UNIT-IV:

GEAR BOXES

Procedure for typical gear box design.

***Step 1*: Selection of spindle speeds:**

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

   (where n= number of speed)

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

3. **Write the structural formulae**

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

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

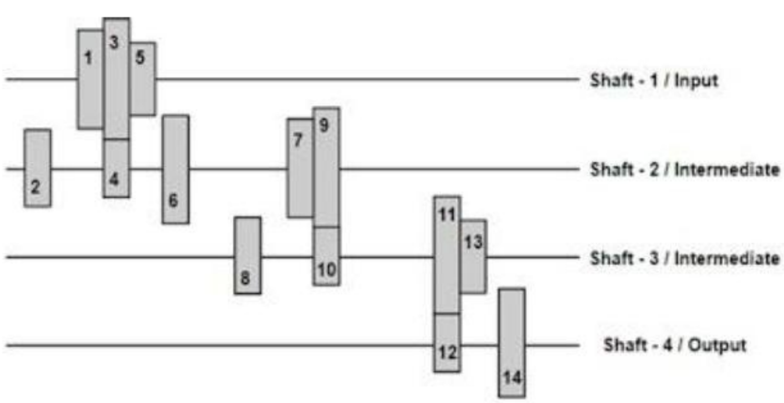
Preferred Structural Formulas	
6 speeds	2 x 3 (or) 3 x 2
8 speeds	2 x 4 (or) 4 x 2 (or) 2 x 2 x 2
9 speeds	3 x 3
12 speeds	3 x 2 x 2 (or) 2 x 2 x 3 (or) 2 x 3 x 2
16 speeds	4 x 2 x 2 (or) 2 x 4 x 2 (or) 2 x 2 x 4

Step 2:

Draw the ray diagram *or* speed diagram

Step 3:

Draw the kinematic diagram.



Step 4:

Calculate the number of teeth.

Step 5: Select the suitable materials

Materials constant PSG.1.15

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

Permissible shear stress(τ) N/mm^2

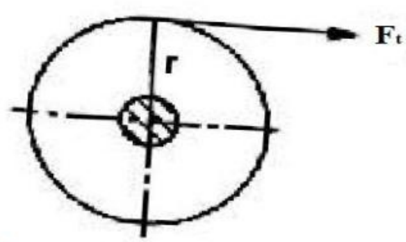
S.No	Shaft materials	(τ) N/mm^2
1.	C14 (as supplied)	25
2.	C45 (case hardened)	30
3.	Low carbon alloyed steel (case hardened)	40
4.	40 Ni 2Cr 1 Mo 28 (hardened and tempered)	55

Step 6: calculation of module

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

$$T = \frac{PX60}{2\pi N}$$

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



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

Gear

[T= F_tx r and r = $\frac{zm}{2}$]

Now calculate the module using the relation

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

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

M = Material constant

Standard module in **PSG data book pg no: 8.2**

<u>Step 7:</u>Calculation of centre distance in all stages:

Calculate the centre distance in each stage by using the relation

$$a = [\frac{Z_x + Z_Y}{2}]m$$

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

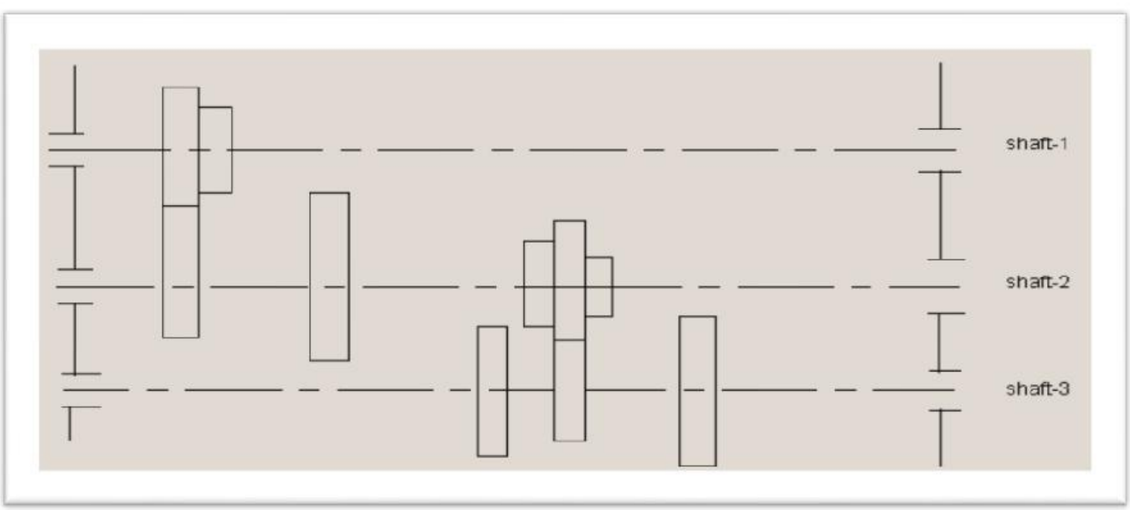
<u>Step 8:</u> calculation of face width: b=10*m

