Index/ Instructor's evaluation of experiment reports

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9	To study various types of gear- Helical, cross helical, worm, bevel gear			
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Experiment No.-1

Aim: To study the different types of centrifugal and inertia governors and demonstrate any one.

REQUIREMENT: Universal Governor Apparatus & Tachometer.

THEORY:

INTRODUCTION:

The function of a governor is to regulate the mean speed of an engine, when there are variations in loads e.g. when load on an engine increase or decrease, obviously its speed will, respectively decrease or increase to the extent of variation of load. This variation of speed has to be controlled by the governor, within small limits of mean speed. This necessitates that when the load increase and consequently the speed decreases, the supply of fuel to the engine has to be increased accordingly to compensate for the loss of the speed, so as to bring back the speed to the mean speed. Conversely, when the load decreases and speed increases, the supply of fuel has to be reduced.

The function of the governor is to maintain the speed of an engine within specific limit whenever there is a variation of load. The governor should have its mechanism working in such a way, that the supply of fuel is automatically regulated according to the load requirement for maintaining approximately a constant speed. This is achieved by the principle of centrifugal force. The centrifugal type governors are based on the balancing of centrifugal force on the rotating balls by an equal and opposite radial force, known as the controlling force.

Governors are broadly classified as:

a) Centrifugal Governors.
b) Inertia Governors.

The centrifugal governors are based on the balancing of centrifugal force on the rotating balls by an equal and opposite radial force, known as controlling force.

In Inertia governors the position of the balls are affected by the forces set by an angular acceleration or deceleration of the given spindle in addition to centrifugal forces on the balls.

DESCRIPTION:

The apparatus is designed to exhibit the characteristics of the spring-loaded governor and centrifugal governor. The experiments shall be performed on following centrifugal type governors:

1. Watt governor
2. Porter governor
3. Proell governor
4. Hartnell governor

WATT GOVERNOR

It is the simplest form of a centrifugal governor, which is known as Watt Governor. It is the original form of the governor used by Watt on early steam engines. It consists of two balls which are attached to the spindle with the helps of links or arms.

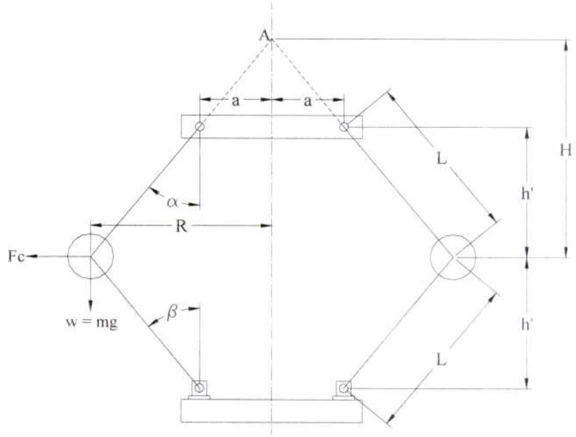


Figure: Watt Governor

The drive unit consists of a DC motor connected to the shaft through V belt. Motor and shaft are mounted on a rigid MS base frame in vertical position. The spindle is supported in ball bearing.

PORTER GOVERNOR

In case of porter governor, a central heavy load is attached to the sleeve. The central load and sleeve moves up & down the spindle.

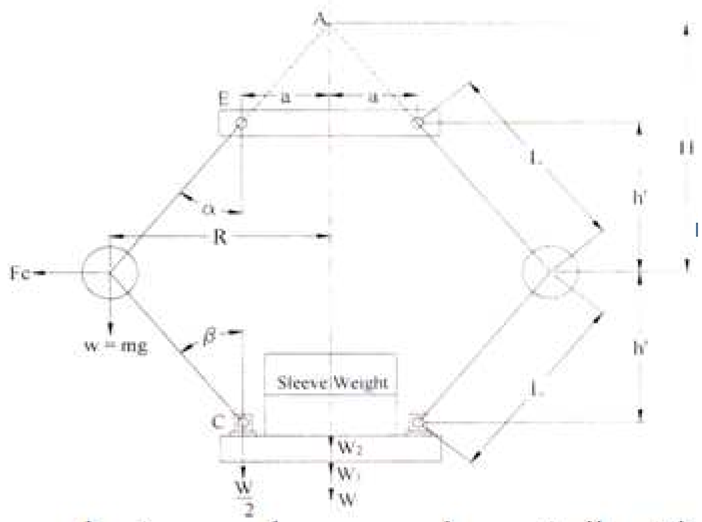


Figure: Porter Governor

The optional governor mechanism can be mounted on spindle. The speed control unit controls the precise speed and speed of the shaft is measured with the help of hand tachometer. A counter sunk has been provided at the topmost bolt of the spindle. A graduated scale is fixed to measure the sleeve lift.

PROELL GOVERNOR

Proell governor is similar to the porter governor having a heavy central load at sleeve. But it differs from porter governor in the arrangement of balls. The balls are carried on the extension of the lower arms instead of at the junction of upper and lower arms.

The center sleeve of the Porter and Proell governors incorporates a weight sleeve to which weights can be added. The Hartnell governor consists of a frame, spring and bell crank lever. The spring tension can be increased or decreased to study the governor.

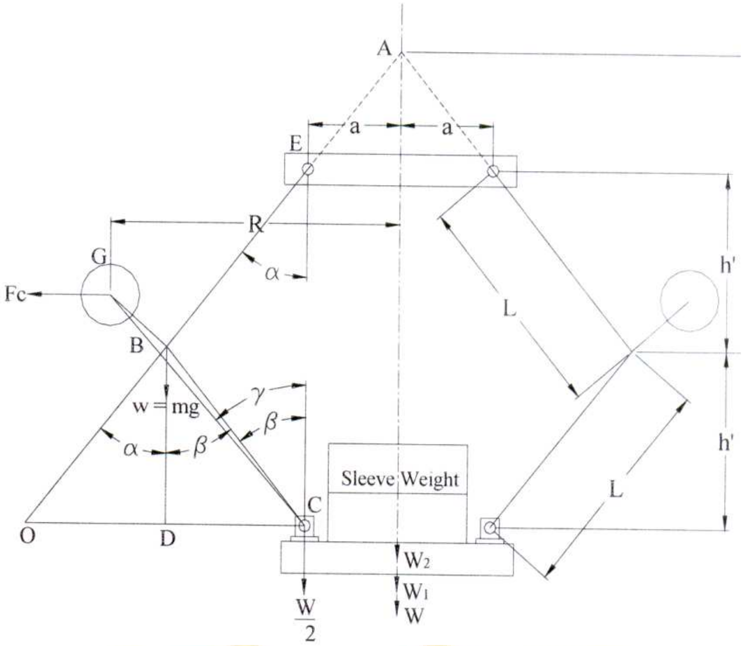


Figure: Proell governor

HARTNELL GOVERNOR

Hartnell governor is spring controlled governor. Two bell crank levers, each carrying a ball at on one end and a roller on the other end. The roller fit into a groove in the sleeve. The frame is attached to the governor spindle and hence rotates with it. A helical spring in compression provides equal downward forces on the two rollers through a collar on the sleeve.

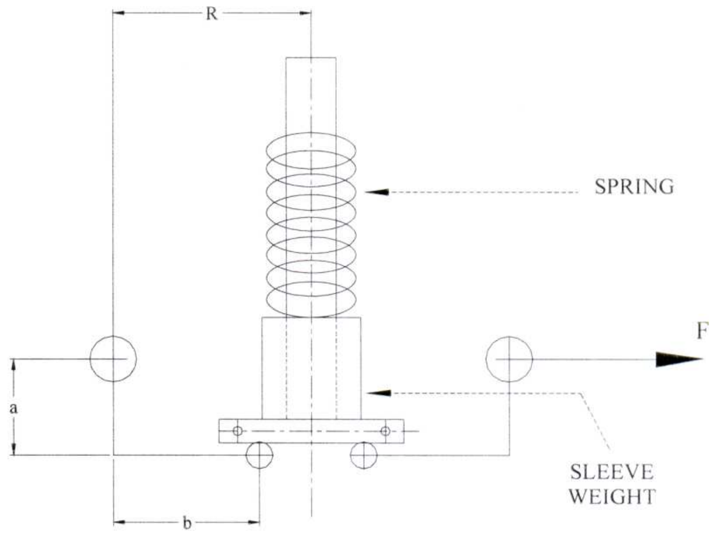


Figure: Hartnell Governor

PROCEDURE:

Starting Procedure:

1. Assemble the governor to be tested.
2. Complete the electrical connections.
3. Switch ON the main power.
4. Note down the initial reading of pointer on the scale.
5. Switch On the rotary switch.
6. Slowly increase the speed of governor until the sleeve is lifted from its initial position by rotating Variac.

7. Let the governor be stabilized.
8. Increase the speed of governor in steps to get the different positions of sleeve lift at different RPM.
9. Increase the speed of governor in steps to get the different positions of sleeve lift at different RPM.

Closing Procedure:

1. Decrease the speed of governor gradually by bringing the Variac to zero position and then switch off the motor.
2. Switch OFF all switches.
3. Disconnect all the connections.
4. Draw the graph for governor as stated further in manual.
5. Repeat the experiment for different type of governor.

WATT GOVERNOR

It is assumed that mass of the arms; links & sleeve are negligible in comparison with the mass of the balls and are neglected in the analysis.

In Figure 1, taking moments about point A

$F_c \times H = mg \times R$

i.e., $m\omega^2 R \times H = mg \times R$

Therefore, $H = \dfrac{g}{\omega^2}$

Also $\omega = \dfrac{2\pi N}{60}$ radians/sec

Therefore, $H = \dfrac{g}{\left(\dfrac{2\pi N}{60}\right)^2}$

$= \dfrac{91.2\ g}{N^2}$

$N = \sqrt{\dfrac{91.2\ g}{H}}$

FORMULAE:

1. Initial reading of pointer on scale, X’ = mm
2. Height gained by sleeve, X = (X” – X’) mm
3. Height, h = $\left\{h - \left(\dfrac{X}{2}\right)\right\}$ mm
4. $\alpha = \cos^{-1}\left(\dfrac{h}{L}\right)$
5. Governor height, H = $\left\{(a) + h\right\}$ mm

$\tan\alpha$

6. Governor speed (theo.), Ntho = $\sqrt{(\underline{91.2 \text{ x } g \text{ x } 1000}}$ RPM
H

7. Radius of rotation, R = $\{a + (L \sin\alpha)\}$ mm

8. Centrifugal force (actual), F_{act} = $\dfrac{w \times R \times \omega^2}{g \times 1000}$ kg

9. Angular Velocity, ω = $\dfrac{2 \times \pi \times N}{60}$ radians/sec

10. Centrifugal force (theo), F_{act} = $\dfrac{w \times R}{H}$ kg

OBSERVATIONS & CALCULATIONS:

1. Length of each link (L) = 105 mm
2. Initial height, (h') = 100 mm
3. Weight of each ball (w) = 0.75 + 0.75 = 1.5 kg
   = 0.37 + 0.37 = 0.74 kg
4. Acceleration due to gravity (g) = 9.81 m/s^2
5. Weight of Aluminium sleeve = 1.04 kg
6. Distance of pivot to center of spindle (a) = 50 mm
7. Reading of pointer on scale at N rpm = X' mm

OBSERVATION TABLE:

Initial reading of pointer on scale, X' = mm

Selected ball weight, w = kg

S. No.	Sleeve displacement, X" mm	Speed, N_{act} RPM
1		

Plot the graph for following curves: -

1. Sleeve (X) vs. N_{tho}
2. Sleeve (X) vs $N_{act.}$

PORTER GOVERNOR:

Porter Governor differs from Watt's Governor only in extra sleeve weight, else is similar to Watt Governor.

