Submission of:

1. **Course Materials: Theory of Machine, Theory (ME-312) for ICD**
2. **Lab Manuals: Theory of Machine, Lab (ME-312) for ICD**
3. **50 MCQ**
4. **30 Short Answer Questions**
5. **20 Descriptive Questions**

Submitted By:

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Theory of Machine, Lab (ME-312) for ICD

List of experiments in sequence from the provided lab manual

1. **Experiment-1**: To study inversions of 4 Bar Mechanisms, Single & double slider crank mechanisms.
2. **Experiment-2**: To determine experimentally, the Moment of Inertia of a Flywheel and Axle and compare with the theoretical values.
3. **Experiment-3**: To plot follower displacement v/s cam rotation for various cam follower systems.
4. **Experiment-4**: To find experimentally the Gyroscopic couple on Motorized Gyroscope and compare with applied couple.
5. **Experiment-5**: To find the coefficient of friction between belt and pulley and plot a graph.
6. **Experiment-6**: To study the different types of centrifugal and inertia governors and demonstrate any one.
7. **Experiment-7**: To study the different types of brakes.
8. **Experiment-8**: To study the compound Screw Jack and to find out Mechanical Advantage, Velocity Ratio, and Efficiency.
9. **Experiment-9**: To determine the value of coefficient of friction for a given pair of surfaces using friction apparatus.
10. **Experiment-10**: To verify the law of moment using Bell Crank Lever.

11.

Experiment-1

AIM: To study inversions of 4 Bar Mechanisms, Single & double slider crank mechanisms.

APPARATUS: Single slider crank mechanism & double slider crank mechanism.

THEORY:

FOUR BAR MECHANISM: A four bar link mechanism or linkage is the most fundamental of the plane kinematics linkages. Basically it consists of four rigid links which are connected in the form of a quadrilateral by four pin joints.

INVERSIONS OF SINGLE SLIDER CRANK CHAIN: Different mechanisms obtained by fixing different links of a kinematics chain are known as its inversions.

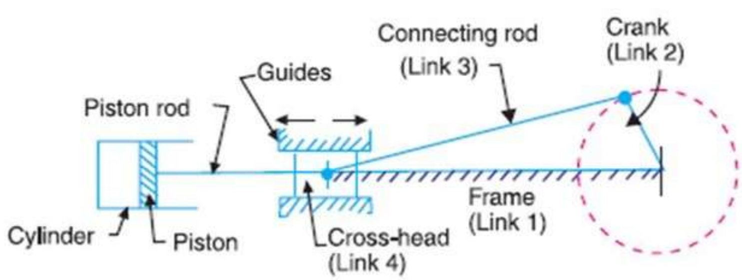
A slider –crank chain has the following inversions:

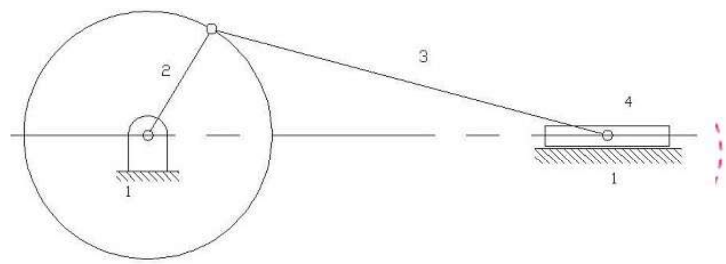
- First inversion (i.e; Reciprocating engine and compressor)
- Second inversion (i.e., Whitworth quick return mechanism and Rotary engine)
- Third inversion (i.e., Oscillating cylinder engine and crank & slotted – lever mechanism)
- Fourth inversion (Hand pump)

INVERSIONS OF DOUBLESLIDER CRANKCHAIN: A fourbar chain having two turning and two sliding pairs such that two pairs of the same kind are adjacent is known as a doubleslidercrank chain. The following are its inversions:

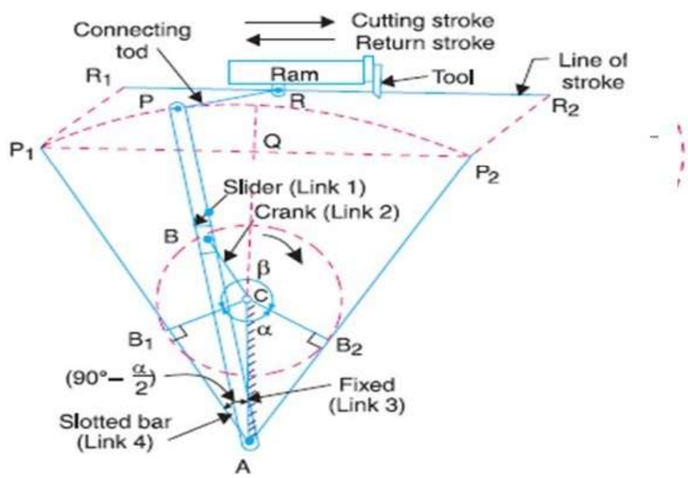
- First inversion (i.e., Elliptical trammel)
- Second inversion (i.e., Scotch yoke)
- Third inversion (i.e., Actual Oldham's coupling)

PROCEDURE: Reciprocating engine mechanism: In the first inversion, the link 1 i.e., the cylinder and the frame is kept fixed. The fig below shows a reciprocating engine.



