ME 8651 DESIGN OF TRANSMISSION SYSTEMS

UNIT I
DESIGN OF FLEXIBLE ELEMENTS

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

Introduction

- To transmit power from flexible elements such as belts, chains and ropes are frequently used.

- Pulleys are mounted on a shaft and a continuous belt or rope is passed over them.

- In belt and ropes, power is transmitted due to friction between them and pulleys. In case of chain drives, sprocket wheels are used.

BELT DRIVES

✓Belt drive is a mechanical drive in which the driving and driven shaft are connected by a loop of flexible material called as belt through pulleys mounted on the shafts

✓The distance between the shaft is large, then belts (or) ropes (or) chains are used

✓It can absorb a good amount of shock and vibration

✓It can take care of some degree of misalignment between the driven and the driver machine shafts

Design

- ✓Material - Leather, Rubber, Plastics, Fabric
- ✓No. of ply and Thickness
- ✓Maximum belt stress per unit width
- ✓Density of Belt material
- ✓Coefficient of friction of the belt material

Types of Belts

- Flat Belts
- V – Belts
- Ribbed Belts
- Toothed or timing belts

Problem:

Design a flat belt drive to transmit 22 KW, at 740 rpm to Aluminium rolling machine, The speed ratio is 3.0; The distance between the centers of pulley is 3 m; The dia. of aluminium rolling pulley is 1.2 m.

Step i) Calculation of Design power: (DP)

$$\boxed{DP = \text{Power} \times \text{Load Correction Factor (LCF)} \times \text{Angle of Contact Factor (ACF)}}$$

LCF = 1.5 (Shock load) ⟶ Page 7.53, for Application rolling machine,

Page 7.53

ACF ⇒

Page 7.54

$$AOC = 180^\circ - \left(\frac{D-d}{C} \times 60^\circ\right)$$

$$AOC = 180^\circ - \frac{1.2 - 0.4}{3} \times 60^\circ = 164^\circ$$

For $\alpha = 164^\circ$, ACF = 1.04 (170°) (or) 1.08 (160°)

$$D.P = 22 \times 1.5 \times 1.08 = 35.64 \text{ kW}$$

Step ii) Belt selection:

I have selected, Dunlop "HI-SPEED" 878g fabric belting for medium duty condition.

Page 7.52

$$V = \frac{\pi d n}{60} = \frac{\pi \times 0.4 \times 740}{60}$$

$$V = 15.5 \text{ m/s} < 22 \text{ m/s}$$

(medium duty)

Step iii) Belt rating: Page 7.54

"HI-SPEED", belt load rating = 0.023 KW/mm/ply

(for $\alpha = 180^\circ$, $v = 10$ m/s)

for $v = 15.5$ m/s & $\alpha = 164^\circ$;

$$B.R = 0.023 \frac{KW}{mm} \times \text{no. of ply} \times \frac{15.5}{10} \times \frac{164^\circ}{180^\circ}$$

no. of plies, = 8, for 15 m/s & d = 500 mm = 0.5 m

$v \simeq 15.5$ m/s $d \simeq 0.4$ m

$$B.R = 0.023 \times 8 \times \frac{15.5}{10} \times \frac{164^\circ}{180^\circ}$$

$$\text{Belt rating} = 0.259 \frac{KW}{\text{mm of width}}.$$

Step IV) Belt width:

$$\text{Belt width} = \frac{\text{Design power (kW)}}{\text{Belt rating (kW/mm)}}$$

$$\text{Belt width} = \frac{35.64}{0.259} = 137.6 \text{ mm}$$

Standard width, for 8 plies (HI-SPEED) 200 for FORT

for HI-SPEED, 200 mm ; Page 7.52

Step v) Length of Belt: Page 7.53

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

$$L = (2 \times 3) + \frac{\pi}{2}(1.2 + 0.4) + \frac{(1.2 - 0.4)^2}{4 \times 3}$$

$$L = 8.56 \text{ m}$$

Tension Compensation: Shortening the Belt length.

for 8 plies; 0.5% lesser Page 7.53

$$L = 8.56 \times (100 - 0.5)\% \ [\text{of } L]$$

$$L = 8.56 \times 99.5\% = 8.56 \times 0.995$$

$$L = 8.51 \text{ m}$$

Step vi) Pulley width: Page 7.54

for Belt width 200 mm, Pulley width $200 + 25 = 225$ mm

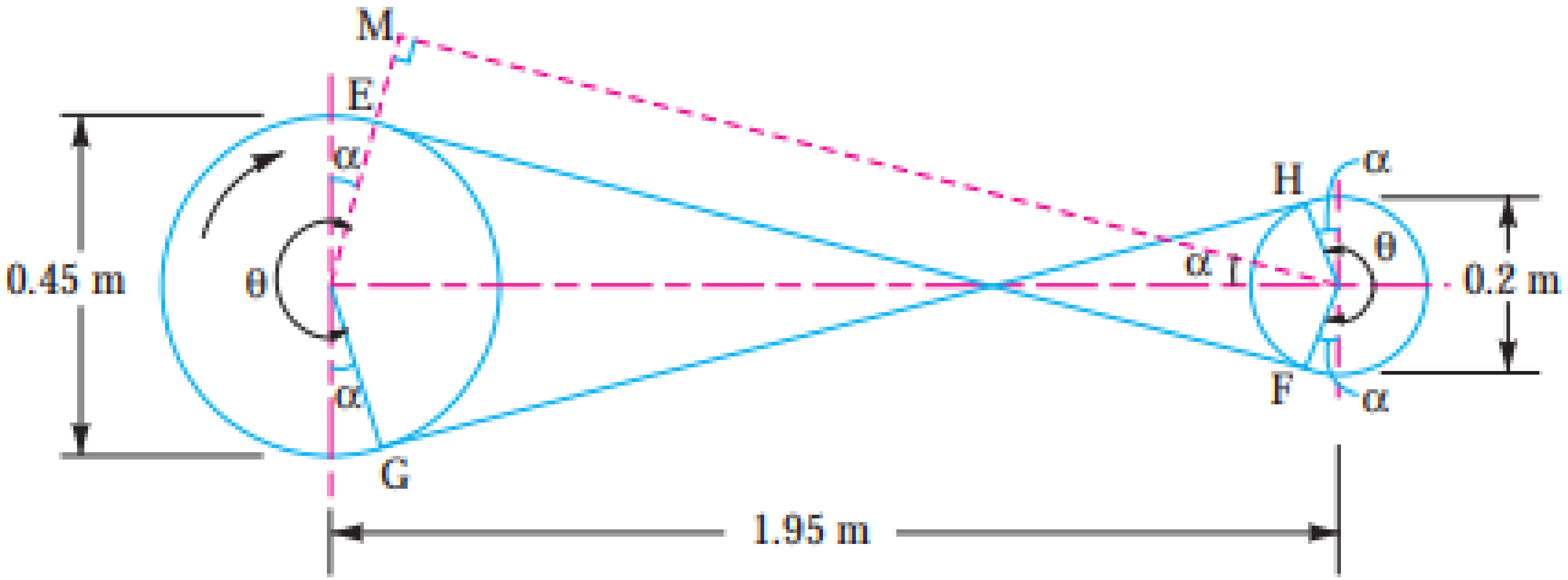
Crowning $h = 2.0$ mm for 200mm width & $D = 1200$ mm

Example 18.2. *Two pulleys, one 450 mm diameter and the other 200 mm diameter, on parallel shafts 1.95 m apart are connected by a crossed belt. Find the length of the belt required and the angle of contact between the belt and each pulley.*

What power can be transmitted by the belt when the larger pulley rotates at 200 rev/min, if the maximum permissible tension in the belt is 1 kN, and the coefficient of friction between the belt and pulley is 0.25?

Solution. Given : d_1 = 450 mm = 0.45 m or r_1 = 0.225 m ; d_2 = 200 mm = 0.2 m or r_2 = 0.1 m ; x = 1.95 m ; N_1 = 200 r.p.m. ; T_1 = 1 kN = 1000 N ; μ = 0.25

The arrangement of crossed belt drive is shown in Fig. 18.17.



Length of the belt

We know that length of the belt,

$$L = \pi (r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$$

$$= \pi (0.225 + 0.1) + 2 \times 1.95 + \frac{(0.225 + 0.1)^2}{1.95}$$

$$= 1.02 + 3.9 + 0.054 = 4.974 \text{ m}$$ **Ans.**

Angle of contact between the belt and each pulley

Let θ = Angle of contact between the belt and each pulley.

We know that for a crossed belt drive,

$$\sin \alpha = \frac{r_1 + r_2}{x} = \frac{0.225 + 0.1}{1.95} = 0.1667$$

$\therefore$ $\alpha = 9.6°$

and $\theta = 180° + 2\alpha = 180 + 2 \times 9.6 = 199.2°$

$$= 199.2 \times \frac{\pi}{180} = 3.477 \text{ rad}$$ **Ans.**

Power transmitted

Let T_1 = Tension in the tight side of the belt, and

T_2 = Tension in the slack side of the belt.

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta = 0.25 \times 3.477 = 0.8693$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.8693}{2.3} = 0.378 \quad \text{or} \quad \frac{T_1}{T_2} = 2.387 \qquad \text{... (Taking antilog of 0.378)}$$

$\therefore$
$$T_2 = \frac{T_1}{2.387} = \frac{1000}{2.387} = 419 \text{ N}$$

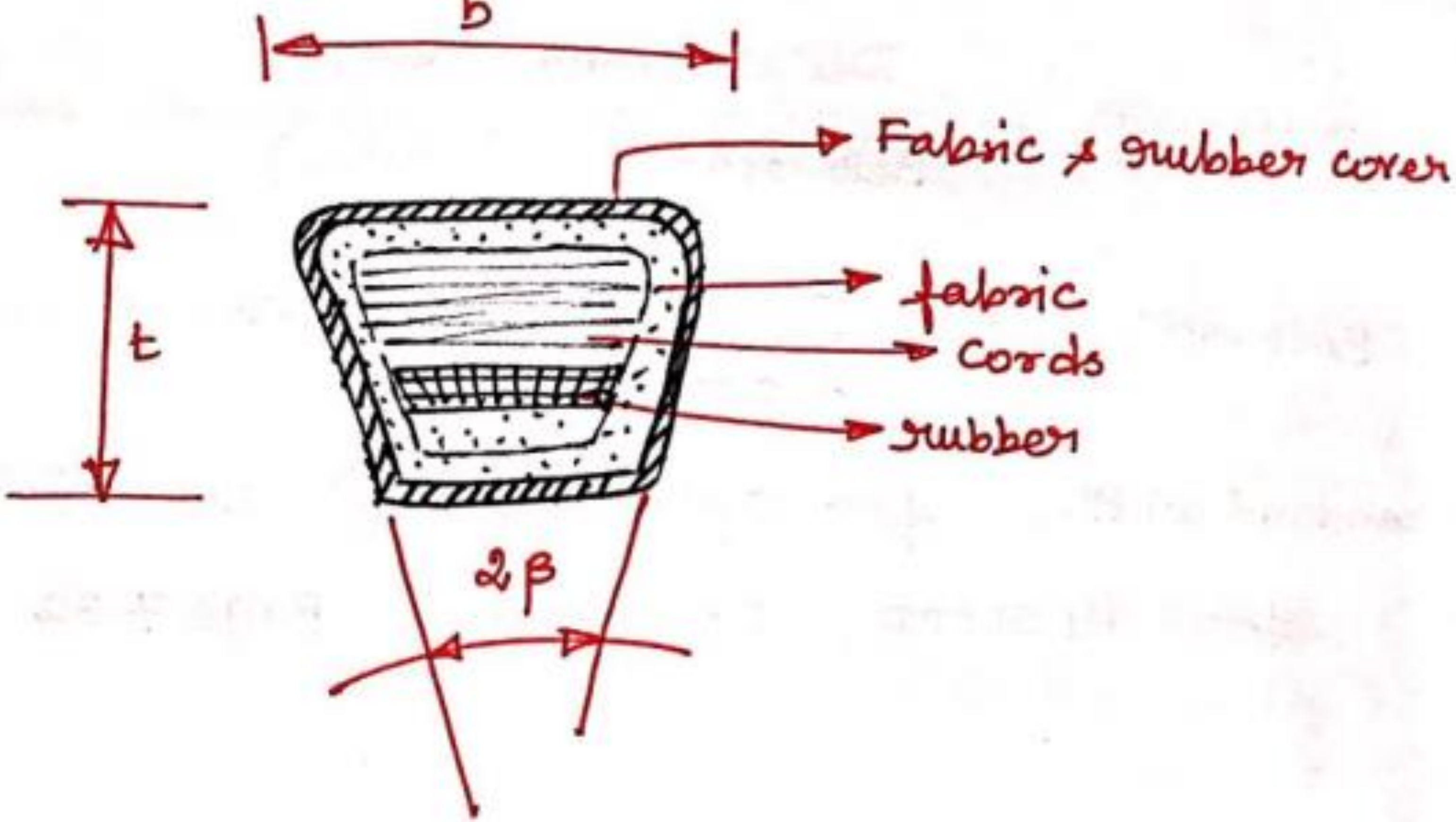
We know that the velocity of belt,

$$v = \frac{\pi\, d_1 N_1}{60} = \frac{\pi \times 0.45 \times 200}{60} = 4.713 \text{ m/s}$$

$\therefore$ Power transmitted,

$$P = (T_1 - T_2)\, v = (1000 - 419)\, 4.713 = 2738 \text{ W} = 2.738 \text{ kW}$$ **Ans.**

V– BELT DRIVE :



Design procedure - V-Belts:

Step i) **Selection of Belt type:**

From page 7.58 Based on given rated power value in kW; The belt type should be selected.

For Ex: $P = 28\ kW$, either 'C' or 'D', can
7.5–75 kW 22–150 kW

be selected;

The width & thickness value should be mentioned.

Step ii) **Calculation of Number of belts :**

Page 7.70 Number of belts, $= \dfrac{P \times F_a}{F_c \times F_d \times kW}$

F_a = Service factor Page 7.69, Based on application, duty time duration.

F_c = length correction factor Page 7.61

Based on length, $L = 2C + \dfrac{\pi}{2}(D+d) + \dfrac{(D-d)^2}{4C}$

standar length & factor values are in 7.58/7.59/7.60 Pages.

F_d - angle of contact factor (ACF), Based on AOC value

AOC - angle of contact.

Page 7.68 $\quad AOC = 180^\circ - \left(\frac{D-d}{c}\right)60^\circ$

Maximum power kW $\quad$ Page 7.62

Ex. for D type,

$$kW = \left(3.22\, S^{-0.09} - \frac{506.7}{d_e} - 4.78\times 10^{-4} S^2\right) S$$

de - equivalent pitch dia. for smaller pulley

$$d_e = d_p \times F_b$$

$d = d_p$ - pitch diameter, F_b - diametrical factor based on D/d ratio

Page 7.62

$$S = \text{speed} = v = \frac{\pi d n}{60} \text{ in } m/s$$

step iii) Corrected centre distance:

$$C = A + \sqrt{A^2 - B}$$

$$A = \frac{L}{4} - \pi \frac{D+d}{8}$$

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

from Page 7.61

$$C_{min} = 0.55(D+d) + T$$

$$C_{max} = 2(D+d)$$

T- nominal thickness from page 7.58.

Due to initial tension; 0.5% to 1% of 'L' should be stretched.

Step iv) calculation of tensions

$$\frac{T_1 - T_c}{T_2 - T_c} = e^{\mu\theta/\sin\beta} \quad ; \quad 2\beta = 40^\circ$$

$$T_t = T_1 + T_c$$

$$T_c = mv^2 \quad ; \quad m \rightarrow \text{step(i) in kgf}$$

Step v) stress calculation:

$$\sigma = \frac{T_c + T_1}{A}$$

$$A = \frac{1}{2}(W+b)T \quad ; \quad W, T \rightarrow \text{step (i)}$$

$$b = W - 2T\tan\beta$$

Problem:

A 30kW, 1440rpm motor is to drive a compressor by means of V-Belts. The diameter of pulleys are 220mm and 750 mm. The centre distance is 1440mm Design a suitable drive.

$P = 30\,kW$, $n = 1440\ rpm$

$d = 220\,mm$, $D = 750\,mm$, $C = 1440\,mm = 1.44\,m$

Step i) Belt selection: (Page 7.58)

Based on given rated power 30kW, 'C' type belt is selected.

width = 22 mm

thickness T = 14 mm

wt/meter = 0.343 kgf/m.length

Step ii) Calculation of number of Belts :

Page 7.70

$$\boxed{\text{number of belts} = \frac{P \times F_a}{kW \times F_c \times F_d}}$$

F_a - service factor - Page 7.69

$F_a = 1$ for compressor

F_c - length correction factor - Page 7.61

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

L - Nominal pitch Length

$$L = \frac{\pi}{2}(750+220) + \frac{(750-220)^2}{4\times1440} + 2\times1440$$

$$L = 4452.4 \text{ mm}$$

Page 7.60 For C-type nearest $L = 4450$ mm

Nominal inside length $L_i = 4394$ mm

$$F_c = 1.04$$

F_d - angle of contact factor Page 7.68

$$AOC = \theta = 180^\circ - \left(\frac{D-d}{C}\right)60^\circ$$

$$\theta = 180^\circ - \left(\frac{750-220}{1440}\right)60^\circ = 157.9 \simeq 158^\circ$$

$$F_d = 0.94$$

KW - Design maximum power Page 7.62

For c-type ;

$$KW = \left[1.47\,S^{-0.09} - \frac{142.7}{d_e} - 2.34 \times 10^{-4} \cdot S^2\right] S$$

$$S = \frac{\pi d n}{60} = \frac{\pi \times 220 \times 10^{-3} \times 1440}{60} = 16.59 \text{ m/s}$$

$$d_e = d_p \cdot F_b$$

$$d_p = d = 220 \text{ mm}$$

$F_b \rightarrow$ based on $D/d = \frac{750}{220} = 3.4$

$F_b = 1.14$ page 7.62

$$d_e = 220 \times 1.14 = 250.8 \text{ mm}$$

$$KW = \left[1.47(16.59)^{-0.09} - \frac{142.7}{250.8} - 2.34 \times 10^{-4} \cdot (16.59)^2 \right] \times 16.59$$

$$KW = 8.4312 \text{ kW}$$

$$\text{Number of belts} = \frac{30 \times 1}{8.4312 \times 1.04 \times 0.94}$$

$$\text{Number of belts} = 3.63 \simeq 4 \text{ no's.}$$

Step iii) Corrected centre distance:

$$C = A + \sqrt{A^2 - B}$$ Page 7.61

$$A = \frac{L}{4} - \pi\left(\frac{D+d}{8}\right) = \frac{4450}{4} - \pi\left(\frac{750+220}{8}\right)$$

$$A = 731.58 \text{ mm}$$

$$B = \frac{(D-d)^2}{8} = \frac{(750-220)^2}{8} = 35112.5 \text{ mm}$$

$$C = 731.58 + \sqrt{731.58^2 - 35112.5}$$

$$C = 1438.75 \text{ mm}.$$

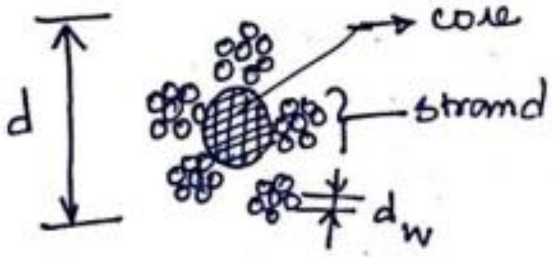
ROPE DRIVE

Why?