<u>Step 9:</u>calculation of distance between the bearings ie., length of shafts:

Calculate the distance between the bearings by using the following assumptions

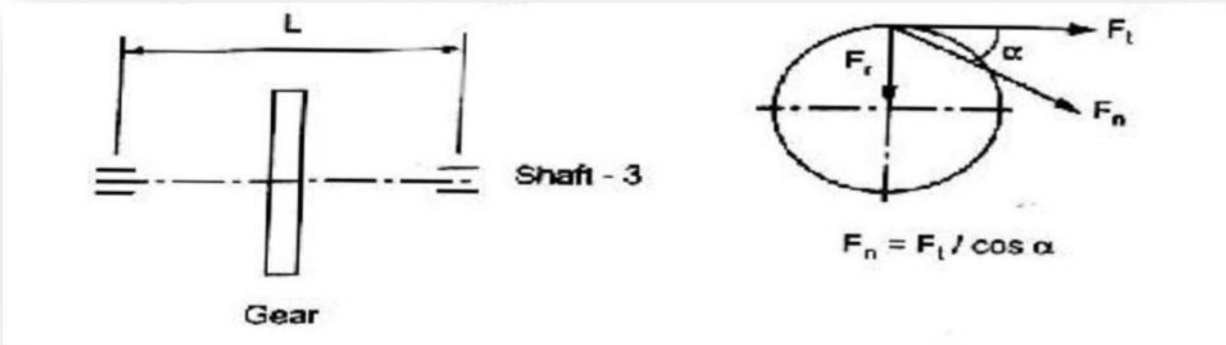
ASSUMPTIONS:
Give 10mm clearance between the gear and the bearings on both sides.
Take the distance between the adjacent groups of gears as 20 mm.
Take the total length for two pairs gear group as 4b and for three pairs gear group as 7b as shown in figure.
Assume the width of the bearings as 25mm

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



***Step 10:*Design of shafts:**

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



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

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

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

***Step 12* Calculate the equivalent torque using the relation**

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

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

***Step 13* Calculate the diameter of the spindle using the relation**

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

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

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

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

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

UNIT– IV: GEAR BOXES (PART - A)

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

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

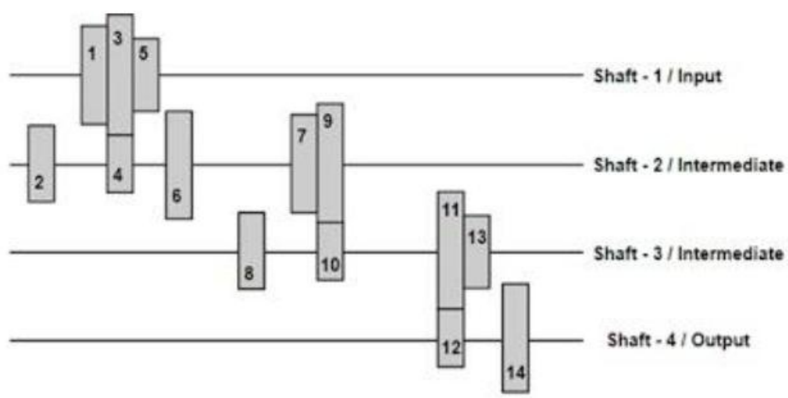
If Nr is;the maximum speed and N, is the minimum speed, then,

$\frac{Nr}{Nr-1}$= (Step ratio) = r-1

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

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

Soln.



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

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

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

Soln.

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

UNIT– IV: GEAR BOXES (PART - A)

5. Which type of gear is used in constant mesh gear box? Justify. ***(Nov/Dec 2012)***

Soln.
Helical gears are used in constant mesh gear boxes to provide quieter and smooth operation

6. What are the possible arrangements to achieve 12 speeds from a gear box?

(April/May2015)

Soln.

S.No	*Number Of Speeds*	*Preferred Structural Formula*
1.	*12speed*	***(i). 3(1) 2(3) 2(6)*** ***(ii). 2(1) 3(2) 2(6)*** ***(iii). 2(1) 2(2) 3(4)***

7. Define the term progression ratio? ***(April /May2015)***

Soln.
When the spindle speeds are arranged in geometric progression, the ratio between the two adjacent speeds is known as step ratio or progression ratio.

8. What are the points to be considered while designing a sliding mesh type of multi speed gear box? (April /May2010)

Soln.
i) The transmission ratio in a gear box is limited by $\frac{1}{4} < i < 2$
ii) Speed ratio of any stage should not be greater than 8.