FORMULAE:

1. Initial reading of pointer on scale, X' = mm

2. Height gained by sleeve, X = (X" – X') mm

3. Height, h = $\left\{h - \left(\frac{X}{2}\right)\right\}$ mm

4. $\alpha = \cos^{-1}\left(\frac{h}{L}\right)$

5. Governor height, H = $\left\{\frac{(a)}{\tan\alpha} + h\right\}$ mm

6. Radius of rotation, R = $\{a + (L \sin\alpha)\}$ mm

7. Centrifugal force (actual), $F_{act} = \frac{w \times R \times \omega^2}{g \times 1000}$ kg

8. Angular Velocity, $\omega = 2 \times \pi \times N/60$ radians/sec

9. Governor speed (theo.), Ntho = $\sqrt{\frac{w + W}{w} \times \frac{91.2\ g \times 1000}{H}}$ RPM

10. Centrifugal force (theo), $F_{act} = \left[w + \frac{W}{2}(1+k)\right] \tan\alpha$ kg

Where, $k = \frac{\tan\beta}{\tan\alpha}$

$\tan\alpha = \frac{r}{h}$

Length of arms is equal to the length of links and the points E and C lies on the same vertical line.

Then $\tan\beta = \tan\alpha$

Or $k = 1$

Therefore, $F_{act.} = [(w + W) \times \tan\alpha]$ kg

OBSERVATION & CALCULATION:

1.	Length of each link (L)	=	105 mm		
2.	Initial height, (h')	=	100 mm		
3.	Radius of rotation ®	=	mm		
4.	Weight of each ball (w)	=	0.7 + 0.7	=	1.4 kg
		=	0.37 + 0.37	=	0.74 kg
5.	Acceleration due to gravity (g)	=	9.81 m/s^2		
6.	Distance of pivot to center of spindle (a)	=	50 mm		
7.	Weight of Cast Iron sleeve (W_1)	=	2.06 kg		
8.	Dead weight applied on sleeve (W_2)	=	0.950 /0.7 kg (one each)		
9.	Total dead weight on sleeve (W)	=	(W_1 + W_2)		
10.	Reading of pointer on scale at N rpm	=	X' mm		

OBSERVATION TABLE:

Initial reading of pointer on scale, X" = mm

Weight applied on sleeve, (W_2) = kg

Selected ball weight, w = kg

S. No.	Sleeve displacement, X" mm	Speed, N_{act} RPM
1		

Plot the graph for following curves: -

1. Sleeve (X) vs. N_{tho}
2. Sleeve (X) vs $N_{act.}$

PROELL GOVERNOR:

FORMULAE:

1. Initial reading of pointer on scale, X' = mm

2.	Height gained by sleeve, X	=	(X” – X’) mm
3.	Height, h	=	$\{h - \frac{X}{2}\}$ mm
4.	α	=	$\cos^{-1} \frac{h}{L}$
5.	Governor height, H	=	$\{\frac{a}{\tan\alpha} + h\}$ mm
6.	′Y	=	[(α - α’) + Y]
7.	Radius of rotation, R	=	{a + (GC sinY)} mm
8.	Centrifugal force (actual), F_{act}	=	$\frac{w \times R \times \omega^2}{g \times 1000}$ kg
9.	Angular Velocity, ω	=	$\frac{2 \times \pi \times N}{60}$ radians/sec
10.	DG	=	(GC cosY) mm
11.	BD	=	{h – (X/2)} = h mm
12.	Centrifugal Force (theo) F_{act}	=	$\{(W+2w)*\frac{BD * \tan\alpha}{GC}\}-\{w*\frac{\tan Y}{\cos Y}\}$ kg
13.	Governor speed (theo.), Ntho	=	$\sqrt{\frac{(W + w)* F_{act}*g*1000}{w * R}}$ RPM

Length of arms is equal to the length of links and the points E and C lies on the same vertical line.

Then tanβ = tanα

OBSERVATION & CALCULATION:

1.	Length of each link (L)	=	105 mm		
2.	Initial height, (h’)	=	100 mm		
3.	Initial angle, α	=	17.753°		
4.	Initial angle, ′Y’	=	23.611°		
5.	Initial Radius of rotation ®	=	mm		
6.	Weight of each ball (w)	=	0.7 + 0.7	=	1.4 kg

= 0.37 + 0.37 = 0.74 kg

7. Acceleration due to gravity (g) = $9.81\ m/s^2$

8. Distance of pivot to center of spindle (a) = 50 mm

9. Displacement between points G & C of lower link

10. Weight of Cast Iron sleeve (W_1) = 2.06 kg

11. Dead weight applied on sleeve (W_2) = 0.950 /0.7 kg (one each)

12. Total dead weight on sleeve (W) = $(W_1 + W_2)$

Reading of pointer on scale at N rpm = X” mm

OBSERVATION TABLE:

Initial reading of pointer on scale, X’ = mm

Weight applied on sleeve, (W_2) = kg

Selected ball weight, w = kg

S. No.	Sleeve displacement, X” mm	Speed, N_{act} RPM
1		
2		
3		
4		

Plot the graph for following curves: -

1. Sleeve (X) vs. N_{tho}
2. Sleeve (X) vs. $N_{act.}$

HARTNELL GOVERNOR:

FORMULAE:

1. Initial reading of pointer on scale, X” = mm

2. Height gained by sleeve, X = (X” – X’) mm

3. Radius of rotation, R = $\left\{R' + X \times \frac{a}{b}\right\}$ mm

4. Centrifugal force (actual), F_{act} = $\frac{w \times R \times \omega^2}{g \times 1000}$ kg

5. Angular Velocity, ω = $\frac{2 \times \pi \times N}{60}$ radians/sec

6. Force exerted by spring, S = $[2 \times F \times a/b] - W$ kg

7. Stiffness of spring, s = $2 * (a/b)^2 * (\underline{F_c - F_c'})$

R-R'

If ω = 0

Then F_c' = 0

Hence, s = $2*2 * (a/b)^2*(F_c)$ kg/mm

R-R'

Length of arms is equal to the length of links and the points E and C lies on the same vertical line.

Then $\tan\beta = \tan\alpha$

OBSERVATION & CALCULATION:

1. Length of arm (a) = 75 mm
2. Length of arm (b) = 120 mm
3. Initial Radius of rutation R' = 182.6 mm
4. Weight of each ball (w) = 0.7 + 0.7 = 1.4 kg

   = 0.37 + 0.37 = 0.74 kg
5. Acceleration due to gravity (g) = 9.81 m/s^2
6. Weight of Cast Iron sleeve (W_1) = 2.06 kg
7. Dead weight applied on sleeve (W_2) = 0.347 gm
8. Total dead weight on sleeve (W) = $(W_1 + W_2)$
9. Initial reading of pointer on scale, X" = mm
10. Reading of pointer on scale at N rpm = X" mm

OBSERVATION TABLE:

Data:

Initial reading of pointer on scale, X"' = mm

Selected ball weight, w = kg

S. No.	Sleeve displacement, X" mm	Speed, N_{act} RPM
1		

Plot the graph for following curve: -

1. Sleeve (X) vs. $N_{act.}$

PRECAUSTIONS:

1. No voltage fluctuation is desirable, as it may hamper results.
2. Always increase the speed gradually.
3. Take the sleeve displacement reading when steady state is achieved.
4. At higher speed the load on sleeve does not hit the upper sleeve of the governor.
5. Always switch off the motor after bringing the variac to zero position.
6. Keep the apparatus free from dust.
7. Before performing any experiment clean the sleeve properly and lubricate it properly.

RESULT:

Studied the different types of centrifugal and inertia governors & graph is plotted between r.p.m & Displacement.

Viva-Voce:

Experiment No.-2

Aim: To study the different types of brakes.

Requirement: Block or shoe brake, Band brake, Band and Block brake, Internal expanding shoe brake models.

Thoery: A brake is an appliance used to apply frictional resistance to a moving body to stop or retard it by absorbing its kinetic energy.

Types of brakes:

The following are the main types of mechanical brakes.

(i) Block or shoe brake
(ii) Band brake
(iii) Band and block brake
(iv) Internal expanding shoe brake

BLOCK OR SHOE BRAKE:

A block or shoe brake consists of a block or shoe which is pressed against a rotating drum. The force on the drum is increased by using a lever. If only one block is used for the purpose, a side thrust on the bearing of the shaft supporting the drum will act. This can be prevented by using two blocks on the two sides of the drum. This also doubles the braking torque.

A material softer than that of the drum or the rim of the wheel is used to make the blocks so that these can be replaced easily on wearing. Wood and rubber are used for light and slow vehicles and cast steel for heavy and fast ones.

Let,

r = radius of the drum

μ = coefficient of friction

F_r = radial force applied on the drum

R_n = normal reaction on the block (= F_r)

F = Force applied at the lever end

F_f = frictional force = μR_n

Assuming that the normal reaction R_n and the frictional force F_f act at the mid-point of the block.

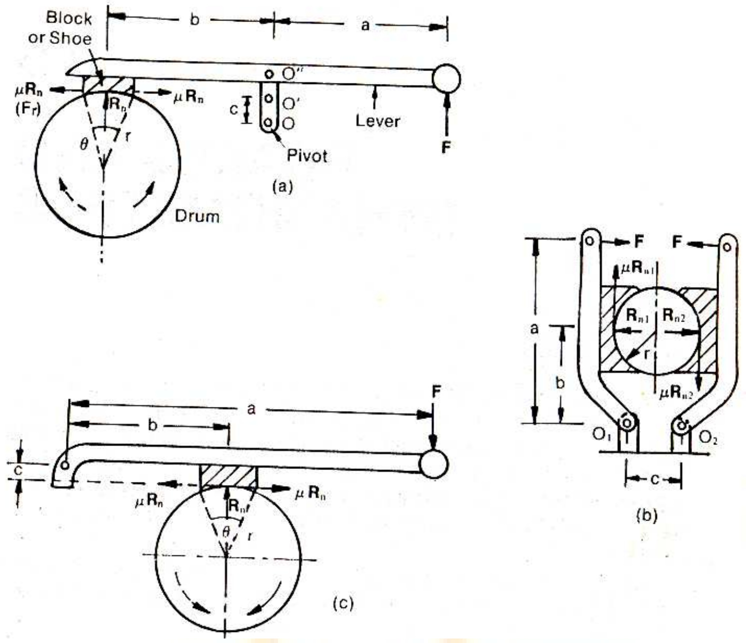


Figure: Different types of brakes

Breaking torque on the drum = frictional force × radius

Or $T_B = \mu R_n \times r$

To obtain R_n, consider the equilibrium of the block as follows.

The direction of the frictional force on the drum is to be opposite to that of its rotation while on the block it is in the same direction. Taking moments about the pivot O

$$F \times a - R_n \times b + \mu R_n \times c = 0$$

$$R_n = \frac{Fa}{b - \mu c}$$

Also

$$F = R_n \left[\frac{b - \mu c}{a}\right]$$

When $b = \mu c$,

$F = 0$,

Which implies that the force needed to apply the brake is virtually zero, or that once contact is made between the block and the drum, the brake is applied itself. Such a brake is known as a self-locking brake.

As the moment of the force F_r about O is in the same direction as that of the applied force F, F_f aids in applying the brake. Such a brake is known as self-energized brake.

If the rotation of the drum is reversed, i.e. it is made clockwise

$F = R_n \frac{[(b + \mu)c]}{a}$ which shows that required force F will be far greater than what it would be when the drum rotates counter-clockwise.

If the pivot lies on the line of action of F_f i.e. O', c = 0 and

$F = R_n \frac{a}{b}$ which is valid for clockwise as well as for counter-clockwise rotation.

If c is made negative, i.e. if the pivot of O"

$$F = R_n \left\{ \frac{b + \mu c}{a} \right\}$$ for counter-clockwise rotation

and

$$F = R_n \left\{ \frac{b - \mu c}{a} \right\}$$ for clockwise rotation.

In case the pivot is provided on the same side of the applied force and the block as shown in fig. 9.1 the equilibrium condition can be considered accordingly.

In the above treatment, it is assumed that the normal reaction and the frictional force act at the mid-point of the block. However, this is true only for small angles of contact. When the angle of contact is more than 40º, the normal pressure is less at the ends than at the center. In that case, μ given by

$$\mu' = \mu \left[\frac{4 \sin\left(\frac{\theta}{2}\right)}{\theta + \sin\theta} \right]$$

BAND BRAKE:

It consists of a rope, belt or flexible steel band (lined with friction material), which is pressed against the external surface of a cylindrical drum when the brake is applied. The force is applied at the free end of a lever.

Brake torque on the drum = $(T_1 - T_2)r$

Where r is the effective radius of the drum.

The ratio of the tight and the slack side tensions is given by $T_1/T_2 = e^{\mu\theta}$ on the assumption that the band is on the point of slipping on the drum.

that the band is on the point of slipping on the drum.