* High power, long distance (upto 150 m)
* Hoisting mechanism.

Ropes → fibre ropes - about 60 m.
Ropes → wire-steel ropes - about 150 m

fibres → hemp, manila, cotton & lubricants tar, tallow ang graphite.



Designation 6×7 rope.

6 → Stands

7 → wires

Design procedure:

Step i) selection of rope drive:

From page 9.1, Rope type is selected based on given application; Min. D/d should be mentioned.

for ex, for hoisting purpose,

6 x 19 rope is selected; class 4; $\frac{D_{min}}{d} = 27$

for 50 m/min speed.

Step ii) Calculation of Design load:

$$\text{Design load} = \text{load to be lifted} \times \underset{[n]}{\text{F.O.S}}$$

Recommended, $\text{F.O.S} = n' \times \text{duty factor}$

From page 9.1

Step iii) wire diameter & sheave/pulley diameter

Page 9.4, 9.5 & 9.6

From page 9.5, based on Design load,

diameter of rope is selected; (d) in mm,

from page 9.1, from $\frac{D_{min}}{d}$ ratio, for 50 mm/min

D_{min}, can be calculated for required Velocity.

$d_w = d/1.5\sqrt{i}$ (or) $d_w = 0.07\,d$ for (6×19 rope)

Step iv) Approximate weight of rope;

From page 9.3, 9.4, 9.5 & 9.6, For design load,

Wt / 1 m Length ; calculate, total wt = Wt/m × length (height to be lifted)

Step v) Calculation of loads:

i) tensile load $\quad F_t = W + w$

W – load acting

w – wt of rope

ii) Bending load:

$$\sigma_b = \frac{F_b}{A}$$

$$F_b = \sigma_b \times A = E_r \frac{d_w}{D} \cdot A \qquad ; A = 0.4 \frac{\pi}{4} d^2$$

$$E = 0.8 \times 10^5 \ N/mm^2$$

iii) Acceleration load :

$$F_a = F_t \times \frac{a}{g}$$; a-acceleration $= \frac{v}{t}$

Total load during acceleration $F = F_t + F_b + F_a$

$$F = \sigma . A$$

$$\sigma = \sigma_u / F.O.S$$

iv) Force at starting :

$$F_{st} = 2 F_t$$

Step v) Corrected Factor of safety for Normal working:

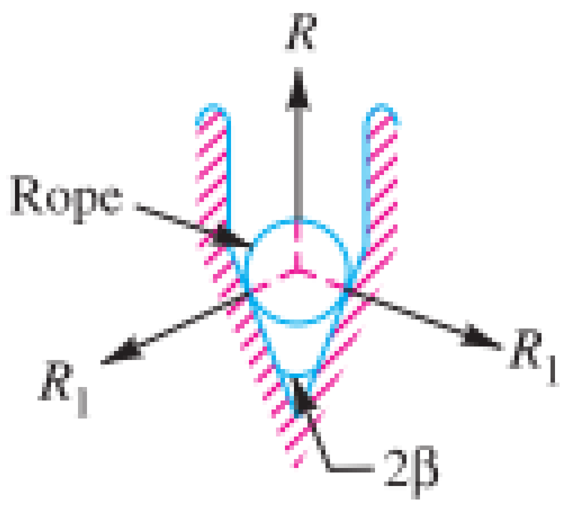
$$n;\ F.O.S = \frac{\text{Design load}}{F}$$

$$\therefore [n] > n$$

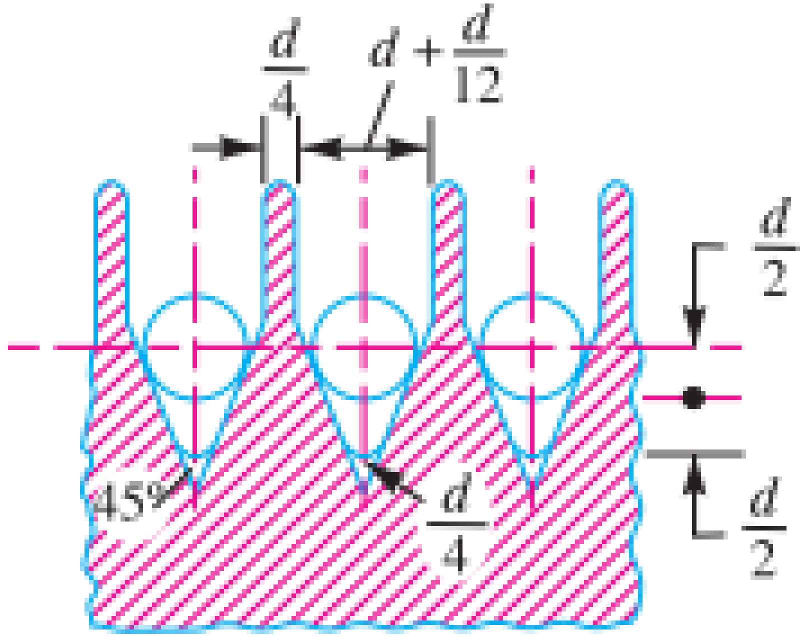
$$\text{No. of iopes} = \frac{[n]}{n}$$

Sheave for Fibre Ropes

- The fibre ropes are usually circular in cross-section as shown in Fig.



(a) Cross-section of a rope.



(b) Sheave (grooved pulley) for ropes.

Ratio of Driving Tensions for Fibre Rope

- A fibre rope with a grooved pulley is shown in Fig.
- The fibre ropes are designed in the similar way as *V*-belts.
- We have discussed in V-belt that the ratio of driving tensions is

$$2.3 \log \left(\frac{T_1}{T_2} \right) = \mu.\theta \operatorname{cosec} \beta$$

where μ, θ and β have usual meanings.

Pbm:1 *A pulley used to transmit power by means of ropes has a diameter of 3.6 m and has 15 grooves of 45° angle. The angle of contact is 170° and the coefficient of friction between the ropes and the groove sides is 0.28. The maximum possible tension in the ropes is 960 N and the mass of the rope is 1.5 kg per metre length. Determine the speed of the pulley in r.p.m. and the power transmitted if the condition of maximum power prevail.*

GIVEN DATA:

$d = 3.6$ m ;

$n = 15$;

$2\beta = 45°$ or $\beta = 22.5°$;

$\mu = 0.28$; $T = 960$ N ; $m = 1.5$ kg / m

Speed of the pulley

Let N = Speed of the pulley in r.p.m.

We know that for maximum power, speed of the pulley,

$$v = \sqrt{\frac{T}{3m}} = \sqrt{\frac{960}{3 \times 1.5}} = 14.6 \text{ m/s}$$

We also know that speed of the pulley (v),

$$14.6 = \frac{\pi d . N}{60} = \frac{\pi \times 3.6 \times N}{60} = 0.19\ N$$

$\therefore$ $N = 14.6 / 0.19 = 76.8$ r.p.m. **Ans.**

Power transmitted

We know that for maximum power, centrifugal tension,

$$T_C = T / 3 = 960 / 3 = 320 \text{ N}$$

$\therefore$ Tension in the tight side of the rope,

$$T_1 = T - T_C = 960 - 320 = 640 \text{ N}$$

Let T_2 = Tension in the slack side of the rope.

We know that

$$2.3 \log\left(\frac{T_1}{T_2}\right) = \mu.\theta \operatorname{cosec} \beta = 0.28 \times 2.967 \times \operatorname{cosec} 22.5° = 2.17$$

$$\therefore \quad \log\left(\frac{T_1}{T_2}\right) = \frac{2.17}{2.3} = 0.9435 \quad \text{or} \quad \frac{T_1}{T_2} = 8.78 \qquad \text{...(Taking antilog of 0.9435)}$$

and $$T_2 = T_1 / 8.78 = 640 / 8.78 = 73 \text{ N}$$

$\therefore$ Power transmitted,

$$P = (T_1 - T_2)\, v \times n = (640 - 73)\, 14.6 \times 15 = 124\,173 \text{ W}$$

$$= 124.173 \text{ kW}$$ **Ans.**

PROBLEM: 1

Select a wire rope for a vertical mine hoist to lift a load of 55 kN from a depth of 350m. A rope speed of 600 m/min is to be attained in 20 seconds.

$W = 55 \times 10^3$ N, $h = 350$ m, $v = 600$ m/min & $t = 20$ s

Step i) Selection of rope drive:

From page 9.1, for mine hoisting ;

6x19 rope is selected ;

$\frac{D_{min}}{d} = 27$ for class 4 ropes for v, 50m/min

for 600 m/min $= 50 + (11 \times 50)$

$= 27 + (11 \times 8\% \text{ of } 27)$

$\frac{D_{min}}{d} = (27) + (27 \times 0.08 \times 11) = 50.76$ mm/mm

Step ii) calculation of Design load:

$$\text{Design load } (P) = \text{load to be lifted} \times \text{F.O.S } [n]$$

$$\text{Recommended F.O.S} = [n] = n' \times \text{duty factor}$$

Page 9.1 ; $n' = 6$ for hoisting

From page 9.2

duty factor $= 1.6$ for class 4.

$$\text{F.O.S} = 6 \times 1.6 = 9.6 \simeq 10$$

$$\text{Design load} = 55 \times 10^3 \times (9.6 \simeq 10) = 55 \times 10^4 \text{ N}$$

$$\boxed{P = 550 \text{ kN} = 55 \text{ tonnes}}$$

Step iii) Wire & Rope diameter, sheave/Pulley diameter :

For 55 tonnes of Design load,

From page 9.5, for $\sigma_u = 160$ to $175\ kgf/mm^2$

Next to 55 Tonnes, 64.5 tonnes

dia of rope, $d = 35\,mm$

dia of wire $d_w = \dfrac{d}{1.5\sqrt{i}} = \dfrac{35}{1.5\sqrt{6 \times 19}} = 2.186\ mm$

Pulley dia, $D \Rightarrow \dfrac{D_{min}}{d} = 50.76$

$$D = 50.76 \times 35 = 1777\ mm$$

step iv> Approximate weight of rope :

From 9.5, for 64.5 tonnes,

$$W_t = 4.55 \ kg/_{1\,m} \text{ length}$$

total wt, $w = 4.55 \times h = 4.55 \times 350 = 1592.5$ kgf

$$\boxed{w = 15925 \ N}$$

Step v) Calculation of loads:

i) tensile load $(F_t) = W + w = 55 \times 10^3 + 15925$

$$F_t = 70925 \text{ N}$$

ii) Bending load $(F_b) = \sigma_b \cdot A$

$$= E_r \cdot \frac{d_w}{D} \cdot A$$

$E_r = 0.8 \times 10^6 \text{ kgf/cm}^2$; $d = 35 \text{mm}$

$A \simeq 0.4 \times \frac{\pi}{4} d^2$ in cm^2 $\quad d = 3.5 \text{ cm}$

$$F_b = 0.8 \times 10^6 \times \left(\frac{2.186}{1777}\right) \times 0.4 \times \frac{\pi}{4} (3.5)^2$$

$$F_b = 3787.37 \text{ kgf}$$

$$F_b = 37873.7 \text{ N}$$

iii) Acceleration load ; $(F_a) = F_t \times \frac{a}{g}$

$$a = \frac{v}{t} = \frac{600}{20 \times 60} = 0.5 \ m/s^2$$

$$g = 9.81 \ m/s^2$$

$$F_a = 70925 \times \frac{0.5}{9.81} = 3614.9 \ N$$

$$F = F_t + F_a + F_b = 70925 + 3614.6 + 37873.7$$

$$F = 112413.6 \ N = 112.4 \ KN$$

Step vi) Factor of safety

$$n = \frac{P}{F} = \frac{550 \text{ KN}}{112.4 \text{ KN}}$$

$$\boxed{n = 4.89}$$

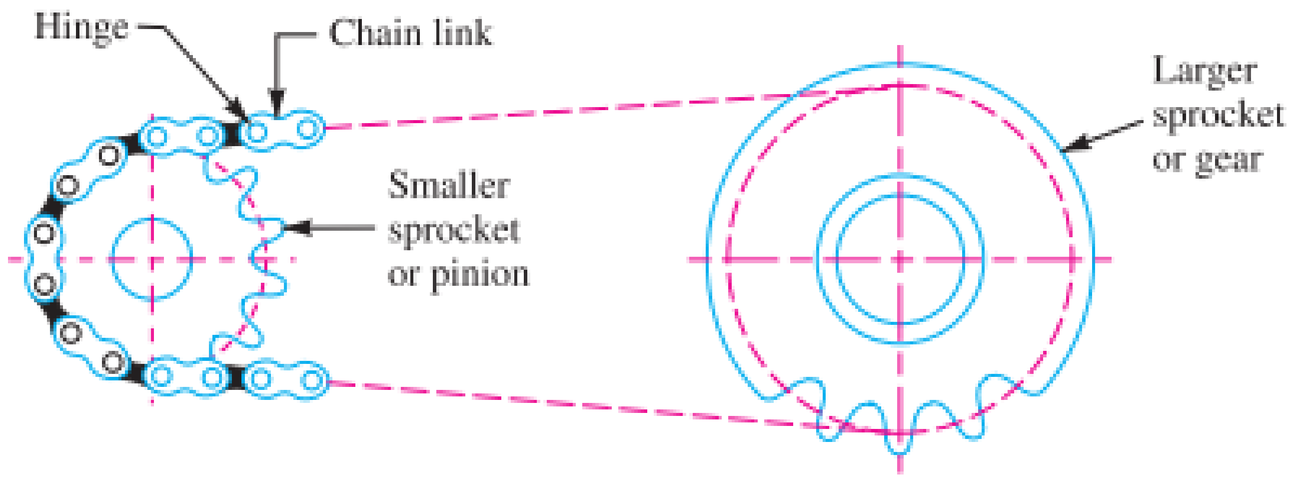
Factor of safety for Normal condition $n = 4.89$ is higher than the recommended FOS $[n] = 10$; so Design is safe.

$$\text{Number of ropes} = \frac{[n]}{n} = \frac{10}{4.89} = 2.044 \simeq 3 \text{ No's}$$

Chain Drive

In order to avoid slipping steel chains are used. The chains are made up of number of rigid links which are hinged together by pin joints

The toothed wheels are known as ***sprocket wheels or simply sprockets.*** The sprockets and the chain are thus constrained to move together without slipping and ensures perfect velocity ratio.



Design procedure:

Step i) calculation of Number of teeth: $(z_1 \& z_2)$

$$\text{Transmission ratio} = \frac{N_1}{N_2} = \frac{z_2}{z_1} = i$$

Page 7.74, Based on 'i' value, z_1 can be choosen & z_2 will be calculated.

Step ii) Calculation of standard pitch: (p)

Page 7.74,

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

$a = 30\, p_{max}$

$a = 50\, p_{min}$

in between p_{max} & p_{min}, p – standard pitch can be selected from page 7.72

Various standard 'p' : 6, 8, 9.525, 12.7, 15.875, 19.05, 25, 40, 31.75 ... etc

Step ii) Calculation of standard pitch : (p)

Page 7.74,

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

$$a = 30\, p_{max}$$

$$a = 50\, p_{min}$$

in between p_{max} & p_{min}, p - standard pitch can be selected from page 7.72

Various standard 'p' : 6, 8, 9.525, 12.7, 15.875, 19.05
25, 40, 31.75 ... etc

Step iii) SELECTION OF CHAIN

Based on pitch value, chain can be selected from Pages 7.71, 7.72 & 7.73.

Ex, for $p = 19.05 mm$

12A-1 chain is selected,

Area $A = 1.05 \ cm^2$

Wt per meter $W = 1.47 \ Kgf$

Breaking load $Q = 3200 \ Kgf$

Step iv) CHECKING FOR BREAKING LOAD :

$$\text{Power} = P = N = \frac{Q v}{102\, n\, k_s} \quad \text{in kW}$$

Page 7.77

$$v = \frac{z_1 N_1 p}{60} \quad \text{in m/s}$$

n = factor of safety from page 7.77

Based on speed of small sprocket

k_s - factors affecting power - $k_1 . k_2 . k_3 . k_4 . k_5 . k_6$

k_1 – Loading condn.

k_2 – supports

k_3 – centre distance

k_4 – Position of sprockets

k_5 – Lubrication

k_6 – Rating

Page 7.76 & 7.77

$$Q = \_\_\_\_ < [Q]$$

For safe design.

Step v) Calculation of safety factor :

Factor of safety $[n] = \dfrac{Q}{\Sigma P}$ Page 7.78

$\Sigma P = P_t + P_c + P_s$

→ Tangential force $P_t = \dfrac{Power}{v}$

→ centrifucal tension $P_c = \dfrac{Wv^2}{g}$

→ Tension due to sag $P_s = kWa$

W - wt pet meter length of chain

a - centre distance

k - Co-efficient of sag , Page 7.78

$[n] > n$

Step vi) Checking of Bearing stress:

Power $P = \frac{\sigma A v}{k_s}$; Page 7.77

$\sigma < [\sigma]$

A - Projected bearing area

σ - induced bearing stress

Step vii) length of chain :

From page 7.75

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

$a_p = \frac{a_0}{p}$; a_0 - centre distance (a)

$l = l_p \times p$ in (m)

Step viii) corrected centre distance:

From page 7.75

$$[a] = \left[\frac{e + \sqrt{e^2 - 8m}}{4}\right] \quad ; \quad m = \left(\frac{z_2 - z_1}{2\pi}\right)^2$$

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

allowance to accomodate initial chain sag;

$$\Delta = 0.5 f$$

$$f = 0.02[a]$$

Step ix) Other important specifications :

From Page 7.78,

i) Pitch diameters,

ii) outer diameters,

iii) roller seating radius

iv) root diameter

v) Tooth flank radius.

The transporter of a heat treatment furnace is given by a 4.5 kW, 1440 rpm induction motor through a chain drive with a speed reduction ratio of 2.4. The transmission is horizontal with bath type of lubrication, rating is continuous with three shifts per day. Design a chain drive for 700 mm centre distance.