9. Where is multi-speed gear boxes employed? (Apr/May2011)

Soln.

(i) Automobiles

(ii) Machine tools and

(iii) Aeronautical.

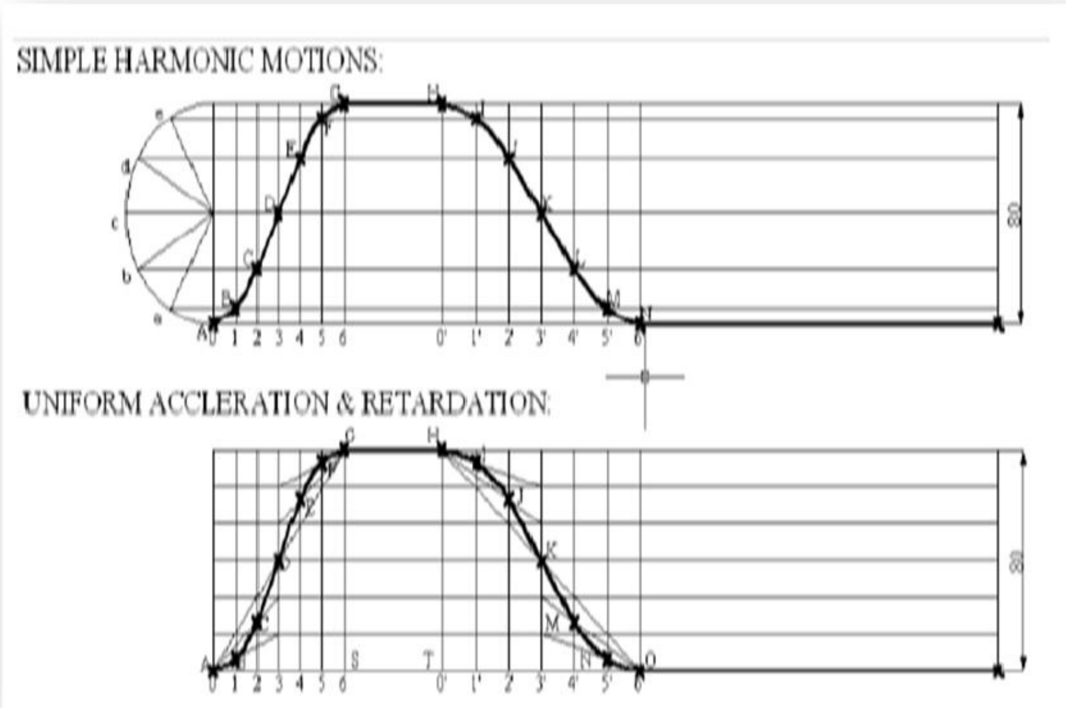
10. Where multispeed gearbox is used? (May/Jun 2016)

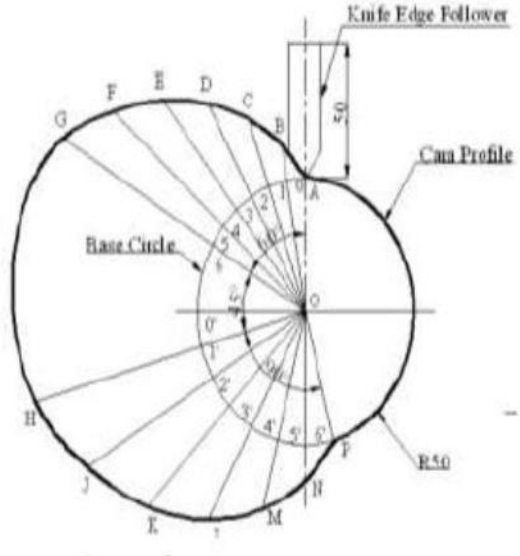
Soln. Multi speed gearbox is used for speed adjustment at constant power level.
Heavy duty Industrial Multi-speed gearboxes are designed for continuous operation.