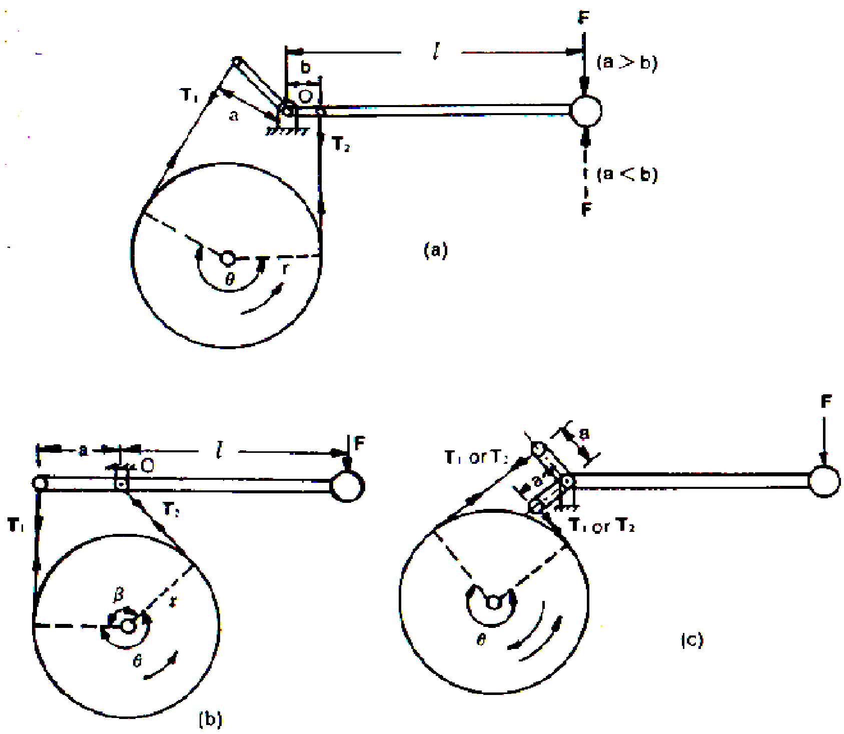


Figure: Band brake

The effectiveness of the force F depends upon

- the direction of rotation of the drum
- the ratio of lengths a and b
- the direction of the applied force F

To apply the brake to the rotating drum, the band has to be tightened on the drum. This is possible if

(i) F is applied in the downward direction when a > b.
(ii) F is applied in the upward direction when a < b.

If the force applied is not as above, the band is further loosened on the drum, which means no braking effect is possible.

(a) Rotation counter-clockwise:
For counter-clockwise rotation of the drum, the tight and the slack sides of the band will be as shown in fig.

Considering the forces acting on the lever and taking moments about the pivot,

$$Fl \ - T_1a + T_2b = 0$$

$$\text{Or} \quad F = \frac{T_1a + T_2b}{l}$$

As $T_1 > T_2$ and a > b under all conditions, the effectiveness of the brake will depend upon the force F.

(b) Rotation clockwise:

In this case, the right and the slack sides are reversed, i.e. T_2 becomes greater than T_1.

Then

$T_1 < T_2$ and $a > b$.

The brake will be effective as long as T_1a is greater than T_2b.

Or $T_2b < T_1a$

Or $$\frac{T_2}{T_1} < \frac{a}{b}$$

i.e. as long as the ratio of T_2 to T_1 is less than the ratio a/b.

When $\frac{T_2}{T_1} \geq \frac{a}{b}$, F is zero or negative i.e. the brake becomes self-locking as no force is needed to apply the brake. Once the brake has been engaged, no further force is required to stop the rotation of the drum.

(iii) a = b, the band cannot be tightened and thus the brake cannot be applied.

(iv) The band brake just discussed is known as differential band brake. However, if either a or b is made zero, a simple band brake is obtained. If b = 0, and F downwards

$$Fl - T_1a = 0$$

$$\text{Or } F = \frac{T_1 a}{l}$$

Similarly, the force can be found for the other cases.

Note that such a brake can neither have self-energizing properties nor it can be self-locked.

(v) The brake is said to be more effective when maximum braking force is applied with the least effort F.

For case (i), when a > b and F is downwards, the force (effort) F required is less when the rotation is clockwise (assuming that $[(T_2/T_1) < (a/b)]$.

For case (ii), when a < b and F is upwards, F required is less when the rotation is counter clockwise (assuming that $[(T_2/T_1) < (a/b)]$.

Thus, for the given arrangement of the differential brake, it is more effective when,

(a) a > b, F downwards, rotation clockwise.

(b) a< b, F upwards, rotation counter-clockwise.

(vi) The advantage of self-locking is taken in hoists and conveyers where motion is permissible in only one direction. If motion gets reversed somehow, the self-locking is engaged which can be released only by reversing the applied force.

(vii) It is seen in (v) that a differential band brake is more effective only in one direction of rotation of the drum. However, a two-way band brake can also be designed which is equally effective for both the directions of rotation of the drum. In such a brake, the two lever arms are made equal.

For both directions of rotation of the drum,

$$Fl - T_1a - T_2a = 0$$

$$F = \frac{(T_1 + T_2)\,a}{l}$$

BAND AND BLOCK BRAKE:

A band and block brake consists of a number of wooden blocks secured inside a flexible steel band. When the brake is applied, the blocks are pressed against the drum. Two sides of the band become tight and slack as usual. Wooden blocks have a higher coefficient of friction, thus increasing the effectiveness of the brake. Also, such blocks can be easily replaced on being worn out.

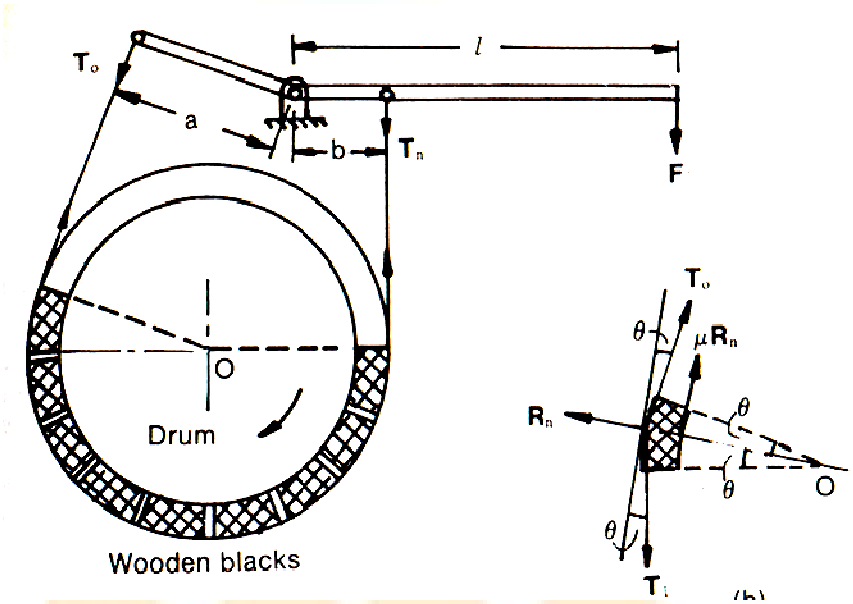


Figure: Band and block brake

Each block subtends a small angle 2θ at the center of the drum. The frictional force on the blocks acts in the direction of rotation of drum. For n blocks on the brake,

Let

T_0 = Tension on the slack side.

T_1 = Tension on the tight side after one block.

T_2 = Tension on the tight side after two block

T_n = Tension on the tight side after n blocks

μ = Coefficient of friction

R_n = Normal reaction on the block

The forces on one block of the brake are shown in fig.

For equilibrium,

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

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

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

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

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

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

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

Similarly,

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

And

$$\frac{T_n}{T_{n-1}} = \frac{1 + \mu\tan\theta}{1 - \mu\tan\theta}$$

$$\frac{T_n}{T_0} = \frac{T_n}{T_{n-1}}\,\frac{T_{n-1}}{T_{n-2}} \cdots \frac{T_2\,T_1}{T_1\,T_0}$$

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

INTERNAL EXPANDING SHOES BRAKE:

Earlier, automobiles used band brakes which were exposed to dirt and water. Their heat dissipation capacity was also poor. These days, band brakes have been replaced by internally expanding shoe brakes having at least one self-energizing shoe per wheel. This results in tremendous friction, giving great braking power without excessive use of pedal pressure.

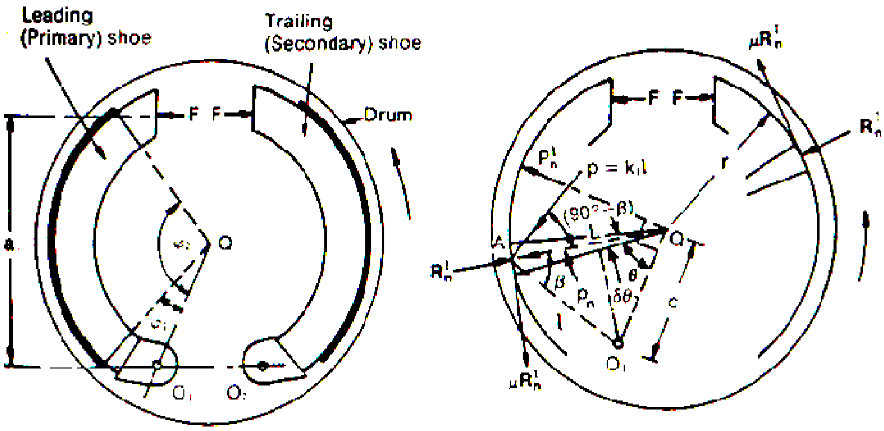


Figure: Internal expending shoe brake

Figure shows an internal shoe automobile brake. It consists of two semi-circular shoes which are lined with a friction material such as ferodo. The shoes press against the inner flange of the drum when the brakes are applied. Under normal running of the vehicle, the drum rotates freely as the outer diameter of the shoes is a little less than the internal diameter of the drum.

The actuating force F is usually applied by two equal-diameter pistons in a common hydraulic cylinder and is applied equally in magnitude to each shoe. For the shown direction of the drum rotation, the left shoe is known as the leading or forward shoe and the right as the trailing or rear shoe.

Assuming that each shoe is rigid as compared to the friction surface, the pressure p at any point A on the contact surface will be proportional to its distance l form the pivots.

Considering the leading shoe,

$\rho \;\alpha\; l \quad = \quad k_1 l$, where k_1 is a constant.

The direction of p is perpendicular to OA.

The normal pressure, $p_n \quad = \quad k_1 l \cos(90^o - \beta) \quad = \quad k_1 l \sin\beta$

$= \quad k_1 c \sin\theta$

$= \quad (OL \quad = \quad l \sin\beta \quad = \quad c \sin\theta)$

$= \quad k_2 \sin\theta$, where k_2 is constant

p_n is maximum when $\theta \quad = \quad 90^o$

Let $p_n \quad = \quad$ Maximum intensity of normal pressure on the leading shoe.

P^l_n max $= \quad P^l_n \quad = \quad k_2 \sin 90^o \quad = \quad k_2$

Or $p_n \quad = \quad P^l_n \sin\theta$

Let ω = width of brake lining

μ = Coefficient of friction

Consider a small element of brake lining on the leading shoe that makes an angle $\delta\theta$ at the center.