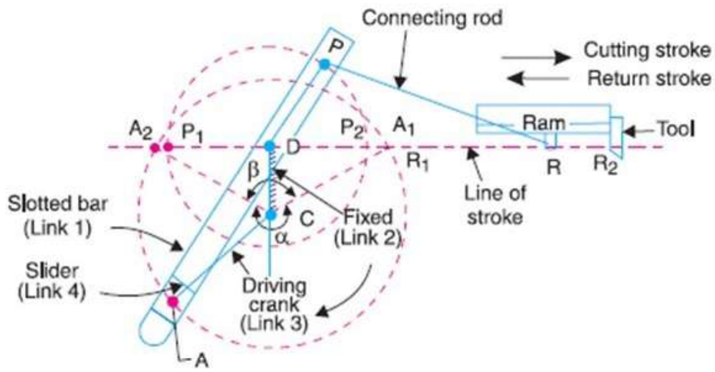


slotted link 1 is fixed. When the crank 2 rotates about O, the sliding piston 4 reciprocates in the slotted link 1. This mechanism is used in steam engine, pumps, compressors, I.C. engines, etc. Crank and slotted lever mechanism: It is an application of second inversion. The crank and slotted lever mechanism is shown in figure below.



This mechanism is used in shaping machines, slotting machines and in rotary engines. In this mechanism link 3 is fixed. The slider (link 1) reciprocates in oscillating slotted lever (link 4) and crank (link 2) rotates. Link 5 connects link 4 to the ram (link 6). The ram with the cutting tool reciprocates perpendicular to the fixed link 3. The ram with the tool reverses its direction of motion when link 2 is perpendicular to link 4. Thus the cutting stroke is executed during the rotation of the crank through angle α and the return stroke is executed when the crank rotates through angle β or 360 – α. Therefore, when the crank rotates uniformly, we get,

$$\frac{\text{Time of cutting stroke}}{\text{Time of return stroke}} = \frac{\beta}{\alpha} = \frac{\beta}{360^\circ - \beta} \quad \text{or} \quad \frac{360^\circ - \alpha}{\alpha}$$



Third inversion is obtained by fixing the crank i.e. link 2. Whitworth quick return mechanism is an application of third inversion. This mechanism is shown in the figure below. The crank OC is fixed and OQ rotates about O. The slider slides in the slotted link and generates a circle of radius CP. Link 5 connects the extension OQ provided on the opposite side of the link 1 to the ram (link 6). The rotary motion of P is taken to the ram R which reciprocates. The quick return motion mechanism is used in shapers and slotting machines. The angle covered during cutting stroke from P1 to P2 in counter clockwise direction is α or 360 2θ. During the return stroke, the angle covered is 2θ or β.

$$\frac{\text{Time of cutting stroke}}{\text{Time of return stroke}} = \frac{\alpha}{\beta} = \frac{\alpha}{360^\circ - \alpha} \quad \text{or} \quad \frac{360^\circ - \beta}{\beta}$$

Experiment- 2

AIM:

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

APPRATUS:

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

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,

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

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

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

$$= 2 \pi n_2 / t$$

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

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,

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

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

$$I = \frac{2\, mg.\, 2\pi r n_1 - mr^2 (4\pi n_2/t)^2}{(4\pi n_2/t)^2 \quad (1 + n_1/n_2)}$$

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

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

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

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 = $\frac{D}{2}$ = -----cm

CALCULATIONS:

(i) From 1st set I = -------- = ------------ g-cm^2
(ii) From 2nd set I = -------- = ------------ g-cm^2
(iii) From 3rd 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 $\Delta I/I$

$\therefore$ Maximum permissible error = $\frac{\Delta I}{I} . 100\%$ = ---%

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.

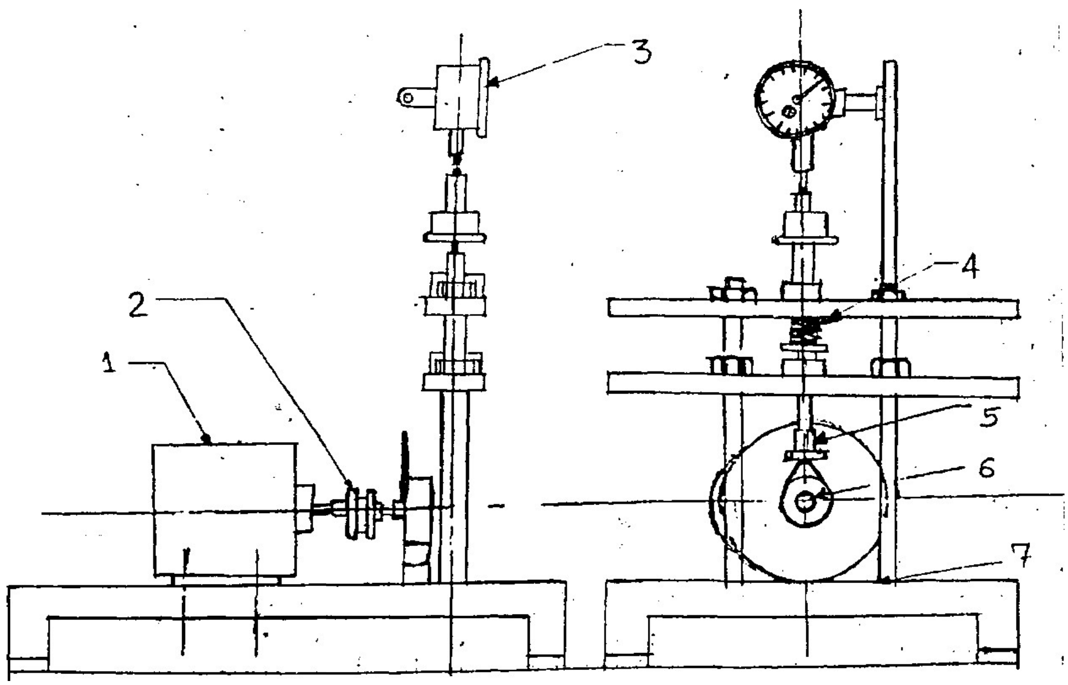
QUESTIONS FOR VIVA-VOCE

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-3

AIM: To plot follower displacement v/s cam rotation for various cam follower systems.