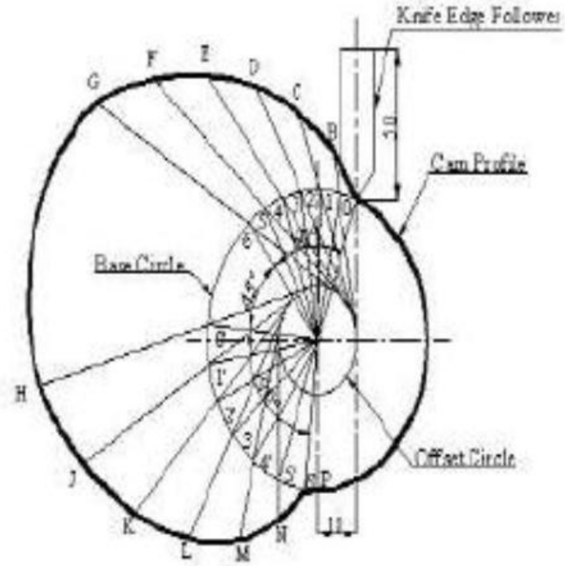
UNIT V

CAMS, CLUTCHES AND BRAKES

CAM TYPES

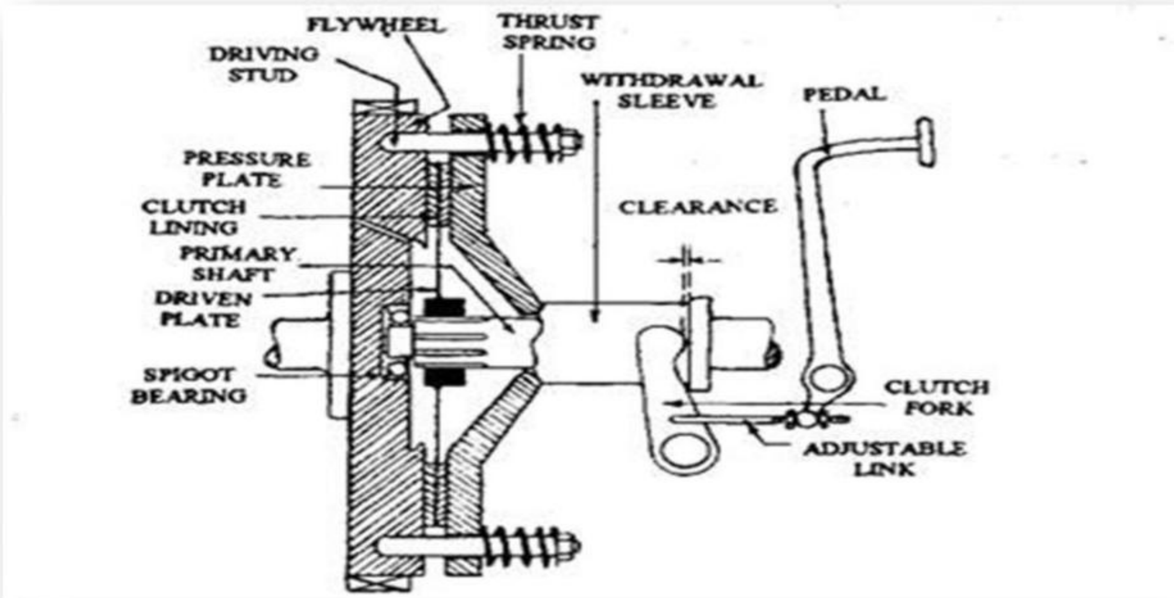



Knife Edge Follower
50
Cam Profile
Base Circle
R50


Knife Edge Follower
Cam Profile
Base Circle
Offset Circle

CLUTCHES

Design of a single plate clutch



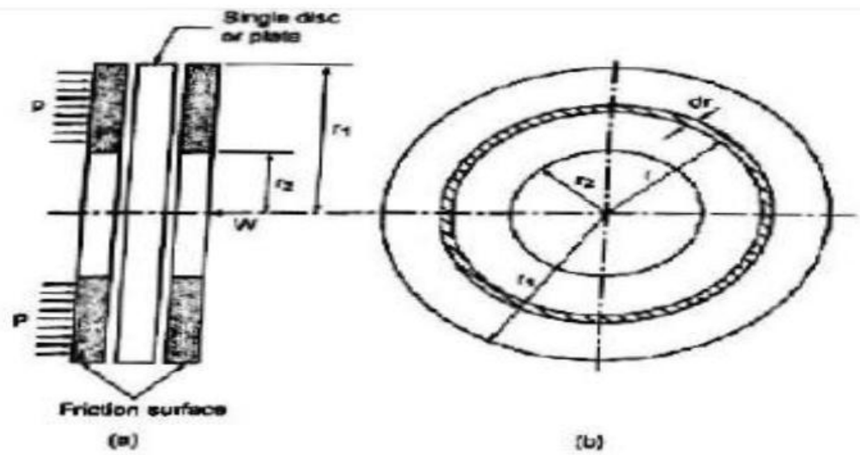


Fig. 10.3. Forces on a single disc or plate clutch

T= Torque transmitted by the clutch,

P= Intensity of axial pressure acting on contact surfaces,

r_1 = External radius of friction surface,

r_2 = Internal radius of friction surface.

