(17) 1

SCHEME OF VALUATION

(Scoring Indicators)

Revision: 2015

Course Title: DESIGN OF MACHINE ELEMENTS

Course Code: (15) 5021

Qst. No.	Scoring Indicator	Split up score	Sub Total	Total
	PART A			10
I (1)	Factor of safety may by defined as the ratio of Yield stress to the working stress for a ductile material. For Brittle materials factor of safety is defined as the ratio of ultimate stress to the working stress.	2	2	
I (2)	Sunk keys, Saddle keys, Tangent keys, Round keys and Splines.	$\frac{1}{2} \times 4$	2	
I (3)	The following stresses are induced in the shafts: (i) Shear stress due to transmission of torque (ii) Bending stress due to mountings such as gears, pulleys etc. on the shaft (iii) Axial tensile or compressive stresses (iv) Surface Stresses due to rubbing on bearing	1 1	2	
I (4)	The value of Bearing Characteristic Number $\left(\frac{ZN}{P}\right)$ corresponding to minimum coefficient of friction (μ) is known as Bearing Modulus. It is denoted by K.	1x2	2	
I (5)	(i) The production cost of chains is relatively high (ii) The chain drive needs accurate mounting and careful maintenance (iii) The chain drive has velocity fluctuations especially when unduly stretched	1x2	2	
	PART B			42
II (1)	When a bolt is designed to withstand shock loading the resilience of the bolt is considered in order to prevent its breaking at the threaded portion. In a normal bolt (fig. a) the effect of the impulsive load is stress concentration under the root of the threads. In other words, the stress in the threaded portion of the bolt will be higher than that in the shank. Hence a major portion of the energy will be absorbed by the threaded portion which may result in fracture.	3	6	

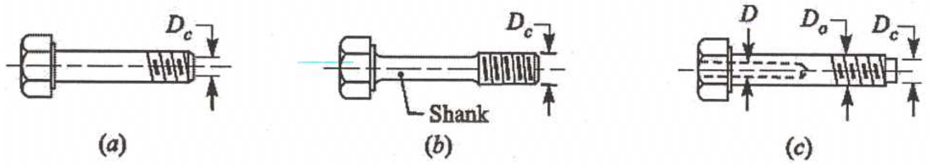
2

Bolts which prevent such stress concentration under shock loading and have a uniform stress throughout its length are called bolts of uniform strength.

This can be achieved by:

(i) Reducing the shank diameter to core diameter of the thread (fig. b),

(ii) Drilling an axial hole through the head to the beginning of thread (fig. c).



(a) (b) (c) fig-3

II (2) Given that: $N = 400$ rpm, $P = 10kW = 10 \times 10^3 W$, $\tau_s = 40\ MPa = 40\ N/mm^2$ — 6

Torque transmitted by the shaft, $T = \frac{P \times 60}{2\pi N} = \frac{10 \times 10^3 \times 60}{2 \times \pi \times 400} = 238.73\ Nm$ — 2

$= 238.73 \times 10^3\ Nmm$

Diameter of the shaft can be found by using the relation, — 2

$$T = \frac{\pi}{16}\tau_s d^3$$

$$238.73 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d = 31.21mm \cong 35mm$$ — 2

II (3) Given that: $N = 240$ rpm, $P = 1MW = 10 \times 10^6 W$ — 6

Torque transmitted by the shaft, $T = \frac{P \times 60}{2\pi N} = \frac{1 \times 10^6 \times 60}{2 \times \pi \times 240}$

$= 39788.73Nm = 39.79 \times 10^6\ Nmm$ — 2

To find diameter of shaft,

$\theta = 1^o = \frac{\pi}{180}\ rad;\ L = 15d;\ G = 80\ GPa = 80 \times 10^3\ N/mm^2$

$$\frac{T}{J} = \frac{G\theta}{L}$$

$$\frac{39.79 \times 10^6}{\left(\frac{\pi}{32}\right) \times d^4} = \frac{80 \times 10^3 \times \left(\frac{\pi}{180}\right)}{(15d)}$$ — 2

$$d^3 = \frac{32 \times 39.79 \times 10^6 \times 15 \times 180}{\pi^2 \times 80 \times 10^3}$$

$$d = 163.29\ mm \cong 165\ mm$$

To find stress induced in the shaft,

$$T = \frac{\pi}{16}\tau d^3$$

Shear stress, $\tau = \frac{T \times 16}{\pi \times d^3} = \frac{39.79 \times 10^6 \times 16}{\pi \times 165^3} = 45.11\ N/mm^2$ — 2

II (4) <u>(i) Trace Point:-</u> The reference point on the follower used to generate the pitch curve. For knife edge follower, the knife edge represents the trace point and the cam profile is the same as the pitch curve. For roller follower, the center of the roller represents the trace point. — 6

<u>(ii) Pressure Angle:-</u> It is the angle between the direction of the follower motion and a normal to the pitch curve. This angle is very important in designing a cam profile. If the pressure angle is too large, a reciprocating follower will jam in its bearings.

<u>(iii) Pitch Point:-</u> It is a point on the pitch curve having the maximum pressure angle.

(iv) Pitch Circle:- It is a circle drawn from the center of the cam through the pitch points.

(v) Pitch Curve:- It is the curve generated by the trace point as the follower moves relative to the cam. For a knife edge follower, the pitch curve and the cam profile are same whereas for a roller follower, they are separated by the radius of the roller.

(vi) Prime Circle:- It is the smallest circle that can be drawn from the centre of the cam and tangent to the pitch curve. For a knife edge and a flat face follower, the prime circle and the base circle are identical. For a roller follower, the prime circle is larger than the base circle by the radius of the roller.