Normal reaction on the differential surface,

$$R^l_n = \text{Area} \times \text{Pressure}$$

$$= (r\,\delta\theta\,\omega)p_n$$

$$= r\,\delta\theta\,\omega\,P^l_n \sin\theta$$

Taking moments about the fulcrum O_1,

$$Fa - \sum R^l_n c \sin\theta + \sum \mu R^l_n (r - c\cos\theta) = 0$$

Where, $\sum R^l_n c \sin\theta$ $= \dfrac{\int rc\,\omega\,P^l_n\, 1\,(1-\cos 2\theta)\,d\theta}{2}$

$$= \int rc\,\omega\,P^l_n \frac{1}{2}\left[\theta - \frac{\sin 2\theta}{2}\right]$$

$$= rc\,\omega\,P^l_n(2\,Ø_2 - 2\,Ø_1 - \sin 2\,Ø_2 + \sin 2\,Ø_1)$$

and

$\sum \mu R^l_n (r - c\cos\theta)$ $= \int \mu r^2\,\omega\,P^l_n \sin\theta\, d\theta - \int \mu rc\omega\,P^l_n \sin\theta\cos\theta\, d\theta$

$$= \mu r^2\,\omega\,P^l_n(-\cos\theta) - \frac{\int \mu rc\omega\,P^l_n \sin\theta\cos\theta\, d\theta}{2}$$

$$= \mu r^2\,\omega\,P^l_n(\cos Ø_1 - \cos Ø_2) - \mu rc\omega\,P^l_n \frac{1}{2}\left(\frac{-\cos 2\theta}{2}\right)$$

$$= \frac{\mu rc\omega\,P^l_n}{4}\,[4r(\cos Ø_1 - \cos Ø_2) - c(\cos 2Ø_1 - \cos 2Ø_2)]$$

Taking moments about the fulcrum O_2 for the trailing shoe,

$$Fa - \sum \mu R^t_n (r - c\sin\theta) - \sum \mu R^t_n (r - c\cos\theta) = 0$$

Where, $\sum \mu R^t_n c\sin\theta = rc\,\omega P^t_n\,[(2\,Ø_2 - 2\,Ø_1 - \sin 2\,Ø_2 + \sin 2\,Ø_1)]$

Thus P^t_n and P^l_n, the maximum intensities on the leading and the trailing shoes, can be determined.

Braking torque,

$T_B \quad = \quad \sum \mu R^l_n r + \sum \mu R^t_n r$

$= \quad \int \mu r^2 \omega P^l_n \sin\theta \, d\theta + \int \mu r^2 \omega C \sin\theta \, d\theta$

$= \quad \mu r^2 \omega (P^l_n + P^t_n) \quad (-\cos Ø)$

$= \quad \mu r^2 \omega (P^l_n + P^t_n) \quad (\cos Ø_1 - \cos Ø_2)$

Note that for the same applied force F on each shoe, P^l_n is not equal to P^t_n and $P^l_n > P^t_n$. Usually, more than 50% of the total braking torque is supplied by the leading shoe.Also note that the leading shoe is self-energizing whereas the trailing shoe is not. This is because the friction forces acting on the leading shoe help the direction of drum rotation, the right shoe will become self-energizing, whereas the left will so any longer.

If the third term exceeds the second term on the LHS. F will be negative and the brake becomes self-locking. A brake should be self-energizing but not self-locking. The amount of self-energizing is measured by the ratio of the friction moment and the normal reaction moment, i.e. the ratio of the third term to the second term. When this ratio is equal to or more than unity, the brake is self-locking. When the ratio is less than unity (more than zero), the brake is self-energizing.

RESULT: Studied the following types of mechanical brakes:

1. Block or shoe brake
2. Band brake
3. Band and block brake
4. Internal expanding shoe brake

Viva-Voce:

Experiment No.-3

Aim: To find experimentally the Gyroscopic couple on Motorized Gyroscope and compare with applied couple.

REQUIREMENT:

Motorised Gyroscope Apparatus, weights, tachometer.

THOERY

AXIS OF SPIN:

If a body is revolving about an axis, the latter is known as axis of spin (refer fig. where OX is the axis of spin).

PRECESSION**:**

Precession means the rotation about the third axis OZ (refer fig. 1) that is perpendicular to both the axis of spin OX and that of couple OY.

AXIS OF PRECESSION:

The third axis OZ is perpendicular to both the axis of spin OX and that of couple OY is known as axis of precession.

GYROSCOPIC EFFECT:

To a body revolving (or spinning) about an axis say OX, (refer fig. 1) if a couple represented by a vector OY perpendicular to OX is applied, then the body tries to process about an axis OZ which is perpendicular both to OX and OY. Thus, the couple is mutually perpendicular. The above combined effect is known as precessional or Gyroscopic effect.

GYROSCOPE:

It is a body while spinning about an axis is free to rotate in other direction under the action of external forces.

GYROSCOPIC COUPLE OF A PLANE DISC:

Let a disc of weight of 'W' having a moment of inertia I be spinning at an angular velocity ω about axis OX in anticlockwise direction viewing from front (refer fig. 2). Therefore, the angular momentum of disc is Iω. Applying right-hand screw rule the sense of vector representing the angular momentum of disc which is also a vector quantity will be in the direction OX.

A couple whose axis is OY perpendicular to OX and is in the plane Z, is now applied to prices the axis OX.

Let axis OX turn through a small angular displacement from OX to OX' in time δt. The couple applied produces a change in the direction of angular velocity, the magnitude & the magnitude remaining constant. This change is due to the velocity of precession.

Therefore, 'OX' represented the angular momentum after time δt.

Change of angular momentum = OX' – OX = XX'

Angular Displacement or rate of change of angular momentum $= \dfrac{XX'}{\delta t}$

$$= \frac{OX \times \delta\theta}{\delta t}$$

As, XX' $= OX \times \delta\theta$

In direction if XX'

Now as rate of change of angular momentum = Couple Applied

C = T

We get $T = \dfrac{OX \times \delta\theta}{\delta t}$

However, OX = I.ω

Where,

I = Moment of inertia of disc

ω = Angular Velocity of disc

$$T = \frac{I.\omega \times \delta\theta}{\delta t}$$

And in the limit δt when is very small

$$T = I.\omega \times \frac{\delta\theta}{dt}$$

We have dθ/dt = ωp
= Angular velocity of precession of yoke about vertical axis. Thus we get T = I x ω x ωp

The direction of couple applied on the body is clockwise when looking in the direction XX' and in the limit this is perpendicular to the axis of ω and of ωp.

The reaction couple exerted by the body on its frame is equal in magnitude to that of C, but opposite in direction.

DESCRIPTION

The set up consists of heavy disc mounted on a horizontal shaft, rotated by a variable speed motor. The rotor shaft is coupled to a motor mounted on a trunion frame having bearing in a yoke frame, which is free to rotate about vertical axis. A weight pan on other side of disc balances the weight of motor. Rotor disc can be move about three axis. Weight can be applied at a particular distance from the center of rotor to calculate the applied torque. The Gyroscopic couple can be determined with the help of moment of inertia, angular speed of disc and angular speed of precession.

RULE NO. 1

"The spinning body exerts a torque or couple in such a direction which tends to make the axis of spin coincides with that of precession".

To study the rule of gyroscopic behavior, following procedure may be adopted:

Balance the initial horizontal position of the rotor. Start the motor by increasing the voltage with the autotransformer, and wait until it attains constant speed.

Presses the yoke frame about vertical axis by an applying necessary force by hand to the same (in the clockwise sense seen from above).

It will be observed that the rotor frame swings about the horizontal AXIS Y Y. Motor side is seen coming upward and the weight pan side going downward.

Rotate the vertical yoke axis in the anticlockwise direction seen from above and observe that the rotor frame swing in opposite sense (as compared to that in previous case following the above rule).

RULE NO. 2

"The spinning body precesses in such a way as to make the axis of spin coincide with that of the couple applied, through 90° turn axis".

Balance the rotor position on the horizontal frame. Start the motor by increasing the voltage with the autotransformer and wait until the disc attains constant speed.
Put weights in the weight pan, and start the stopwatch to note the time in seconds required for precession, through 90° or 180° etc.
The vertical yoke precesses about OZ axis as per the rule No. 2.

PROCEDURE:

1. Set the rotor at zero position.
2. Start the motor with the help of rotary switch.
3. Increase the speed of rotor with dimmer state & stable it & measure the rpm with the help of tachometer (optional).
4. Put the weight on weight pan than yoke rotate at anticlockwise direction.
5. Measure the rotating angle (30°, 40°) with the help of stopwatch.
6. Repeat the experiment for the various speeds and loads.
7. After the test is over set dimmer to zero position and switch off main supply.

FORMULAE:

$$T_{theo} = I\omega\,\omega p$$

$$I = \frac{W}{G} \times \frac{r^2}{2}\ \text{Kgm sec}^2$$

$$\omega = \frac{2 \times \pi \times N}{60}\ \text{rad/sec}$$

$$\omega p = \frac{d\theta}{dt} \times \frac{\pi}{180}\ \text{rad/sec}$$

$$T_{act} = wL$$

OBSERVATION & CALCULATION:

DATA:

Density of Rotor	=	7817 Kg/m^3		
Rotor Diameter	=	300 mm	=	0.3m
Rotor Thickness	=	10 mm	=	0.001 m
Weight of disc	=	5.42 kg		
Weights	=	0.500 kg, 1.0 kg, 2.0 kg		
Distance of bolt of Weight pan from disc Center	=	225 mm	=	0.225 m

OBSERVATION TABLE:

S. No.	**Speed (RPM)**	**Weight (Kg)**	**dθ (degree)**	**Dt (Sec)**

CALCULATION:

S. No.	**I Kg.-m.-sec^2**	**ω rad/sec**	**ωp rad/sec**	**T_{act} (Kg.m)**	**T_{th} (Kg.m)**

NOMENCLATURE:

I = Mass Moment of inertia of disc, Kg.-m.-sec^2

ω = Angular velocity of disc

W = Weight of rotor disc, in kg

R = Radius of disc, in meter

G = Acceleration due to gravity, in m/sec^2

N = RPM of disc spin

ωp = Angular velocity of precession of yoke about vertical axis.

dθ = Angle of precession

dt = Time required for this precessions.

T = Gyroscopic couple, Kg. M

W = Weight of pan

L = Distance of weight

PRECAUTIONS

1. Before start the motor dimmer state at zero position
2. Increase the speed gradually.
3. Do not run the motor at low voltage i.e. less than 180 volts.

RESULT: Compared experimentally the gyroscopic couple on Motorized Gyroscope with applied couple.

Viva-Voce:

Experiment No.-4

Aim: To find out critical speed experimentally and to compare the Whirling Speed of a shaft with theoretical values.

Requirement:

Whirling of shaft Apparatus, tachometer, and shafts of different diameters.

THEORY:

In actual practice, a rotating shaft carries different mountings and accessories in the form of gears, pulleys etc. when the gears or pulleys are put on the shaft, the center of gravity of the pulley or gear does not coincide with the center line of the bearings or with the axis of the shaft, when the shaft is stationary. This means that the center of gravity of the pulley or gear is at a certain distance from the axis of rotation and due to this, the shaft is subjected to centrifugal force. This force will bend the shaft, which will further increase the distance of center of gravity of the pulley or gear from the axis of rotation. The bending of shaft not only depends upon the value of eccentricity (distance between center of gravity of the pulley and axis of rotation) but also depends upon the speed at which the shaft rotates.

The speed, at which the shaft runs so that the additional deflection of the shaft from the axis of rotation becomes infinite, is known as critical or whirling speed.

DESCRIPTION:

The apparatus consists of a DC motor as the driving unit, which drives the shaft supported in fixing ends. Fixing ends can slide and adjust according to the requirement on the guiding pipes. Motor is connected to the shaft through flexible coupling. The shafts of different diameters can be replaced easily with the help of fixing ends. A dimmer stat is provided to increase or decrease the rpm of the motor. The whole arrangement is fixed on M.S. frame. Guards are provided to protect the user from accident.

PROCEDURE:

1. Fix the shaft to tube tested in the fixed ends.
2. Supply the main power to the motor through dimmer stat.
3. Gradually increase the speed of motor until the first mode of vibration is not arrived.
4. Study the first mode of vibration and note down the corresponding speed of the shaft with the help of hand tachometer.
5. Gradually increase the speed of motor again, until the second mode of vibration is not arrived.
6. Study the second mode of vibration and note down the corresponding speed of the shaft with the help of hand tachometer.
7. Reduce the speed gradually and when shaft stop rotating, cut off the main power supply.
8. Repeat the experiment for the shafts of different diameters.

FORMULAE:

1. Moment of inertia of shaft, I = $(\pi/64) \times D^4\ m^4$
2. Mass of the shaft = Area x Length x density
3. Area of shaft: = $(\pi/4) \times D^2\ m^2$

When,

Static deflection of shaft due to mass of the shaft,

$$\delta_s = \frac{WL^4}{384\ EI} \text{ meter}$$

Frequency of transverse vibration,

$f_n = \dfrac{0.5616}{\sqrt{\{\delta_s/1.27\}}}$ Hz

Critical or Whirling speed of the shaft (in r.p.s.) is equal to the frequency of transverse vibration in Hz.