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

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

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

(i) Considering uniform pressure:

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

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

T= μ WR

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

(ii) Considering uniform wear

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

$P_1.r_1 = p_2.r_2 = c$

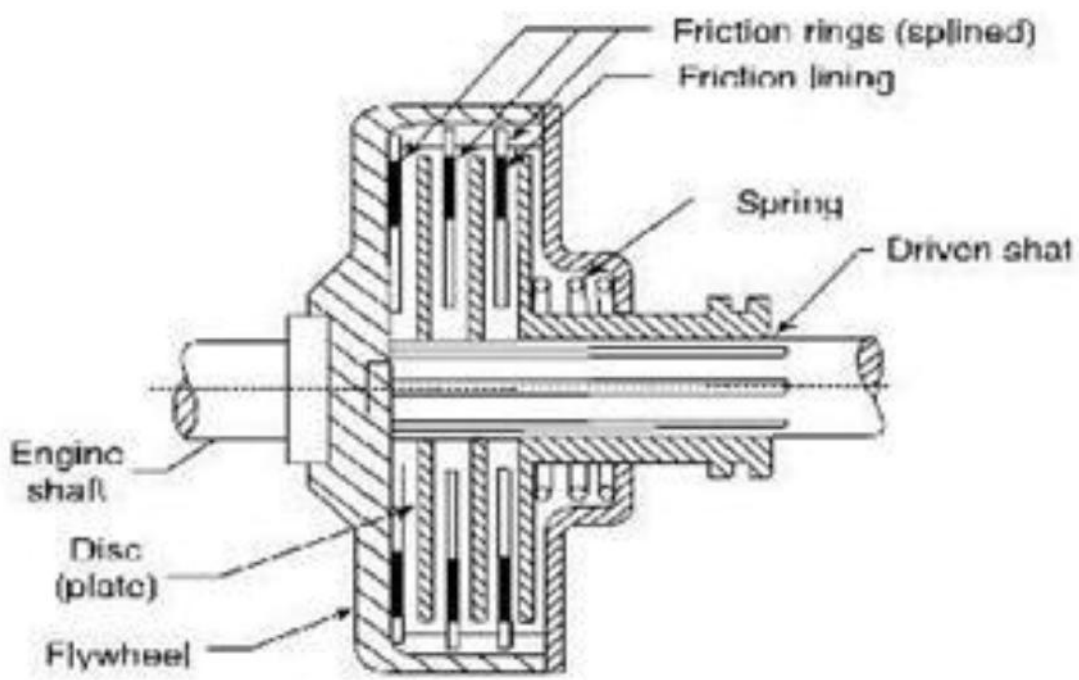
$$C = \frac{W}{2\pi(r_1 - r_2)}$$

T= μ WR

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

Design of a Multiplate clutch

(Torque transmitted on multiplate clutch)



n_1 = Number of discs on the driving shaft, and

n_2 = Number of discs on the driven shaft.

Number of pair of contact surfaces,

$$n = n_1+n_2-1$$

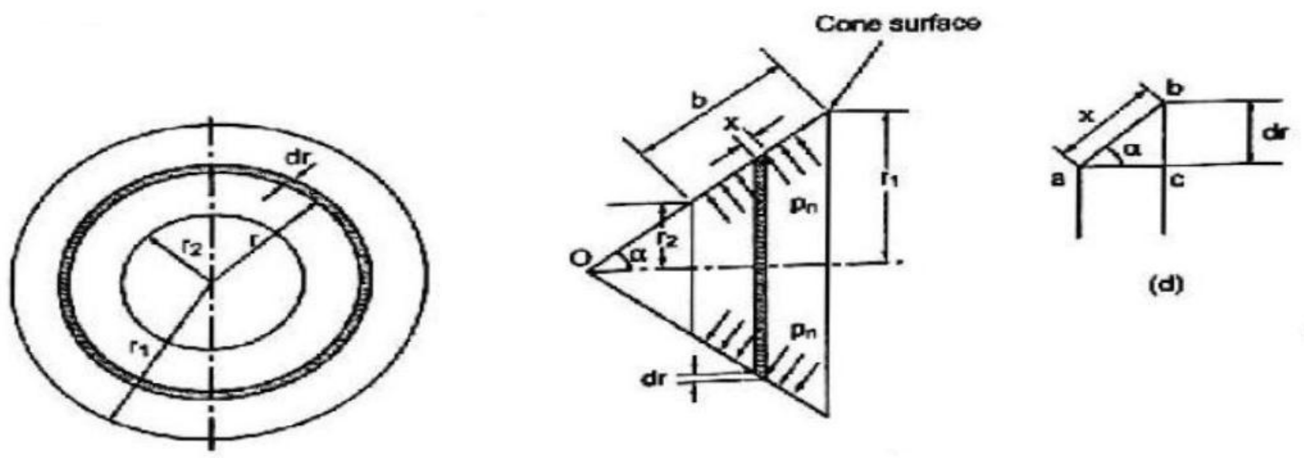
Total frictional torque on the clutch is given by