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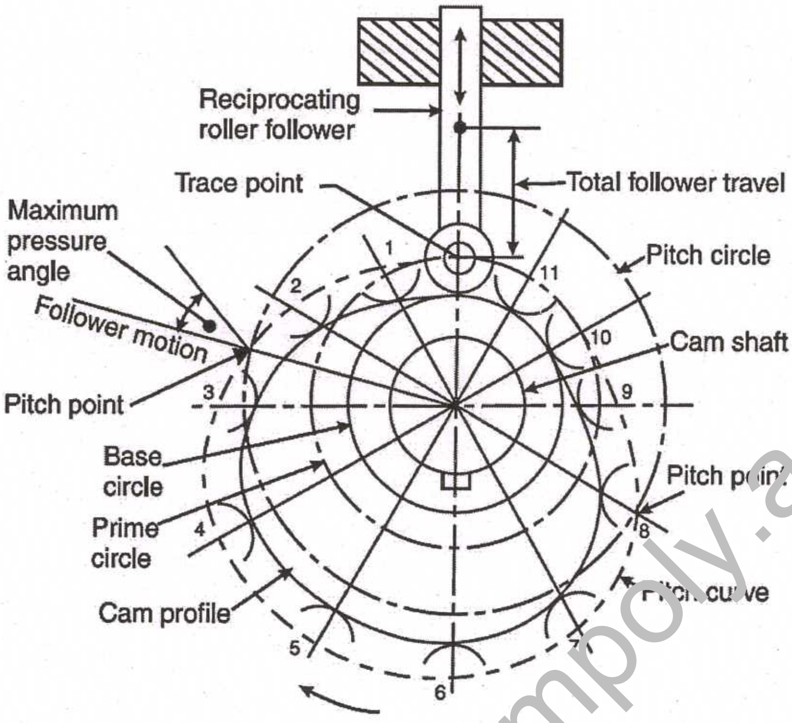

Terms used in radial cams.

Fig-3

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II (5) The function of a governor is to regulate the mean speed of an engine, when there are variations in the load. When the load on an engine increases, its speed decreases, therefore it becomes necessary to increase the supply of fuel to the engine and vice versa. 6

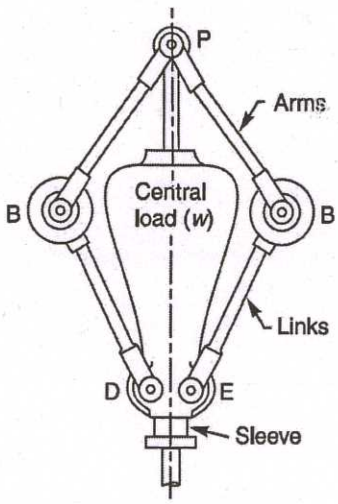


Fig-2

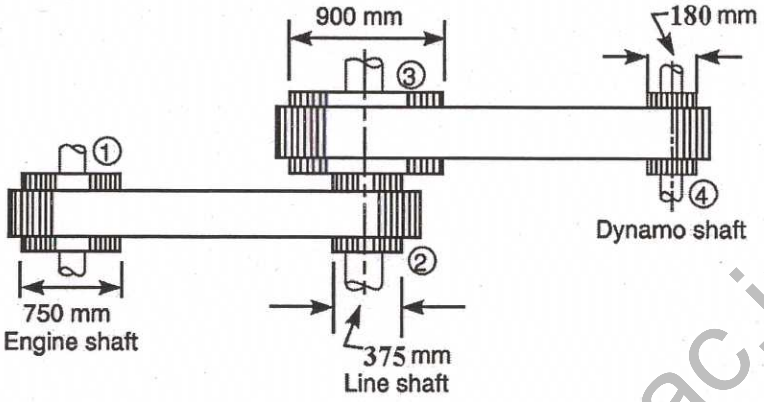
The porter governor consists of a sleeve which rotates along with the engine, a pivot, arms, balls, links, and a central mass.

The Porter governor is a modification of a Watt's governor, with central load attached to the sleeve as shown in figure. It is basically a conical pendulum with links attached to a sleeve with the cental mass. The load moves up and down the central spindle. This additional downward force increases the speed of revolution required to enable the balls to rise to any predetermined level. 4

4

Either method of resolution of forces or instantaneous centre method is used to determine the relation between the height of the governor and the angular speed of the balls.

II (6) Given that: $N_1 = 160\,rpm$; $d_1 = 750\,mm$; $d_2 = 375\,mm$; $d_3 = 900\,mm$; $d_4 = 180\,mm$; $N_4 =$ Speed of dynamo shaft. 6



1. When there is no slip,

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4}$$

$$\frac{N_4}{160} = \frac{750 \times 900}{375 \times 180} = 10$$

$$N_4 = 10 \times 160 = 1600\,rpm$$

3

2. When there is a slip of 2% at each drive,

$$\frac{N_4}{N_1} = \frac{d_1 \times d_3}{d_2 \times d_4}\left(1 - \frac{s_1}{100}\right)\left(1 - \frac{s_2}{100}\right)$$

$$\frac{N_4}{160} = \frac{750 \times 900}{375 \times 180}\left(1 - \frac{2}{100}\right)\left(1 - \frac{2}{100}\right) = 9.6$$

$$N_4 = 9.6 \times 160 = 1536\,rpm$$

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II (7) Unlike other gear trains in an epicyclic gear train, the axes of the shafts, over which the gears are mounted, may move relative to a fixed axis. A simple epicyclic gear train is shown in figure. 2 6

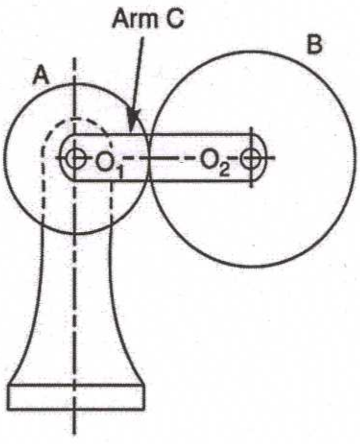

fig-2

Epicyclic gear train.

	From the figure, a gear A and the arm C have a common axis at O1 about which they can rotate. The gear B meshes with gear A and has its axis on the arm at O2, about which the gear B can rotate. If the arm is fixed, the gear train is a simple gear train and gear A can drive gear B or vice- versa, but if gear A is fixed and the arm is rotated about the axis of gear A (i.e. O1), then the gear B is forced to rotate upon and around gear A. Such a motion is called epicyclic and the gear trains arranged in such a manner that one or more of their members move upon and around another member are known as epicyclic gear trains. The epicyclic gear trains may be simple or compound. The epicyclic gear trains are useful for transmitting high velocity ratios with gears of moderate size in a comparatively lesser space. The epicyclic gear trains are used in the back gear of lathe, differential gears of the automobiles, hoists, pulley blocks, wrist watches etc.	2		

PART C

III (a) — 8 — 15

Given that: $D = 340\ mm$; $p = 1.25\ N/mm^2$; $\sigma_t = 30\ MPa = 30\ N/mm^2$

Let d = Nominal diameter of studs,

d_c = Core diameter of studs, and

n = Number of studs. — 1

Upward force acting on the cylinder cover,

$$P = \frac{\pi}{4} \times D^2 \times p = \frac{\pi}{4} \times 340^2 \times 1.25 = 113490\ N \ \ldots\ldots \text{(i)}$$

Assume that studs of nominal diameter of 24 mm are used.