APPARATUS: Cam analysis apparatus provided with different sets of cam and followers.



1. Motor	2. Coupling	3. Dial gauge	4. Spring	5. Follower	6. Cam	7. Base frame

CAM ANALYSIS APPARATUS

INTRODUCTION:
A cam is a rotating device that transforms rotary motion into reciprocating motion. A rotating cam drives a follower & gives it some specified type of oscillating motion.

CONSTRUCTION: The machine is a motorized unit consisting of a cam shaft on free end of which a cam can be easily mounted & a follower (according to type of cam) is properly guided A graduated circular protector is fitted coaxially with the shaft & a dial gauge fitted to note the follower displacement for the angle of cam rotation as shown:

Different type of cam and follower pair used here are:
(i) Circular arc cam with mushroom follower.
(ii) Tangent cam with roller follower.
(iii) An eccentric cam with knife-edge follower.

PROCEDURE:

1. Select a suitable cam and follower combination say a tangent cam with a roller follower.
2. Set the cam on the cam shaft. Set a dial gauge for getting follower movements & set it to zero position.
3. Choose suitable amount of weight to be added to the follower.
4. Note down the readings of the follower displacements w.r.t. angular positions of the cam.
5. Draw the required cam profile.

OBSERVATION TABLE:

θ = Cam angle
α= Follower displacement

Sr. No.	Cam Angle = θ	Displacement α
1	30^0	
2	60^0	
3	90^0	
4	120^0	
5	150^0	
6	180^0	
7	210^0	
8	240^0	
9	270^0	
10	300^0	
11	330^0	
12	360^0	

RESULT: To plot a graph between θ *vs* α.

Experiment-4

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

APPRATUS:
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.) that is perpendicular to both the axis of spin OX and that of couple OY.

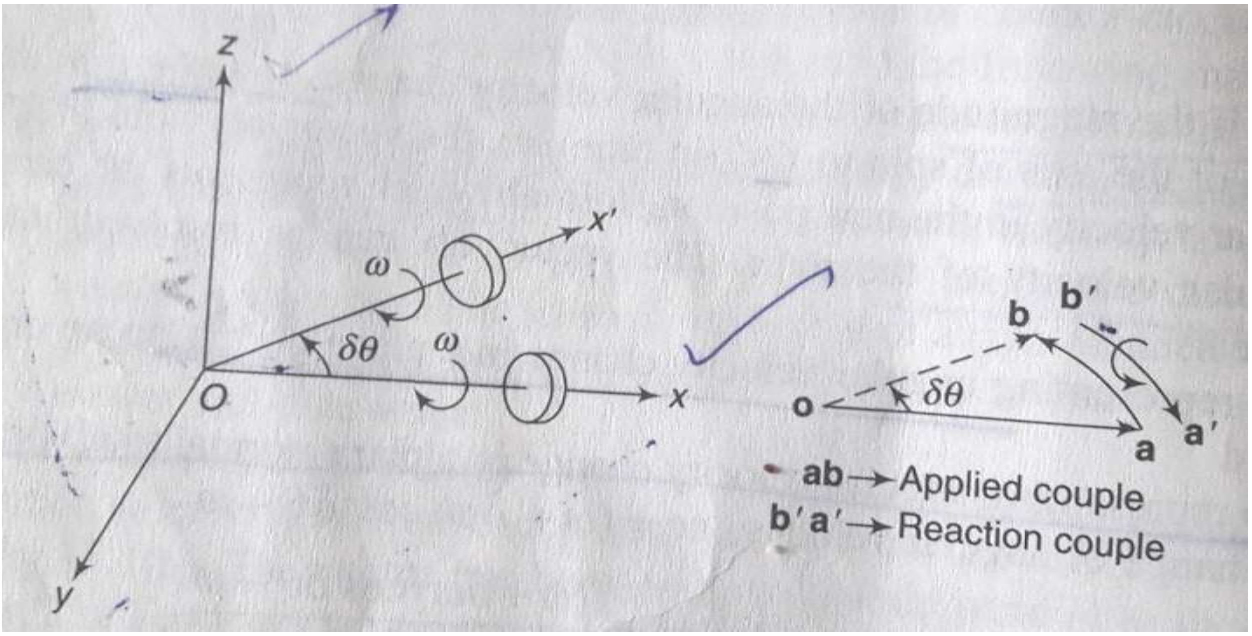
AXIS OF PRECESSION:
The third axis OZ is perpendicular to both the axis of spin OX and that of couple OY is known as axis of precession.

GYROSCOPIC EFFECT:
To a body revolving (or spinning) about an axis say OX, (refer fig.) 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.). Therefore, the angular momentum of disc is $I\omega$. 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.

Figure



Change of angular velocity = ab = I ω*δθ
Then angular acceleration = α = I*ω * δθ/δt

I = Moment of inertia of disc

ω = Angular Velocity of disc

We have dθ/dt = ωp = Angular velocity of precession of yoke about vertical axis.

Thus we get T = I *ω * 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 . 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:

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

$$I = (W * r^2)_{Kg.m2}$$

Where

$$\omega = (2 * \pi * N) / 60 \text{ rad/sec}$$

$$\omega p = (d\theta/dt) * (\pi/180) \text{ rad/sec}$$

$$T_{act} = w*L$$

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.-sec²**	**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.