Given: $N_1 = 1440$ rpm, $P = 45$ kW, $i = 2.4$ & $a = 700$ mm

Step i) calculation of Number of teeth.

$$i = \frac{N_1}{N_2} = \frac{D_2}{D_1} = \frac{Z_2}{Z_1}$$

for $i = 2.4$, range $(2-3)$; $Z_1 = 27$ to 25

Let $Z_1 = 25$ from page 7.74

$$Z_2 = iZ_1 = 2.4 \times 25 = 60$$

Step ii) Calculation of standard pitch:

From page 7.74, $a = (30 \text{ to } 50)\,p$

$$p_{min} = \frac{a}{50} = 700/50 = 14\,mm$$

$$p_{max} = \frac{a}{30} = 700/30 = 23.33\,mm$$

p is in the range from 14 to 23.3 mm

Let $p = 15.375$ mm   from page 7.72

Step iii) Calculation of Breaking load:

Power $N = \frac{Q\,v}{102\,\eta\,K_s}$

$N = 4.5\ kW$

$v = \frac{Z_1 N_1 P}{60} = \frac{25 \times 1440 \times 15.875 \times 10^{-3}}{60} = 9.525\ m/s$

K_s = Service factor $= K_1 . K_2 . K_3 . K_4 . K_5 . K_6$

K_1 - constant load = 1

K_2 - adjustable supports = 1

K_3 - $a = (30 \text{ to } 50)p$ = 1

K_4 - horizontal drive = 1

K_5 - Bath type of lubrication = 0.8

K_6 - 3 shifts (continuous running) = 1.5

$K_s = 1 \times 1 \times 1 \times 1 \times 0.8 \times 1.5 = 1.2$

n - factor of safety

Let $n = 13$ for around 1600 rpm speed

$$4.5 = \frac{Q \times 9.525}{102 \times 13 \times 1.2}$$

$$\boxed{Q = 751.75 \text{ kgf}}$$

Step iv) selection of chain:

From page 7.72,

I select 10A1 - R50 for $p = 15.875$ & $[Q] > 750$ kgf

roller dia = 10.16 mm

width = 9.55 mm

Bearing area = $0.7\ cm^2$

wt/m = 1.01 kgf

Breaking load = 2220 kgf

Step v) Actual factor of safety:

From Page 7.78,

$$[n] = \frac{Q}{\Sigma P} \; ; \quad \Sigma P = P_t + P_c + P_s$$

Tangential force $\quad P_t = \frac{102\, N}{v} = \frac{102 \times 4.5}{9.525} = 48.18 \text{ kgf}$

Centrifugal tension $\quad P_c = \frac{w v^2}{g} = \frac{1.01 \times 9.525^2}{9.81} = 9.34 \text{ kgf}$

Tension due to sagging $\quad P_s = k w a \Rightarrow$

$$k = 6 \text{ (horizontal drive)}$$

$$P_s = 6 \times 1.01 \times 700 \times 10^{-3} = 4.242 \text{ kgf}$$

$$[n] = \frac{2220}{(48.18 + 9.34 + 4.242)} = 35.9 \simeq 36 > \underset{n=13}{n}$$

Step vi) Checking for Bearing stress :

From Page 7.77

power $N = \dfrac{\sigma A v}{102\, k_s}$ in kW

$$4.5 = \frac{\sigma \times 0.7 \times 10^2 \times 9.52}{102 \times 1.5}$$

$\sigma = 0.8265$ kgf/mm^2 $< [\sigma]$ from page 7.77

$\sigma = 1.85$ kgf/mm^2

for 1600 rpm speed

step vii) calculation of length of the chain.

(Pitches) length $l_p = 2a_p + \frac{z_1+z_2}{2} + \frac{\left(\frac{z_2-z_1}{2\pi}\right)^2}{a_p}$

$$a_p = \frac{a_o}{p} = \frac{700}{15.875} = 44.09$$

$$l_p = 2 \times 44.09 + \left(\frac{25+60}{2}\right) + \frac{\left(\frac{60-25}{2\pi}\right)^2}{44.09}$$

$$l_p = 131.39 \simeq 132$$

$$l = l_p \times p = 132 \times 15.875 = 2095.5 \text{ mm}$$

From page 7.75

Step viii) Calculation of corrected centre distance.

From page 7.75

$$[a] = \frac{e + \sqrt{e^2 - 8m}}{4} \cdot p$$

Constant, $m = \left(\frac{z_2 - z_1}{2\pi}\right)^2$ (or) from 7.76

for $z_2 - z_1 = (60-25) = 35$, $m = 31.0$

$$e = l_p - \left(\frac{60+25}{2}\right) = 89.5 \text{ mm}$$

$\therefore [a] = 704.85$ mm

allowance $\Delta = 0.5 f = 0.5 \times (0.02a) = 0.5 \times 0.02 \times 704.85$

$f = 14.09$ mm ; $\Delta = 7.045$ mm

Step ix) Other specifications:

i) Pitch diameters, (Page 7.78)

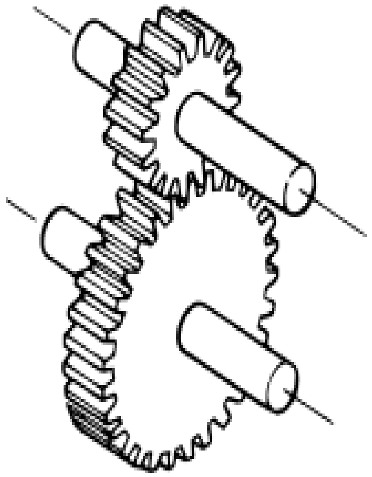
$d_1 = P/\sin\left(\frac{180}{z_1}\right) = 126.6\,mm$

$d_2 = P/\sin(180/z_2) = 303.3\,mm$

UNIT II
SPUR GEARS AND PARALLEL AXIS HELICAL GEARS

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

Spur Gear



Teeth are straight and parallel to shaft axis. Transmits power and motion between rotating two parallel shafts.

〔Features〕

(1) Easy to manufacture.

(2) There will be no axial force.

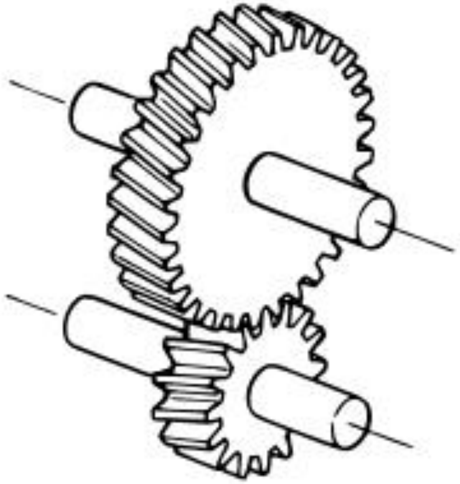
(3) Relatively easy to produce high quality gears.

(4) The commonest type.

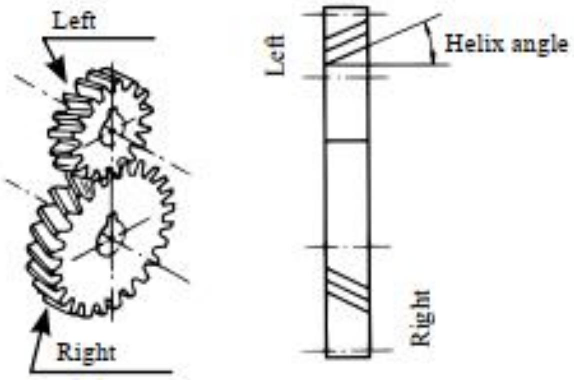
〔Applications〕

Transmission components

Helical Gear



Teeth are twisted oblique to the gear axis.



The hand of helix is designated as either left or right.

Right hand and left hand helical gears mate as a set. But they have the same helix angle.

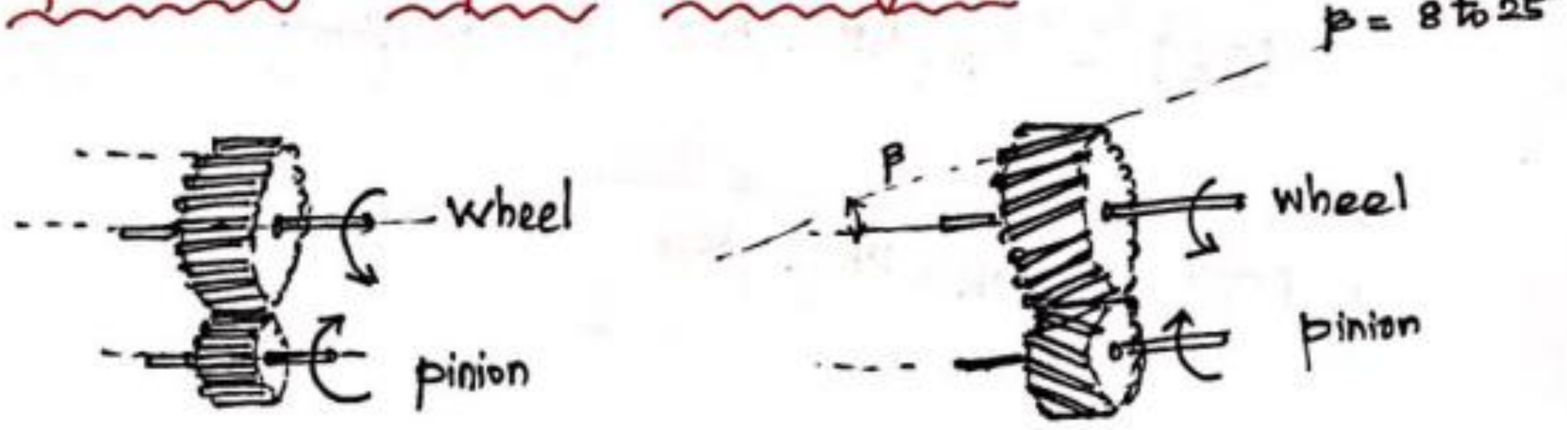
〔Features〕

(1) Has higher strength compared with spur gear.
(2) Effective in reducing noise and vibration compared with spur gear.
(3) Gears in mesh produce thrust forces in the axial directions.

〔Applications〕

Transmission components, automobile, speed reducers etc.

Design procedure - spur & helical gear:



SPUR.   HELICAL.

Step i) Material selection for pinion & wheel

When the materials are given, design for the material from page 8.5, for $[\sigma_b]$ & $[\sigma_c]$ design stresses.

case i) When life of the drive is not given:

a) Pinion & wheel are same material : Carbon steel C45

b) Pinion : C45 (or) 15Ni2Cr1Mo15 from page 8.5

wheel : cast iron grade 35

$[\sigma_b]$, $[\sigma_c]$ for pinion

$[\sigma_b]$, $[\sigma_c]$ for wheel, should be notified.

Pinion - High strength material - to withstand more loading cycles.

case ii) when life in hours is given;

$[\sigma_b]$ & $[\sigma_c]$ values for pinion & wheel should be calculated for the selected materials.

Page 8.16 :

$$[\sigma_c] = C_B . HB . K_{cl} , \quad kgf/cm^2$$

or

$$\boxed{[\sigma_c] = C_R . HRC . K_{cl}} , \quad kgf/cm^2$$

C_B (or) C_R - Co-efficients of Hardness; (table 16)

HB (or) HRC – Brinell (or) Rockwell Hardness number (table 16)

K_{cl} - life factor (table 17) based on hardness & life in number of cycles.

Step ii) Calculation of minimum Centre distance:

Spur:

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

Helical:

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

E- Equivalent Young's modulus; $E_{eq} = \dfrac{2E_1E_2}{E_1 + E_2}$

E_1 → Pinion, E_2 → wheel

Page 8.14:

$\psi = b/a$: from page 8.14

$\psi = 0.3$ spur

$\psi = 0.5$ Helical

$[\sigma_c]$ - Design surface stress, [crushing strength] Kgf/cm^2
for weaker material, (minimum value of $[\sigma_c]$, among pinion & wheel)

i - speed ratio | gear ratio

$[M_t] = M_t \cdot K_d \cdot k$; Design Twisting moment.

$$M_t = 71620 \cdot \frac{hp}{n} = 97420 \cdot \frac{kW}{n}$$; n - speed in rpm
kW - power

$K_d k$ = Initial assumption load factors = 1.3

$K_d K = 1.3$ (symmetric scheme)

Step iii) calculation of minimum module :

Page 8.13A)

Spur: $$m \geq 1.26 \sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m\, z_1}}$$

Helical: $$m_n \geq 1.15 \cos\beta \sqrt[3]{\frac{[M_t]}{y_v\,[\sigma_b]\,\psi_m\, z_1}}$$

$\psi_m = b/m = 10$, Page 8.14

$[\sigma_b]$ - bending strength, for weaker material,
(minimum value, amoung pinion & wheel)

Y_v - Form factor from page 8.18
Based on z_1 (no. of teeth on pinion)
& addendum modification X , (X = 0)

$[M_t] = M_t \cdot k_d k$

For helical, β = 8° to 25° ; Let $\beta = 15°$ (Intermittent)

<u>Spur</u>: $z_1 = 20$ (assume) ; $z_2 = i z_1$

<u>Helic.</u> : $z_v = z_1 / \cos^3\beta$; page 8.22

m - standard module can be selected from
Page 8.2

Step iv) Corrected centre distance: [a]

Page 8.22

Spur : $a = m \dfrac{z_1+z_2}{2}$

Helical : $a = \dfrac{m_n}{\cos\beta} \cdot \dfrac{z_1+z_2}{2}$

$[a] \geq [a]_{minimum}$ from (step ii)

step v) calculation of face width (b)

From contomt values ; ψ & ψ_m Page 8.14

$\psi = b/a = 0.3$ & 0.5 ; $b = \psi \cdot a$

for spur helical

$\psi_m = b/m = 10$; $b = \psi_m \cdot m$

select max. 'b' from above values.

Step vi) Load factors k_d & k ; Corrected Design twisting moment.

$$\underset{\text{actual}}{[M_t]} = M_t \cdot k_d k \qquad \rightarrow \text{From page 8.15}$$

k_d - from page 8.16, dynamic load factor, based on $v = \frac{\pi d_1 n_1}{60}$

k - from page 8.15, Load correction factor

based on $\psi_p = b/d_1$; $d_1 = m z_1$ from 8.22

Step vii) Checking for induced stresses:

Page 8.13:

Spur : $$\sigma_c = 0.74\,\frac{i \pm 1}{a}\sqrt{\frac{i \pm 1}{ib}\,E\,[M_t]} \leq [\sigma_c]$$

Helical : $$\sigma_c = 0.7\,\frac{i \pm 1}{a}\sqrt{\frac{i \pm 1}{ib}\,E\,[M_t]} \leq [\sigma_c]$$

Page 8.14 :

Spur : $$\sigma_b = \frac{i \pm 1}{amby}\,[M_t] \leq [\sigma_b]$$

Helical : $$\sigma_b = 0.7\,\frac{i \pm 1}{a\,b\,m_n\,y_v}\,[M_t] \leq [\sigma_b]$$

m, a, $[M_t]$ → corrected values should be used.

step viii) other specifications:

Spur

Helical

Tip diameter = $d_{a_1} = (z_1 + 2f_0)m$; $d_{a_1} = \left(\frac{z_1}{\cos\beta} + 2f_0\right)m_n$

$d_{a_2} = (z_2 + 2f_0)m$; $d_{a_2} = \left(\frac{z_2}{\cos\beta} + 2f_0\right)m_n$

; $f_0 = 1$

Root diameter = d_{f_1}, d_{f_2} d_{f_1}, d_{f_2}

$= (z_1, z_2 - 2f_0)m - 2c$; $= \left(\frac{z_1, z_2}{\cos\beta} - 2f_0\right)m_n - 2c$

$c = 0.25m$

Problem:1

Design a spur gear drive to transmit 22 kW at 900 rpm, speed reduction is 2.5, pressure angle 20°

i) Assume suitable material for pinion and wheel, Design the spur gear drive.

ii) Materials for pinion & wheel are C15 steel & C.I grade 30; working life of gears as 10,000 hours.

Given: $P = 22 \times 10^3$ W , $i = 2.5$, $N_1 = 900$ rpm , $\alpha = 20°$

Step i) Material selection for pinion & wheel.

I have selected the materials, for pinion – 15 Ni 2 Cr 1 Mo 15 steel and for wheel gear – C 45 steel ; From page 8.5

Pinion : 15 Ni 2 Cr 1 Mo 15 ; $[\sigma_b] = 3200\ kgf/cm^2$

$[\sigma_c] = 9500\ kgf/cm^2$

Wheel : C 45 steel : $[\sigma_b] = 1400\ kgf/cm^2$

$[\sigma_c] = 5000\ kgf/cm^2$

Step ii) Calculation of minimum centre distance:

Page 8.13

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

$i = 2.5$ (given)

$[\sigma_c] = 5000 \text{ kgf/cm}^2$ (minimum)

$E_{eq} = \frac{2E_1E_2}{E_1 + E_2}$; E_1 - for 15Ni2Cr1Mo15 steel

E_2 - for C45 steel

$E_1 = E_2 = 2.15 \times 10^6 \text{ kgf/cm}^2$

$E_q = 2.15 \times 10^6 \text{ kgf/cm}^2$

$\psi = 0.3$ from page 8.14;

$[M_t] = M_t \cdot k_d k$ from page 8.15;

$$M_t = 97420 \frac{kW}{n} = 97420 \times \frac{22}{900} = 2381.38 \text{ kgf.cm}$$

kW - power = 22 kW

n - speed = 900 rpm

$k_d k = 1.3$ from page 8.15;

$$[M_t] = 2381.38 \times 1.3 = 3095.8 \text{ kgf.cm}$$

$$\boxed{a \geq 20.275 \text{ cm}}$$

Step iii) Calculation of minimum module:

Spur: $$m \geq 1.26 \sqrt[3]{\frac{[M_t]}{y\,[\sigma_b]\,\psi_m\,z_1}}$$

$[\sigma_b] = 1400$ Kgf/cm² from minimum value.

$\psi_m = 10$ from page 8.14.

$z_1 = 20$ (assume) Number of teeth of pinion can be taken (18 to 30).