Using Design dimensions of screw threads, bolts and nuts according to IS: 4218 (Part III) 1976 (Reaffirmed 1996), core diameter of the stud, $d_c = 20.32\ mm$.

Resisting force,

$$P = \frac{\pi}{4} \times (d_c)^2 \times \sigma_t \times n = \frac{\pi}{4} \times (20.32)^2 \times 30 \times n = 9728.78n\ N \ \ldots\ldots\ldots \text{(ii)}$$ — 2

From equations (i) and (ii),

$$n = \frac{113490}{9728.78} = 11.66 \cong 12$$

Diameter of stud hole, $d_1 = 25\ mm$; and thickness of the cylinder wall, $t = 10\ mm$.

Pitch circle diameter of the studs,

$$D_p = D + 2t + 3d_1 = 340 + 2 \times 10 + 3 \times 25 = 435\ mm$$

Circumferential pitch of the studs,

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 435}{12} = 113.9\ mm$$ — 2

Minimum Circumferential pitch of the studs for a leak-proof joint,

$$= 20\sqrt{d_1} = 20 \times \sqrt{25} = 100\ mm$$

Maximum circumferential pitch of the studs for a leak-proof joint,

$$= 30\sqrt{d_1} = 30 \times \sqrt{25} = 150\ mm$$

Since the circumferential pitch of the studs obtained lies within the range of 100 mm to 150 mm, the size of the stud chosen is satisfactory. — 3

∴ Size of the stud = M 24

III (b) — 7

The sunk keys are provided half in the keyway of the shaft and half in the keyway of the hub or boss of the pulley.

The usual proportions of a rectangular sunk key are:

Width of key, $w = \frac{d}{4}$; and thickness of key, $t = \frac{2w}{3} = \frac{d}{6}$ — 2

Where d is the diameter of the shaft.

The key has a taper of 1 in 100 on the top side only.

The design of key using these proportions is called empirical design.

The two types of forces acting on keys are: (i) forces due to the fit of the key in the keyway, F_1. These forces produce compressive stresses in the key which are difficult to tetermine in magnitude; (ii) forces due to the torque transmitted by the shaft, F. These forces produce shearing and crushing (compressive) stresses in the key.

Usually the forces due to fit of the key are neglected and it is assumed that the distribution of forces along the length of the key is uniform.

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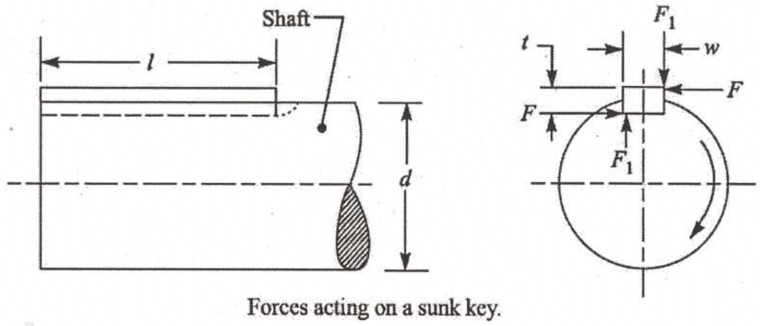

Forces acting on a sunk key.

fig-3

Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft,

$$F = Area\ resisting\ shearing \times shear\ stress = l \times w \times \tau$$

where, l = length of the key, w = width of the key, and τ = shear stress induced in the key.

Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \times w \times \tau \times \frac{d}{2} \text{ .... (1)}$$

where, d = diameter of the shaft.

Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft,

$$F = Area\ resisting\ crushing \times crushing\ stress = l \times \frac{t}{2} \times \sigma_c$$

where, t = the thickness of the key, and σ_c = crushing stress induced in the key.

Torque transmitted by the shaft,

$$T = F \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2} \text{ .... (2)}$$

Now by equating the shearing strength of the key to the torsional shear strength of the shaft we can get another analytical equation.

Shearing strength of key,

$$T = l \times w \times \tau \times \frac{d}{2}$$

Torsional shear strength of the shaft,

$$T = \frac{\pi}{16} \times \tau_1 \times d^3$$

where, τ_1 = the shear stress induced in the shaft.

Equating the above two equations, 2

$$l \times w \times \tau \times \frac{d}{2} = \frac{\pi}{16} \times \tau_1 \times d^3 \text{ .... (3)}$$

We can find the dimensions of the key from equations (1), (2), and (3). This method is called analytical design.

IV (a)	Given that, $d = 25\ mm,\ N = 600\ rpm,\ P = 7\ kW, \tau = 60\ MPa = 60\ N/mm^2$, $\sigma_c = 120\ MPa = 120\ N/mm^2$. Torque acting on the key, $T = \frac{P \times 60}{2\pi N} = \frac{7 \times 10^3 \times 60}{2 \times \pi \times 600} = 111.41\ Nm = 111.41 \times 10^3 Nmm$	1	8	15 7
	Using a rectangular sunk key. Determining the dimensions of the key from the proportions. Width of the key, $w = \frac{d}{4} = \frac{25}{4} = 6.25mm$ Thickness of the key,$t = \frac{d}{6} = \frac{25}{6} = 4.166mm \cong 4.2mm$	2		
	To find the length of key, torsional strength of the shaft is equated tot shear strength of the key. $\frac{\pi}{16} \times \tau_1 \times d^3 = l \times w \times \tau \times \frac{d}{2}$ $\frac{\pi}{16} \times 40 \times 25^3 = l \times 6.25 \times 40 \times \frac{25}{2}$ $l = \frac{\pi}{8} \times \frac{25^2}{6.25} = 39.26mm \cong 40mm$	2		
	To check the key for shear, $T = l \times w \times \tau \times \frac{d}{2}$ $111.41 \times 10^3 = 40 \times 6.25 \times \tau \times \frac{25}{2}$ $\tau = 35.65\ N/mm^2$ Since the induced shear stress is less than the permissible value of 60 N/mm^2 the key is safe under shear. To Check the key for crushing, $T = l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$ $111.41 \times 10^3 = 40 \times \frac{4.2}{2} \times \sigma_c \times \frac{25}{2}$ $\sigma_c = 106.1\ N/mm^2$ Since the induced crushing stress is less than the permissible value of $120 N/mm^2$ the key is safe under crushing. The dimensions of the key are $w = 6.25mm, t = 4.2mm, and\ l = 40mm$	3		
IV (b)	Given: $W = 75\ kN = 75 \times 10^3\ N,\ v = 300\ mm/min,\ p = 6mm,\ d_o = 40\ mm,$ $\mu = \tan\phi = 0.1$ Mean diameter of the screw,	1	7	
	$d = d_o - \frac{p}{2} = 40 - \frac{6}{2} = 37mm$ $\tan\alpha = \frac{p}{\pi d} = \frac{6}{\pi \times 37} = 0.0516$ Tangential force required at the circumference of the screw, $P = W\tan(\alpha + \phi) = W\left[\frac{\tan\alpha + \tan\phi}{1 - \tan\alpha\tan\phi}\right]$ $= 75 \times 10^3 \left[\frac{0.0516 + 0.1}{1 - 0.0516 \times 0.1}\right] = 11.43 \times 10^3 N$	2		
	Torque required to operate the screw, $T = P \times \frac{d}{2} = 11.43 \times 10^3 \times \frac{37}{2} = 211.45 \times 10^3 Nmm = 211.45\ Nm$ Speed of the screw in revolutions per minute (r.p.m.),	2		