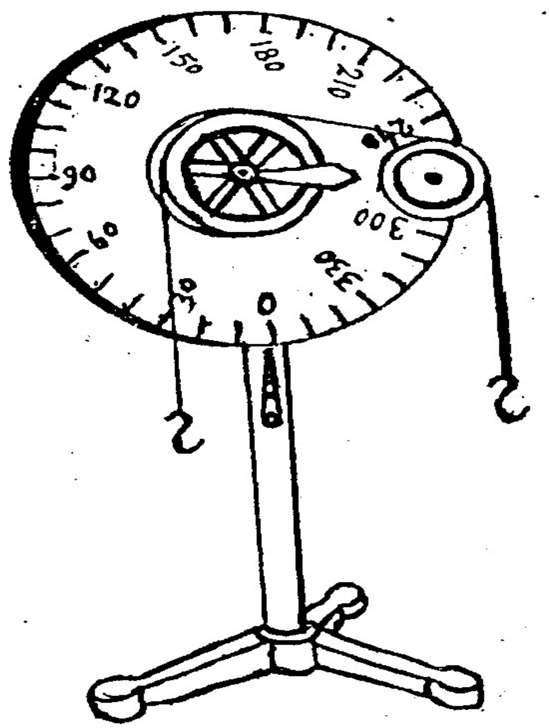
RESULT:

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

Experiment-5

AIM: **To find the coefficient of friction between belt and pulley and plot a graph between $\log_{10} T_1/T_2$ *v/s* θ.**

APPARATUS: Belt and Pulley apparatus spirit level, weights.



BELT AND PULLEY APPARATUS

PART OF APPRATUS:-

1. Heavy cast iron base with holes for foundation fitting.
2. Central rod dia. One end is fixed on base and the other end to-square block.
3. Cast iron pipe fitted with pointer.
4. Square block with stem.
5. Large wooden graduated dial.
6. Large Cast iron Double Groove pulley
7. Small Cast iron Double Groove pulley.
8. Key with handle.
9. Axle for rotating the wooden disc & pulley.

THEORY:

When a rough belt goes round a rough pulley or a rough string goes round any rough curve, the angle subtended at the center of the pulley by the surface of the belt in contact with the pulley is called the angle of contact (θ).

The driving side of the belt is called as tight side and the side which is being driven is called as slack side. If T_1 is the tight side tension and T_2 is the slack side tension then the coefficient of friction (μ.) between belt and pulley is given by the formula. $2.3 \log_{10} T_1/T_2 = \mu\,\theta$.

PROCEDURE:-

1. The graduation on the large disc attached to the apparatus is meant to set any angle of contact (θ). To test of graduations are rightly set, pass the belt over the pulley. Suspend some weight on the one end of the belt which is vertical. The other end of the belt may also be made to carry some weight. Adjust the rotating disc in such a manner that if one part of the belt is vertical, the other is horizontal, so that angle of lap is 90°. This can be tested by placing the spirit level on the horizontal part of the belt and adjusting the disc in such a manner that the bubble comes between two marks on the spirit level. See that the pointer rests against 90° graduation on the large disc, if not it may be taken as 90° and accordingly any angle (greater or less) can be set.
2. Put more weights on the two ends of the belt and set any angle less than 90°. Keeping the weights on the slack side as fixed add more weights on the tight side so that the tight side just begins to drag the slack side. The weights on the tight side give the value of T_1 and those on the slack side given the value T_2. The angle of contact is θ. Note: The slack side part of the belt passes.
3. Over a smooth movable pulley so that tensions on both sides of it are equal. The smooth movable pulley has been provided to set different values of θ. Otherwise, only an angle of 180° could be set.
4. Change the angle of contact θ. Keep the slack side tension (T_2) as same, find the tight side tension (T_1), as before. Take in this way about seven or eight readings for different values of θ.

OBSERVATION TABLE:

Sr. No.	Angle of Contact (θ)	θ (rad)	T_1 (gms)	T_2 (gms)	$\mu = \frac{2.3 log 10 T1}{T2.\theta}$	μ_{mean}
1	90^0					
2						
3						
4						
1	100^0					
2						
3						
4						
1	110^0					
2						
3						
4						

Experiment-6

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

APPRATUS: 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.

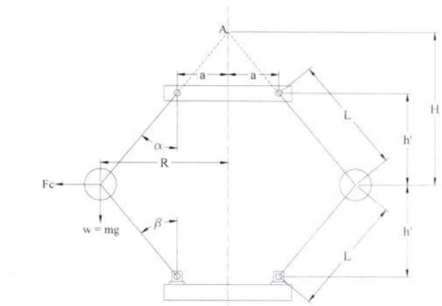
DESCRIPTION:

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

1. **WATT GOVERNOR**

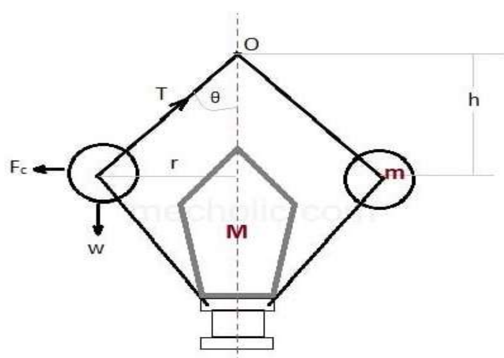
It is the simplest form of a centrifugal governor, which is known as Watt Governor. It is the original form of the governor used by Watt on early steam engines. It consists of two balls which are attached to the spindle with the helps of links or arms.



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.