$[M_t] = 3095.8$ kgf.cm

$y = 0.389$ (for $z_1 = 20$ & $x = 0$)

$$m \geq 1.26 \sqrt[3]{\frac{3095.8}{0.389 \times 1400 \times 10 \times 20}}$$

$$m \geq 0.384 \text{ cm} \geq 3.84 \text{ mm} \simeq 4 \text{ mm}$$ standard module from page 8.2

Step iv) Corrected centre distance:

$$a = m\left(\frac{z_1+z_2}{2}\right)$$

For $m = 0.4\,cm$

$a = 0.4\left(\frac{20+50}{2}\right)$

$[a] = 14\,cm > (a)_{minimum}$

Not safe;

$\frac{z_2}{z_1} = 2.5$; $z_2 = 2.5 \times 20 = 50$

$$a = 0.4\left(\frac{20+50}{2}\right) = 14\,cm \quad \text{Not safe}$$

Select $m = 6$ mm from page 8.2

$$a = 0.6\left(\frac{20+50}{2}\right)$$

$$\boxed{[a] = 21\ cm} > a_{min} = 20.275\,cm$$

calculated centre distance $[a]$ is higher than minimum a value;

Step v) calculation of face width:

$$\varphi = b/a = 0.3 \quad ; \quad b = 0.3 \times 21 = 6.3\,cm$$

$$\varphi_m = b/m = 10 \quad ; \quad b = 10 \times .6 = 6.0\ cm$$

Select the maximum $\boxed{`b' = 6.3\ cm}$

Step vi) Calculating Design twisting Moment:

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

$M_t = 2381.38$ kgf.cm

K = from page 8.15 ; $\psi_p = b/d_1$

$d_1 = m z_1 = 0.6 \times 20 = 12$ cm from 8.22

$\psi_p = b/d_1 = \frac{6.3}{12} = 0.525 \simeq 0.6$

$K = 1.06$ for $\psi_p = 0.6$, symmetrical;

K_d = from page 8.16 ; $v = \frac{\pi d_1 n_1}{60} = \frac{\pi \times 0.12 \times 900}{60}$

$v = 5.65$ m/s

$K_d = 1.55$ $\left\{ \begin{array}{l} v \text{ upto } 8\,m/s \text{ , spur,} \\ HB \leq 350 \text{ Hardness} \\ \text{IS Quality 8, Cylinderical gear} \end{array} \right.$

$[M_t] = 2381.38 \times 1.06 \times 1.55 = 3912.6$ kgf.cm

Step vii) checking for induced stresses:

Crushing: $\sigma_c = 0.74 \frac{(i+1)}{a}\sqrt{\frac{(i+1)}{ib} E.[M_t]}$

$i = 2.5$

$[a] = 21\ cm$

$b = 6.3\ cm$

$E = 2.15 \times 10^6\ kgf/cm^2$

$[M_t] = 2381.38 \times 1.06 \times 1.55$

$[M_t] = 3912.6\ kgf.cm$

$$\sigma_c = 0.74 \frac{2.5+1}{21} \sqrt{\frac{2.5+1}{2.5 \times 6.3} \times 2.15 \times 10^6 \times 3912.6}$$

$$\sigma_c = 5332.44 \ kgf/cm^2 \quad > [\sigma_c]$$

Design is not safe; The induced stress is higher than the $[\sigma_c]$ strength.

so I can modify the prefered module (m) to next standard one, to make satisfactory design.

bending: $\sigma_b = \dfrac{i+1}{a\ m\ b\ y}[M_t]$

$$\sigma_b = \frac{2.5 + 1}{21 \times 0.6 \times 6.3 \times 0.389} \times 3912.6$$

$$\sigma_b = 443.48 \ \text{kgf/cm}^2 < [\sigma_b]$$

step viii) other specifications :

i) Tip diameter ; $d_{a_1} = (z_1 + 2f_0)m$; $f_0 = 1$

$$d_{a_1} = \left(20 + (2\times1)\right)0.6$$

$$\boxed{d_{a_1} = 13.2\,cm}$$

$$d_{a_2} = (z_2 + 2f_0)m = (50+2)\times 0.6$$

$$\boxed{d_{a_2} = 31.2\,cm}$$

ii) Root diameter $d_{f_1} = (z_1 - 2f_0)m - 2c$; $c = 0.25$, $f_0 = 1$

$$d_{f_1} = (20 - 2\times1)\,0.6 - (2\times0.25)$$

$$\boxed{d_{f_1} = 10.3\,cm}$$

$$d_{f_2} = (z_2 - 2f_0)m - 2c$$

$$d_{f_2} = (50 - 2\times1)\,0.6 - (2\times0.25)$$

$$\boxed{d_{f_2} = 28.3\,cm}$$

Helical Gears

- A helical gear has teeth in form of helix around the gear.

- Two such gears may be used to connect two parallel shafts in place of spur gears.

- The helixes may be right handed on one gear and left handed on the other.

29.5 Proportions for Helical Gears

Though the proportions for helical gears are not standardised, yet the following are recommended by American Gear Manufacturer's Association (AGMA).

Pressure angle in the plane of rotation,

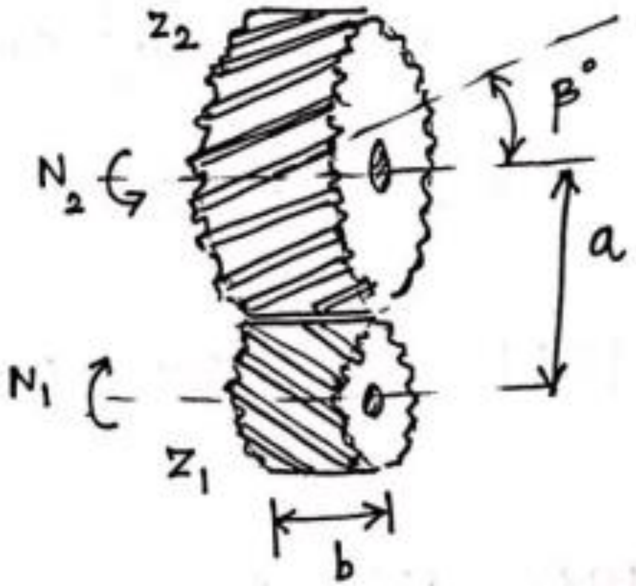
$$\phi = 15° \text{ to } 25°$$

Helix angle,	$\alpha = 20°$ to $45°$
Addendum	$= 0.8\ m$ (Maximum)
Dedendum	$= 1\ m$ (Minimum)
Minimum total depth	$= 1.8\ m$
Minimum clearance	$= 0.2\ m$
Thickness of tooth	$= 1.5708\ m$

A pair of helical gears for a turbine has a transmission ratio of 10:1. The pinion rotates at 5000 rpm & made of carbon steel & the gear wheel is made of high graded cast iron. Power transmitted is 90 kW; Select the suitable gear materials for the life of 12000 hrs.

$\beta = 8^\circ - 25^\circ$

Let $\beta = 15^\circ$



Step i> Material selection for 12000 hrs:

Pinion : C45

Gear : C.I.-35 grade

comparitively wheel material is weaker than pinion material; so let me calculate the strengths $[\sigma_c]$ & $[\sigma_b]$ for wheel - CI grade 35.

From page 8.16 ;

$$\boxed{[\sigma_c] = C_B \ HB \ k_{cl}} \quad ; \ kgf/cm^2$$

Hardness HB, for CI35; ... Page 8.16

$HB \Rightarrow 200$ to $260 \simeq 230$

$C_B = 23$

$k_{cl} \rightarrow$ for cast iron $\sqrt[6]{10^7/N}$

$N = 60nT = 60 \times 5000 \times 12000$

$N = 3.6 \times 10^9$ cycles $\Rightarrow k_{cl} = \sqrt[6]{10^7/3.6\times10^9}$

$k_{cl} = 0.375$

$[\sigma_c] = 23 \times 230 \times 0.375 = 1983.4 \ kgf/cm^2$

from page 8.18;

$$[\sigma_b] = \frac{K_{bl}}{n \cdot K_\sigma} \cdot \sigma_{-1} \qquad \ldots \text{Page 8.18}$$

$$K_{bl} = \sqrt[9]{10^7 / 3.6\times10^9} = 0.5199$$

F.O.S $n = 2.5$ (CI – No Heat treated)

$K_\sigma = 1.2$ (CI – $x=0$ (addendum modification))

$$\sigma_{-1} = 0.45\,\sigma_u = 0.45 \times 3600 = 1620 \text{ kgf/cm}^2$$

$$[\sigma_b] = 281 \text{ kgf/cm}^2$$

Step ii) Calculation of minimum centre distance:

Page 8.13

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

$E_{eq} = \dfrac{2E_1E_2}{E_1+E_2}$; E_1 for C45 = 2.1×10^6 kgf/cm²

E_2 for CI 35 = 1.4×10^6 kgf/cm²

$E_{eq} = 1.7 \times 10^6$ kgf/cm² ... Page 8.14

$i = 10$

$\psi = 0.5$ (Helical gears)

$[M_t] = M_t \cdot K_d K$ ... Page 8.15

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

... Page 8.15

$$M_t = 97420 \cdot \frac{kw}{n} = 97420 \times \frac{90}{5000} = 1753.56 \text{ kgf.cm}$$

$$[M_t] = 1753.56 \times 1.3 = 2279.63 \text{ kgf.cm}$$

$$a \geq (10+1) \sqrt[3]{\left[\frac{0.7}{1983.4}\right]^2 \times \frac{1.7\times10^6 \times 2279.63}{10 \times 0.5}}$$

$$\boxed{a \geq 50.46 \text{ cm}}$$

$$m_n \geq 1.15 \cos\beta \sqrt[3]{\frac{[M_t]}{y_v [\sigma_b] \psi_m z_1}}$$

$\psi_m = 10$ --- Page 8.14

z_1 ; Let assume $z_1 = 26$ --- Page 8.18

$$z_2 = i z_1 = 10 \times 26 = 260$$

$$z_v = \frac{z_1}{\cos^3\beta} = 28.84 \simeq 30$$ Next value from form-factor table

--- Page 8.22

y_v value for $z_v = 30$ & $x = 0$

$$y_v = 0.44$$ --- Page 8.18

$$m_n \geq 1.15 \times \cos 15^\circ \sqrt[3]{\frac{2279.63}{0.44 \times 281 \times 10 \times 26}}$$

$m_n \geq 0.459\ cm \simeq m_n = 5\ mm = 0.5\ cm$

--- Page 8.2

Step iv) calculation of Corrected Centre distance:

$$a = \frac{m_n}{\cos\beta}\left(\frac{z_1+z_2}{2}\right) \qquad \text{---- 8.22 page}$$

$m_n = 0.5$ cm

$z_1 = 26$; $z_2 = 260$

$a = 74.02$ cm $> [a]_{min}$

Design is satisfactory.

step v) calculation of face width :

--- Page 8.14

$\psi = 0.5 = b/a$; $b = 0.5 \times 74.02 = 37.01\,cm$

$\psi_m = 10 = b/m_n$ $b = 0.5 \times 10 = 5\,cm$

face width $= 37.01$ cm

step vi) calculation of corrected design torque $[M_t]$

$[M_t] = M_t \cdot K_d \cdot K$ --- Page 8.15

$M_t = 1753.56$ Kgf.cm

$K_d \Rightarrow$ Based on $v = \dfrac{\pi d_1 n_1}{60} \Rightarrow$ $d_1 = \dfrac{m_n z_1}{\cos\beta} = 13.46\,cm$

$v = 35.23$ m/s

K_d for maximum speed 1.3

K based on $\psi_p = b/d_1 = 37.01/13.46 = 2.74$

$K = 1.25$ for maximum ψ_p value.

$[M_t] = 1753.56 \times 1.3 \times 1.25$

$\boxed{[M_t] = 2849.5 \text{ kgf.cm}}$

Step vii) Checking for induced stresses:

Crushing stress: ... Page 8.13

$$\sigma_c = 0.7 \; \frac{i+1}{a} \sqrt{\frac{i+1}{i.b} \; E . [M_t]}$$

$$\sigma_c = 0.7 \; . \; \frac{10+1}{74.02} \sqrt{\frac{10+1}{10 \times 37.01} \times 1.7 \times 10^6 \times 2849.5}$$

$$\sigma_c = 1248.22 \text{ kgf/cm}^2 \quad < [\sigma_c]$$

$$1983.4 \text{ kgf/cm}^2$$

UNIT III
BEVEL, WORM AND CROSS HELICAL GEARS

Straight bevel gear: Tooth terminology, tooth forces and stresses, equivalent number of teeth. Estimating the dimensions of pair of straight bevel gears.

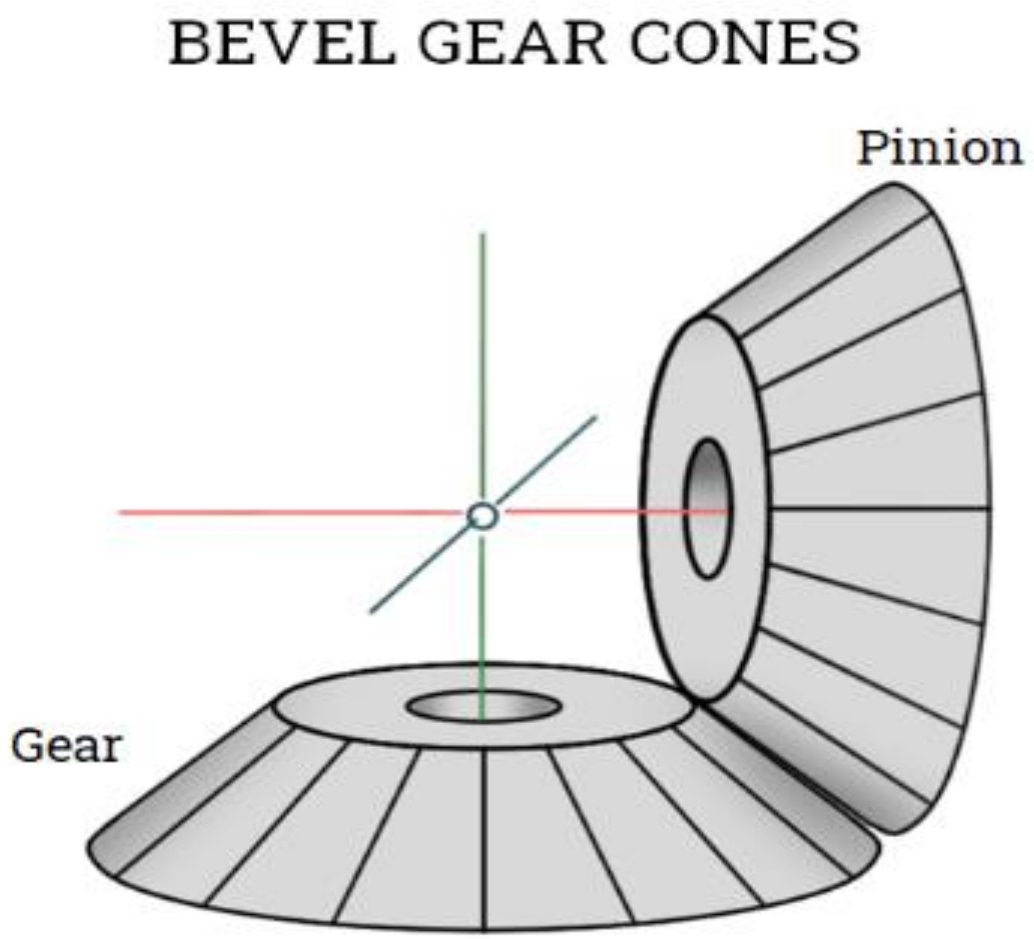
Worm Gear: Merits and demerits terminology. Thermal capacity, materials-forces and stresses, efficiency, estimating the size of the worm gear pair.

Cross helical: Terminology-helix angles-Estimating the size of the pair of cross helical gears.

Bevel Gear

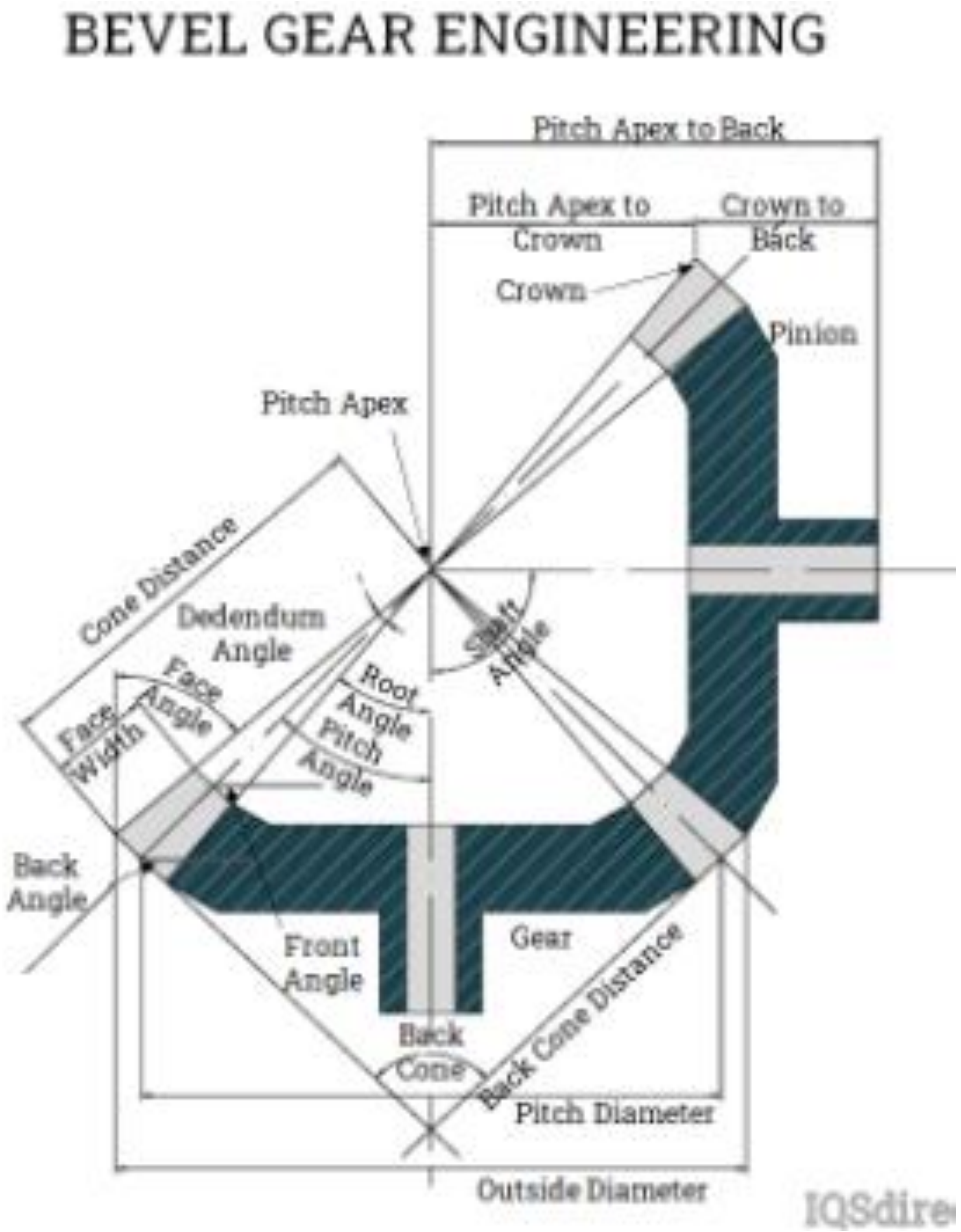
A bevel gear is a toothed rotating machine element used to transfer mechanical energy or shaft power between shafts that are intersecting, either perpendicular or at an angle.

This results in a change in the axis of rotation of the shaft power. A side from this function, bevel gears can also increase or decrease torque while producing the opposite effect on the angular speed.



Geometry and Terminologies

To better understand gears and gear systems, one must first look at its terminologies. Below are some of the terms used to describe gears and their tooth profile.



Pinion

The smaller bevel gear in a bevel gear set.

Gear

The larger bevel gear in a bevel gear set.

Pitch

Also known as circular pitch, is the distance from one point on a tooth to the corresponding point of the adjacent tooth on the same gear.

Pitch diameter

The diameter of the pitch circle. This is a predefined design dimension where other gear characteristics such as tooth thickness, pressure angles, and helix angles are determined.

Diametral pitch

The ratio of the number of teeth and the pitch diameter.

Pitch angle

The angle between the face of the pitch surface and the shaft axis.

Addendum

The upper outline of the gear teeth.

Dedendum

The bottom outline of the gear teeth.

Total depth

The radial distance between the addendum and dedendum circles of a gear. Note that the teeth of a bevel gear are slightly tapered, thus the total depth is not constant along the tooth. Because of this, the addendum and dedendum angles are used to describe the teeth instead of the addendum and dedendum circles.

Addendum angle

The angle between the face of the upper surface of the teeth or top land and the pitch surface.

Dedendum angle

The angle between the bottom surface of the teeth or bottom land and the pitch surface.