$$N = \frac{Speed\ in\ mm/min}{Pitch\ in\ mm} = \frac{300}{6} = 50\ r.p.m.$$

Angular speed,

$$\omega = \frac{2\pi N}{60} = \frac{2\pi \times 50}{60} = 5.24\ rad/s$$

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Power of the motor, 2

$$= T \times \omega = 211.45 \times 5.24 = 1108\ W = 1.108\ kW$$

V (a) Given: $d_o = d$; $d_i = \frac{d_o}{2}$; or $k = \frac{d_i}{d_o} = \frac{1}{2} = 0.5$ 8 15

<u>Comparison of Weight</u>

Weight of the hollow shaft,

$$W_H = \frac{\pi}{4}[(d_o)^2 - (d_i)^2] \times length \times density \text{ .... (i)}$$

Weight of the Solid Shaft,

$$W_S = \frac{\pi}{4} \times d^2 \times length \times density \text{ .... (ii)}$$

Dividing equation (i) by equation (ii),

$$\frac{W_H}{W_S} = \frac{(d_o)^2 - (d_i)^2}{d^2} = \frac{(d_o)^2 - (d_i)^2}{(d_o)^2}$$ 2

$$= 1 - \frac{(d_i)^2}{(d_o)^2} = 1 - k^2 = 1 - (0.5)^2 = 0.75$$

<u>Comparison of strength</u>

Strength of the hollow shaft,

$$T_H = \frac{\pi}{16} \times \tau \times (d_o)^3 \times (1 - k^4) \text{ .... (iii)}$$

Strength of the solid shaft,

$$T_S = \frac{\pi}{16} \times \tau \times d^3 \text{ .... (iv)}$$

Dividing equation (iii) by equation (iv),

$$\frac{T_H}{T_S} = \frac{(d_o)^3(1 - k^4)}{d^3} = \frac{(d_o)^3(1 - k^4)}{(d_o)^3} = 1 - k^4$$ 3

$$= 1 - (0.5)^4 = 0.9375$$

<u>Comparison of Stiffness,</u>

$$Stiffness = \frac{T}{\theta} = \frac{G \times J}{L}$$

Stiffness of hollow shaft,

$$S_H = \frac{G}{L} \times \frac{\pi}{32}[(d_o)^4 - (d_i)^4] \text{ .... (v)}$$

Stiffness of solid shaft,

$$S_S = \frac{G}{L} \times \frac{\pi}{32} \times d^4 \text{ .... (vi)}$$

Dividing equation (v) by equation (vi),

$$\frac{S_H}{S_S} = \frac{(d_o)^4 - (d_i)^4}{d^4} = \frac{(d_o)^4 - (d_i)^4}{(d_o)^4} = 1 - \frac{(d_i)^4}{(d_o)^4}$$ 3

$$= 1 - k^4 = 1 - (0.5)^4 = 0.9375$$

V (b) Given: $P = 25\ kW = 25 \times 10^3\ W$; $N = 360\ rpm$ 1 7

$S_{yt} = S_{yc} = 400\ N/mm^2$; $S_{ut} = 200\ N/mm^2$

F.S. for shaft and key = 4; F.S. for sleeve = 6

Safe crushing stress for shaft and key, $\sigma_{cs} = \frac{400}{4} = 100\ N/mm^2$

Safe shear stress for shaft and key, $\tau_s = \frac{400}{4\times2} = 50\ N/mm^2$

Safe shear stress for sleeve, $\tau_c = \frac{200}{6\times2} = 16.67\ N/mm^2$

	1. Design for shaft Torque transmitted, $$T = \frac{P \times 60}{2\pi N} = \frac{25 \times 10^3 \times 60}{2\pi \times 360} = 663.14\ Nm = 663.14 \times 10^3\ Nmm$$ To find diameter of the shaft, $$T = \frac{\pi}{16} \times \tau_s \times d^3$$ $$663.14 \times 10^3 = \frac{\pi}{16} \times 50 \times d^3$$ $$d = 40.72\ mm \cong 45\ mm$$	2		9
	2. Design for sleeve Outer diameter of the muff, $$D = 2d + 13mm = 2 \times 45 + 13 = 103\ mm \cong 105\ mm$$ Length of the muff, $$L = 3.5d = 3.5 \times 45 = 157.5 \cong 160\ mm$$ To check the shear stress induced in the muff, $$T = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right]$$ $$663.14 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{105^4 - 45^4}{105}\right]$$ $$\tau_c = 3.02\ N/mm^2$$ Since the induced stress is less than the permissible stress of 16.67 N/mm^2 the design of muff is safe.	2		
	3. Design of key Selecting a rectangular sunk key, Width of key, $w = \frac{d}{4} = \frac{45}{4} = 11.25 \cong 12mm$ Thickness of key, $t = \frac{d}{6} = \frac{45}{6} = 7.5mm$ Length of each key, $l = \frac{L}{2} = \frac{160}{2} = 80\ mm$ To check shear stress induced in the key, $$T = l \times w \times \tau_s \times \frac{d}{2}$$ $$663.14 \times 10^3 = 80 \times 12 \times \tau_s \times \frac{45}{2}$$ $$\tau_s = 30.7\ N/mm^2$$ To check the crushing stress induced in the key. $$T = l \times \frac{t}{2} \times \sigma_{cs} \times \frac{d}{2}$$ $$663.14 \times 10^3 = 80 \times \frac{7.5}{2} \times \sigma_{cs} \times \frac{45}{2}$$ $$\sigma_{cs} = 98.24\ N/mm^2$$ Since the induced shear and crushing stresses are less than the permissible values, the design of key is safe.	2		
VI (a)	Given: $P = 15\ kW = 15 \times 10^3 W$; $N = 900\ rpm$; $Service\ Factor = 1.35$; $\tau_s = \tau_b = \tau_k = 40\ MPa = 40\ N/mm^2$; $\sigma_{cb} = \sigma_{ck} = 80\ MPa = 80\ N/mm^2$; $\tau_c = 8\ MPa = 8\ N/mm^2$. 1. Design for hub Torque transmitted, $$T = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2\pi \times 900} = 159.13\ Nm$$		8	15