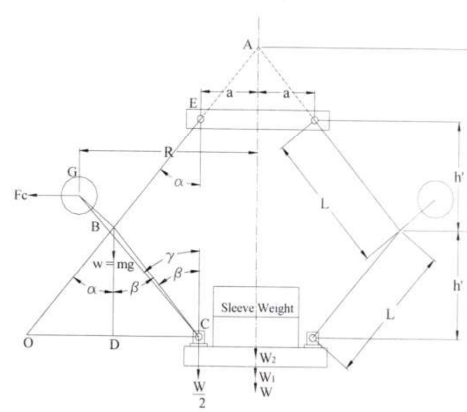
2. **PORTER GOVERNOR**

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



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.

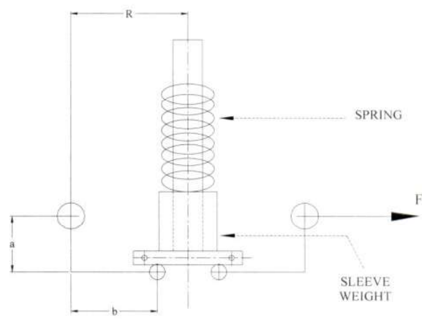
3. **PROELL GOVERNOR**



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.

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.



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, $H = \frac{g}{\omega^2}$

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

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

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

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

FORMULAE:

1. Initial reading of pointer on scale, X' = mm
2. Height gained by sleeve, X = (X" – X') mm
3. Height, h = $\left\{ h - \left(\frac{X}{2}\right) \right\}$ mm
4. $\alpha = \cos^{-1}\left(\frac{h}{L}\right)$
5. Governor height, H = $\left\{ \frac{a}{\tan \alpha} + h \right\}$ mm
6. Governor speed (theo.), Ntho = $\sqrt{\left(\frac{91.2 \times g \times 1000}{H}\right)}$ RPM
7. Radius of rotation, R = $\{ a + (L \sin\alpha) \}$ mm
8. Centrifugal force (actual), $F_{act} = \frac{w \times R \times \omega^2}{g \times 1000}$ kg
9. Angular Velocity, $\omega = \frac{2 \times \pi \times N}{60}$ radians/sec
10. Centrifugal force (theo), $F_{act} = \frac{w \times R}{H}$ 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 = $\{h - (\underline{X})\}$ mm
2
4. α = $\cos^{-1} (\underline{h})$
L
5. Governor height, H = $\{(\underline{a}) + h\}$ mm
$\tan\alpha$
6. Radius of rotation, R = $\{a + (L \sin\alpha)\}$ mm

7. Centrifugal force (actual), F_{act} = $\underline{w \times R \times \omega^2}$ kg
$g \times 1000$
8. Angular Velocity, ω = $2 \times \pi \times N/60$ radians/sec
9. Governor speed (theo.), Ntho = $\sqrt{}$ (w + W x 91.2 g x 1000) RPM
w H
10. Centrifugal force (theo), F_{act} = $[w + \underline{W} (1+k)] \tan\alpha$ kg
2

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

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

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 = $\left\{h - \frac{X}{2}\right\}$ mm
4. α = $\cos^{-1}\left(\frac{h}{L}\right)$
5. Governor height, H = $\left\{\frac{a}{\tan \alpha} + h\right\}$ mm

6. Ύ = [(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 = $\left\{h - \frac{X}{2}\right\}$ = h mm
12. Centrifugal Force (theo) F_{act} = $\left\{(W+2w) \frac{BD \tan\alpha}{GC \cos Y}\right\} - \{\mathbf{w} \tan Y\}$ kg
13. Governor speed (theo.), Ntho = $\sqrt{\left\{\frac{(W + w) F_{act} \, g \, 1000}{w R}\right\}}$ 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 = $\{R' + X \times \frac{a}{b}\}$ 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 = $2 * (a/b)^2 * \frac{(F_c - F_c')}{R - R'}$

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 tanβ = tanα

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.

Experiment-7

AIM: To study the different types of brakes.

APPRATUS:
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.

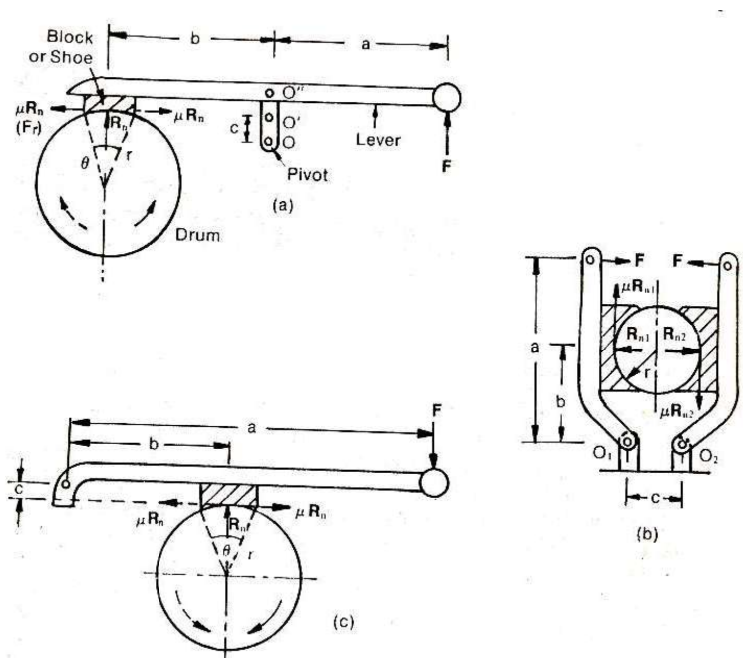


Fig 9.1

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

$$F = R_n \left[b - \mu c \right]$$

a

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 40°, the normal pressure is less at the ends than at the center. In that case, μ given by