Problem 3.13 *Design a bevel - gear drive to transmit 7.5 kW. at 1440 rpm. Gear ratio is 3; pinion and gear are made of C45 steel; Life of gears 10,000 hrs.* *(Apr./May 2009)*

Step i) Selection of materials: (Page 8.16 – 8.20 PSG Data book)

For both pinion & wheel : C45 steel

Design Surface (contact compressive) Stress $[\sigma_c]$

$[\sigma_c] = C_B$ HB k_{cl}, kgf/cm²	...(5.1)	C_B or C_R coefficient depending on the surface hardness, from table 16
$[\sigma_c] = C_R$ HRC k_{cl}, kgf/cm²	...(5.2)	HB or HRC, Brinell or Rockwell 'C' hardness number
		k_{cl}, life factor from table 17

For C45 steel, $\boxed{[\sigma_c] = C_R \cdot HCR \cdot K_{cl}}$ From page 8.16

Page 8.16

Coefficients, C_B & C_R *Table 16*

Wheel Material	Heat Treatment	Surface Hardness	Coefficient C_B or C_R
Carbon Steels and Alloy Steels of any type	Normalised or Hardened and Tempered	HB $\leqslant$ 350	$C_B=25$
High strength Alloy Nickel Chromium Steels	Case Hardened	HRC=55 to 63	$C_R=310$
Alloy Steels	,,	,,	$C_R=280$
Carbon & Manganese Steels C15; C20; C15 Mn 85; C20 Mn 85	,,	,,	$C_R=220$
Alloy Steels, Carbon Steels C40; C45	Hardened & Tempered	HRC=40 to 55	$C_R=265$
,,	Surface Hardened	,,	$C_R=230$
Cast Iron, Grade 20, 25	—	HB=170 to 200	$C_B=20$
Cast Iron, Grade 30, 35	—	HB=200 to 260	$C_B=23$

For C45 steel,

$\boxed{C_R = 230}$

HRC = 40 to 55

$\boxed{\text{Let HRC} = 50}$

Page 8.17

Equivalent mean life

Condition	Formula	Notation
Constant Loading	$N = 60nT$ ...(6.1)	N, Life in number of cycles n, rpm T, Life in hours
Variable Loading	$N = \frac{60}{M^3_{u}} \sum M^3_{u} T_i n_i$ ...(6.2)	

Page 8.17

Life Factor for Surface (Contact Compressive) Strength $k_{cl} = \sqrt[6]{\frac{10^7}{N}}$ *Table 17*

Material	Surface Hardness, HB	Life in number of cycles	Life Factor, k_{cl}
Steel	≤ 350	$\geqslant 10^7$	1
		$< 10^7$	$\sqrt[6]{\frac{10^7}{N}}$
	> 350	$\geqslant 25 \times 10^7$	0.585
		$< 25 \times 10^7$	$\sqrt[6]{\frac{10^7}{N}}$
Cast Iron			$\sqrt[6]{\frac{10^7}{N}}$

Life in number of cycles,

$$\boxed{N = 60 \cdot n \cdot T}$$

$$N = 60 \times 1440 \times 10000$$

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

Let consider,

HB of C45, > 350 HB

$k_{cl} = 0.585$ for N

$86.4 \times 10^7 \geq 25 \times 10^7$

$$[\sigma_c] = k_{cl} \cdot HRC \cdot C_R = 0.585 \times 50 \times 230$$

$$\boxed{[\sigma_c] = 6727.5 \ kgf/cm^2}$$

$[\sigma_B]$ for C45 steel from Page 8.18

$$\boxed{[\sigma_B] = \frac{1.4 \, k_{bL}}{n \, k_\sigma} \cdot \sigma_{-1}}$$

Design Bending Stress (Tension) $[\sigma_b]$

Rotation in one direction only,

$$[\sigma_b] = \frac{k_{bl}}{n.k_\sigma} \sigma_o = \frac{1.4 \ k_{bl}}{n \ k_\sigma} \sigma_{-1} \quad \ldots(7.1)$$

Rotation in both directions,

$$[\sigma_b] = \frac{k_{bl}}{n.k_\sigma} \sigma_{-1} \quad \ldots(7.2)$$

k_{bl}, life factor for bending, table 22

k_σ, fillet stress concentration factor, table 21

$\sigma_o = 1.4 \ \sigma_{-1}$

σ_o, Endurance limit stress in bending for repeated stress, kgf/cm^2

σ_{-1}, Endurance limit stress in bending for complete reversal of stresses, kgf/cm^2, table 19

n, factor of safety, table 20

Value of σ_{-1} *Table 19*

Material of mating gear	σ_{-1}, Endurance limit stress in bending for complete reversal of stresses. kgf/cm²
Forged Steels	$\sigma_{-1}=0.25\ (\sigma_u+\sigma_y)+500$
Cast Steels	$\sigma_{-1}=0.22\ (\sigma_u+\sigma_y)+500$
Alloy steels	$\sigma_{-1}=0.35\ \sigma_u+1200$
Cast Iron	$\sigma_{-1}=0.45\ \sigma_u$

σ_u, ultimate tensile stress, kgf/cm²
σ_y, yield stress, kgf/cm²

From page 8.19,

For C45 (Forged Steel),

$$\boxed{\sigma_{-1} = 0.25(\sigma_u+\sigma_y)+500}$$

kgf/cm²

CARBON STEELS WITH SPECIFIED CHEMICAL COMPOSITION AND RELATED MECHANICAL PROPERTIES

Designation	% C	% Mn	Tensile strength kgf/mm²	% Minimum Elongation (Gauge length 5.65 $\sqrt{a^*}$, Cylindrical test piece)	Yield stress kgf/mm²	Izod impact value min. (If specified) kgf m	Brinell hardness (If specified) HB
C 07	0.12 max	0.50 max	32–40	27	20	—	—
C 10†	0.15 max	0.30–0.60	34–42	26	21	5.5	—
C 14†	0.10–0.18	0.40–0.70	37–45	26	22	5.5	137
C 15	0.20 max	0.30–0.60	37–49	25	24	—	137
C 15 Mn 75	0.10–0.20	0.60–0.90	42–50	25	25	—	163
C 20	0.15–0.25	0.60–0.90	44–52	24	26	—	156
C 25	0.20–0.30	0.30–0.60	44–54	23	28	—	170
C 25 Mn 75	0.20–0.30	0.60–0.90	47–57	22	28	—	207
C 30⊕	0.25–0.35	0.60–0.90	50–60	2[illegible]	30	5.5	179
C 35	0.30–0.40	0.30–0.60	52–62	20	31	—	187
C 35 Mn 75⊕	0.30–0.40	0.60–0.90	55–65	20	32	5.5	223
C 40⊕	0.35–0.45	0.60–0.90	58–68	18	33	4.1	217
C 45⊕	0.40–0.50	0.60–0.90	63–71	15	36	4.1	229
C 50⊕	0.45–0.55	0.60–0.90	66–78	13	38	—	241
C 50 Mn 1	0.45–0.55	1.10–1.40	72 min	11	40	—	255

From page 1.9

$\sigma_u = 63 - 71$ kgf/mm²

$\sigma_y = 36$ kgf/mm²

Let $\sigma_u = 70$ kgf/mm²

$\sigma_u = 7000$ kgf/cm²

$\sigma_y = 3600$ kgf/cm²

$$\therefore \sigma_{-1} = 0.25(7000 + 3600) + 500 = 3150 \text{ kgf/cm}^2$$

Factor of Safety, n *Table 20*

Material	Mode of Manufacture	Heat Treatment	Factor of Safety, n
Steel, Cast Iron	Cast	No heat treatment	2.5
		Tempered or Normalised	2.0
Steel	Cast or Forged	Case Hardened	2.0
	Forged	Surface Hardened	2.5
		Normalised	2.0

From page 8.19,

For C45 (forged & surface hardened)

F.O.S, $\boxed{n = 2.5}$

From page 8.19,

For C45, surface hardened, $0 \leq x \leq 0.1$

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

Stress Concentration Factor for the Fillet, k_σ *Table 21*

Material and Heat Treatment	Addendum Modification Coefficient, X		
	X<0	0 ≤ X ≤ 0.1	X>0.2
Steel, Normalised, Surface Hardened	1.4	1.5	1.6
Steel Case Hardened	1.1	1.2	1.3
Cast Iron	1.2	1.2	1.3

From page 8.20;

Life Factor for Bending, k_{bl} *Table 22*

Material	Surface Hardness HB	Life in Number of Cycles, N	k_{bl}
Steel	$\leqslant 350$	$\geqslant 10^7$	1
		$< 10^7$	$\sqrt[9]{\frac{10^7}{N}}$
	> 350	$\geqslant 25\times10^7$	0.7
		$< 25\times10^7$	$\sqrt[9]{\frac{10^7}{N}}$

For, Steel (C45)

Core hardness, < 350 HB

Life $86.4 \times 10^7 \geq 10^7$

$$\boxed{k_{bl} = 1}$$

$$\therefore [\sigma_b] = \frac{1.4\, k_{bl}}{n \cdot k_\sigma} \cdot \sigma_{-1} = \frac{1.4 \times 1}{2.5 \times 1.5} \times 3150 = 1176\ \frac{kgf}{cm^2}$$

Step ii) Calculation of Cone distance (minimum):

From page 8.13,

$$R = \psi_y \cdot \sqrt{i^2+1} \cdot \sqrt[3]{\left[\frac{0.72}{(\psi_y - 0.5)[\sigma_c]}\right]^2 \frac{E[M_t]}{i}}$$

Cylindrical Gears		Bevel Gears	
FOR DESIGNING			
Spur		Straight and Spiral	
$a \geq (i \pm 1) \sqrt[3]{\left(\frac{0.74}{[\sigma_c]}\right)^2 \frac{E[M_t]}{i\psi}}$	...(1.1)	$R \geq \psi_y \sqrt{i^2+1} \sqrt[3]{\left(\frac{0.72}{(\psi_y - 0.5)[\sigma_c]}\right)^2 \frac{E[M_t]}{i}}$	...(1.3)
Helical and Herringbone			
$a \geq (i \pm 1) \sqrt[3]{\left(\frac{0.7}{[\sigma_c]}\right)^2 \frac{E[M_t]}{i\psi}}$	...(1.2)		

$$R = 2\psi_y \cdot \sqrt{i^2+1} \cdot \sqrt[3]{\left[\frac{0.72}{(\psi_y-0.5)[\sigma_c]}\right]^2 \frac{E[M_t]}{i}}$$

i = speed ratio/gear ratio

i = 3 (given)

$\psi_y = R/b$ (From page 8.15)

Type of gear Transmission			$\psi_y = \frac{R}{b}$
Speed Reducers	a. Housed in roller bearings	$i = 1$ to 4	3
		$i = 4$ to 6	4
	b. Housed in Journal & Thrust bearings	$i = 6$	5

$\psi_y = 3$ (for $i = 1$ to 4)

$[\sigma_c]$ = From step ①, in kgf/cm²

$[\sigma_c] = 6727.5$ kgf/cm²

E – Eq. young's modulus

For Steel C45, $E = 2.16 \times 10^6$ kgf/cm²

From page 8.14

$[M_t] = M_t \cdot k_d k$ from page 8.15;

Design Twisting Moment, $[M_t]$

$[M_t] = M_t \, k_d \, k$ ...(4.0)

Initially assume for symmetric scheme

$k_d \, k = 1.3$ ...(4.1)

For unsymmetric and over-hanging scheme

$k_d \, k = 1.5$ ...(4.2)

$M_t = 71620 \dfrac{hp}{n} = 97420 \dfrac{kW}{n}$ ...(4.3)

M_t, nominal twisting moment transmitted by the pinion, kgf. cm

hp, nominal horsepower transmitted

n, speed of rotation of pinion, rpm

k_d, dynamic load factor, table 15

kW, nominal power transmitted in kW

M_t = Twisting moment $= 97420 \cdot \dfrac{kW}{N} = 97420 \times \dfrac{7.5}{1440} = 507.3$ kgf. cm

$K_d \cdot K = 1.3$ (symmetric)

$[M_t] = 507.3 \times 1.3 = 659.61$ $kg_f \cdot cm$

$$R = \cancelto{3}{\psi_y} \cdot \sqrt{\cancelto{3}{i}^2+1} \cdot \sqrt[3]{\left[\frac{0.72}{(\cancelto{3}{\psi_y}-0.5)\,\cancelto{(6727.5\ \frac{kgf}{cm^2})}{[\sigma_c]}}\right]^2 \frac{\cancelto{(2.15\times10^6\ kgf/cm^2)}{E}\;\cancelto{(659.61\ kgf.cm)}{[M_t]}}{\cancelto{3}{i}}}$$

$$\boxed{R = 9.043\ cm}$$ (minimum cone distance)

Step iii) Calculation of minimum module :

From page 8.38,

Nomenclature	Symbol	Formulae for straight and spiral bevel gears
transverse module	m_t	$m_t = m_m \frac{\Psi_y}{\Psi_y - 0.5} = m_m + \frac{b \sin \delta}{Z}$
mean module	m_m	
normal module	m_n	$m_n = m_t \cos \beta_m$
Cone distance	R	$R = 0.5\, m_t\, Z_1 \sqrt{i^2+1} = 0.5\, m_t \sqrt{Z_1^2+Z_2^2}$
		$= \frac{m_t Z_1}{2 \sin \delta_1} = \frac{m_t Z_2}{2 \sin \delta_2}$

Transverse module m_t ⇒ $R = 0.5\, m_t\, Z_1 \sqrt{i^2+1}$

R - Cone distance in Cm

Z_1 - No. of. teeth on pinion

i - gear ratio

$m_t \Rightarrow$ $\boxed{R = 0.5\, m_t\, Z_1 \sqrt{i^2+1}}$

Assume;
$Z_1 \Rightarrow 18$ to 28

$$\cancelto{9.043\text{ cm}}{R} = 0.5\, m_t\, \cancelto{\text{let } 20}{Z_1} \sqrt{\cancelto{3^2}{i^2}+1}$$

$m_t = 0.2859$ cm $\simeq 2.9$ mm

Standard module from page 8.2

$\boxed{\text{Let } m_t = 4\text{ mm}}$

Preferred (I)
1
1.25
1.5
2
2.5
3
4
5
6
8
10
12
16
20

Step iv) Corrected Cone distance

From page 8.38,

Nomenclature	Symbol	Formulae for straight and spiral bevel gears
transverse module	m_t	$m_t = m_m \frac{\Psi_y}{\Psi_y - 0.5} = m_m + \frac{b \sin \delta}{Z}$
mean module	m_m	
normal module	m_n	$m_n = m_t \cos \beta_m$
Cone distance	R	$R = 0.5\, m_t\, Z_1 \sqrt{i^2+1} = 0.5\, m_t \sqrt{Z_1^2 + Z_2^2}$
		$= \frac{m_t Z_1}{2 \sin \delta_1} = \frac{m_t Z_2}{2 \sin \delta_2}$

$$R = 0.5\, \cancelto{0.4\,cm}{m_t}\, \cancelto{20}{Z} \sqrt{\cancelto{3^2}{i^2} + 1} = 0.5 \times 0.4 \times 20 \times \sqrt{3^2 + 1} = 12.649\ cm$$

$$R = 12.649\ cm > [R_{min} = 9.043\ cm]$$

Design is Satisfactory:

Step v) Calculation of average module : (m_{av})

Mean / Average module m_{av} (m_m) $\Rightarrow$ $m_m = \left(m_t - b\dfrac{\sin\delta_1}{z_1}\right)$

from page 8.13A

Bevel Gears

FOR DESIGNING

Straight

$$m_{av} \geq 1.28\sqrt[3]{\frac{[M_t]}{y_v[\sigma_b]\psi_m Z_1}} \quad \ldots(2.1)$$

$$m_t = m_{av} + \frac{b}{Z}\sin\delta \quad \ldots(2.3)$$

mean / average module, $m_m = m_t - \dfrac{b}{z_1}\sin\delta_1$

face width $b \Rightarrow$ $\psi_y = R/b$; $3.0 = \dfrac{12.649 \text{ cm}}{b}$

$b = 4.216$ cm

$\psi_y = 3$
from step (2)

$$\left.\begin{aligned} \tan\delta_2 &= i \\ \delta_1+\delta_2 &= 90^\circ \end{aligned}\right\}$$ from page 8.39 ; δ_1 & δ_2 - reference angles

$$\tan\delta_2 = i = 3 \Rightarrow \delta_2 = \tan^{-1}(3) = 71.56^\circ$$

$$\delta_1 = 90^\circ - \delta_2 = 90 - 71.56 = 18.43^\circ$$

$$m_{av} = m_m = \left(\cancelto{0.4\,cm}{m_t} - \frac{\cancelto{4.216cm}{b}}{\cancelto{20}{z_1}} \sin \cancelto{18.43^\circ}{\delta_1}\right) = 0.333cm$$

$$\boxed{m_{av} = 3.33\ mm}$$

Step vi) Corrected Design torque $[M_t]$

For page 8.15; $\boxed{[M_t] = M_t k_d k}$

$M_t = 507.3$ kgf.cm (step 2)

k - load Correction factor page 8.15

Based on $\boxed{\psi_p = b/d_1}$; $b = 4.216$ cm

$d_1 = z_1 m_m = 20 \times 0.333$

$d_1 = 6.667$ cm

$\psi_p = 4.216/6.667 = 0.633 \leq 1$

Bevel Gears

Surface hardness of Gears HB	b/d$_{av}$ ratio ≤1	1 to 1.6	1.6 to 1.8
>350 for both the Gears	1.6	—	—
≤350 for both the gears or at-least for wheel	1.1	1.2	1.3

$\boxed{\text{so } k = 1.6}$

k_d - dynamic load factor from page 8.16

$$V = \frac{\pi d_1 n_1}{60} = \frac{\pi \times 0.0667 m \times 1440 rpm}{60} = 5.026 \ m/s$$

Page 8.3

Peripheral Speed of Gears *Table 2*

IS Quality	Preferred Quality	Speed of gears in metres per second: Cylindrical Gears	Straight Bevel	Spiral Bevel
High Precision 3 & 4	4	Above 15	upto 9	upto 18
Precision 5 & 6	6	Above 8 & upto 15	„ 6	„ 12
Medium 7, 8 & 9	8	Above 1 & upto 8	„ 3	„ 7
Coarse 10 & 12	10, 12	upto 1	„ 2	„ 4

Page 8.16

Dynamic Load Factor, k_d *Table 15*

IS quality: Cylindrical gear	IS quality: Conical gear	Pinion Surface Hardness HB	Spur & Straight bevel (Pitch line velocity, m/s, upto) 1.0	3.0	8.0	12.0	Helical & Spiral Bevel (Pitch line velocity, m/s, upto) 3.0	8.0	12.0	18.0
5	—	⩽ 350	—	—	1.2	1.4	—	1	1.1	1.2
		> 350	—	—	1.2	1.3	—	1	1.0	1.1
6	5	⩽ 350	—	1.25	1.45	—	1	1	1.2	1.3
		> 350	—	1.2	1.3	—	1	1	1.1	1.2
8	6	⩽ 350	1	1.35	1.55	—	1.1	1.3	1.4	—
		> 350	—	1.3	1.4	—	1.1	1.2	1.3	—
10	8	⩽ 350	1.1	1.45	—	—	1.2	1.4	—	—
		> 350	—	1.4	—	—	1.2	1.3	—	—

For $V = 5.02 \ m/s$; IS quality $= 6$

$$\boxed{k_d = 1.4}$$

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

$$= 507.3 \times 1.4 \times 1.6$$

$$\boxed{[M_t] = 1136.5 \ kgf.cm}$$

Step vii) checking for stresses

From page 8.13 & 8.13 A

FOR CHECKING

Spur $\sigma_c = 0.74 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{ib} E[M_t]} \leq [\sigma_c]$ ...(1.4)	Straight and Spiral $\sigma_c = \frac{0.72}{(R-0.5b)} \sqrt{\frac{\sqrt{(i^2+1)^3}}{ib} E[M_t]}$ ..(1.6)
Helical and Herringbone $\sigma_c = 0.7 \frac{i \pm 1}{a} \sqrt{\frac{i \pm 1}{ib} E[M_t]} \leq [\sigma_c]$ ...(1.5)	