T= nμWR

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

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

Design of a cone clutch



Torque transmitted on the cone clutch is given by

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

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

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

Axial force required at the engagement of clutch is given by

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

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

$$W_c = Wn\,(\sin\alpha-\mu\cos\alpha)$$

BRAKE

SINGLE BLOCK OR SHOE BRAKE

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

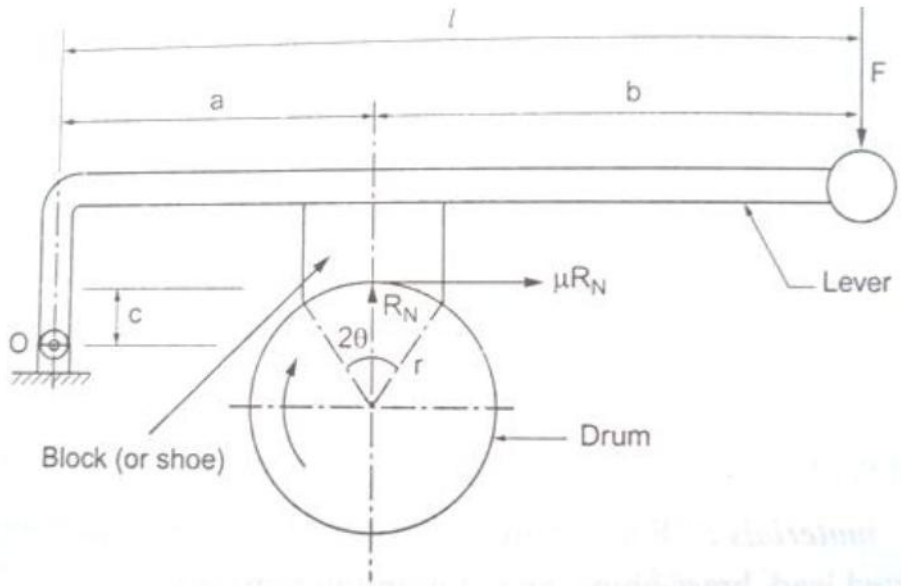


Fig. 11.2. Clockwise rotation of brake drum

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

Let r = Radius of the drum

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

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

F = Force applied at the lever end

μ = Coefficient of friction

μR_N = Frictional force

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

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

DOUBLE BLOCK OR DOUBLE SHOE BRAKE

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

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

R = Radius of drum.

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

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

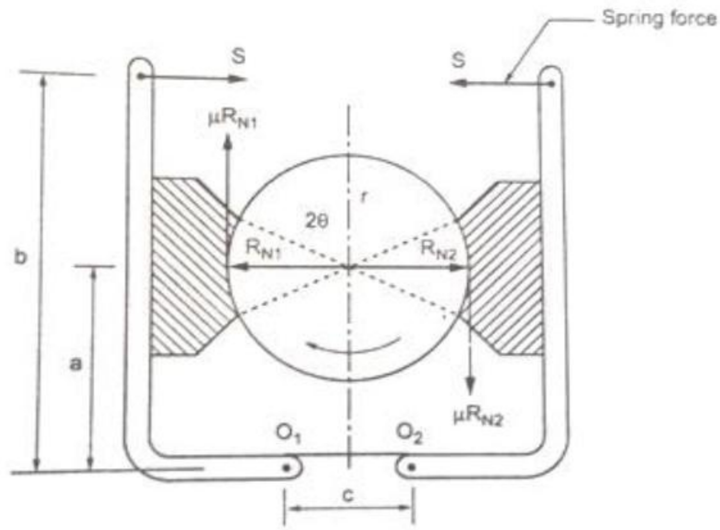


Fig. 11.7. Double shoe brake

DESIGN PROCEDURE FOR BLOCK BRAKE

***Step 1:** calculate the total energy absorbed by the brake*

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

***Step 2:** calculate the torque capacity by using the relation*

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

N_i- Initial speed of brake drum and

t = Time of application of brake

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

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

***Step 4:**Select the brake drum diameter*

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

Table 11.1 properties of brake lining materials

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

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

Table 11.2 Limiting values of pv (from PSG 7.130)