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

$\theta + \sin\theta$

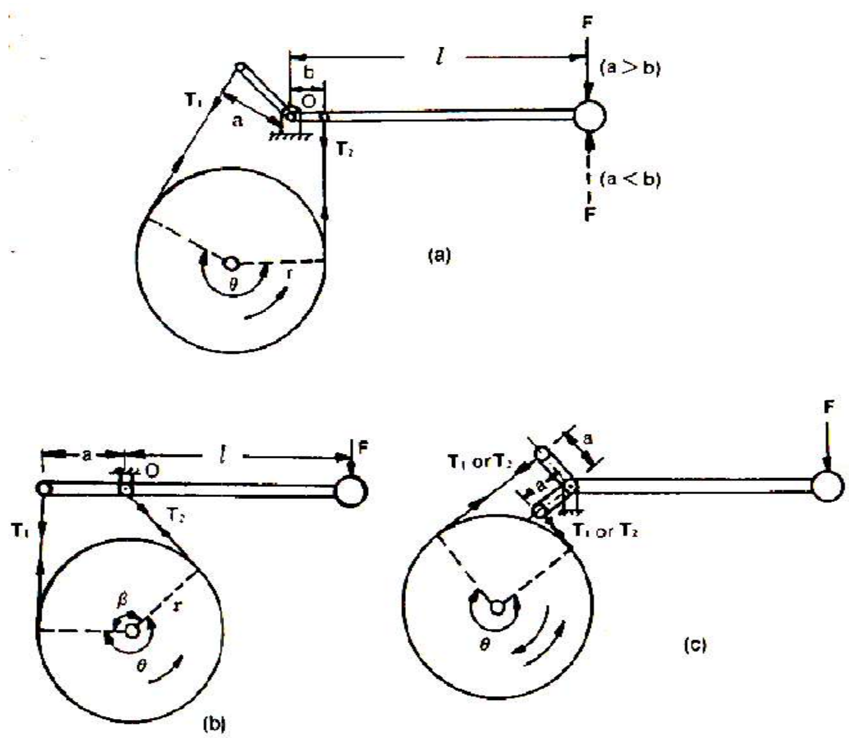
BAND BRAKE:

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.



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,

$$Fl - T_1 a + T_2 b = 0$$

$$\text{Or} \quad F = \frac{T_1 a + T_2 b}{l}$$

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 $T_1 a$ is greater than $T_2 b$.

$$\text{Or} \quad T_2 b. < T_1 a$$

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

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

$$Fl - T_1 a = 0$$

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

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.

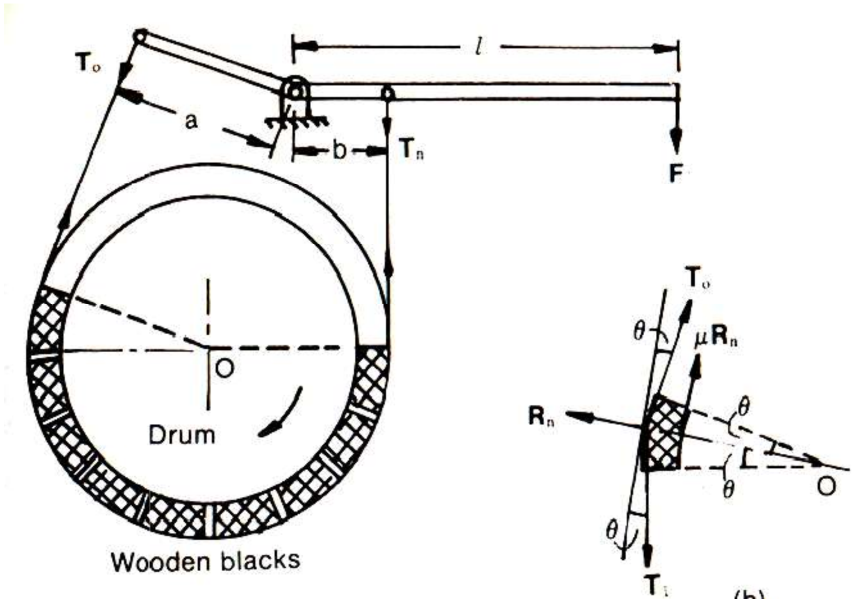
For both directions of rotation of the drum,

$$Fl - T_1 a - T_2 a = 0$$

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

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.



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,

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

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

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

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

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

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

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

Similarly,

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

And

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

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

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

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.

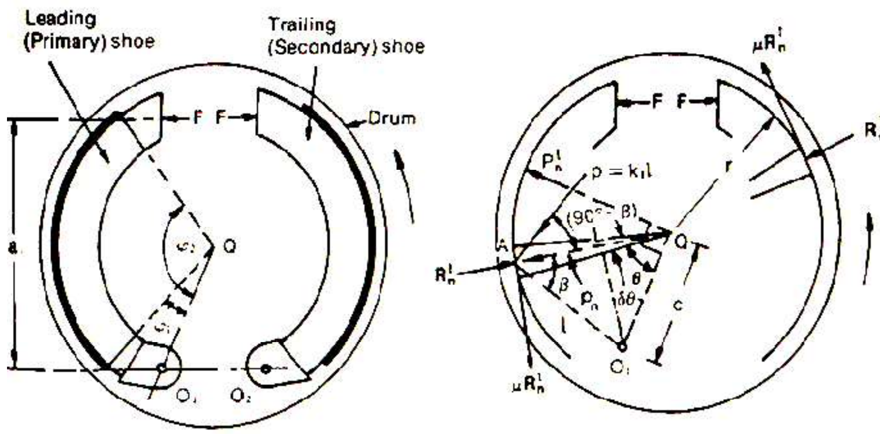


Figure shows an internal shoe automobile brake. It consists of two semi-circular shoes which are lined with a friction material such as ferodo. The shoes press against the inner flange of the drum when the brakes are applied. Under normal running of the vehicle, the drum rotates freely as the outer diameter 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, $p_n = k_1 l \cos(90^\circ - \beta) = k_1 l \sin\beta$