N_c (r.p.s.) $= f_n$ (Hz)

N_c (rpm) $= 60 \times f_n$

When the both ends are supported:

State deflection of shaft due to mass of the shaft,

$\delta_s = \dfrac{5WL^4}{384\ EI}$ meter

Frequency of transverse vibration,

$f_n = \dfrac{0.56196}{\sqrt{\{\delta_s/1.27\}}}$ Hz

Critical or Whirling speed of the shaft (in r.p.s) is equal to the frequency of transverse vibration in Hz.

N_c (r.p.s.) $= f_n$ (Hz)

N_c (rpm) $= 60 \times f_n$

When one end is fix and other is supported:

State deflection of shaft due to mass of the shaft,

$\delta_s = \dfrac{WL^4}{185\ EI}$ meter

Frequency of transverse vibration,

$f_n = \dfrac{0.56196}{\sqrt{\{\delta_s/1.27\}}}$ Hz

Critical or Whirling speed of the shaft (in r.p.s) is equal to the frequency of transverse vibration in Hz.

N_c (r.p.s.) $= f_n$ (Hz)

N_c (rpm) $= 60 \times f_n$

OBSERVATION & CALCULATION:

GIVEN DATA:

1. Diameter of shaft 1, d_1 = ___ mm = _______ m
2. Diameter of shaft 2, d_2 = ___ mm = _______ m
3. Diameter of shaft 3, d_3 = ___ mm = _______ m
4. Length of shaft 1, L_1 = ___ meter
5. Length of shaft 2, L_2 = ___ meter
6. Length of shaft 3, L_3 = 1 meter
7. Young's Modulus of Elasticity, E = 2×10^6 kg/cm^2
8. Density of shaft material = 0.0078 kg/cm^2
9. Acceleration due to gravity, g = 9.81 m/s^2

When both the ends are fixed:

S. No.	**Actual whirling speed, N_a**	**Theoretical Whirling speed, N_t**

When both the ends are supported

S. No.	**Actual whirling speed, N_a**	**Theoretical Whirling speed, N_t**

When one end is fix and other end is supported:

S. No.	Actual whirling speed, N_a	Theoretical Whirling speed, N_t

Possible Experiments with Elastic Rods:

Experiment No.	End Fixing	Mode of Whirl
1	One supported other Fixed	1st Mode
2	One supported other fixed	2nd Mode
3	Both ends supported	1st Mode
4	Both ends Supported	2nd Mode
5	Both end Fixed	1st Mode

PRECAUTIONS

1. If the revolutions of an unloaded shaft are gradually increased, it will be found that a certain speed will be reached at which violent instability will occur, the shaft.
2. Deflecting in a single bow and whirling round like a skipping rope. If this speed is maintained the deflection will become so large that the shaft will be fractured.
3. It is advisable to increase the speed of shaft rapidly and pass through the critical speeds first rather than observing the 1st critical speed which increases the speed of rotation slowly. In this process, there is a possibility that the amplitude of vibration will increase suddenly bringing the failure of the shaft.
4. It is destructive test of shaft and it is observed that the elastic behavior of the shaft material changes a little after testing it for a few times and it is advisable to use fresh shaft afterwards.
5. Fix the apparatus firmly on the suitable foundation.
6. Do not run the motor at low voltage i.e. less than 180 volts.
7. Always keep apparatus free from dust.

RESULT:

Actual and theoretical whirling speed of the shaft compared.

Viva-Voce:

Experiment No.-5

AIM:
To determine experimentally, the Moment of Inertia of a Flywheel and Axle compare with the theoretical values.

REQUIREMENT:
A flywheel mounted in the laboratory, a set of slotted weights and hanger, stop watch, vernier calipers, a strong but thin cord, meter rod etc.

THEORY:
Moment of Inertia:
The body tends to rotate at a uniform rate about the given axis and opposes the change in its angular velocity of the inherent tendency. Moment of inertia is also referred as rotational inertia.
Mathematically, moment of inertia of a given body about a given axis equal to the sum of the products of the mass and the square of the distances from axis of rotation of different particles of which the body is composed.
Moment of inertia of a scalar quality and depends upon mass of body as well as the distribution of its mass about the axis of rotation.
Unit of moment of inertia is kg x m^2 in S.I. system and gram x cm^2 in C.G.S. system of units.
Moment of inertia is the rotational analogue of mass.
The radius of gyration of a given rotating body about a given axis of rotation is defined as that imaginary distance from the axis rotation at which if whole mass of the body is supposed to be concentrated then its moment of inertia will be exactly the same as is with actual distribution of its mass.
Mathematically radius of gyration (denoted by k) is the root mean square distance of the particles of given body from the axis of rotation.
Flywheel and determination of its moment of inertia:

A flywheel is a heavy metallic wheel with a long axle supported in bearings. It can remain at rest in any position. In other words, its center of mass lies on the axis of rotation itself. The flywheel is so shaped that most of the material is concentrated at the rim. It increases the moment of inertia of flywheel about its axis. A flywheel is generally used in stationary engines to obtain a uniform motion even with a non-uniform drive.

To determine its moment of inertia we mount the given flywheel in a horizontal position. Its axle is supported on ball bearings to reduced friction. The ball bearings are in turn, fitted in fixed rigid supports. We take a fine but strong cord and one end of cord is tied loosely to a peg (or pin) P on the axle and the cord is wound several times, say n_1, round the axle, A known mass m is attached to other end of the cord.

When the mass is allowed to descend, the cord is unwound wheel rotates about the axle. Thus, mass loses potential energy in its descent. The lost energy is converted partly into the kinetic energy of translation of the falling mass itself and partly into the kinetic energy of rotation of flywheel. A part of it is also used up in doing work against the friction of the supporting bearings.

Let h be the vertical distance through which the mass m falls. Thus, the potential energy lost by the mass = mgh.

Let v be the velocity acquired by the mass and ω the angular velocity acquired by the wheel at the instant when the cord has unwound from the axle and just leaves the axle. Then the kinetic energy gained by the given mass $1/2\ mv^2 = 1/2\ mr^2\ \omega^2$

$[\because\ v = r\omega]$, where r is the radius of the axle.

Further more, the kinetic energy gained by the flywheel $= 1\ I\ \omega^2$

Where I is the moment of inertia of flywheel about its axis of rotation.

Again if f be the constant work done against the frictional force present at the bearings during each revolution of wheel and n_1 be the number of revolutions made of windings of cord on the axle, then total energy spent in overcoming the friction is $n_1 f$.

Applying the law of conservation of energy, we can say that loss of potential energy of mass is equal to the sum of

(i) Kinetic energy of falling mass
(ii) Rotational kinetic energy of wheel and
(iii) Work done against friction. Mathematically,

$$mgh = 1/2\, mr^2 \omega^2 + 1/2\, I \omega^2 + n_1 f \text{ -----------------------------(i)}$$

After the cord leaves the axle, the wheel makes a definite number of revolution (say n_2) before finally coming to rest due to the presence of frictional forces at the bearings. Thus, the kinetic energy (= $I\omega^2/2$) of the wheel is spent in overcoming the

Friction in n_2 revolutions i.e.,

$$\frac{1}{2} I \omega^2 = n_2 f$$

$$\text{or } f = \frac{I \omega^2}{2\, n_2}$$

or substituting this value of f in equation (i), we get,

$$mgh = \frac{1}{2} mr^2 \omega^2 + \frac{1}{2} I \omega^2 + n_1 \frac{I \omega^2}{2\, n_2}$$

or $mgh = 1/2\, mr^2 \omega^2 + 1/2\, I \omega^2 \; (1 + n_1/n_2)$

or rearranging the terms and on simplifying, we get,

$$I = \frac{2mgh - mr^2 \omega^2}{\omega^2 \; (1 + n_1/n_2)}$$

We already know that the wheel is finally brought to rest by the frictional forces. If the frictional forces are throughout constant, the wheel is uniformly retarded. At the instant, when the cord leaves the axle it has an angular velocity ω which finally becomes zero. Therefore,

The average angular velocity = $(\omega + 0)/2 = \omega/2$

If t be the time taken by the wheel before coming to rest then the average angular velocity

$$\omega/2 = \text{Total angle described in } n_2 \text{ revolutions}/2$$

$$= 2\pi n_2/t$$

$$\omega = 4\pi n_2/t \quad \text{---------(iii)}$$

Again, h is the height through which the mass m falls and, in turn, it is equal to the length of the cord wound up on the axle in n_1 windings. Thus,

$$h = 2\pi r n_1 \quad \text{-------------(iv)}$$

Substituting values of ω from (iii) and h from (iv) in equation (ii) we get,

$$I = 2\,mg.\,2\pi r n_1 - mr^2 (4\pi n_2/t)^2$$

$$(4\pi n_2/t)^2 \quad (1 + n_1/n_2)$$

$$= \frac{mgr\,n_1 t^2}{4\pi n_2 (n_1 + n_2)} \qquad \frac{mr^2 n_2}{(n_1 + n_2)} \qquad \text{--(v)}$$

In practice, it is found that in equation (v) numerical value of first factor

$\left\{ \dfrac{mgrn_1 t^2}{4\pi n_2 (n_1 + n_2)} \right\}$ is very large as compared to the second factor $\left\{ \dfrac{mr^2 n_2}{(n_1 + n_2)} \right\}$

coming on the right hand side. Thus, to the first approximation, we may leave the second factor and in that case we may write that

$$I = \frac{mgr\,n_1 t^2}{4\pi n_2 (n_1 + n_2)} \quad \text{(approximately)} \quad \text{--------(vi)}$$

Use of flywheel:

A flywheel forms an important part of stationary engines. A flywheel of large moment of inertia is joined to the shaft of the engine. The force driving the engine changes between maximum and minimum values and hence the motion of the machine coupled to the engine is not uniform. The flywheel stores the power in the form of fits its kinetic energy of rotation when the driving force is minimum. Thus, it makes the motion of the machine almost uniform.

Formula used:

$$I = \frac{mgr\,n_1 t^2}{4\pi n_2 (n_1 + n_2)} - \frac{mr^2 n_2}{(n_1 + n_2)}$$

or

$$I = \frac{mgr\,n_1 t^2}{4\pi n_2 (n_1 + n_2)} \qquad \text{(to the first approximation)}$$

where,

I	=	moment of inertia of given flywheel,
r	=	radius of axle,
m	=	mass attached to the cord

n_1 = number of revolutions of the cord wound on the axle,

n_2 = number of revolutions made by the wheel after detaching of mass a

t = time for completing n_2 vibrations.

PROCEDURE:

1. Examine the flywheel and see carefully that there is least possible friction in the bearings. If necessary, oil the bearings. In that case, rotate the wheel for some time by hand so that the oil spreads in whole bearings.
2. Take a strong but thin cord having length slightly less than the height of the axle of flywheel from the floor. Make a simple loose loop at one end of cord and attach a hanger of known mass at its other end. Put some slotted weights in the hanger so that total mass of hanger and weights is about 250 grams.
3. Now slip the loop formed over the peg P, projecting on the axle of flywheel, and gently rotate the flywheel with the hand so that the cord is wrapped uniformly round the axle. When almost the whole core has been wound, stop at a stage where the projecting peg is just horizontal.
   In this position, mark a reference point, using a chalk, on the rim of flywheel opposite to the horizontal pointer N fixed to the flywheel structure. Count the number of turns wound round the axle. Let it be n_1.
4. Hold a stopwatch in your hand and allow the mass to descend. The cord is unwrapped from the axle and after completing n_1 revolutions; the cord is detached from the peg. Consequently, the mass falls on the floor is heard, start the stopwatch. Count the number of revolutions n_2 made by the flywheel before coming to rest and the time taken t for it. If n_2 and t are found to be too small then increase the mass m suspended from the cord. However, if n_2 and t are too large and after detachment of mass, the flywheel rotates so rapidly that you cannot count the number of revolutions made by wheel then reduces the mass. In this manner, adjust the suitable mass to be suspended from the cord.
5. Repeat the step no. 4 and accurately note the value of n_2 and t. Sometime, it so happens that in addition to the complete rotations made by the wheel there may be a fraction of the rotation too. To find out the value of fractional rotation, measure the distance along the circumference of the wheel find the fraction of rotation. Add this fraction to the number of complete rotations to find the correct value of n_2. For same mass m and same number of windings, n_1 on the axle repeat this step 2 – 3 times and find mean value of n_2 and t.
6. Repeat the step no. 3 and 5 at least two times by changing mass m or changing n_1, the number of revolutions of the cord on the axle.
7. Using vernier calipers, measure the diameter of the axle at 3 – 4 places in two mutually perpendicular directions. Also, measure, using a thread and metre scale, the circumference of wheel.

