222.8\ Nm = 3222.8\times10^3\ Nmm$$

By using uniform wear Method.

(i) Find Diameter of the shaft,

$$d > \left[\frac{16T}{\pi\tau}\right]^{1/3}$$

Assume

τ - Permissible shear stress for shaft

$\tau = 50\ N/mm^2$

$$\therefore d > \left[\frac{16\times 3222.8\times10^3}{\pi\times 50}\right]$$

$$d > 68.97\ mm$$

Take $d = 68mm$

The Inner radius of the clutch Plate

$R_i = 2d = 2\times 68 = 136mm$

Outer radius of clutch R_o

$R_o = 1.25 R_i = 1.25\times 136 = 170mm$

Mean radius

$$R_M = \frac{R_i + R_o}{2} = \frac{136+170}{2} = 153mm$$

Face with of clutch Plate

$b = R_o - R_i$

$= 170 - 136 = 34\,mm$

~~b = 34 mm~~

$b = 34\,mm$

Thickness $= 5\,mm$ (Assumed)

Minimum number of pairs of friction surfaces required

$$n_{Min} = \frac{T_d}{2\pi (P)\, b\, \mu\, R_m^2}$$

Here, T_d - design Torque

(P) - Allowable press $= K.P_b$

$\mu = 0.3$

K = Speed factor (This depend on the velocity at the max radius of clutch plate).

$T_d = T \times K_s$

K_s - Service factor

Refer PSG Design Data book Pg No: 7.9

$K_s = 1.25$ (for steady load)

$T_d = T \times K_s = 3222 \times 10^3 \times 1.25$

$T_d = 4027.5 \times 10^3$

Velocity at the maximum radius,

$$V = \frac{\pi\, D_{max}\, N}{60 \times 1000} = \frac{\pi \times 600 \times 400}{60 \times 1000}$$

$V = 12.56\ m/s.$

Refer PSG Databook Pg No 7-89

$\therefore K = 1$

$P_b = 3$ kgf/cm² (assume)

$P_a = K \cdot P_b = 1 \times 3$ kgf/cm²

$P_a = 3$ kgf/cm²

$$n_{min} = \frac{4027.5 \times 10^3}{2\pi \times 3 \times 34 \times 0.3 \times (153)^2}$$

$= 0.89$

$= 1$ (Say)

$\therefore$ No of plates $= 1$

If 6 spring are used,

$K_s = K_1 + K_2 + K_3 + K_4 + K_5 + K_6$

$= 1.25 + 1.25 + 1.25 + 1.25 + 1.25 + 1.25 = 7.5$

$T_d = T \times K_s = 3222 \times 10^3 \times 7.5$

$T_d = 24165 \times 10^3$ Nmm

Diameter of the shaft, Assume $\tau = 50$ N/mm²

$$d = \left[\frac{16\, T_d}{\pi \tau}\right]^{1/3}$$

$$= \left[\frac{16 \times 24165 \times 10^3}{\pi \times 50}\right]^{1/3}$$

$\boxed{d = 135 \text{ mm}}$

3. A power of 20KW is to be transmitted through a cone clutch at 500rpm. For uniform wear condition, find the main dimensions of clutch and shaft. Also determine the axial force required to engage the clutch. Assume co-efficient of friction as 0.25, the maximum normal pressure on the friction surface is not to exceed 0.08MPa and take the design stress for the shaft material as 40MPa.

Given Data

$P = 20\,KW \quad N = 500rpm \quad \mu = 0.25$

$P_{max} = 0.08MPa \quad P_{shaft} = 40\,MPa$

To find: ~~Main~~ Axial force required to engage the clutch(W)

diameter of the shaft.

Solution

To find: Main dimension of clutch: It may be assumed that the face width b is equal to half of the main radius of the friction surface R.