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Maximum torque transmitted by the shaft,

$$T_{max} = T \times service\ factor = 159.13 \times 1.35 = 215\ Nm = 215 \times 10^3\ Nmm$$

To find the diameter of the shaft,

$$T_{max} = \frac{\pi}{16} \times \tau_s \times d^3$$

$$215 \times 10^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$d = 30.1\ mm \cong 35\ mm$$

Outer diameter of the hub,

$$D = 2d = 2 \times 35 = 70\ mm$$

Length of hub, 2

$$L = 1.5d = 1.5 \times 35 = 52.5\ mm$$

To check the shear stress induced in the hub,

$$T_{max} = \frac{\pi}{16} \times \tau_c \times \left[\frac{D^4 - d^4}{D}\right]$$

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_c \times \left[\frac{70^4 - 35^4}{70}\right]$$

$$\tau_c = 3.4\ N/mm^2 = 3.4\ MPa$$

Since the induced shear stress in the hub is less than the permissible value of 8 MPa the design of hub is safe.

2. Design of key

Since the crushing stress for the key is twice its shear stress a square key may be used.

Width of key, $w = \frac{d}{4} = \frac{35}{4} = 8.75 \cong 10\ mm$

Thickness of key, $t = w = 10\ mm$

Length of key, $l = L = 52.5\ mm$

To check the shear stress induced in the key,

$$T_{max} = l \times w \times \tau_k \times \frac{d}{2}$$

$$215 \times 10^3 = 52.5 \times 10 \times \tau_k \times \frac{35}{2}$$

$$\tau_k = 23.4\ N/mm^2$$

To check the crushing stress induced in the key,

$$T_{max} = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$$

$$215 \times 10^3 = 52.5 \times \frac{10}{2} \times \sigma_{ck} \times \frac{35}{2}$$

$$\sigma_{ck} = 46.8\ N/mm^2$$

Since the induced shear and crushing stresses in the key are less than the permissible values, the design for the key is safe.

3. Design of Flange

The thickness of flange,

$$t_f = 0.5d = 0.5 \times 35 = 17.5\ mm$$

To check the shear stress induced in the flange,

$$T_{max} = \pi D \times t_f \times \tau_c \times \frac{D}{2}$$

$$215 \times 10^3 = \pi 70 \times 17.5 \times \tau_c \times \frac{70}{2}$$

$$\tau_c = 1.6\ N/mm^2$$

Since the shear stress in the flange is less than 8 MPa, the design of flange is safe under shear. 2

4. Design for bolts

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Pitch circle diameter of bolts,

$$D_1 = 3d = 3 \times 35 = 105\ mm$$

Let d_1 = Nominal diameter of bolts.

n = Number of bolts

$n = 3$, since the diameter of shaft is 35 mm

To find nominal diameter of the bolts,

$$T_{max} = \frac{\pi}{4} \times (d_1)^2 \times \tau_b \times n \times \frac{D_1}{2}$$

$$215 \times 10^3 = \frac{\pi}{4} \times (d_1)^2 \times 40 \times 3 \times \frac{105}{2}$$ — 2

$$d_1 = 6.6\ mm$$

Assuming coarse threads, the nearest standard size of bolt is M 8.

Outer diameter of the flange,

$$D_2 = 4d = 4 \times 35 = 140\ mm$$

Thickness of the protective circumferential flange,

$$t_p = 0.25d = 0.25 \times 35 = 8.75\ mm$$

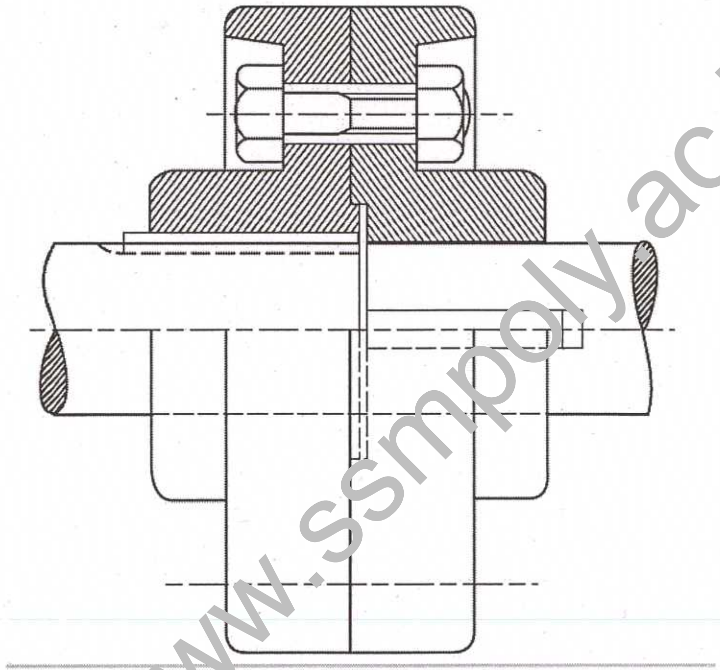


fig-2

VI (b) — 7

Given: $P = 600\ kW = 600 \times 10^3\ W;\ N = 500\ rpm;$ — 1

$\tau = 62.4\ MPa = 62.4\ N/mm^2;\ d_o = 2d_i;\ k = \frac{d_i}{d_o} = 0.5$

Torque, $T = \frac{P \times 60}{2\pi N} = \frac{600 \times 10^3 \times 60}{2\pi \times 500} = 11.5 \times 10^6\ Nmm$ — 2

$$T_{max} = 1.2T = 1.2 \times 11.5 \times 10^6 = 13.8 \times 10^6\ Nmm$$

$$T_{max} = \frac{\pi}{16} \times \tau \times {d_o}^3 \times (1 - k^4)$$ — 2

$$13.8 \times 10^6 = \frac{\pi}{16} \times 62.4 \times {d_o}^3 \times (1 - 0.5^4)$$