FOR CHECKING

Spur $\sigma_b = \frac{i \pm 1}{a\, m\, b\, y} [M_t] \leq [\sigma_b]$ ..(2.5)	Straight $\sigma_b = \frac{R\sqrt{i^2+1}\, [M_t]}{(R-0.5b)^2\, b\, m\, y_v} \cdot \frac{1}{\cos\alpha} \leq [\sigma_b]$ ...(2.8)
Helical $\sigma_b = 0.7 \frac{i \pm 1}{a\, b\, m_n\, y_v} [M_t] \leq [\sigma_b]$ ...(2.6)	Spiral

$$\sigma_c = \frac{0.72}{(R-0.5b)} \sqrt{\frac{(i^2+1)^{3/2} \cdot E[M_t]}{ib}}$$

$$\sigma_b = \frac{R\sqrt{i^2+1} \cdot [M_t]}{(R-0.5b)^2\, b\, m\, y_v \cdot \cos\alpha}$$

$$\sigma_c = \frac{0.72}{\left(\cancelto{12.649\,cm}{R} - 0.5\,\cancelto{4.216\,cm}{b}\right)} \sqrt{\frac{\left(\cancelto{3^2}{i^2}+1\right)^{3/2} \cdot \cancelto{2.15\times10^6\,kgf/cm^2}{E}\,\cancelto{1136.5\,kgf.cm}{[M_t]}}{\cancelto{3}{i}\,\cancelto{4.216\,cm}{b}}}$$

$$\boxed{\sigma_c = 5338 \ \ kgf/cm^2 \ \leq \ [\sigma_c] = 6727.5 \ kgf/cm^2}$$

from step 1

Design is satisfactory.

$$\sigma_b = \frac{\overset{12.649\,cm}{R}\sqrt{\overset{3^2}{i^2}+1}\cdot\overset{1136.5\,kgf\cdot cm}{[M_t]}}{(\underset{12.649\,cm}{R}-0.5\underset{4.216\,cm}{b})^2\,\underset{4.216\,cm}{b}\,\overset{0.4\,cm}{m_t}\,y_v\cdot\cos\overset{20^\circ\text{ (pressure angle)}}{\alpha}}$$

y_v = form factor from page 8.18, based on Z_v ;

$$Z_v = \frac{Z_1}{\cos\delta_1} = \frac{20}{\cos 18.43} = 21.08 \simeq 22$$

$Z_v = 22$, $y_v = 0.402$

Form Factor, y for $\alpha = 20°$ and $f_o^* = 1$

(y includes cos α term)

Z or Z_v	External Pinion, Wheel & Internal Pinion: Addendum Modification Coefficient, X								Annulus
	−0.6	−0.4	−0.2	0	+0.2	+0.4	+0.6	+1.0	
12	—	—	0.239	0.308	0.378	—	—	—	—
14	—	—	0.266	0.330	0.392	0.458	—	—	—
16	—	—	0.302	0.355	0.408	0.461	—	—	—
18	—	—	0.330	0.377	0.424	0.470	—	—	—
20	—	—	0.348	0.389	0.431	0.471	0.513	—	—
22	—	—	0.367	0.402	0.437	0.473	0.509	—	—

$$\sigma_b = \frac{\overset{12.649\,cm}{\cancel{R}}\sqrt{\overset{3^2}{\cancel{i^2}}+1}\ \cdot \left[\overset{1136.5\,kgf\cdot cm}{\cancel{M_t}}\right]}{(\underset{12.649\,cm}{\cancel{R}}-0.5\underset{4.216\,cm}{\cancel{b}})^2\ \underset{4.216\,cm}{\cancel{b}}\ \overset{0.4\,cm}{\cancel{m_t}}\ \underset{0.402}{\cancel{y_v}}\cdot \cos\overset{20^\circ\,(\text{pressure angle})}{\cancel{\alpha}}}$$

$$\sigma_b = 642.401\ kgf/cm^2 \leq [\sigma_b] = 1176\ kgf/cm^2$$

step ①

UNIT IV GEAR BOXES

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

What is Gearbox?

- An automobile requires high torque when climbing hills and when starting, even though they are performed at low speeds.

- On other hand, when running at high speeds on level roads, high torque is not required because of momentum. So requirement of a device is occur, which can change the vehicle's torque and its speed according to road condition or when the driver need.

- This device is known as transmission box.

Introduction to Gearbox

- Gearbox often referred as transmission is a unit that uses gears and gear trains to provide speed and torque conversions from a rotating power source to another device. Gearboxes are employed to convert input from a high speed power sources to low speed(E.g. Lift, Cranes and Crushing Machine) or into a many of speeds(Lathe, Milling Machine and Automobiles).

- A gearbox that converts a high speed input into a single output it is called a single stage gearbox. It usually usually has two gears and shafts.

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

automobile

- The transmission box which is also known as the gear box is the second element of the power train in an automobile. It is used to change the speed and torque of vehicle according to variety of road and load condition. Transmission box change the engine speed into torque when climbing hills and when the vehicle required. Sometimes it is known as torque converter. Main functions of a gear box is as follow:

1. Provide the torque needed to move the vehicle under a variety of road and load conditions. It does this by changing the gear ratio between the engine crankshaft and vehicle drive wheels.

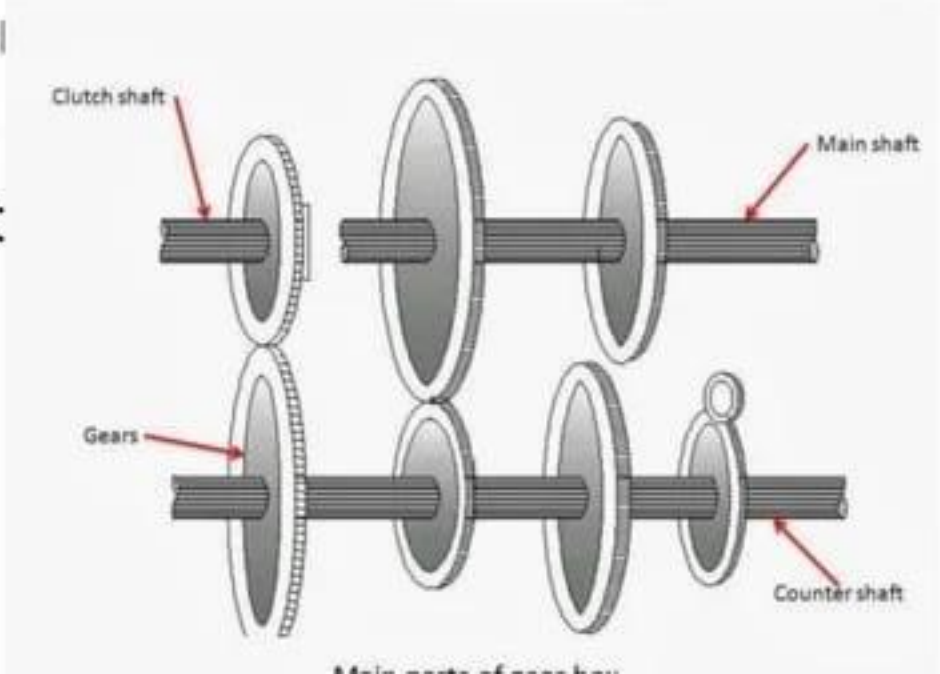
2. Be shifted into reverse so the vehicle can move backward.

3. Be shifted into neutral for starting the engine.

Main components of a gear box

- In any device two or more component works together and fulfills the required function. In a transmission box four components are required to fulfill its fu

❖Counter shaft
❖Main shaft
❖Gears
❖Bearings



1. Counter shaft:

- Counter shaft is a shaft which connects with the clutch shaft directly. It contains the gear which connects it to the clutch shaft as well as the main shaft. It may be run runs at the engine speed or at lower than engine speed according to gear ratio.

2. Main shaft:

- It is the shaft which runs at the vehicle speed. It carries power form the counter shaft by use of gears and according to the gear ratio, it runs at different speed and torque compares to counter shaft. One end of this shaft is connects with the universal shaft.

3. Gears:

- Gears are used to transmit the power form one shaft to another. They are most useful component of transmission box because the variation is torque of counter shaft and main shaft is depend on the gear ratio.

- The gear ratio is the ratio of the driven gear teeth to the driving gear teeth. If gear ratio is large than one, the main shaft revolves at lower speed than the counter shaft and the torque of the main shaft is higher than the counter shaft. On other hand if the gear ratio is less than one, than the main shaft revolves at higher speed than the counter shaft and the torque of the main shaft is lower than the counter shaft. A small car gear box contains four speed gear ratio and one reverse gear ratio.

4.* Bearings*:

- Whenever the rotary motion, bearings are required to support the revolving part and reduce the friction. In the gear box both counter and main shaft are supported by the bearing.

Working of a principle gear box

- In a gear box, the counter shaft is mashed to the clutch with a use of a couple of gear. So the counter shaft is always in running condition. When the counter shaft is bring in contact with the main shaft by use of meshing gears, the main shaft start to rotate according to the gear ratio.
- When want to change the gear ratio, simply press the clutch pedal which disconnect the counter shaft with engine and change connect the main shaft with counter shaft by another gear ratio by use of gearshift lever.
- In an gear box, the gear teeth and other moving metal must not touch. They must be continuously separated by a thin film of lubricant. This prevents excessive wear and early failure. Therefor a gearbox runs partially filled with lubricant oil.

Laws of stepped regulation of speeds in multi-speed gearbox

- Arithmetic Progression
- Geometric Progression
- Harmonic Progression

	Parameter		Progression	
1	Definition	In arithmetic progression, the difference between any two successive spindle speeds is constant.	In geometric progression, the ratio of any two successive spindle speeds is constant.	In harmonic progression, the difference between reciprocal of any two successive speeds is constant.
2	Z^{th} Spindle Speed	$n_z = \frac{n_{max} - nmin}{(z-1)}$	$\emptyset = \left[\frac{n_{max}}{n_{min}}\right]^{\frac{1}{z-1}}$	$nz = \frac{n_{min}}{[1-(z-1)Cn_{max}]}$
3	Good in	High spindle speed range	High spindle speed range	Low spindle speed range
4	Poor in	Low spindle speed range	Low spindle speed range	High spindle speed range

Design procedure of gear box (sliding gear type)

A. For designing a stepped drive

- Following informations are necessary

- ✓ Highest output speed
- ✓ Lowest output speed
- ✓ Number of steps (Z)
- ✓ Number of stages to achieve the required number of speed steps.

B. Break up of speed steps

- The number of steps (Z) should be so selected that it can be broken into the multiples of 2 & 3. Thus, selected values of Z are : 6,8,9,10,12,14,16 & 18.

C. Structural diagram

- It gives the information about
  - Number of shafts in the speed box
  - Number of gears on each shaft
  - The order of changing transmission in individual groups to obtain the desired speed.
  - Transmission range
  - Group characteristics

While drawing the structural diagram, following points should be considered

a) Number of gears on the last shaft should be as minimum as possible.

b) The speed reduction between the spindle and preceding shaft should be as maximum as possible.

c) Number of gears on any shaft should not be more than three. It can be four in exceptional case.

d) $i_{max} \times imin = 1$ is for least radial dimensions of gear box. This is possible by making the axes of adjacent shafts coincident i.e., co axial

- $i_{max} \times imin = 1$ is possible when maximum speed reductions equals the maximum speed increase.

- But considering the importance of reduction of axial dimensions of gear in machine tool with a traversing spindle head, the above point does not favour it, because small axial dimensions of traversing units are critical.

- Structural formula represented by the special form of graphs is called as structural diagram.

Method of drawing structural diagrams

1. If n= no of transmission groups then draw(n+1) vertical lines at a convenient distance. Here the first vertical line represents the transmission from motor shaft, and the rest represents the transmission group of speed box.

2. Draw an any of horizontal lines intersecting the vertical lines at a distance of logØ from each other. The number of horizontal lines are equal to the number of speed steps(Z). The spacing between the horizontal line should be equal so that interval between the spindle speeds is content. In practice the distance between adjacent horizontal lines is taken equal to Ø, but not log for convenience.

Cont...

3. Draw a line joining the first shaft of known speed & the second shaft for calculated input speed. The speed reduction between the first and second shaft is usually through belt derive.

4. From second shaft at the input speed point of diverging lines joining the third shafts. The number of lines will be equal to the number of transmission. The maximum spacing between the lines on third shaft, will be according to the calculated transmission range between these two shafts say ϕ^1 ϕ^2 ϕ^4 ϕ^6 ......etc.

5. From the third shafts for all groups draw the diverging lines, having maximum spacing on the fourth shaft as per calculated value of transmission range of groups.

Speed chart

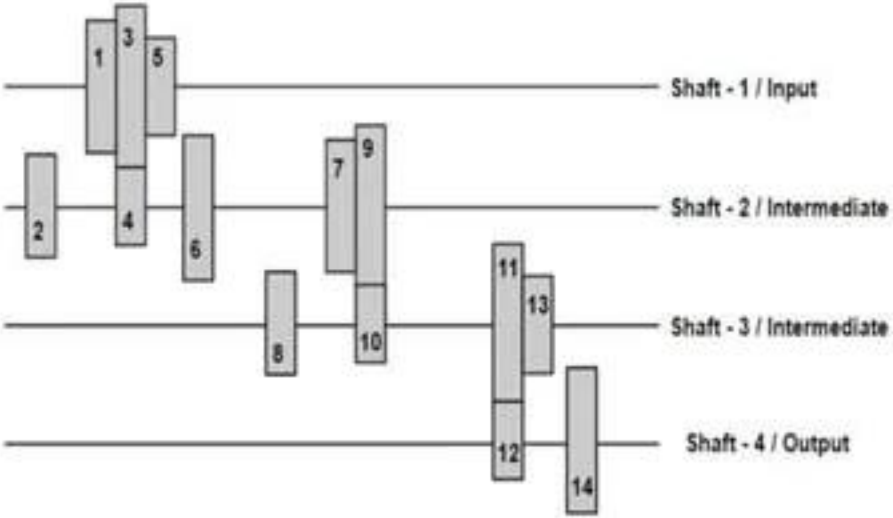
- The speed chart depicts the transmission ratios. The structural diagram depicts only range ratio so speed chart must be plotted to depict the transmission ratios

- Horizontal line corresponds to transmission ratio, i = 1. (no speed)
- Line inclined upwards corresponds to transmission ratio, I >1 (increase in speed)
- Line inclined downwards corresponds to transmission ratio, i <1 (decrease in speed)

- While plotting the speed chart it is desirable to have minimum transmission ratio i.e maximum speed reduction in the last transmission group. The remaining shafts run at relatively higher speed and so subjected to less torques

Kinematic Diagram

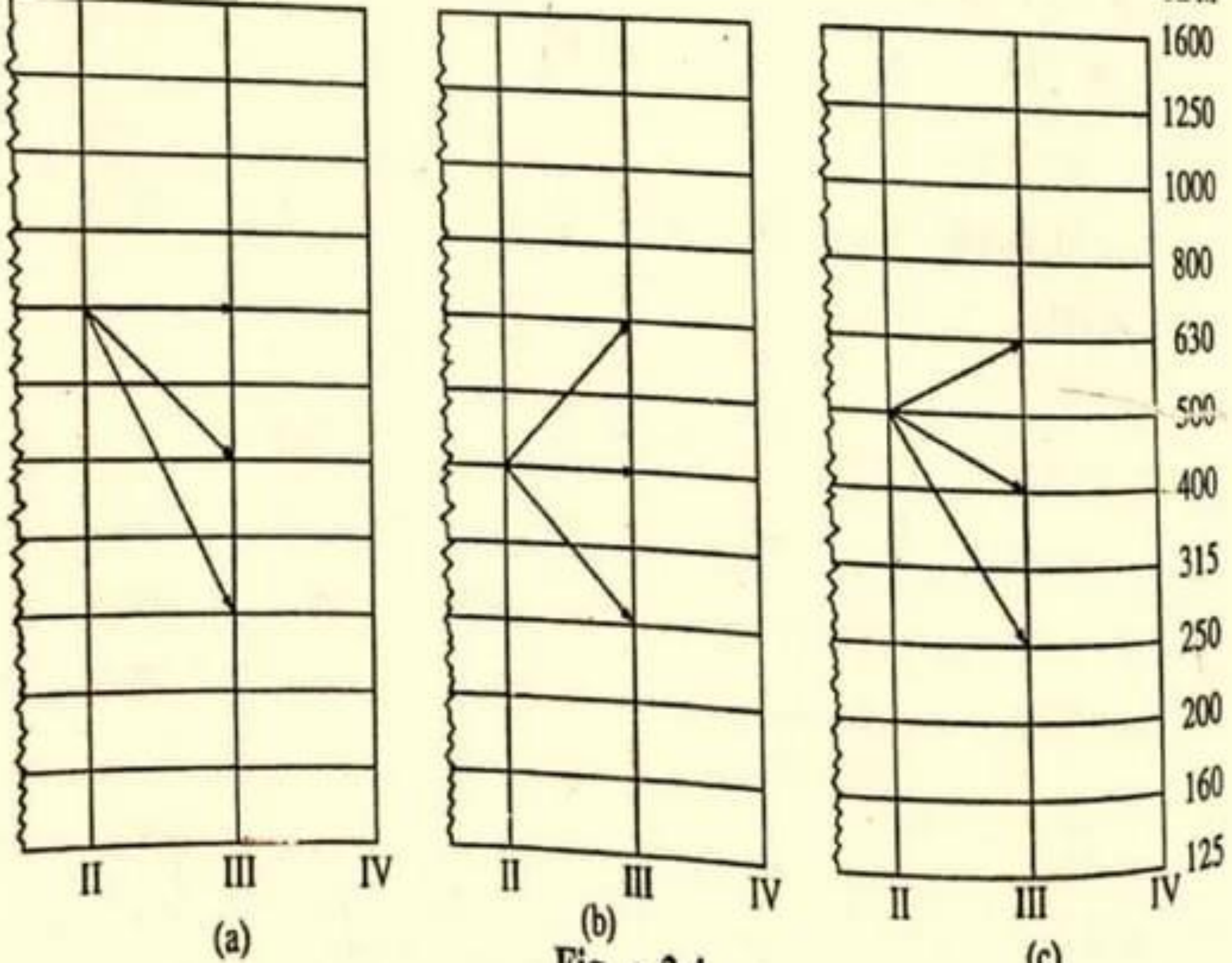
- A kinematic layout is a pictorial representation of gearbox, describing the arrangement of gears.
- It provides information like number of stages, number of shafts used, number of gear pairs and its arrangement.



Ray diagram

- A ray diagram is a representation of structural formula. It provides information such as speed in each stage, the transmission ratio in each stage, The total number of speeds and its values.

- As seen in fig. (a) the maxi speed and minimum speed both are higher for shaft. This requires smaller size of shafts due to reduced torque. But as fig(b) the maximum speed and minimum speed both are lower which requires lager size of shaft due to increased torque. The version indicated in fig(c) is the middle situation. One will definitely prefer the version(b) , but if the machine cost of shafts is not a criterion , then version(a) is preferred.