OBSERVATIONS:

Circumference of the wheel x = ------cm

Table for $n_{1,}$ n_2 and t.

S. No	Total mass applied m (g)	No.of Rev. Made by Cord on axle n_1	No. Of revolutions n_2 made by the Wheel after detachment of mass				Mean n_2	Time t(s)	Mean t(s)
			Complete Rev. a	Distance of chalk mark pointer y (cm)	Fractional revolution b = y/x	Total number n_2=a+b			
1		(a) (b) (c)							
2		(a) (b) (c)							
3		(a) (b) (c)							

Least count of the vernier calipers = -------cm

Zero error of the vernier calipers = -------cm

(with proper sign)

Table for radius of axle:

S. No.	Observed diameter (cm)			Corrected diameter (cm)
	In one direction	In perpendicular direction	Mean	

Mean diameter of axle, D = --------cm

Mean radius of axle, r =D = -----cm

2

CALCULATIONS:

(i) From 1^{st} set I = -------- = ------------ g-cm^2

(ii) From 2^{nd} set I = -------- = ------------ g-cm^2

(iii)From 3^{rd} set I = -------- = ------------ g-cm^2

Determination of maximum permissible error:

If radius r of the axle is small then according to the approximate formula for moment of inertia, we have,

$$I = \frac{mgr\, n_1 t^2}{4\, \pi n_2 (n_1 + n_2)}$$

Taking log and differentiating, we get

$$\frac{\Delta I}{I} = \frac{\Delta m}{m} + \frac{\Delta r}{r} + \frac{\Delta n_1}{n_1} + 2 . \frac{\Delta t}{t} + \frac{\Delta n_2}{n_2} + \frac{\Delta(n_1 + n_2)}{n_1 + n_2}$$

Here all the terms have been assigned +ve sign so as to calculate the maximum permissible (log) error. Substitute various values and find value of Δ I/I

$\therefore$ Maximum permissible error = ΔI . 100% = ---%

Precautions:

1. The friction at the bearings should be least possible and the mass fixed to the end of the cord should be sufficient to be capable of overcoming friction at the bearings. In other words, the mass should be start falling on its own accord.
2. The length of the cord should be less than the height of the axle of the flywheel from the floor.
3. The loop of cord slipped over the peg should be quite loose so that the cord leaves the axle immediately on unwinding itself. If the loop is not loose, the cord will have a tendency to rewind on the axle in the opposite direction.
4. The cord should be uniformly wound on the axle. It means that neither there should be neither overlapping nor a gap between successive turns.
5. The cord used should be thin enough as compared to the diameter of axle. However, if the cord used is of appreciable thickness then add half of its thickness to the radius of axle to get the correct value of r.
6. There should be whole number of turns of cord wound on the axle. For this purpose, the winding of cord should be stopped at a point where the projecting peg is horizontal.
7. Be careful to start the stopwatch at the instant when the cord is just detached from the peg.
8. The diameter of the axle should be measured at different points of the axle in two mutually perpendicular directions.

SOURCES OF ERROR:

1. The exact instant at which the mass drops off (i.e. the instant at which the cord detaches itself from the peg) cannot be correctly found out.
2. Angular velocity of the wheel has been calculated on the assumption that frictional force throughout remains constant. However, this assumption is not true and in practice, friction increases as the speed of rotation of wheel decreases.

RESULT:

Moment of inertia of given flywheel about its axis of rotation = ------- g-cm^2.

VIVA-VOCE

1. What is flywheel?
2. Where flywheel is used
3. Define Moment of inertia
4. Define Radius of gyration
5. Define co-efficient of fluctuation of speed & co-efficient of fluctuation of energy

Experiment No.-6

Aim: To calculate the torque on a Planet Carrier and torque on internal gear using epicyclic gear train and holding torque apparatus.

REQUIREMENT: Epicyclic Gear Train Apparatus.

THOERY:

INTRODUCTION:

Any combination of gear wheels by means of which motion is transmitted from one shaft to another shaft is called a gear train. In case of epicyclic gear trains the axes of the shafts on which the gears are mounted may move relatively to a fixed axis.

The gear trains are useful for transmitting high velocity ratio 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 automobiles, wristwatches.

EPICYCLIC GEAR TRAIN:

A simple gear train (shown in fig. 1) is a train in which a gear A and the arm C have a common axis at O_1 about which they can rotate. The gear B meshes with gear A and has its axis on the arm at O_2, about which the gear B can rotate. If the arm is fixed, the gear train is simple and gear A can drive gear B or vice versa, but if gear A is fixed and arm is rotated about the axis of gear A (i.e. O_1), then the gear B is forced to rotate upon and around gear A. Such a motion is called epicyclic and the gear trains arrangement 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.

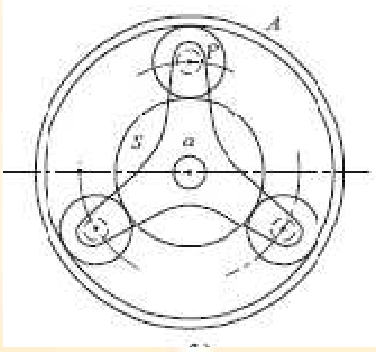


Figure: Epicyclic gear train

VELOCITY RATIO OF EPICYCLIC GEAR-TRAIN

The following two methods may be used for finding out the velocity ratio of an epicyclic gear train.

1. Tabular Method
2. Algebraic Method

A compound epicyclic gear train (internal type) consists of two co-axial S_1 and S_2 sun gear (A), an arm (H), three planetary gears (B,C,and E) and an annular gear (D) as shown in Fig.. Wheel A has 10 external teeth, B,C and E have 12 external teeth. The annular gear has 35 internal teeth. The sun gear A is fixed on the input shaft S_1. Three planetary compound gear (B,C,E) are mesh with sun gear A and annular gear D. The planetary gears are free to revolve on the pins of arm H.

Step No.	Condition of Elements	Revolution of Elements			
		Arm H	Gear A	Compound gear B, C, E	Gear D
1	Arm fixed gear A rotates through + 1 revolution (i.e. 1 rev. in anticlockwise)	0	+1	$-\frac{T_A}{T_B}$	$-\frac{T_A}{T_B} \times -\frac{T_B}{T_D} = \frac{T_A}{T_D}$
2	Arm fixed –gear A rotates through + X revolution	0	+X	$-X\frac{T_A}{T_B}$	$-X\frac{T_A}{T_D}$
3	Add + Y revolution to all elements	+Y	+Y	+Y	+Y
4	Total Motion	+Y	X + Y	$Y - X\frac{T_A}{T_B}$	$Y - X\frac{T_A}{T_D}$

Speed of Gear :-

If we know that the speed of arm is 271 R.P.M.

Therefore, y = 271 R.P.M.

And the gear D is fixed due to holding

Therefore, $Y - X\frac{T_A}{T_D} = 0$

$$271 - X\frac{10}{35} = 0$$

$$X = \frac{271 \times 35}{10}$$

$$= 948.5 \text{ R.P.M.}$$

Speed of A = X + Y

= 948.5 + 271

= 1219.5 R.P.M.

1219.5 R.P.M. in the direction of arm

$$\text{Speed Ratio} = \frac{\text{Speed of Driver}}{\text{Speed of Driven}}$$

$$= \frac{\text{Speed of Sun gear A}}{\text{Speed of Arm H}}$$

$$= \frac{1219.5}{271}$$

$$= 4.5$$

Let d_A, and d_B and d be the pitch circle diameter of sun gear A, planet gear B and internally toothed gear D respectively. Assuming the pitch of all the gears to be same therefore from the fig. Geometry

$d_A + 2\,d_B$ = d

The numbers of teeth are proportional to their pitch circle diameters, therefore

$T_A + 2\,T_{\mathbf{B}}$ = T_D

$2T_{\mathbf{B}} + 10$ = 35

$2\,T_{\mathbf{B}}$ = 35 – 10 = 25

$T_{\mathbf{B}}$ = 25/2 = 12.5 = 12

The numbers of teeth of planetary gears B, C, E are

T_B = T_C = T_E = 12

TORQUE IN EPICYCLIC GEAR TRAIN (INTERNAL TYPE):-

If the parts of an epicyclic gear train are all moving at uniform speeds, so that no angular acceleration are involved, the algebraic sum of all external torque applied to the train must be zero or there are at least external torques for the train, and in many cases there are only three. There are:-

T_i The input torque on the driving member, Arm.

T_o The resisting, or load, torque on the driven member.

T_{he} The holding, or braking torque on the fixed member.

If there is no acceleration,

$T_i + T_o + T_h$ = 0 ---------------(1)

Further, if there are no internal friction losses at the bearing and at the contact surfaces of the wheel teeth, the net energy dissipated by the train must be zero, or

$T_i\,\omega_i + T_o\,\omega_o + T_h\,\omega_h$ = 0 ---------------(2)

Where, ω_I, ωo and ω_h are the angular velocities of the three members to which the external torques are applied. But for the

Fixed member, ω_h = 0, so that

$T_i\,\omega_i + T_o\,\omega_o$ = 0 --------------(3)

From the resisting or load, torque

T_o = $\dfrac{-T_i\,\omega_I}{\omega_o}$ -------------(4)

and from equation (1)

Th = $-(T_i + T_o)$

= $T_i\,(\omega_i/\omega_o - 1)$ --------(5)

These equations may be used to find the values of T_o and Th when the input torque T_i applied to the driving members is known. In addition, for complex trains they may be used to find the tooth loads or torques on all the intermediate members through which power is transmitted.

PROCEDURE:

1. Check the experiment set-up.
2. Give supply to Motor from control panel.
3. Adjust the RPM of Input shaft to some fix value.
4. Apply holding torque just to hold the drum. This must be done carefully.
5. Take the readings of loads of the holding drum & output drum as well as take readings input & output RPM.
6. Take the next reading to apply load on output drum. By applying load on output drum, holding start to rotate.
7. Repeat the same procedure for next reading.

FORMULAE:

1. Gear ratio = $\frac{\text{Speed of driver shaft}}{\text{Speed of driven shaft}}$

2. Input Torque, T_i = $\frac{V \times I \times \eta \times 4500}{746 \quad 2\pi N}$ kg m

3. Holding Torque, T_h = $(T_1 - T_2) \times R_1$ kgm

4. Output Torque, T_o = $(T_3 - T_4) \times R_2$ kg m

*** Take Efficiency (η) of motor = 80% = 0.80**

OBSERVATION & CALCULATION DATA:

1. Number of teeth of SUN gear = 10 Teeth
2. Number of teeth of PLANET gear = 12 Teeth
3. Number of teeth of ANNULAR gear (Internal gear) = 35 Teeth
4. Diameter of holding drum = 186 mm = 0.186 m
5. Radius of holding drum, R_1 = 93 mm = 0.093 m
6. Diameter of output brake drum = 186 mm = 0.186 m
7. Radius of output brake drum, R_2 = 93 mm = 0.093 m

OBSERVATION TABLE:

S. No.	Voltage (DC) V	Current (DC) A	Speed I/P shaft N1	Speed O/P shaft N2	O/P Spring Balance Reading, $(T_1 - T_2)$ kg	Holding spring balance reading $(T_1 - T_2)$ kg

CALCULATION:

S. No.	Actual gear ratio	Theo gear ratio	Input Torque (kg m)	Holding Torque (kg m)	Output Torque (kg m)

PRECAUTIONS & MAINTENANCE INSTRUCTIONS:

1 Do not run the motor at low voltage i.e. less than 180 volts.

2 Before starting the motor both the ropes in loose position.
3 Before starting the motor with rotary switch, the dimmer state at zero position.
4 Increase speed gradually.
5 Always keep apparatus free from dust.