$= k_1 c \sin\theta$

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

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

p_n is maximum when $\theta = 90^\circ$

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

P^l_n max = $P^l_n = k_2 \sin 90^\circ = k_2$

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

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

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

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

and

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

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

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

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

$\overline{\quad 4 \quad}$

Taking moments about the fulcrum O_2 for the trailing shoe,

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

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

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

Braking torque,

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

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

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

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

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

Experiment-8

AIM: To study the compound Screw Jack and to find out Mechanical Advantage, Velocity Ratio and Efficiency.

APPARATUS: Compound Screw Jack App. weights

THEORY:
Screw Jack is a device employed for lifting heavy loads with help of a small effort applied at its handle. The loads are usually centrally loaded upon it. Screw jacks of three types:-

1. Simple screw jack.
2. Compound Screw jack.
3. Differential Screw jack.

A simple screw jack consists of a nut, a screw square threaded and a handle fitted to the head of the screw. The nut also forms the body of the jack.
The load to be lifted is placed on the head of the screw. Here the axial distance between corresponding points on two consecutive threads is known as pitch. If ‘p’ be the pitch of the screw and ‘t’ is the thickness of thread, then p = 2t.

V.R. = Distance moved by the effort/Distance moved by the load = $\frac{2\pi\ell}{P}$

Now Mechanical Advantage = $\frac{W}{P}$

Compound Screw jack

SPECIFICATION OF COMPOUND SCREW JACK:

The nut of the screw is fitted on a pedestal bearing and is keyed to a worm gear of **40 teeth** operated by a worm screw. The spindle of the worm screw is provided with an effort wheel of **15 cm** diameter. Weights are not included.

THEORY OF COMPOUND SCREW JACK: - In which the velocity ratio is further intensified with the help of a geared screw jack, in which the screw is lifted with the help of worm and worm wheel, instead of effort at the end of a lever. Now consider a worm geared screw jack.

Let, ℓ = Radius of the effort wheel = 65mm

þ = pitch of the screw = 3mm
P = effort applied to lift the load = ……gm
W = Load lifted and = …….gm
T = No. of teeth on the worm wheel. = 40

We know that distance moved by effort in one revolution of wheel = 2πℓ

If the worm is single threaded then the worm wheel move through 1/T revolution.
Therefore distance moved by the load = þ / T

Velocity Ratio (V.R.) = 2πℓ / þ / T

Mechanical Advantage (M.A) = W/P
Efficiency μ = M.A. / V.R.

OBSERVATION TABLE OF COMPOUND SCREW JACK:

S. No.	Load (W) in gm.	Effort (P) in Nt.	Distance moved by effort	Distance moved by load	M.A.= W/P	V.R. = 2πℓ/ þ/T	Efficiency μ = M.A. / V.R.
1							
2							
3							
4							
5							

CALCULATION:-

M.A. = W/P
V.R. = Distance moved by effort/Distance moved by load
Efficiency = M.A. / V.R.

PRECAUTIONS: -

1. Rope should not be overlap.
2. Carefully measure pitch of screw.
3. Effort handle move smoothly do not applied suddenly or jerking.
4. Oiling & greasing should be properly.
5. Effort arm measure very carefully.

Experiment-9

AIM: To determine the value of coefficient of friction for a given pair of surfaces using friction apparatus.

APPARATUS:

Horizontal/Inclined Apparatus, Sprit level weights, Slider: - Wood & Aluminum.

THEORY:

When two bodies are in contact and one is sliding or has a tendency to slide over the other when there is a resistant force which opposes the sliding is called (resistance) frictional resistance which is self-adjusting by nature .The minimum value of it occurs when the bodies are just on the point of sliding and is called limiting frictional resistance which is proportional to the normal reaction. At this moment motion set in this resistance. The figure showing the force acting on a block at the point of sliding is on left hand side.

This being the state of affairs with horizontal board apparatus the pull F is supplied by weights suspended by setting. In case of inclined plane apparatus, the slider is made to slide down the inclined plane by the component of weight acting down the plane. If at the law of Friction is true , the slider should slide down the plane at the same inclination of the board (when the two surfaces are Int. changed) Irrespective of the loading of the slider and the coefficient of friction (F) being given by Tangent of the angle of the board with horizontal when the friction is limiting P.

<u>Inclined Plane Friction</u> **<u>Horizontal Plane Friction</u>**

$P = \mu R + W\sin\alpha$ -------- I

$R = W\cos\alpha$ ----------- II

Put II in I

$P = \mu W \cos\alpha + W\sin\alpha$

$P - W\sin\alpha = \mu W\cos\alpha$

$\mu = \frac{P - W\sin\alpha}{W\sin\alpha}$

$\mu = \frac{P/W - \sin\alpha}{\cos\alpha}$

$\mu = \tan\theta$

$\theta = \tan^{-1}\mu$ (Angle of Friction)

EXPERIMENTAL SETUP: A plane consisting of inclined load made of timber (with any other desired facing material under test) is hinged at one to the base of apparatus. The inclination of this plane can be measured on the graduated sector attached to the base. The coefficient of friction of material is to be determined with respect to the other for in spacing at the bottom of box type required degree.