1600
1250
1000
800
630
500
400
315
250
200
160
125
II
III
IV
II
III
IV
II
III
IV
(a)
(b)
(c)

Example

- Design a gear box for a head stock to give 16 speeds ranging from 50 rpm to 1600 rpm. The power is supplied by an electric motor of 10 kw, running at 1440 rpm, through a V-belt drive with a speed reduction of 2:1

Find (I) No. of teeth on each gears.
(II) Percentage variation in speed.

Solution

1. Selection of standard speed

$N_{max} = 1600$
$N_{min} = 50$
$Z = 16$

$$\emptyset = \left[\frac{N_{max}}{N_{min}}\right]^{\frac{1}{Z-1}}$$

$$\emptyset = \left[\frac{1600}{50}\right]^{\frac{1}{16-1}}$$

$$= 1.259 \approx 1.25$$

2. From $\emptyset = 1.25$, the standard speeds are

50, 63, 80, 100, 125,
160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250, 1600 rpm

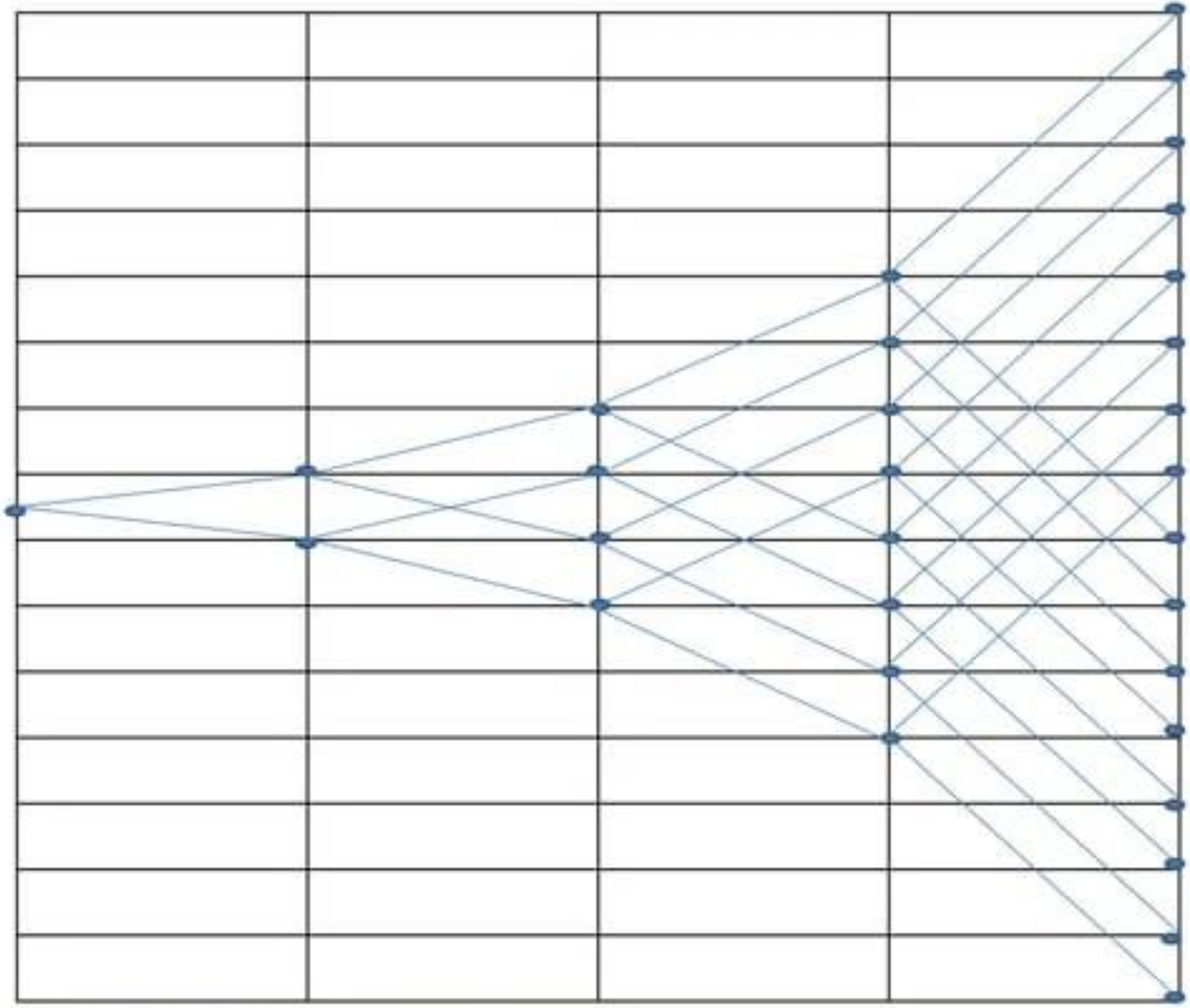
Cont…

3. Structural Diagram

here $Z=16 = 2*2*2*2$
$P_1=P_2=P_3=P_4=2$
here $X_1=1$, $X_2=P_1=2$, $X_3=P_1P_2=4$
$X3=P_1P_2P_3=8$

$\therefore$ Structural Formula

$Z= 2(1)\ 2(2)\ 2(4)\ 2(8)$



Cont...

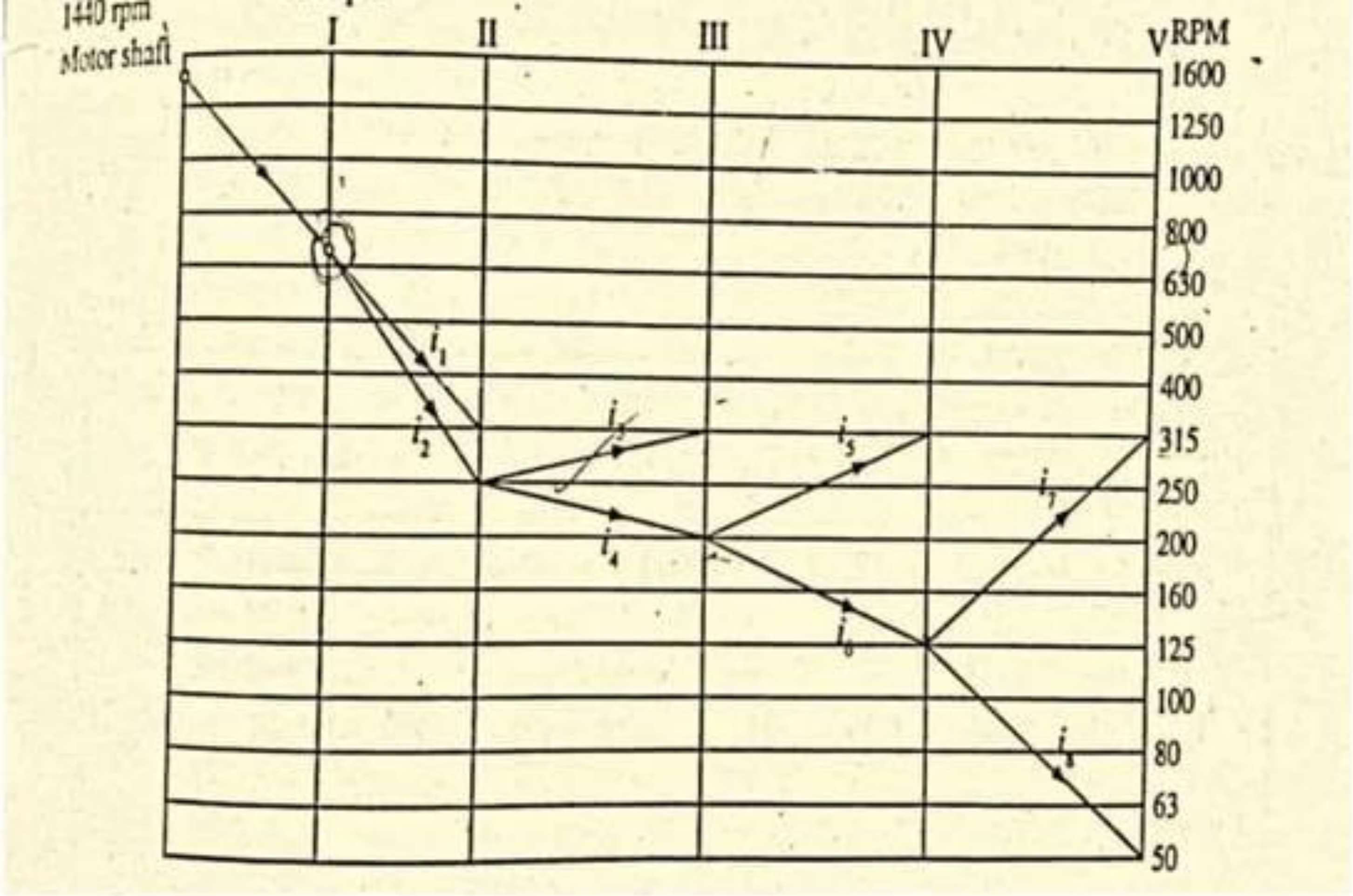
4. Speed Chart

Here the power is supplied to input shaft through a belt

drive

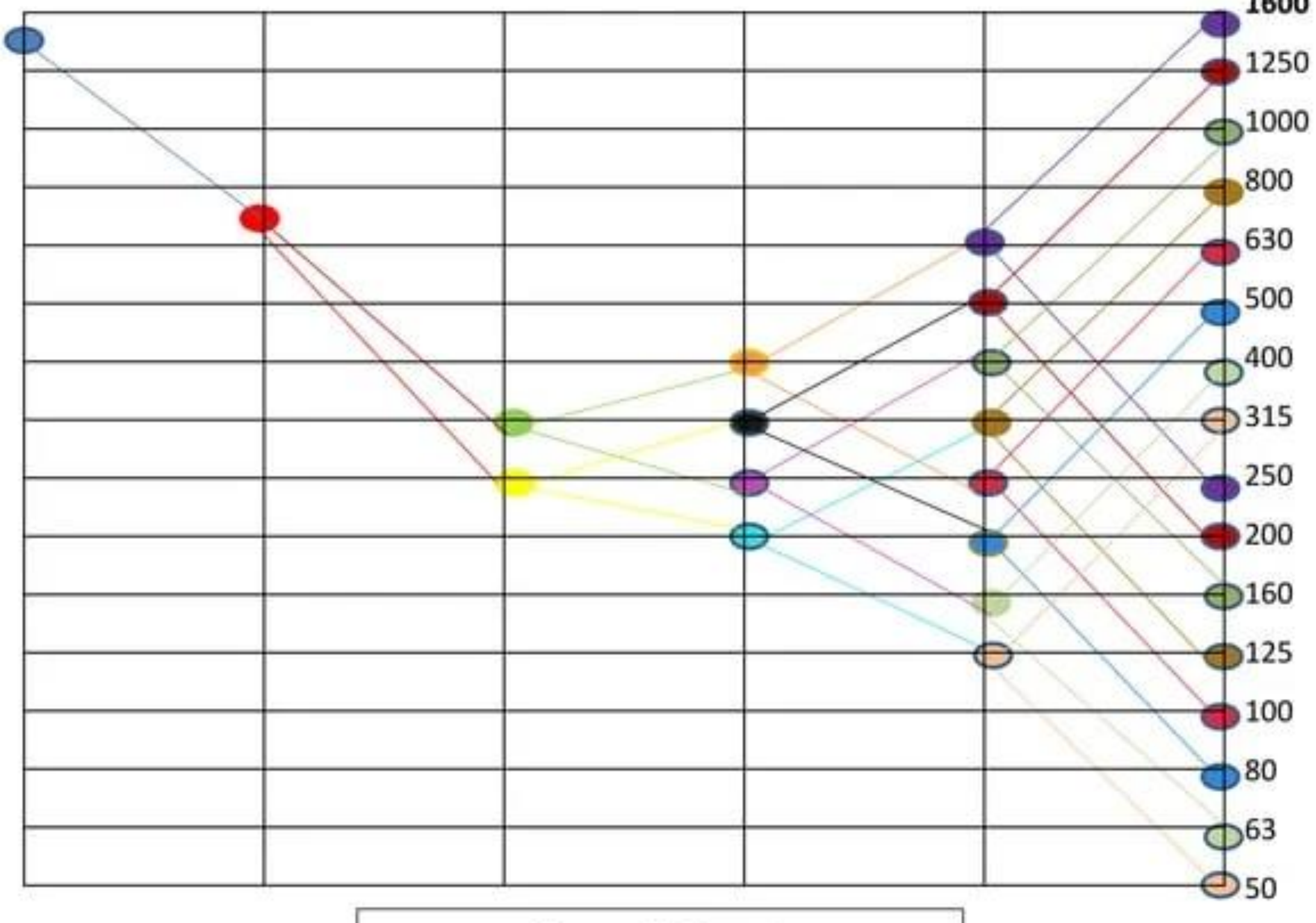
Speed of input shaft

$$= \frac{N_{motor}}{Speed\ Ratio} = \frac{1440}{2} = 720$$



1440 rpm
Motor shaft
I
II
III
IV
V
RPM
1600
1250
1000
800
630
500
400
315
250
200
160
125
100
80
63
50
i_1
i_2
i_3
i_4
i_5
i_6
i_7
i_8

Ray Diagram



1600
1250
1000
800
630
500
400
315
250
200
160
125
100
80
63
50

Cont...

5. Determination of number of teeth on gears
   Between shaft 1 & 2

$$i_1 = \frac{315}{720} = \frac{1}{2.285} = \frac{21}{47.98} = \frac{22}{48} = \frac{Z_1}{Z_1'}$$

$$Z_1 = 22 \;\; Z_1' = 48 \;\; (Z_1 + Z_1' = 70)$$

$$i_2 = \frac{250}{720} = \frac{1}{2.88} = \frac{18}{51.84} = \frac{18}{52} = \frac{Z_2}{Z_2'}$$

$$Z_2 = 18 \;\; Z_2' = 52$$

Cont…

- As the same like above,
  Between shaft 2 & 3

  $Z_3 = 32\ Z_3' = 25\ \ (Z_3 + Z_3' = 57)$

  $Z_4 = 25\ \ Z_4' = 32\ \ (Z_4 + Z_4' = 57)$

- Between shaft 3 & 4

  $Z_5 = 32\ \ Z_5' = 20\ \ (Z_5 + Z_5' = 52)$

  $Z_6 = 20\ \ \ Z_6' = 32\ (Z_6 + Z_6' = 52)$

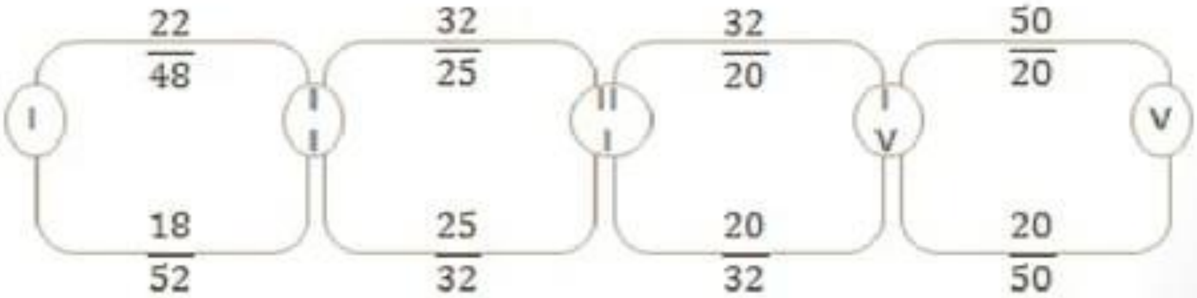
Cont...

- Between shaft 4 & 5

$$Z_7 = 50 \;\; Z_7' = 20 \;\; (Z_7 + Z_7' = 70)$$

$$Z_8 = 20 \;\; Z_8' = 50 \;\; (Z_8 + Z_8' = 70)$$

6.



7. Percentage Variation in Speeds

Sr No	Available Speed (rpm)	Selected Speed	% Variation
1	$720 * \frac{22}{48} * \frac{32}{25} * \frac{32}{20} * \frac{50}{20} = 1689.6$	1600	+5.06
2	$720 * \frac{18}{52} * \frac{32}{25} * \frac{32}{20} * \frac{50}{20} = 1276$	1250	+2.08
3	$720 * \frac{22}{48} * \frac{32}{25} * \frac{32}{20} * \frac{20}{50} = 270.32$	250	+8.12

No	Speed (rpm)	Speed	Variation
4	204.1	160	+2.05
5	1031.25	1000	+3.125
6	165	160	+3.125
7	778.84	800	-0.145
8	124.6	125	-0.32
[illegible]	[illegible]	[illegible]	[illegible]

No	Speed (rpm)	Speed	Variation
10	105.6	100	+5.6
11	498.46	500	-0.30
12	79.75	80	-0.31
13	402.83	400	+0.707
14	64.45	63	+2.30
15	304.2	315	-3.66
16	48.67	50	-2.66

Cont…

8. Now the permissible speed variation

   is $= \pm 10[\emptyset - 1]\%$

   $= \pm 10\ [1.25\text{-}1]$

   $= \pm 2.5\%$

   i.e. Maximum 5% variation

Here the percentage variation variation is more or less within

the permissible value.

UNIT V CAMS, CLUTCHES AND BRAKES

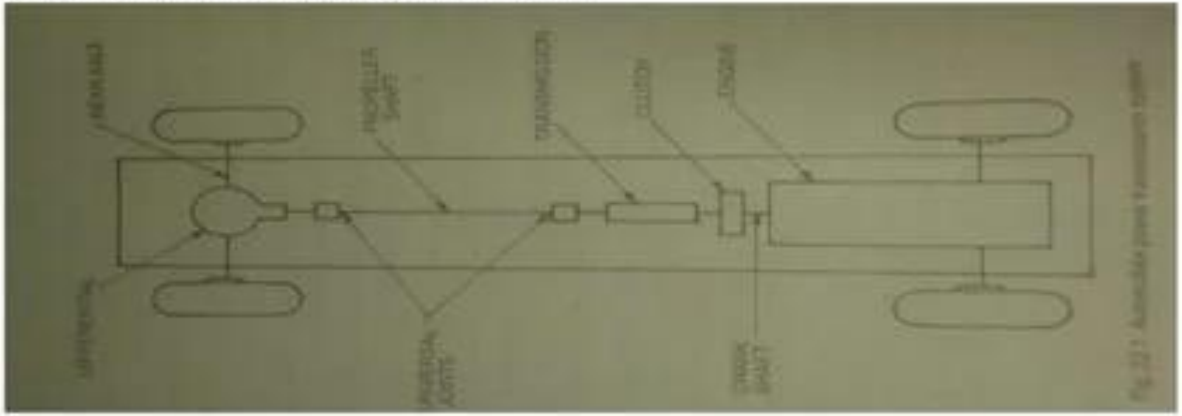
Cam Design: Types-pressure angle and under cutting base circle determination-forces and surface stresses. Design of plate clutches – axial clutches-cone clutches-internal expanding rim clutches Electromagnetic clutches. Band and Block brakes - external shoe brakes – Internal expanding shoe brake.

Clutch

- Defination:-

Clutch is a device used in the transmission system of a motor vehicle to engage and disengage the engine to the transmission.