$$d_o = 106.3\ mm = 110\ mm$$ — 2

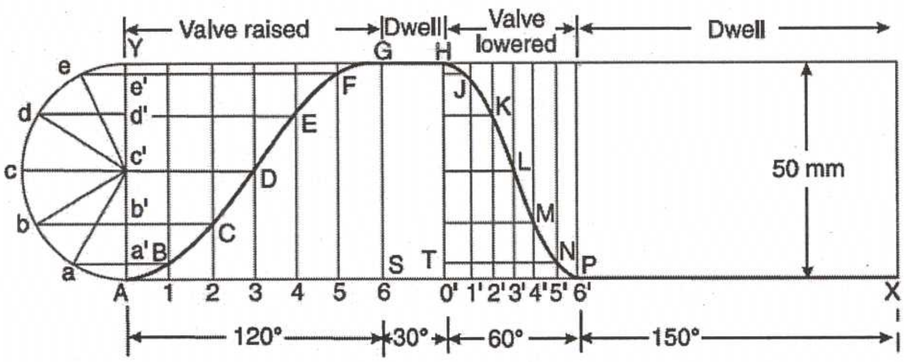
$$d_i = 0.5 \times d_o = 0.5 \times 110 = 55\ mm$$

12

VII (a)

Displacement Diagram

8



Displacement diagram

4

Cam profile

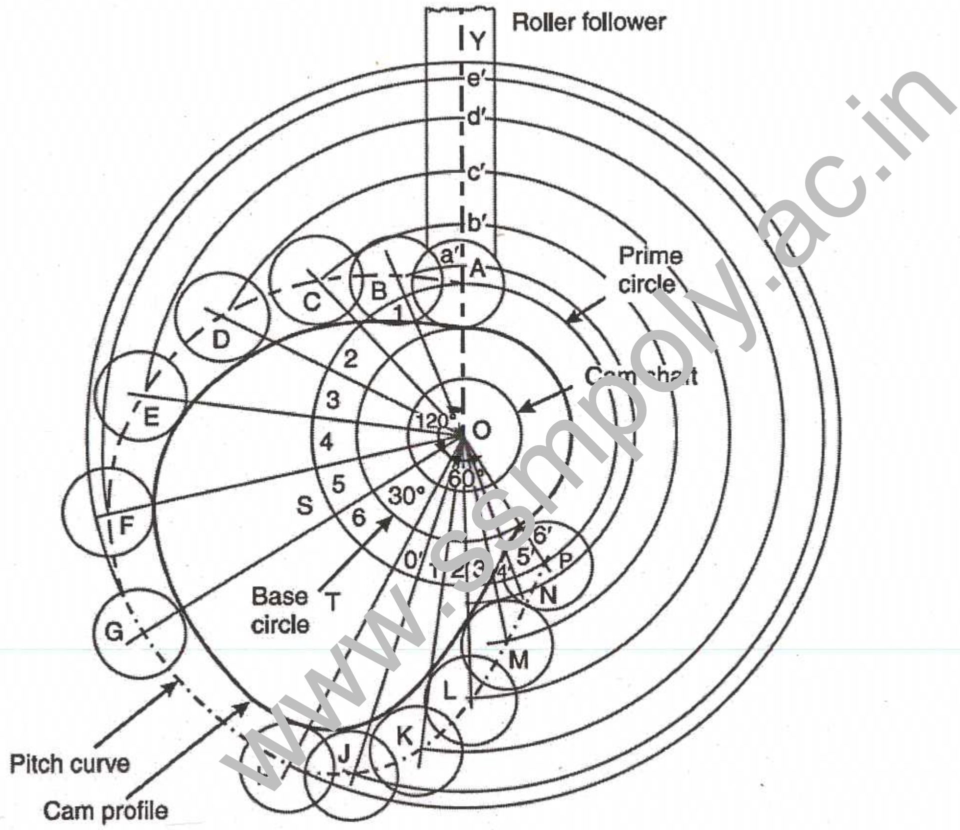


fig-4

VII (b)