$\therefore b = \frac{R}{2}$ Also take semicone angle $\alpha = 15°$

For cone clutch, $b = \frac{r_1 - r_2}{\sin\alpha}$

Mean radius $R = \frac{r_1 + r_2}{2}$

$$\frac{r_1 - r_2}{\sin\alpha} = \frac{\left(\frac{r_1 + r_2}{2}\right)}{2} \qquad \left[\because b = \frac{R}{2}\right]$$

$$r_1 - r_2 = \frac{\sin\alpha\,(r_1 + r_2)}{4}$$

$$0.934 r_1 = 1.064 r_2 \text{ (or)}$$

$$r_1 = 1.139 r_2 \longrightarrow (1)$$

Power transmitted $P = \frac{2\pi NT}{60}$

$$20\times10^3 = \frac{2\pi\times500\times T}{60}$$

$$T = 382\ Nm$$

Assume

Service factor $k_s = 2.5$

Design torque $[T] = T\times k_s = 382\times2.5 = 955\ Nm$

For uniform wear, $[T] = 2\pi\mu P_{max} R^2 b$

$$955\times10^3 = 2\pi(0.25)\ 0.08\ R^2\cdot\frac{R}{2}$$

$$R = 247.7mm$$

$$R = \frac{r_1+r_2}{2} = 247.7$$

$$r_1+r_2 = 495.4 \longrightarrow ②$$

$$r_1 = 1.139r_2 \longrightarrow ①.$$

Solving equ ① & ②.

Inner radius $r_1 = 264\ mm$

Outer radius $r_2 = 231.6\ mm$

Face width $b = \frac{R}{2} = \frac{247.7}{2} = 123.85mm$

To find axial force required to engage the clutch (W)

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

$$= 2\pi\times0.08\times10^6\times0.2316(0.264-0.2316)$$

$$= 3771.84N$$

To find the diameter of the shaft.

W.K.T

$$[T] = \frac{\pi}{16} d_s^3 P_{shaft}$$

$$955 = \frac{\pi}{16} d_s^3 \times 40 \times 10^6$$

Diameter of the shaft

$$d_s = 0.0495 \text{ m (or) } 49.5 \text{ mm}$$

4. A single plate clutch transmits 25KW at 900 rpm. The maximum pressure intensity between the plates is 85kN/m². The ratio of radii is 1.25. Both the sides of the plate are effective and the coefficient of friction is 0.25. Determine (i) the Inner diameter of the plate (ii) the axial force to engage the clutch. Assume theory of uniform wear.

Given Data

$P = 25\text{KW} = 25 \times 10^3 \text{ W}$ $N = 900 \text{rpm}$ $\frac{r_1}{r_2} = 1.25$

$P_{max} = 85 \text{ kN/m}^2 = 85 \times 10^3 \text{ N/m}^2$ $\mu = 0.25$

$n = 2$

Solution : (i) The Inner diameter of the Plate

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

$$25 \times 10^3 = \frac{2\pi \times 900 \times T}{60}$$

$$T = 265.26 \text{ Nm}$$

Since the intensity of pressure is maximum at the Inner radius (r_2)

$P_{max}\, r_2 = C$ or $C = 85 \times 10^3 r_2$ N/mm

The axial thrust transmitted to the frictional surface

$$W = 2\pi C(r_1 - r_2) = 2\pi \times 85 \times 10^3 \, r_2 (1.25 r_2 - r_2)$$ $[\because r_1 = 1.25 r_2]$

$$W = 1.335 \times 10^5 (r_2)^2$$

The mean radius for uniform wear is given by

$$R = \frac{r_1+r_2}{2} = \frac{1.25r_2+r_2}{2} = 1.125\, r_2$$

Torque transmitted

$$T = n.\mu.W.R$$

$$265.26 = 2\times 0.25\times 1.335\times 10^5 (r_2)^2 \times 1.125 r_2$$

$$265.26 = 75.104\times 10^3 \times r_2^3$$

$$r_2 = 0.1523\ m \quad \text{or } 152.3\ mm$$

$$r_1 = 1.25\, r_2 = 1.25\times 152.3 = 190.375\, mm$$

(ii) The axial force to engage the clutch

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

$$= 1.335\times 10^5 \times (r_2)^2 = 1.335\times 10^5\ (0.1523)^2$$

$$W = 3096.57\ N$$

5. A multiplate clutch has three discs on the driving shaft and two on the driven shaft. The outside diameter of the contact surfaces is 240mm and Inside diameter is 120mm. Assume uniform wear co-efficient of friction 0.3. find the maximum axial intensity of pressure between the discs for transmitting 25kW at 1575 rpm.