To this slider is attached to a string (which goes over smooth pulley) and the free end which hinges the apparatus area as on inclined board, no string is attached to the slider (this being made to slide due to self-weight down the plane.) This being the state of affairs with horizontal plane apparatus the pull F is supplied by weights suspended by setting. In case of inclined plane apparatus, the slider is made to slide down the inclined plane by the component of weight acting down the plane. If the law of friction is true, the slider should slide down the plane at the same inclination of the board (when the two surfaces are interchanged) irrespective of the loading of the slider and the coefficient of friction given by tangent of the angle of the board with horizontal when the friction is limiting.

PROCEDURE:

1. Take the inclined board with a glass surface.
2. Keep it horizontal initially and put the slider with steel base on it. Increase the inclination of the inclined board gradually till slider just setting to slide on it.
3. Note the angle in this position. This is the angle of friction. Let it be 9.
4. Now place some weights in the slider and repeats the experiment before. Let the angle of inclination in this case be.
5. Repeat the experiment as before with different weights in the slider each time. Note the corresponding angle of inclination of the inclined board and take their mean. Let it be this is the angle of friction.

Then the coefficient of friction (μ) given by $\mu = \tan\theta$

The experiment may be repeated with the other two different surfaces to find the coefficient of friction between them.

OBSERVATION TABLE:

Wt. of Wooden Slider = 77 gms.

Wt. of Aluminum slider = 102 gms.

Wt. of Pan = 29 gms.

θ = Angle of Friction

Coefficient of friction (μ) given by $\mu = \tan\theta$

FOR HORIZONTAL SURFACE

Table 1: For Glass Surface + Wooden Slider

Sr. No.	W = Wt. of Slider + Wt. added (gms)	P = Wt. of Pan + Wt. added (gms)	$\mu = P/W$	$\theta = \tan^{-1}\mu$
1				
2				
3				
4				

Table 2: For Glass Surface + Aluminum Slider

Sr. No.	W = Wt. of Slider + Wt. added (gms)	P = Wt. of Pan + Wt. added (gms)	$\mu = P/W$	$\theta = \tan^{-1}\mu$
1				
2				
3				
4				

FOR INCLINED SURFACE WITH PULLEY

Table 1: For Glass Surface + Wooden Slider

Sr. No.	Wt. of Slider + Wt. added = W (gms)	Wt. of Pan + Wt. added = P (gms)	Angle of Surface (α)	$\mu = \frac{P/W - \sin\alpha}{\cos\alpha}$	$\theta = \tan^{-1}\mu$
1					
2					
3					

4					

Table 2: For Glass Surface + Aluminum Slider

Sr. No.	Wt. of Slider + Wt. added = W (gms)	Wt. of Pan + Wt. added = P (gms)	Angle of Surface (α)	$\mu = \frac{P/W - \sin\alpha}{\cos\alpha}$	$\theta = \tan^{-1}\mu$
1					
2					
3					
4					

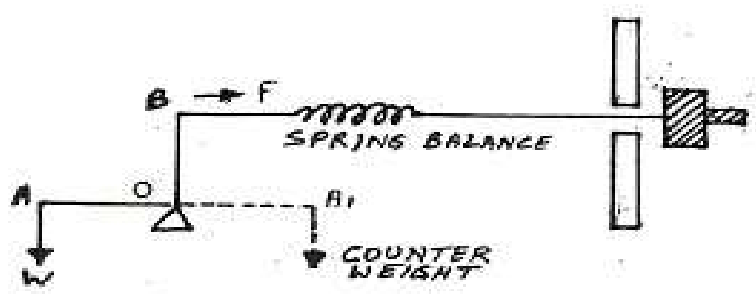
PRECAUTION:
1. Clean the two surfaces so that there is no grease
2. Increase the angle very slowly
3. Particular region of the inclined board may be used.
4. The block should just begin to move. It should not move.

RESULTS: Under experimental error it is verified that limiting frictional resistance is proportional to normal reaction between two surfaces that is law of friction.

Experiment-10

AIM: To verify the law of moment using Bell Crank Lever.

APPARATUS: The bell crank consists of a lever with two arms (OA) & (OB) at right angle to each other and a counter weight C is mounted on the fulcrum O by a chain through a spring balance to a vertical post at 0. The arm OB is kept vertical (this can be checked by the indicator point by tightening the adjusting nut and under these conditions the chain and the arm OA are horizontal) a sliding load is suspended in the arm OA



BELL CRANK LEVER

THEORY: The equilibrium according to the Law of Moment states

$(W\times OA) = (F\times OB)$

W = Weight Suspended from arm OA

F = Reading of force in the spring balance

Thus to prove the principle of moment we have to show that both the sides of the equation are equal.

PROCEDURE: The weight (W) is placed at some place along the arm OA (OA being measured) and then make the arm OB vertical and find out the pull F in the chain, OB is also measured. Change the position of W and note the corresponding value of F. In case the counter balance is faulty use the method of difference i.e. study the change in moment (W×OA) due to shifting of W and corresponding change in moment (F×OB) due to change in the value of F. if the two change are same the principle is verified.

OBSERVATION TABLE:

S. No.	W (gm)	OA (cm)	W×OA (gm. cm)	F (gm)	OB (cm)	F×OB (gm. cm)	Change in moment (gm×cm)

RESULT:

Since the change in moment (W×OA) is equal to the corresponding change in moment (F×OB), thus the Law of moment is verified.

PRECAUTIONS:

1. Note all the reading carefully.
2. Check spring balance at zero before taking reading.
3. Increase weight gradually without impact.

Experiment 11

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

APPRATUS:
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.

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

3. 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.
4. 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.

QUESTIONS FOR 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 …………………………..