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

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

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

A= Breadth X Width

SIMPLE BAND BRAKES

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

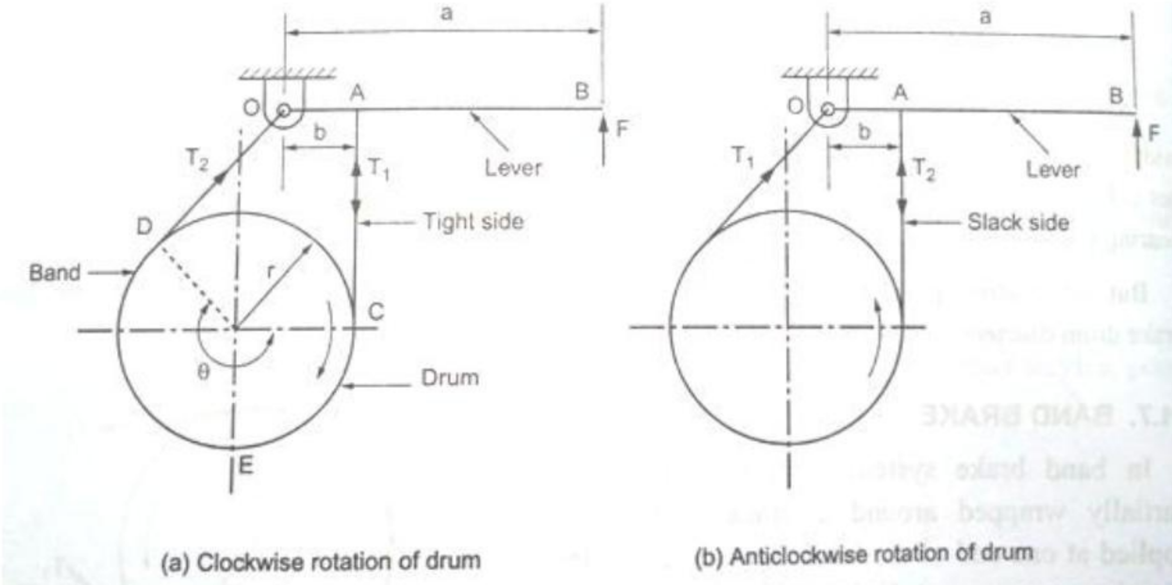


Fig. 11.11. Simple band brake

Let F = Force applied on the lever,

R = Radius of the drum,

T = Thickness of the band,

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

a= Length of lever = OB

b = Distance between the fulcrum O and point A.

For clockwise rotation of drum

$$F.a = T_1 . b$$

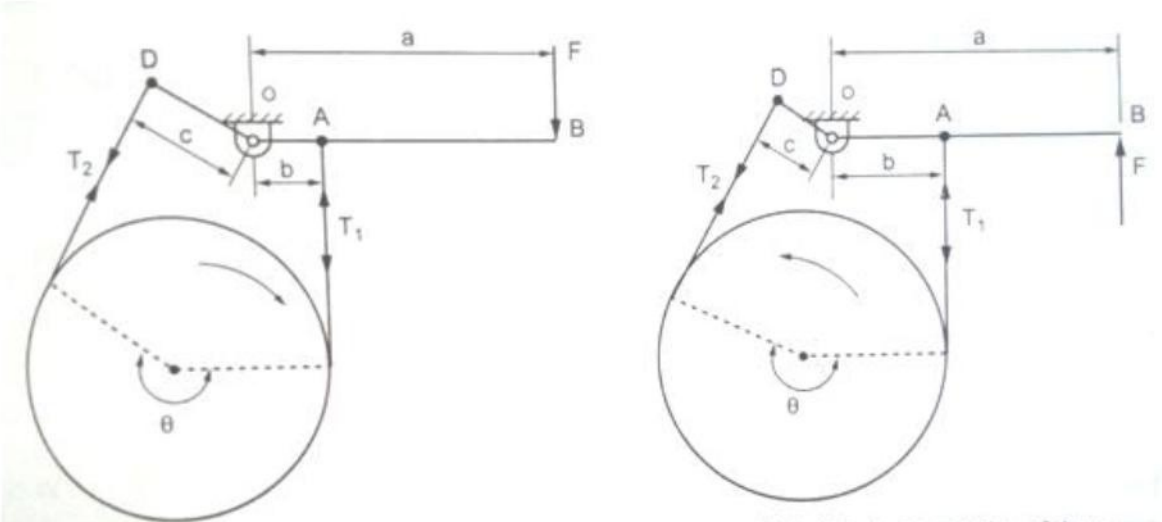
For anticlockwise rotation of drum

$$F.a = T_2 . b$$

Braking torque $T_B = (T1- T2).r = \left[T_1 - \frac{T_1}{e^{\mu\theta}}\right] r = F \times \frac{a}{b} \left[1 - \frac{1}{e^{\mu\theta}}\right] r$

DIFFERENTIAL BAND BRAKE

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



(a) Clockwise rotation of drum (b) Anticlockwise rotation of drum

Fig. 11.14. Differential band brake

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

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

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

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

Thus c>b for the system to work satisfactorily.

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

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

Taking moments about O, we get

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

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

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

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

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

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

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

Thus $T_1\ b > T_2\ c$

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

(iv)Upwards froce on lever for clockwise rotatio of drum:

Taking Moment about O, we get,

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

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

Condition for self-locking:

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

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

DESIGN PROCEDURE FOR BAND BRAKES

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

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

Table 11.3 Dimensions of brake drum (from PSG 7.98)

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

Step 3: Determine the tight and slack side tensions.