RESULT:

Torque on planet carrier = N-m
Torque on internal gear = N-m

VIVA-VOCE

1. Define Gear drive
2. What is the difference between spur & helical gear.
3. Double helical gear is also called………………………… gear.
4. What is difference between simple gear train & epicyclic gear train?
5. What is sun & planet gears
6. What are the applications of Epicyclic Gear Train

Experiment No.-7

Aim: To perform the experiment of Balancing of rotating parts and find the unbalanced couple and forces.

REQUIREMENTS:

Static & Dynamic Balancing Apparatus.

THEORY:

Conditions for Static and Dynamic Balancing:

If a shaft carries a number of unbalanced masses such that center of mass of the system lies on the axis of rotation, the system is said to statically balance. The resultant couple due to all the inertia forces during rotation must be zero. These two conditions together will give complete dynamic balancing. It is obvious that a dynamically -balanced system is also statically balanced, but the statically balanced system is not dynamically balanced.

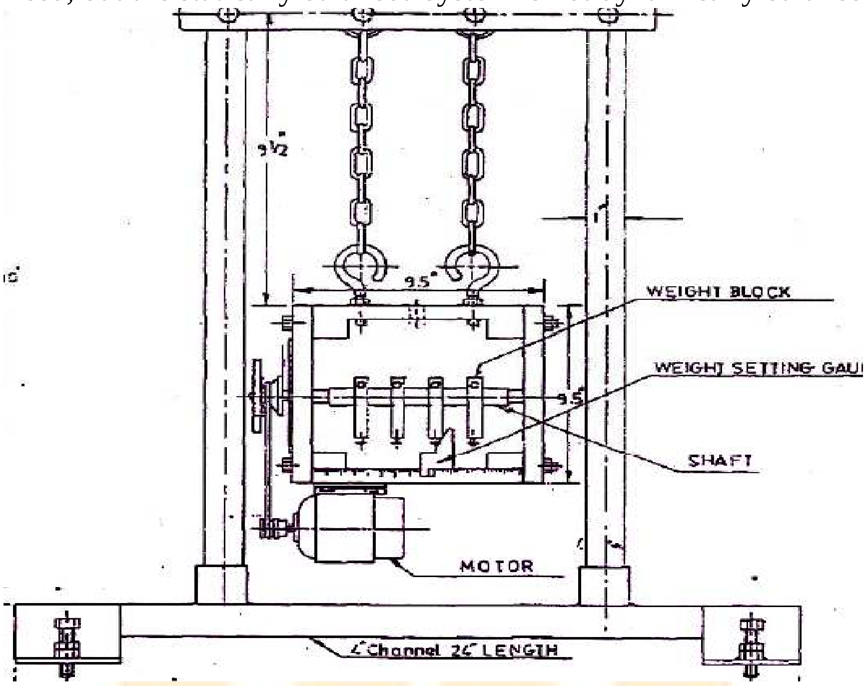


Figure: Dynamic balancing apparatus

Balancing of Several Masses Rotating in Different Planes:

When several masses revolve in different planes, they may be transferred to a reference plane (written as RP), which may be defined as the plane passing through a point on the axis of rotation and perpendicular to it. The effect of transferring a revolving mass (in one plane) to a reference plane is to cause a force of magnitude equal to centrifugal force of the revolving mass to act in the reference plane, together with a couple of magnitude equal to the product of the force and the distance between the plane of rotation and the reference plane. In order to have a complete balance of the several revolving masses in different planes, the following conditions must be satisfied:

1. The forces in the reference plane must balance i.e. the resultant force must be zero.
2. The couple about the reference plane must balance, i.e. the resultant couple must be zero.

Let us now consider four masses m_1, m_2, m_3 and m_4 revolving in plane 1, 2, 3 and 4 shown in fig. The relative angular position and position of the balancing mass m1 in plane may be obtained as discussed below:

1. Take one of the plane, say 1 as the reference plane (R.P). The distance of all the other planes to the left of the reference plane may be regarded as negative, and those to the right as positive.
2. Tabulate the data as in table. The planes are tabulated in the same order i.e. 1, 2,

Plane	Weight No.	Mass (m)	Radius r	Angle (θ)	Mass moment Mr	Distance from plane 1 (L)	Couple mrL

1. The position of plane 4 from plane 2 may be obtained by drawing the couple polygon with the help of data given in column no. 8.
2. The magnitude and angular position of mass m1 may be determined by drawing the force polygon from the given data of column no.5 & column no.6 to some suitable scale. Since the masses are to be completely balanced, therefore the force polygon must be closed figure. The closing side of force polygon is proportional to the m1r1.

The angular position of mass m1 must be equal to the angle in anticlockwise measured from the R.P. to the line drawn in the fig. Parallel to the closing side of the polygon.

Description

The apparatus consists of a steel shaft mounted in ball bearings in a stiff rectangular main frame. A set of four blocks of different weights is provided and may be detached from the shaft.

A disc carrying a circular protractor scale is fitted to one side of the rectangular frame. A scale is provided with the apparatus to adjust the longitudinal distance of the blocks on the shaft. The circular protractor scale is provided to determine the exact angular position of each adjustable block.

The shaft is driven by 230 volts, single phase, 50 cycles electric motor mounted under the main frame, through a belt.

For static balancing of weights the main frame is suspended to support frame by chains then rotate the shaft manually after fixing the blocks at their proper angles. It should be completely balanced. In this position, the motor driving belt is removed.

For dynamic balancing of the rotating mass system the main frame is suspended from the support frame by two short links such that the main frame and supporting frame are in the same plane. Rotate the statically balanced weights with the help of motor. If they rotate smoothly and without vibrations, they are dynamically balanced.

PROCEDURE:

1. Insert all the weights in sequence 1-2-3-4 from pulley side.
2. Fix the pointer and pulley on shaft.
3. Fix the pointer on 0° (θ2) on the circular protractor scale.
4. Fix the weight no.1 in horizontal position.
5. Rotate the shaft after loosening previous position of pointer and fix it on θ3.
6. Fix the weight no. 2 in horizontal position.
7. Loose the pointer and rotate the shaft to fix pointer on θ4.
8. Fix the weight no.3 in horizontal position.
9. Loose the pointer and rotate the shaft to fix pointer on θ1.

10. Fix the weight no. 4 in horizontal position.
11. Now the weights are mounted in correct position.
12. For static balancing, the system will remain steady in any angular position.
13. Now put the belt on the pulleys of shaft and motor.
14. Supply the main power to the motor through dimmer stat.
15. Gradually increase the speed of the motor. If the system runs smoothly and without vibrations, it shows that the system is dynamically balanced.
16. Gradually reduced the speed to minimum and then switch off the main supply to stop the system.

Data:

Mass of 1	=	m_1 gms =	Plane	1	=	Weight No.	= 4
Mass of 2	=	m_2 gms =	Plane	2	=	Weight No.	= 1
Mass of 3	=	m_3 gms=	Plane	3	=	Weight No.	= 2
Mass of 4	=	m_4 gms =	Plane	4	=	Weight No.	= 3

Radius 1, 2, 3, 4= r cm. (Same radius)

Angle between 2 & 3 = θ3
Angle between 2 & 4 = θ4
Angle between 2 & 1 = θ1

Observation & calculations:

Plane	Weight No.	Mass (m)	Radius r	Angle (θ)	Mass moment mr	Distance from plane 1 (L)	Couple mrL

***PRECAUTIONS & MAINTENANCE INSTRUCTIONS*:**
1. Do not run the motor at low voltage i.e. less than 180 volts.
2. Increase the motor speed gradually.
3. Experimental set up is proper tightly before starting experiment.
4. Always keep apparatus free from dust.
5. Before starting the rotary switch, check the needle of dimmer stat at zero position.

RESULT:
Statically and dynamically balanced the rotating parts.

VIVA-VOCE
1. Why balancing is necessary for high speed engine
2. What is difference between Static & Dynamic Balancing?
3. What are effects of partial balancing in locomotives?
4. What are the practical applications of balancing?
5. Secondary balancing force is given by relation …………………………..

Experiment No.-8

AIM: To find out experimentally the coriolli's and component of acceleration and compare with theoretical values.

REQUIREMENT: Coriolli's Component of Acceleration apparatus.

THEORY:

INTRODUCTION:

The apparatus has been designed to enable students to measure the various parameters comprising the Coriolli's of Acceleration.

To maintain this acceleration long enough for measurements to be taken the conventional slider mechanism is replaced by two streams of water flowing radially outwards from an inverted 'T' shaped tube, which is rotated about its vertical axis so that the water in passing along the tube is subjected to a Coriolli's Components of Acceleration.

The total acceleration of a point with respect to another point in a rigid link is the vector sum of its centripetal and tangential components. This holds true when the distance between two points is fixed and the relative acceleration of the two points on a moving rigid link has been considered. If the distances between two points vary, that is the second point, which was stationary, now slides, the total acceleration will contain one additional component, known as Coriolli's component of acceleration.

A mechanism (shown in fig.) consisting translating pair i.e. block B, which is free to slide in straight path fixed in direction. If the translating pair itself revolves, its acceleration will include the Coriolli's component of acceleration due to change in relative distance between two points.

Let link OA oscillate about the fixed center O with constant angular velocity ω, from OA to OA' in time dt, angle between OA and OA' being dθ. The link consists of a slider B that interval of time. Now the slider can be considered to have moved from B to E follows:

From B to C due to outward velocity v of the slider.

D to E due to acceleration perpendicular to the rod.

The third movement of the slider is due to Coriolli's acceleration, which can be analyzed as under: -

Arc DE = arc EF – arc FD

= Arc EF – arc BC

= FO x dθ - BO x dθ

= dθ(FO – BO)

= BF x dθ

= CD x dθ

Now linear displacement

CD = v. dt

And angular displacement

dθ = ω. Dt

Arc DE = (v. dt) (ω. dt

= $v \cdot \omega \cdot (dt)^2$

However, DE= $\frac{1}{2} f^{cc} \cdot (dt)^2$ (If f^{cc} the acceleration of the particle is constant).

$\therefore \frac{1}{2} f^{cc} \cdot (dt)^2 = (dt)^2$

or $f^{cc} = 2 v \cdot \theta$

This is the required Coriolli's component of acceleration and is always perpendicular to the link.

HYDRAULIC ANALOGY:

Consider a short column of the fluid of length δr at distance r from the axis of rotation of the tube. Then if the velocity of the fluid relative to the tube is v and the angular velocity of the tube is ω the Coriolli's component of acceleration of the column is 2vω in a direction perpendicular to, and in the plane of rotation of the tube. The torque δT applied by the tube to produce this acceleration is then-

$$\frac{\delta w}{g} \times 2 v\omega$$

Where δw is the weight of fluid of the short column. If (w) is the specific weight of the fluid and (a) is the cross-section area of the tube outlet, then:

$$\delta w = wa \times \delta r$$

$$\delta T = 2v\omega - \frac{w}{g} \times ar \times \delta r$$

and the complete torque applied to a column of length L is given by

$$T = 2v\omega - \frac{w}{g} \times a \times \frac{L^2}{2}$$

$$T = Cc \times \frac{w}{g} \times a \times L^2$$

or Coriolli's of acceleration

$$Cc = \frac{2\,g\,T}{Wa\,L^2}$$

(Considering both tubes)

DESCRIPTION:

The Apparatus consists of two stainless steel tunes, projecting radially from a central Perspex tube are rotated by a DC swinging field motor, mounted vertically in a pillow blocks and bearings. A spring balance attached to a fixed swinging field motor with a fixed armed length measures the torque supplied by the motor.

A digital rpm indicator is provided to measure the speed of the motor. Water from the pump flows to Perspex tube through the control valve. The water flow rate is measured with the help of rotameter. The water leaving the radial tube circulate continuously by the water pump. The splash tank and all the accessories are mounted on a fabricated M.S. frame.

PROCEDURE:

1. Supply the main power to the control panel.
2. Switch on the motor and then increase the speed of the motor with the help of Variac up to certain speed.
3. Now start the water pump and maintain a constant water level in the vertical Perspex tube with the help of bye pass valve.
4. Note down the readings of spring balance for actual torque.
5. Note down the discharge and RPM from rotameter and digital RPM indicator respectively.
6. Now switch off the water pump and adjust the speed of the motor to its previous value.
7. Note down the reading on the spring balance for frictional torque.
8. Repeat the procedure for the different speeds and flow rates.