Given Data: $n_1 = 3 \quad n_2 = 2$

$d_1 = 240mm \quad r_1 = 120mm = 0.12m$

$d_2 = 120mm \quad r_2 = 60mm = 0.06m$

$\mu = 0.3 \quad P = 25\ kW = 25\times 10^3\ W$

$N = 1775\ rpm.$

Solution.

Number of pairs of contact Surfaces

$$n = n_1 + n_2 - 1 = 3 + 2 - 1 = 4$$

Power transmitted

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

$$25 \times 10^3 = \frac{2\pi \times 1575 \times T}{60}$$

$$T = 151.6 \text{ Nm}$$

For uniform wear, Torque transmitted is given by

$$T = n.\mu.W\left(\frac{r_1 + r_2}{2}\right)$$

$$151.6 = 4 \times 0.1 \times W\left(\frac{0.12 + 0.06}{2}\right)$$

$$W = 1404 \text{ N}$$

The axial force exerted (W)

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

$$W = 2\pi \; P_{max} \times r_2 (r_1 - r_2)$$

$$W = 2\pi \; P_{max} \times 0.06 \; (0.12 - 0.06)$$

$$1404 = 2\pi \times P_{max} \times 0.06$$

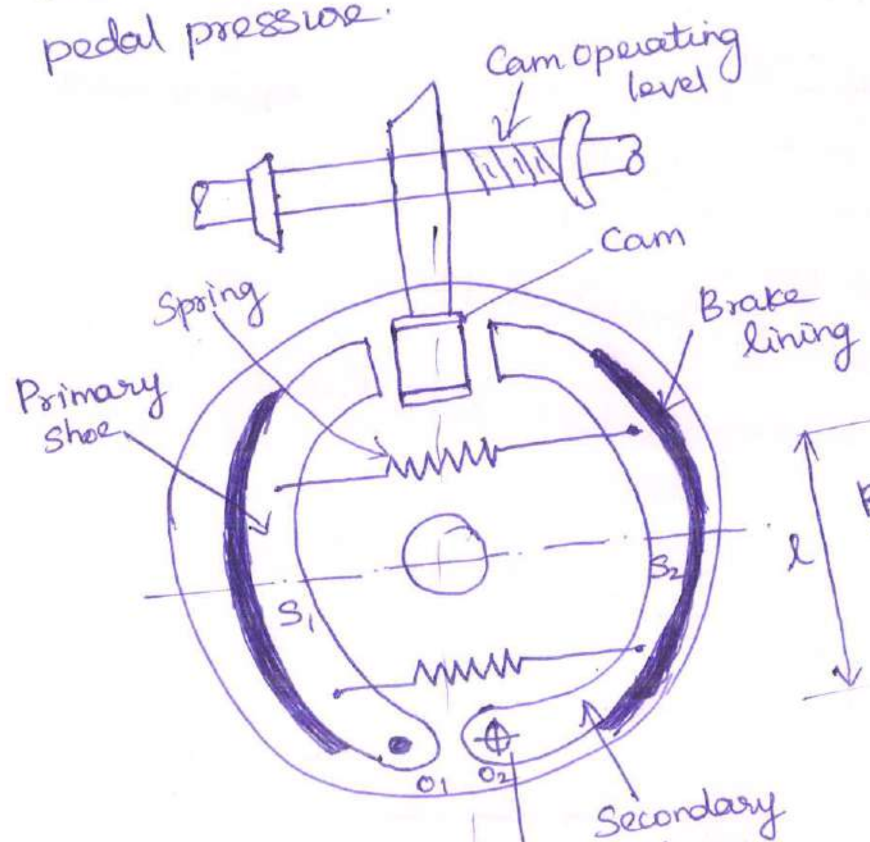
$$P_{max} = 62.07 \times 10^3 \text{ N/m}^2$$