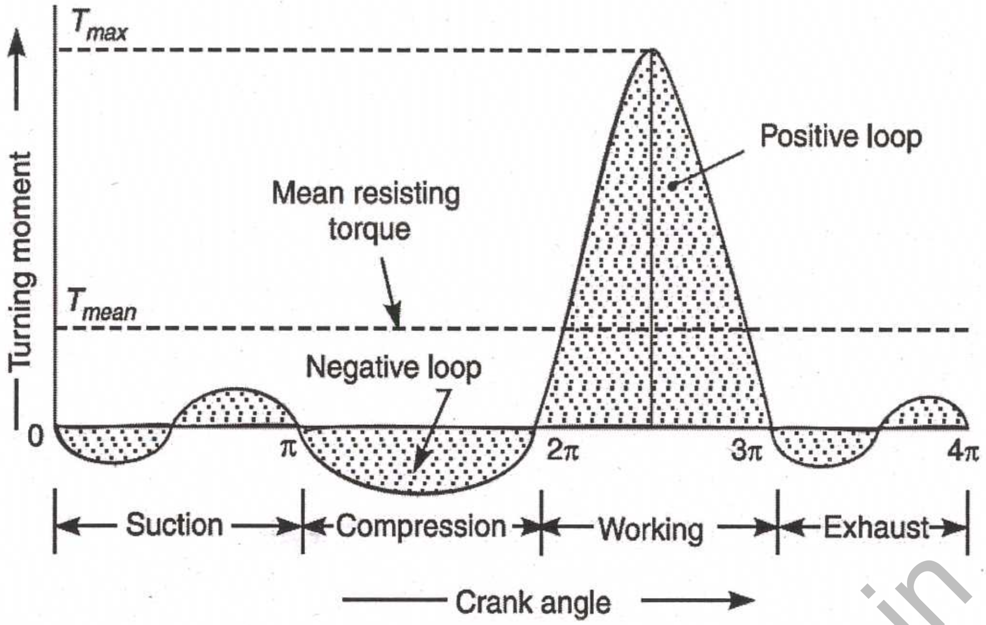
7

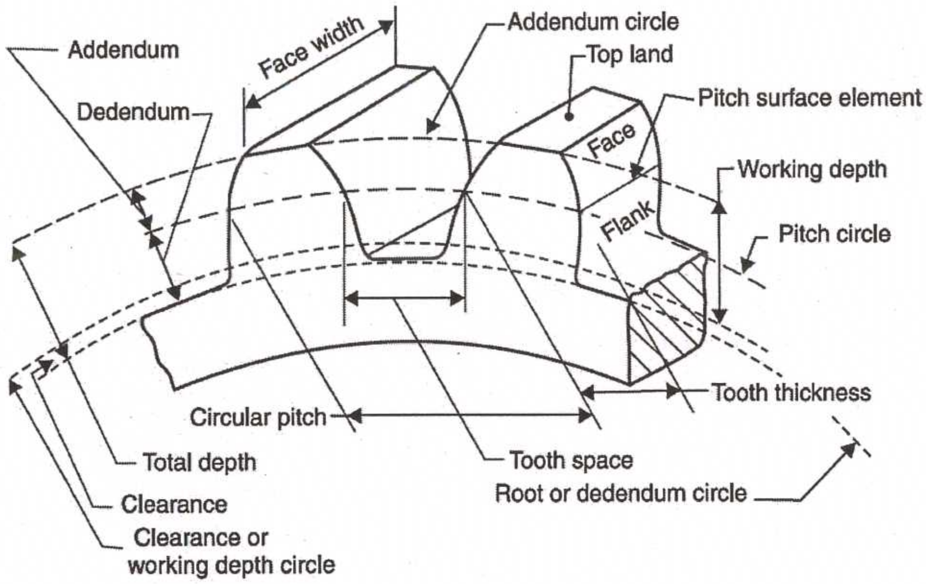
(i) Height of governor:- It is the vertical distance from the centre of the ball to a point where the axes of the arms (or arms produced) intersect on the spindle axis. It is usually denoted by h. 1

(ii) Equilibrium speed:- It is the speed at which the governor balls, arms etc., are in complete equilibrium and the sleeve does not tend to move upwards or downwards. 1

(iii) Sensitiveness:- Consider two governors A and B running at the same speed. When this speed increases or decreases by a certain amount, the lift of the sleeve of governor A is greater than the lift of the sleeve of governor B. It is then said that the governor A is more sensitive than the governor B. 1

	In general, the greater the lift of the sleeve corresponding to a given fractional change in speed, the greater is the sensitiveness of the governor. It may also be stated in another way that for a given lift of the sleeve, the sensitiveness of the governor increases as the speed range decreases.	2		13
	The sensitiveness is defined as the ratio of the difference between the maximum and minimum equilibrium speeds to the mean equilibrium speed. $$Sensitiveness = \frac{2(\omega_2 - \omega_1)}{\omega_1 + \omega_2}$$			
	(iv) Stability:- A governor is said to be stable when for every speed within the working range there is a definite configuration i.e. there is only one radius of rotation of the governor balls at which the governor is in equilibrium. For a stable governor, if the equilibrium speed increases, the radius of governor balls must also increase.	2		
	~~(v) Hunting:- A governor is said to be hunt if the speed of the engine fluctuates continuously above and below the mean speed. This is caused by a too sensitive governor which changes the fuel supply by a large amount when a small change in the speed of rotation takes place.~~	~~2~~		
VIII (a)	Given : $W = 150$ kN $= 150\times10^3$ N; $d = 300$ mm $= 0.3$ m; $N = 1800$ r.p.m. ; $p = 1.6$ N/mm^2; Z = 0.02 kg / m-s; $c = 0.25$ mm	1	8	15
	1. Length of the bearing Projected area, $$A = l \times d = l \times 300 = 300l\ mm^2$$ Allowable bearing pressure, $$p = 1.6 = \frac{W}{A} = \frac{150 \times 10^3}{300l} = \frac{500}{l}$$ $$l = \frac{500}{1.6} = 312.5\ mm$$	2		
	2. Amount of heat to be removed by the lubricant Coefficient of friction for the bearing, $$\mu = \frac{33}{10^8}\left(\frac{ZN}{p}\right)\left(\frac{d}{c}\right) + k = \frac{33}{10^8}\left(\frac{0.02 \times 1800}{1.6}\right)\left(\frac{300}{0.25}\right) + 0.002$$ $$= 0.009 + 0.002 = 0.011$$	2		
	Rubbing Velocity, $$V = \frac{\pi dN}{60} = \frac{\pi \times 0.3 \times 1800}{60} = 28.3\ m/s$$ Amount of heat to be removed by the lubricant, $$Q_g = \mu WV = 0.011 \times 150 \times 10^3 \times 28.3 = 46695\ J/s$$ $$= 46.7\ kW$$	3		
VIII (b)	A turning moment diagram for a four stroke cycle internal combustion engine is shown in figure. In a four stroke cycle internal combustion engine, there is one working stroke when the crank has turned through two revolutions, i.e. 720° (or 4π radians).		7	
	Since the pressure inside the engine cylinder is less than the atmospheric pressure during the suction stroke, therefore a negative loop is formed as shown in figure. During the compression stroke, the work is done on the gases, therefore a higher negative loop is obtained. During the expansion or working stroke, the fuel burns and the gases expand, therefore a large positive loop is obtained. In this stroke, the work is	3		

	done by the gases. During exhaust stroke, the work is done on the gases, therefore a negative loop is formed. 	fig-4		14
IX (a)	Given: $d_1 = 450\ mm = 0.45\ m;\ r_1 = 0.225\ m;\ d_2 = 200\ mm = 0.2\ m;$ $r_2 = 0.1\ m;\ x = 1.95\ m;\ N_1 = 200\ rpm;\ T_1 = 1\ kN = 1000\ N;\ \mu = 0.25$ Speed of the belt, $v = \frac{\pi d_1 N_1}{60} = \frac{\pi \times 0.45 \times 200}{60} = 4.714\ m/s$	1	8	15
	Length of the belt, $L = \pi(r_1 + r_2) + 2x + \frac{(r_1 + r_2)^2}{x}$ $= \pi(0.225 + 0.1) + 2 \times 1.95 + \frac{(0.225 + 0.1)^2}{1.95} = 4.975\ m$	2		
	Angle of contact between the belt and each pulley, $\sin\alpha = \frac{r_1 + r_2}{x} = \frac{0.225 + 0.1}{1.95} = 0.1667$ $\alpha = 9.6^o$ $\theta = 180 + 2 \times \alpha = 180 + 2 \times 9.6 = 199.2^o$ $= 199.2 \times \frac{\pi}{180} = 3.477\ rad$	2		
	To find T_2, $2.3 \log\left(\frac{T_1}{T_2}\right) = \mu \times \theta = 0.25 \times 3.477 = 0.8692$ $\frac{T_1}{T_2} = 2.387$ $T_2 = \frac{T_1}{2.387} = \frac{1000}{2.387} = 419\ N$	2		
	Power Transmitted, $P = (T_1 - T_2) \times v = (1000 - 419) \times 4.714 = 2740\ W = 2.74\ kW$	3		