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

Step 5: calculate the band width (w)

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

T_1 = Tight side tension in the band,

W = width of the band

t = Thickness of the band = 0.005 D

σ_t = permissible tensile stress = 50 to 80 N/mm^2

Step 6: check for bearing pressure

$$p_{max} = \frac{T_1}{W.r}$$

r = Radius of the drum

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

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

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

BAND AND BLOCK BRAKE

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

Let T_n = Tension in the band on tight side,

T_o = Tension in the band on slack side,

T_1 = Tension in band between the first and secon block,

T_2 = Tension in band between second and third block,

T_3 = Tension in band between third and fourth blocks.

n = Number of wooden blocks,

μ = Coefficient of friction between block and drum.

2θ = Angle subtended by each block at the drum centre.

R_N = Normal reaction on the block.

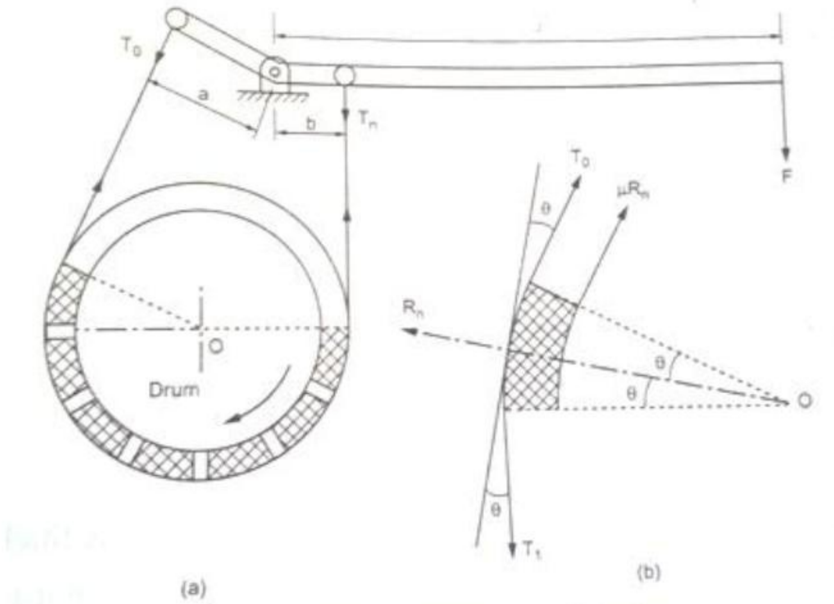


Fig. 11.19. Band and block brake

Resolving the forces radially, we get

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

Resolving the forces tangentialy, we get

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

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

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

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

or

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

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

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

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

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

$$\frac{T_n}{T_0} = \frac{T_1}{T_0} x \frac{T_2}{T_1} \text{ x } \frac{T_3}{T_2} \text{ x.....x } \frac{T_n}{T_{n-1}} = [\frac{1+\mu tan\theta}{1-\mu tan\theta}]^n$$

Braking torque on the drum is given by

$$T_B = (T_1 - T_2) \text{ r}$$

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

DESIGN OF INTERNAL EXPANDING SHOE BRAKE

Moment of normal force, $M_N = \frac{1}{2} p_1.b.r.oo_1 [(\theta_1 . \theta_2) + \frac{1}{2}(sin2\theta_1 \; sin2\theta_2)]$

Moment of frictional force, $M_F = \mu \; p_1.b.r. [r(cos\,\theta_1 - cos\,\theta_2) + \frac{oo_1}{4}(COS2\theta_2 - COS2\theta_1)]$

Braking torque in Block or shoe brake is given by

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

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

where T_B - Braking torque,

r = Radius of drum,
F = Force applied at lever end,
μ=Coefficient of friction, and

a, c & l = Dimensions of lever.

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

$$\mu^1 = \frac{4\mu sin\theta}{2\theta + sin2\theta}$$

where 2θ = Angle of contact

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

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

where r = Radius of drum,
RN_1 & RN_2 = Normal reaction on the left & right hand side shoes

In **Band brake system,**

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

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

T_1 and T_2 = Tension in the band on tight and slack sides respectively
θ = Angle of lap

r = Radius of drum,

Force applied on the lever in Simple band brake is given by
(i) $F = T_1 \left(\frac{b}{a}\right)$. [For clockwise rotation of the drum]

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

Tension ratio in Band and block brake is given by

$$\frac{T_n}{T_0} = \left[\frac{1+\mu\tan\theta}{1-\mu\tan\theta}\right]^n$$

where T_n = Tension in the band on tight side (maximum tension),
T_o = Tension in the band on slack side (minimum tension),
2θ - Angle sub tended by each block at the drum centre, and

n = Number of wooden blocks.

Actuating force on leading (or left hand) shoe. $F1 = \frac{M_N - M_F}{l}$

Actuating force on Trailing (or right hand) shoe. $F2 = \frac{M_N + M_F}{l}$

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

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

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

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