$$\boxed{P_{max} = 62 \text{ kN/m}^2}$$

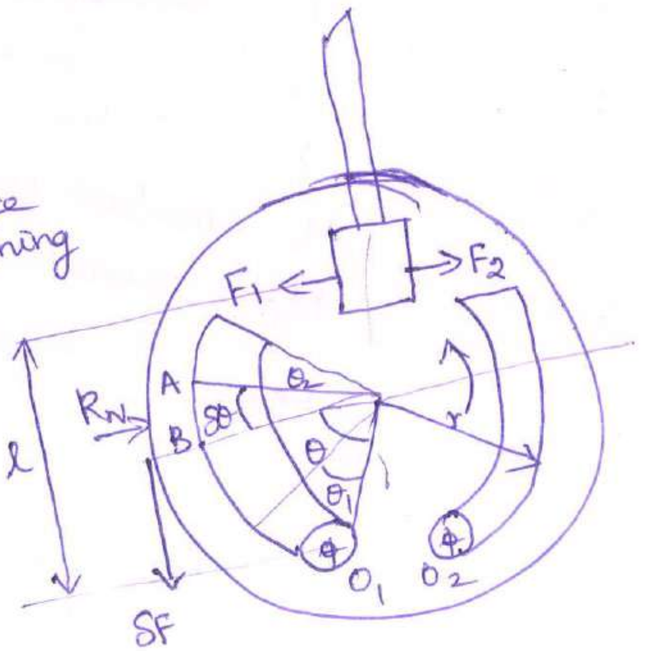
Brake

1. Describe with the help of a neat sketch the design Procedure of an internal expanding shoe brake. Also deduce the expression for the braking torque.

This type of brake is provided internally on the brake drum. In olden days band brakes were used in automobiles, which were exposed to drit and water. Their heat dissipation capacity was also poor. These days, band brakes have been replaced by internally expanding shoe brakes having atleast one self-energizing shoe per wheel. This result in tremendous friction, giving great braking power without excessive use of pedal pressure.



Internal Expanding brake.



b) Forces on brake.

It consisits of two semi-circular shoes S_1 & S_2 which are lined with a frictional material such as ferrodo. When brakes are applied, cam rotates which pushes the shoes outwards to press the brake lining against the rim of the drum. As soon as the brakes are off, the shoes are pushed inside by the spring

Determination of pressure and Brake Torque

Consider the forces on the brake when the drum rotates in anticlockwise direction, as shown in fig.

Let P_1 = Maximum intensity of normal pressure

P_N - Normal pressure

r - Internal radius of the drum

b - width of brake lining

T_B - Braking Torque

F_1 - Force exerted by the cam on the primary shoe

F_2 - Force exerted by the cam on the secondary shoe

R_N - Normal force

F - Frictional force

μ - co-efficient of friction between shoe & drum

M_N - moment of normal force

M_F - moment of frictional force

Braking torque = Frictional force x radius

$$= \delta F \cdot r$$

$$T_B = \mu P_1 \, b r^2 (\cos\theta_1 - \cos\theta_2)$$

Total moment of frictional force

$$M_F = \mu P_1 \cdot b \cdot r \left[r(\cos\theta_1 - \cos\theta_2) + \frac{OO_1}{4}(\cos 2\theta_2 - \cos 2\theta_1) \right]$$

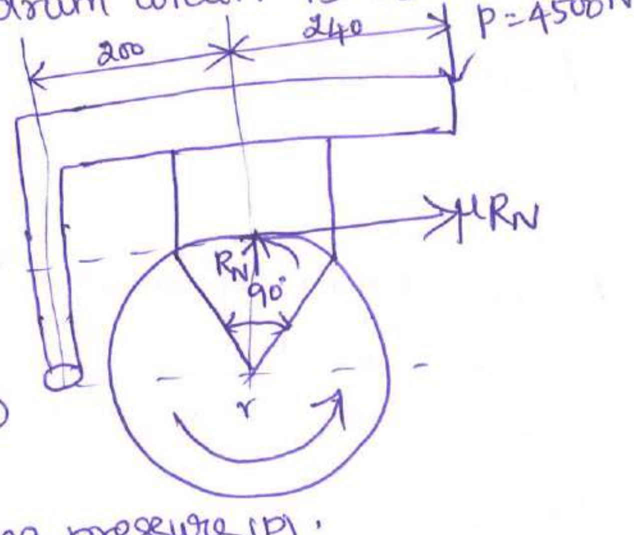
For leading shoe, take moments about O_1,