IX (b)	 1. Pitch circle:- It is an imaginary circle which by pure rolling action, would give the same motion as the actual gear. 2. Pitch circle diameter:- It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also known as pitch diameter. 3. Pitch point:- It is a common point of contact between two pitch circles. 4. Pitch surface:- It is the surface of the rolling discs which the meshing gears have replaced at the pitch circle. 5. Pressure angle or angle of obliquity:- It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point. 6. Addendum:- It is the radial distance of a tooth from the pitch circle to the top of the tooth. 7. Dedendum:- It is the radial distance of a tooth from the pitch circle to the bottom of the tooth. 8. Addendum circle:- It is the circle drawn through the top of the teeth and is concentric with the pitch circle. 9. Dedendum circle:- It is the circle drawn through the bottom of the teeth. It is also called root circle. 10. Circular pitch:- It is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth. It is usually denoted by p_c. Mathematically, circular pitch, $p_c = \frac{\pi D}{T}$ where D = Diameter of the pitch circle, and T = Number of teeth on the wheel. 11. Diametral pitch:- It is the ratio of number of teeth to the pitch circle diameter in millimeters. 12. Module:- It is the ratio of the pitch circle diameter in millimeters to the number of teeth. 13. Clearance:- It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top of the meshing gear is known as clearance circle. 14. Total depth:- It is the radial distance between the addendum and the dedendum circles of a gear. It is equal to the sum of the addendum and dedendum. 15. Working depth:- It is the radial distance from the addendum circle to the clearance circle.	fig-4 3 any six terms	7	15

16. Tooth thickness:- It is the width of the tooth measured along the pitch circle.
17. Tooth space:- It is the width of space between the two adjacent teeth measured along the pitch circle.

X (a) Given: $N_1 = 200\,rpm$; $N_2 = 300\,rpm$; $P = 6\,kW = 6 \times 10^3\,W$; $b = 100\,mm$; $t = 10\,mm$; $x = 4\,m$; $d_2 = 0.5\,m$; $\mu = 0.3$ — 1 — 8 — 15

To find the diameter of the larger pulley.

$$\frac{N_2}{N_1} = \frac{d_1}{d_2}$$

$$d_1 = \frac{N_2 \times d_2}{N_1} = \frac{300 \times 0.5}{200} = 0.75\,m$$ 1

To find velocity of belt.

$$v = \frac{\pi d_2 N_2}{60} = \frac{\pi \times 0.5 \times 300}{60} = 7.855\,m/s$$

$$\sin\alpha = \frac{r_1 - r_2}{x} = \frac{d_1 - d_2}{2x} = \frac{0.75 - 0.5}{2 \times 4} = 0.03125\,rad = 1.8^o$$ 2

Angle of contact,

$$\theta = 180^o - 2\alpha = 180 - 2 \times 1.8 = 176.4^o$$

$$= 176.4 \times \frac{\pi}{180} = 3.08\,rad$$ 1

Considering the ratio of belt tensions,

$$2.3\log\left(\frac{T_1}{T_2}\right) = \mu \times \theta = 0.3 \times 3.08 = 0.924$$

$$\log\left(\frac{T_1}{T_2}\right) = \frac{0.924}{2.3} = 0.4017$$

$$\frac{T_1}{T_2} = 2.52 \,\ldots\ldots\ldots\, (i)$$ 1

Power transmitted,

$$P = (T_1 - T_2) \times v$$

$$6 \times 10^3 = (T_1 - T_2) \times 7.855$$

$$T_1 - T_2 = \frac{6\times10^3}{7.855} = 764\,N \,\ldots\ldots\ldots\, (ii)$$

Solving Equations (i) and (ii),

$$T_1 = 1267\,N, T_2 = 503\,N$$

To find the stress induced, 2

$$T_1 = \sigma \times b \times t$$

$$1267 = \sigma \times 100 \times 10$$

$$\sigma = 1.267\ N/mm^2 = 1.267\,MPa$$

X (b) — 7

When the axes of the first gear (i.e. first driver) and the last gear (i.e. last driven or follower) are co-axial, then the gear train is known as reverted gear train.

The gear 1 (i.e. first driver) drives the gear 2 (i.e. first driven or follower) in the opposite direction. Since the gears 2 and 3 are mounted on the same shaft, therefore they form a compound gear and the gear 3 will rotate in the same direction as that of gear 2. The gear 3 (which is now the second driver) drives the gear 4 (i.e. the last driven or follower) in the same direction as that of gear 1. Thus in a reverted gear train, the motion of the first gear and the last gear is like. 2

Let

T1 = Number of teeth on gear 1,
N1 = Speed of gear 1 in r.p.m.

Similarly,

T2, T3, T4 = Number of teeth on respective gears, and

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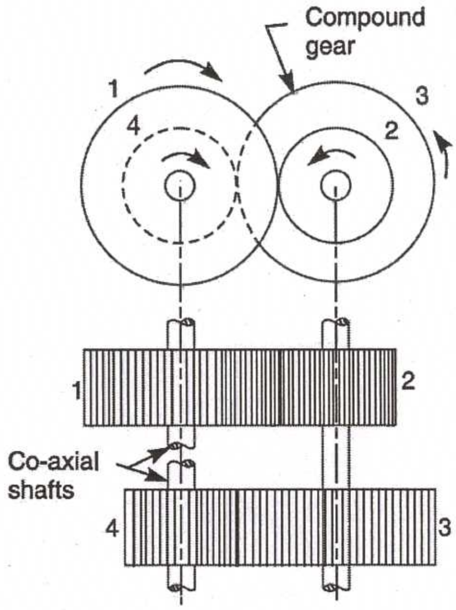
N2, N3, N4 = Speed of respective gears in r.p.m.



Reverted gear train.

fig-2

The circular pitch or module of all the gears is assumed to be same, therefore number of teeth on each gear is directly proportional to its circumference or radius.

T1 + T2 = T3 + T4 .... (ii)

and

$$Speed\ Ratio = \frac{Product\ of\ number\ of\ teeth\ in\ driven\ gears}{Product\ of\ number\ of\ teeth\ in\ driver\ gears}$$

$$\frac{N_1}{N_4} = \frac{T_2 \times T_4}{T_1 \times T_3}$$

The reverted gear trains are used in automotive transmissions, lathe back gears, industrial speed reducers, and in clocks (where the minute and hour hand shafts are co-axial).

3