A clutch is a mechanical device that engages and disengages the power transmission, especially from driving shaft to driven shaft.



Function of Clutch

1. When the clutch is engaged, the power flows from the engine to the wheels through the transmission system and the vehicle moves.
2. When the clutch is disengaged, the power is not transmitted to the wheels and the vehicles stops while the engine is still running.
3. The clutch is kept engaged when the vehicle is moving.
4. The clutch also permits the gradual taking up of the load. When properly operated, it prevents jerky motion of the vehicle.
5. The clutch is disengaged :-

   i) when starting the engine.

   ii) when shifting the gears.

   iii) when stopping the vehicle.

   iv) when idling the engine.

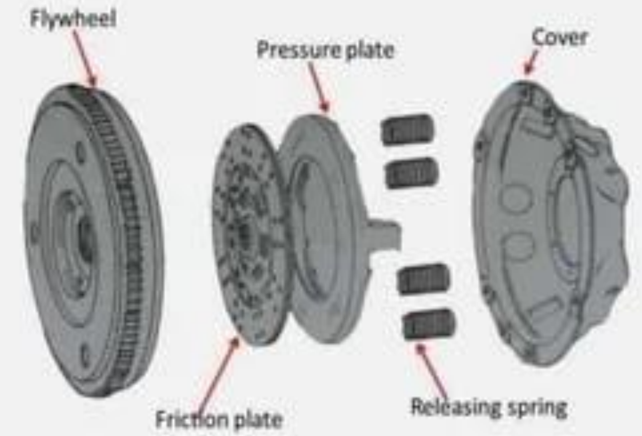
Principle of Operation

- The clutch works on the principles of friction.
- When two friction surfaces are brought in contact with each other and pressed they are united due to the friction between them.
- The friction between the three surfaces depends upon:-

  i) Area of the surfaces.

  ii) applied pressure.

  iii) co-efficient of friction.
- The two surfaces can be separated and brought into contact when required.
- One surface is considered as driving member and other as driven member.
- The driving member is kept rotating.



Main parts of clutch

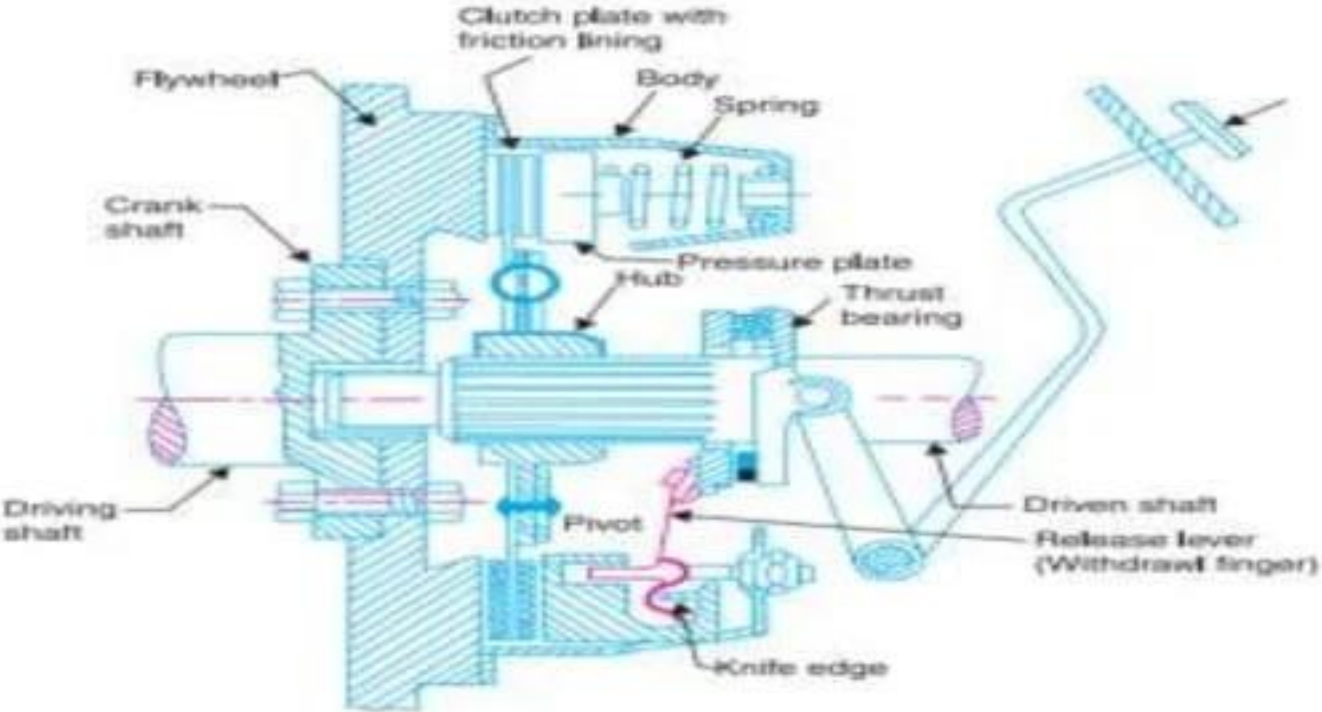
Requirements of a clutch

- Torque transmission
- Gradual engagement
- Heat dissipation
- Dynamic balancing
- Vibration damping
- Size
- Free pedal play
- Easy in operation
- Lightness

Main parts of a clutch

1. Driving member
2. Driven member
3. Operating member

- Driving member has a flywheel which is mounted on the engine crankshaft. A disc is bolted to flywheel which is known as pressure plate or driving disc.
- The driven member is a disc called clutch plate. This plate can slide freely to and fro on the clutch shaft.
- The operating member consists of a pedal or lever which can be pressed to disengage the driving and driven plate.



Clutch plate with friction lining
Flywheel
Body
Spring
Crank shaft
Pressure plate
Hub
Thrust bearing
Driving shaft
Pivot
Driven shaft
Release lever (Withdrawl finger)
Knife edge

Types of clutch

1. Friction clutch :-

    (a) Single plate clutch

    (b) Multi plate clutch -

    i) Wet

    ii) Dry

    (c) Cone clutch –

    i) External

    ii) Internal

2. Centrifugal clutch

4. Electromagnetic Clutch

5 Vacuum Clutch

6. Hydraulic clutch

Single plate clutch

- It has only one clutch plate which is mounted on the splines of the clutch shaft.
- The flywheel is mounted on the engine crankshaft and rotates with it.
- The pressure plate is bolted to the flywheel through clutch springs. It is free to slide on the clutch shaft when the clutch pedal is operated.

Working of single plate clutch

- **When the clutch is engage:-**

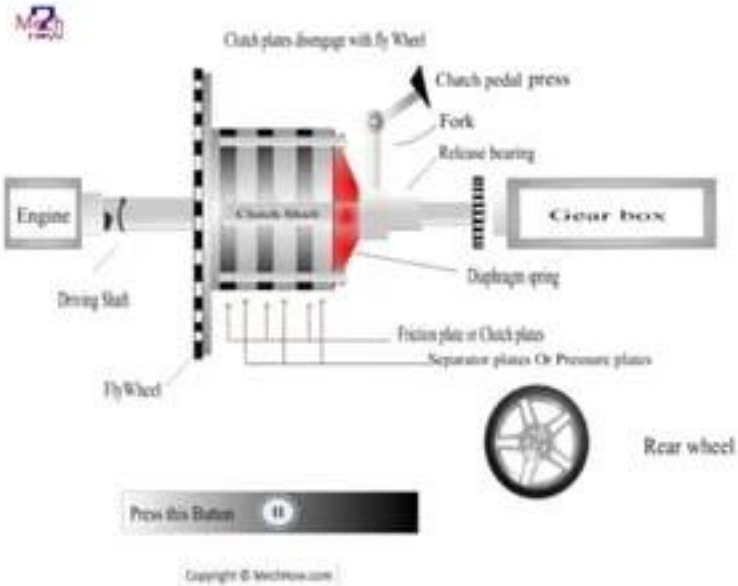
The clutch plate is gripped between the flywheel and the pressure plate. Due to the friction between the flywheel, clutch plate and pressure plate, the clutch plate revolves with the flywheel. As the clutch plate revolves, the clutch shaft also revolves. Clutch shaft is connected to the transmission. Thus, the engine power is transmitted to the crankshaft to the clutch shaft.

- **When the clutch is disengage:-**

When the clutch is pressed, the pressure plate moves back against the force of the springs, and the clutch plate becomes free between the flywheel and the pressure plate. Thus, the flywheel remains rotating as long as engine is running and the clutch shaft speed reduces slowly and finally it stops rotating. As soon as the clutch pedal is pressed, the clutch is said to be disengaged.

Multiplate clutch

- Multiplate clutch consists of a number of clutch plate, instead of only one clutch plate as in the case of single clutch plate.
- The increased number of friction surfaces obviously increases the capacity of the clutch to transmit torque.
- The plates are alternately fitted to the engine shaft and gear box shaft.
- Each of the alternate plate slides in grooves on the flywheel and the other slides on splines on the pressure plate.



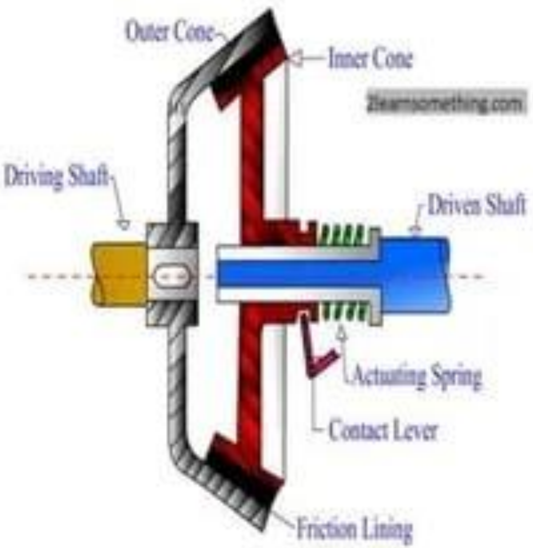
Working of multiplate clutch

- Clutch plate engaged with the flywheel and torque is transmitted flywheel from the through friction facing(clutch plates) to the transmission clutch shaft(clutch shafts).Hence real wheel of the car also rotates.
- When the clutch pedal is pressed the release bearing acts on the pressure plates diaphragms and move the pressure plates away from the flywheel.
- This release bearing the clamping force on the facings plate and separator plate and allows flywheel to freely without turning the clutch shaft.
- Now the clutch plate disengage with the flywheel, and drive is no longer transmitted.
- When the pedal is released, the spring tension forces the pressure plates, clutch plates and separator plates against the flywheel, clamping all components together.

Cone clutch

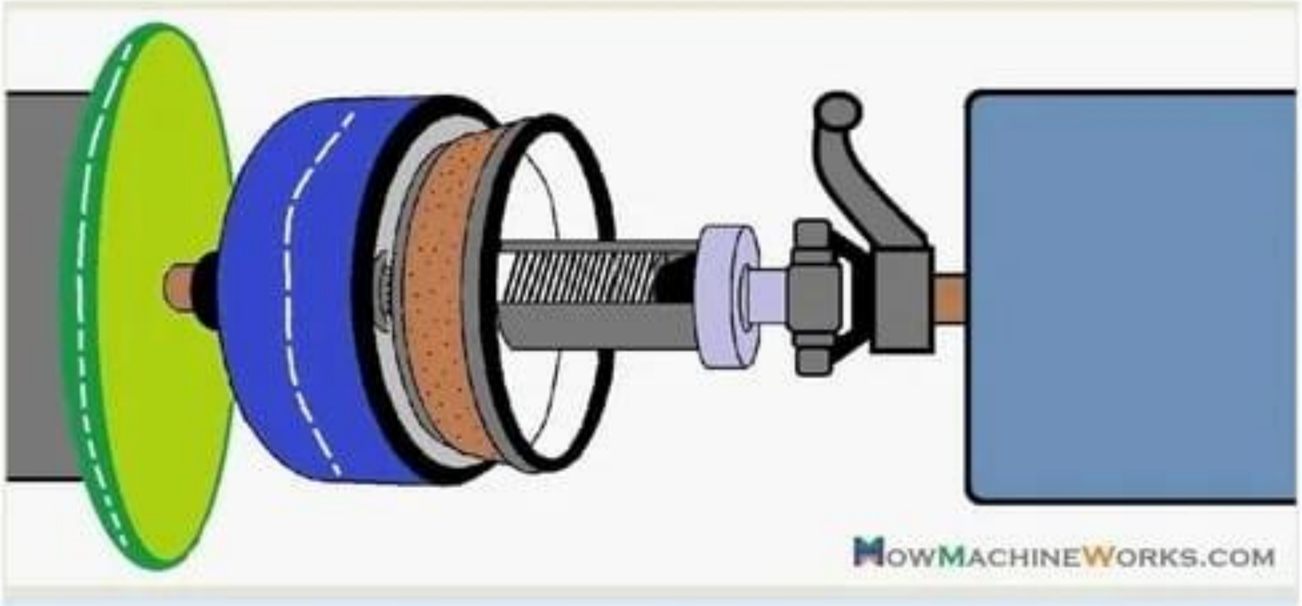
Cone clutch consists of friction surfaces in form of cone. The engine shaft consists of a female cone. The male cone is mounted on the splined clutch shaft. It has friction surfaces on the conical portion. The male cone can slide on the clutch shaft.

- ❖ When the clutch is engaged the friction surfaces of the male cone are in contact with that of the female cone due to the force of spring.
- ❖ When the clutch pedal is pressed, the male cone slides against the spring force and the clutch is disengaged.



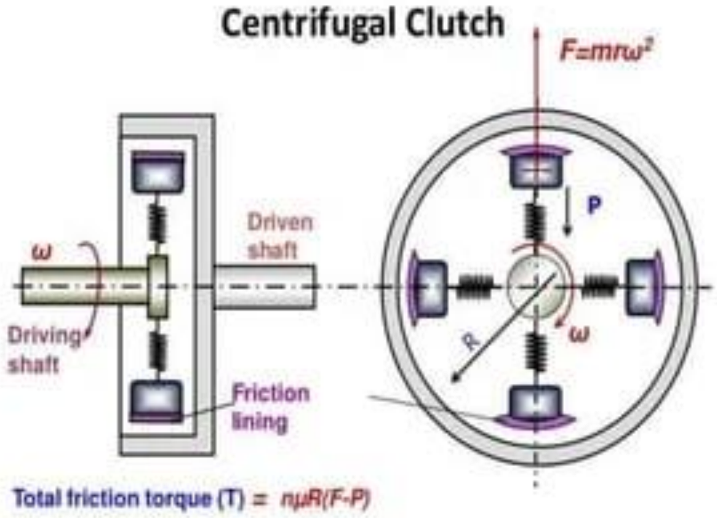
Cone clutch

How to works a cone clutch



Centrifugal clutch

- The centrifugal clutch uses centrifugal force, instead of spring force for keeping it in engaged position. Also, it does not require clutch pedal for operating the clutch.
- The clutch is operated automatically depending upon the engine speed.
- The vehicle can be ted in any gear by pressing the accelerator pedal. stopped in gear without stalling the engine.
- The vehicle can be start.



Electromagnetic clutch

❖ In this type of clutch, the flywheel consists of winding.The current supplied in the winding from the battery or dynamo. When the current passes through the winding, it produces an electromagnetic field which attracts the pressure plate, thereby engaging the clutch. When the supply is cut-off, the clutch is disengaged.

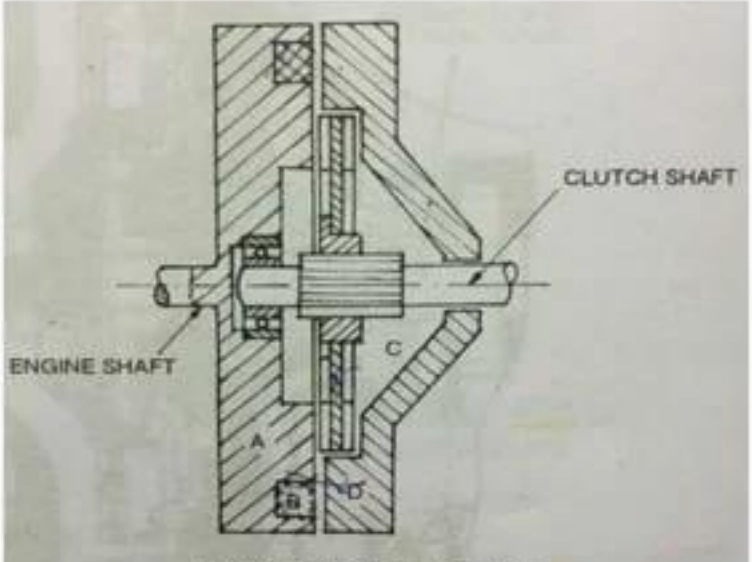
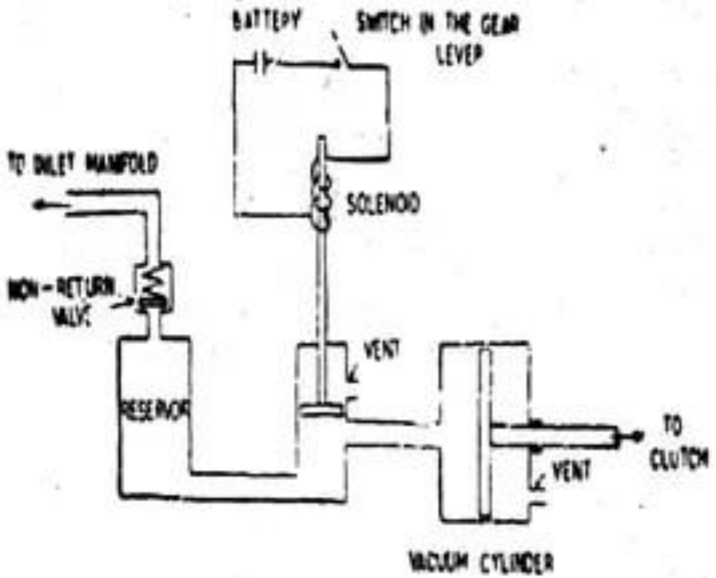


Fig. 22.18. Electromagnetic clutch.

Vacuum Clutch

- It consists of a vacuum cylinder with piston, solenoid operate valve, reservoir and a non-return valve. The reservoir is connected to the engine manifold through a non-return valve. Vacuum cylinder is connected to the reservoir through solenoid operated valve. The solenoid is operated form the battery and the circuit incorporates a switch which is placed in the gear lever.
- Movement of the piston is transmitted by a linkage to the clutch, causing it to disengage. When the driver is not operating the gear lever, the switch is open and the clutch remains engaged due to the force of springs.



Hydraulic Clutch

- The hydraulic clutch is operated in the same way as the vacuum clutch. Only the difference is that it is operated by oil pressure whereas the vacuum clutch is operated by vacuum.
- The pump is operated by the engine itself. The oil from the reservoir is pumped into the accumulator tank. The accumulator tank is connected to the cylinder through the control valve. The control valve is electrically controlled by a switch in the gear lever.
- When the driver holds the gear lever to change the gears, the switch is operated to open the control valve admitting the oil under pressure to the cylinder. Due to the oil pressure, the piston moves causing the clutch to the disengaged.
- As soon as the driver leaves the gear lever, the switch is open which closed the control valve and the clutch is engaged.