$$F_1 = \frac{M_N - M_F}{l}$$

and for trailing shoe, take moments about O_2,

$$F_2 = \frac{M_N + M_F}{l}$$

2. Calculate the average bearing pressure and the initial and average braking powers for the block shoe shown in fig. The diameter of the drum is 400mm and it rotates at 200rpm. The co-efficient of friction is 0.2 and drum width is 75mm.



Given data

$d = 200mm$ or $r = 100mm$

$P = 4500N$

$2\theta = 90° = \frac{\pi}{2}$ rad

$N = 200 rpm$ $W = 75mm$

Solution (i) To find average bearing pressure (P):

Taking moments about pivot O, we get

$R_N = 200 = [4500 \times (200+240)] + (\mu R_N \times 0)$

$R_N = 9900N$

W.K.T

$R_N = P \times b \times W$

$9900 = P \times (2 \times 75) \times 75$

$P = 0.88\ N/mm^2$

(ii) To find initial and average braking powers

Initial velocity of the drum $v_1 = \frac{\pi d N}{60} = \frac{\pi \times 0.4 \times 200}{60}$

$v_1 = 4.19 m/s$

Final velocity of the drum $v_2 = 0$,

Average velocity $v = \frac{v_1 + v_2}{2} = \frac{4.19+0}{2} = 2.095 m/s$

W.K.T

Initial braking power = Frictional force x Initial Velocity

$= \mu R_N \times v_1 = 0.3 \times 9900 \times 4.19 = 12.44\,KW$

Average braking Power = $\left.\begin{matrix}\text{Frictional} \\ \text{force}\end{matrix}\right\} \times \left.\begin{matrix}\text{Average} \\ \text{velocity}\end{matrix}\right\}$

$= \mu . R_N \times v = 0.3 \times 9900 \times 2.095 = 6.22\,KW$

3. A rope drum of an elevator having 650mm diameter is fitted with a brake drum of 1m diameter. The brake drum is provided with four cast iron brake shoes each subtending an angle of 45°. The mass of the elevator when loaded is 2000 kg and moves with a speed of 2.5 m/s. The brake has a sufficient capacity to stop a elevator in 2.75 metres. Assuming the coefficient of friction between the brake drum and shoes as 0.2 find (i) width of the shoe if allowable pressure on the brake shoe is limited to 0.3 N/mm² and ii) heat generated in stopping the elevator.

Given data $d_e = 650mm$ or $r_e = 325mm$ $n = 4$ $2\theta = 45°$

$d = 1m$ or $r = 0.5\,m$ $\theta = 22.5°$

$m = 2000\,kg$ $v = 2.5\,m/s$ $h = 2.75\,m$

$\mu = 0.2$ $P_b = 0.3\,N/mm^2$

To find (i) Width of the shoe

(ii) Heat generated in stopping the elevator

Solution (i) To find the width of the shoe (w)

acceleration of the rope (a)