FORMULAE:

1. Coriolli's Component of acceleration (actual) = $\dfrac{2gt}{2 \times pw \times a\, L^2}$ m/s^2

2. Torque, T = F x R kgm
3. Coriolli's Component of acceleration (theo) = $2\,\omega v\ m/s^2$
4. Angular velocity, ω = $\dfrac{2\,\pi\,N}{60}$ rad/sec

Velocity, v = Q/2a m/s

OBSERVATION & CALCULATIONS:

DATA:

g	=	Acceleration due to gravity	=	$9.81\ m/s^2$
pw	=	Density of water	=	$1000\ kg/m^3$
d	=	Internal diameter of pipe	=	$6.0\ mm = 6.0 \times 10^{-3}\ m$
a	=	Cross sectional area of pipe	=	$2.827 \times 10^{-5}\ m^2$
L	=	Length of pipe	=	300 mm = 0.300 m
R	=	Swinging field arm	=	137 mm = 0.122 m
1 LPH	=	$2.77 \times 10^{-7}\ m^3/sec$		

OBSERVATION TABLE:

S. No.	Speed N	LPH	Force Kg
1			
2			
3			

CALCULATIONS:

S. No.	Torque (T) Kgm	Angular Velocity (ω) Rad/sec	Velocity of Water (v) m/s	Cor. Comp. of acceleration (theo) m/sec^2	Cor. Comp. of acceleration (act.) m/sec^2
1					
2					
3					

PRECAUTIONS & MAINTENANCE INSTRUCTIONS:

1. Don't exceed the discharge more than 2500 LPH.
2. To control the overflow in the central tube, increase the speed of the motor simultaneously as the discharge increases.
3. Bye pass valve should fully open before the experiment start.
4. Variac should be at zero position before starting the experiment.

TROUBLESHOOTING:

1. If pump gets jam, open the back cover of pump and rotate the shaft manually.

2. If pump gets heat up, switch off the main power for 15 minutes and avoid closing the flow control valve and bye pass valve simultaneously during operation.

RESULT:

Compared experimentally the coriolli's component of acceleration with theoretical values.

Viva-Voce:

Experiment No.-9

AIM :- To study various types of gear- Helical, cross helical, worm, bevel gear.

APPARATUS USED :- Arrangement of gear system.

THEORY :-

CLASSIFICATION OF GEAR :- Gears can be classified according to the relative position of their shaft axis are follows:

A: PARALLEL SHAFT

(i) Spur gear

(ii) Spur rack and pinion

(iii) Helical gears or Helical spur gear

(iv) Double- helical and Herringbone gear

B: INTER SECTING SHAFT

  (i) Straight bevel gear

  (ii) Spiral bevel gear

  (iii) Zerol bevel gear

C: SKEW SHAFT

(i) Crossed- helical gear

(ii) Worm gears(Non-throated, Single throated, Double throated)

SPUR GEAR:- They have straight teeth parallel to the axes and thus are not subjected to axial thrust due to teeth load. Spur gears are the most common type of gears. They have straight teeth, and are mounted on parallel shafts. Sometimes, many spur gears are used at once to create very large gear reductions. Each time a gear tooth engages a tooth on the other gear, the teeth collide, and this impact makes a noise. It also increases the stress on the gear teeth. ***Spur gears*** are the most commonly used gear type. They are characterized by teeth, which are perpendicular to the face of the gear. Spur gears are most commonly available, and are generally the least expensive.

Figure: Spur gear

HELICAL GEARS:- In helical gears, the teeth are curved, each being helical in shape. Two mating gears have the same helix angle, but have teeth of opposite hands. At the beginning of engagement, contact occurs only at the point of leading edge of the curved teeth. As the gears rotate, the contact extends along a diagonal line across the teeth. Thus the load application is gradual which result in now impact stresses and reduction in noise. Therefore, the helical gears can be used at higher velocities then the spur gears and have greater load – carrying capacity. The teeth on helical gears are cut at an angle to the face of the gear. When two teeth on a helical gear system engage, the contact starts at one end of the tooth and gradually spreads as the gears rotate, until the two teeth are in full engagement. This gradual engagement makes helical gears operate much more smoothly and quietly than spur gears. For this reason, helical gears are used in almost all car transmission. Because of the angle of the teeth on helical gears, they create a thrust load on the gear when they mesh. Devices that use helical gears have bearings that can support this thrust load.

Figure: Helical gear

DOUBLE HELICAL AND HERRING BONE GEARS :- A- double- helical gear is equivalent to a pair of helical gears secured together, one having a right – hand helix and the other a left hand helix. The tooth of two raw is separated by a grooved used for too run out.

If the left and the right inclinations of a double – helical gear meet at a common apex and there is no groove in between, the gear is known as herring bone gear.

CROSSED – HELICAL GEAR :- The used of crossed helical gear or spiral gears is limited to light loads. By a suitable choice of helix angle for the mating gears, the two shaft can be set at any angle.

WORM GEAR :- Worm gear is a special case of spiral gear in which the larger wheel, usually, has a hollow or concave shape such that a portion of the pitch diameter is the other gear is enveloped on it. The smaller of two wheels is called the worm which also has larger spiral angle. ***worm gear**:* Worm gears are used when large gear reductions are needed. It is common for worm gears to have reductions of 20:1, and even up to 300:1 or greater.

Figure: Worm gear

BEVEL GEAR :- Kinematically, the motion between two intersecting shafts is equivalent to the rolling of two cones, assuming no slipping. The gears, in general, are known as bevel gear. When teeth formed on the cones are straight, the gear are known as straight bevel and when inclined, they are known as spiral or helical bevel.

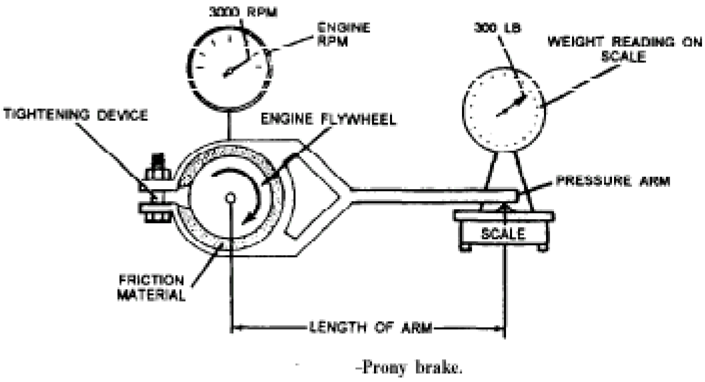
Result:- Different types of gear have been studied.

Viva-Voce:

EXPERIMENT NO:-10

AIM : - To study the various types of dynamometers.
APPARATUS USED : - Models of dynamometer.
THEORY:- The dynamometer is a device used to measure the torque being exerted along a rotating shaft so as to determine the shaft power.
Dynamometers are generally classified into:
1) Absorption dynamometers (i.e. Prony brakes, hydraulic or fluid friction brakes, fan brake and eddy current dynamometers)
2) Transmission dynamometers (i.e. Torsion and belt dynamometers, and strain gauge dynamometer)
3) Driving dynamometers (i.e. Electric cradled dynamometer)
PRONY BRAKE : - The prony and the rope brakes are the two types of mechanical brakes chiefly employed for power measurement. The prony brake has two common arrangements in the block type and the band type. Block type is employed to high speed shaft and band type measures the power of low speed shaft.
BLOCK TYPE PRONY BRAKE DYNAMOMETER :- The block type prony brake consists of two blocks of wood of which embraces rather less than one half of the pulley rim. One block carries a lever arm to the end of which a pull can be applied by means of a dead weight or spring balance. A second arm projects from the block in the opposite direction and carries a counter weight to balance the brake when unloaded. When operating, friction between the blocks and the pulley tends to rotate the blocks in the direction of the rotation of the shaft. This tendency is prevented by adding weights at the extremity of the lever arm so that it remains horizontal in a position of equilibrium.
Torque, $T = W*l$ in Nm
Power $P = 2\pi N* T/60$ in N-m/s
$= 2\pi N * W*l/60* 1000$ in kW
Where, W= weights in Newton
l = Effective length of the lever arm in meter and
N = Revolutions of the crankshaft per minute.

BAND TYPE PRONY BRAKE DYNAMOMETER: - The band type prony brake consists of an adjustable steel band to which are fastened wooden block which are in contact with the engine brake-drum. The frictional grip between the band the brake drum can be adjusted by tightening or loosening the clamp. The torque is transmitted to the knife edge through the torque arm. The knife edge rests on a platform or communicates with a spring balance.
Frictional torque at the drum = F*r
Balancing torque = W*l
Under equilibrium conditions, T = F*r = W*l in Nm.
Power = $2\pi N* T/60$ in N-m/s
$= 2\pi N * W*l/60* 1000$ in kW



-Prony brake.

Figure: Prony brake dynamometer

ROPE BRAKE DYNAMOMETERS: - A rope brake dynamometers consists of one or more ropes wrapped around the fly wheel of an engine whose power is to be measured. The ropes are spaced evenly across the width of the rim by flywheel. The upward ends of the rope are connected together and attached to a spring balance, and the downward ends are kept in place by a dead weight. The rotation of flywheel produces frictional force and the rope tightens. Consequently a force is induced in the spring balance.

Effective radius of the brake R = (D+ d)/2
Brake load or net load = (W-S) in Newton
Braking torque T = (W-S) R in Nm.
Braking torque =2πN* T/60 in N-m/s
= 2πN * (W-S)R/60* 1000 in kW
D= dia. Of drum
d = rope dia.
S = spring balance reading

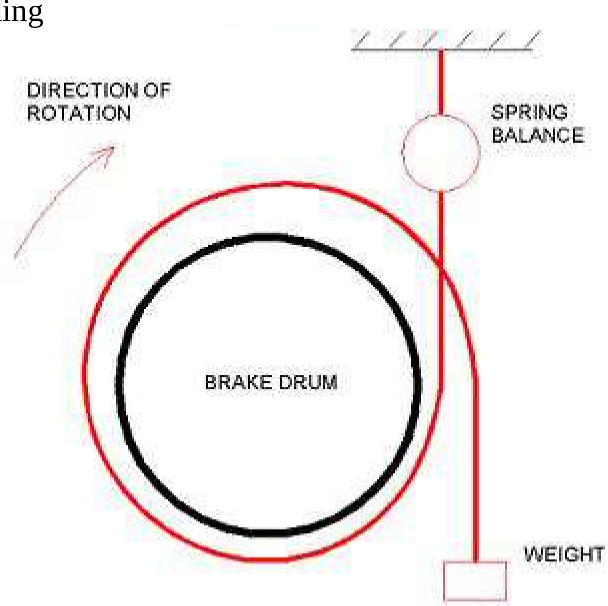


Figure: Rope brake dynamometer

FLUID FRICTION (HYDRAULIC DYNAMOMETER):- A hydraulic dynamometer uses fluid-friction rather than friction for dissipating the input energy. The unit consists essentially of two elements namely a rotating disk and a stationary casing. The rotating disk is keyed to the driving shaft of the prime-mover and it revolves inside the stationary casing. When the brake is operating, the water follows a helical path in the chamber. Vortices and eddy currents are set-up in the water and these tend to turn the dynamometer casing in the direction of rotation of the engine shaft. This tendency is resisted by the brake arm and balance system that measure the torque.

Brake power = W*N/k,

Where W is weight as lever arm, N is speed in revolutions per minute and k is dynamometer constant.

Approximate speed limit = 10,000rpm
Usual power limit = 20,000kW

BEVIS GIBSON FLASH LIGHT TORSION DYNAMOMETER: - This torsion dynamometer is based on the fact that for a given shaft, the torque transmitted is directly proportional to the angle of twist. This twist is measured and the corresponding torque estimated the relation:

T = Ip* C*θ / l

Where Ip = $\pi d^4/32$ = polar moment of inertia of a shaft of diameter d
θ = twist in radians over length l of the shaft

C = modulus of rigidity of shaft material

APPLICATIONS:-

i) For torque measurement.

ii) For power measurement.

VIVA-QUESTIONS:-

1. How many types of method of shaft power measurement ?
2. How many types of mechanical brakes ?
3. Which type mechanical brake use for high speed and low speed shaft ?
4. What is mean by effective radius of the brake drum?
5. Which types of bearing is same as the friction torque transmitted by a disc or plate clutch?