$V^2 - u^2 = 2ah$ $(2.5)^2 - 0 = 2a \times 2.75$

$a = 1.136\,m/s^2$

Accelerating force $= m \times a = 2000 \times 1.136 = 2272\,N$

$\therefore$ Total load acting on the rope while moving

$W =$ load the elevator + Accelerating force

$= (2000 \times 9.81) + 2272 = 21892\,N$

W.K.T

Torque acting on the drum

$T = W \times r_e = 21892 \times 0.325 = 7115\,Nm$

Tangential force acting on the drum

$= \frac{T}{r} = \frac{7115}{0.5} = 14230\,N$

The braking drum is provided with four cast iron shoes

therefore tangential force acting on each rope

$F_t = \frac{14230}{4} = 3557.5\,N$

Normal load on each shoe

$R_N = \frac{F_t}{\mu} = \frac{3557.5}{0.2} = 17787.5\,N$

The projected bearing area of each shoe,

$A_b = W(2r\sin\theta) = W \times (2 \times 500 \sin 22.5^\circ)$

$= 382.7\,W\ mm^2$

Bearing ~~brea~~ pressure

$0.3 = \frac{R_N}{A_b} = \frac{17787.5}{382.7W} = \frac{46.5}{W}$

$W = \frac{46.5}{0.3} = 155\,mm$

(ii) To find the heat generated in stopping the elevator

Heat generated in stopping the elevator

Heat generated in stopping the elevator by the brake

$= \text{Total Energy absorbed}$

$= K.E + P.E$

$= \frac{1}{2} m v^2 + mgh$

$= \frac{1}{2} \times 2000 \times 2.5^2 + (2000 \times 9.81 \times 2.75)$

$= 60205 \text{ Nm}$

$= 60.205 \text{ kJ}$

4. A single block brake, the diameter of the drum is 250mm and the angle of contact is 90°. The operating force of 700N is applied at the end of lever which is at 250mm from the center of the brake block. The co-efficient of friction between the drum and the lining is 0.35. Determine the torque that may be transmitted. Fulcrum is at 200mm from the centre of brake block with an offset of 50mm from the surface of contact.

Given data

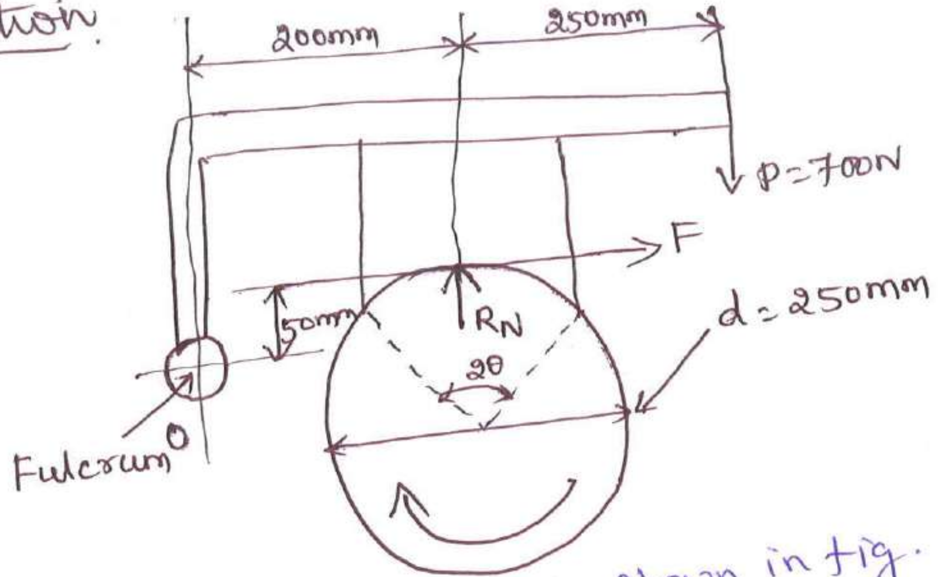
$d = 250\text{mm}$ or $r = 125\text{mm}$

$2\theta = 90° = \frac{\pi}{2} \text{ rad}$

$P = 700\text{N}$

$\mu = 0.35$

Solution



The given arrangement shown in fig.

Since $2\theta > 90°$, therefore equivalent co-efficient of friction

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

$$= \frac{4 \times 0.35 \times \sin 45°}{\frac{\pi}{2} + \sin 90°}$$

$$\mu' = 0.385$$

Taking moments about the fulcrum O, we get

$$700(250+200) + F \times 50 = R_N \times 200 = \frac{F}{\mu'} \times 200$$

$$315000 + 50F = \frac{F}{0.385} \times 200 = 520F$$

$$F = 670N$$

$\therefore$ Torque transmitted by the block brake $\Big\}$ $T_B = F \cdot r = 670 \times 0.125$

$$= 83.75 \text{ Nm}.$$