LAB MANUAL

THEORY OF MACHINES (MED 372)

G.S. Mandal's

**MAHARASHTRA INSTITUTE OF TECHNOLOGY,
AURANGABAD**

DEPARTMENT OF MECHANICAL ENGINEERING

NAME OF LABORATORY : THEORY OF MACHINES	
LABORATORY MANUAL	
CLASS: THIRD YEAR	**COURSE CODE: MED 372**
NAME OF COURSE : THEORY OF MACHINES	

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Vision of Institute:

MIT aspires to be a leader in Techno-Managerial education at national level by developing students as technologically superior and ethically strong multidimensional personalities with a global mindset.

Mission of Institute:

We are committed to provide wholesome education in Technology and Management to enable aspiring students to utilize their fullest potential and become professionally competent and ethically strong by providing,

- Well qualified, experienced and Professionally trained faculty
- State-of-the-art infrastructural facilities and learning environment
- Conducive environment for research and development.
- Delight to all stakeholders.

Vision of Mechanical Engineering Department

To be a center of excellence in the field of Mechanical Engineering where the best of teaching, learning and research synergize and serve the society through innovation and excellence in teaching.

Mission of Mechanical Engineering Department

To provide world-class under-graduate and graduate education in Mechanical Engineering by imparting quality techno-managerial education and training to meet current and emerging needs of the industry and society at large.

Program Educational Objectives (PEOs):

PEO 1	Graduates will apply the tools and skills acquired during their undergraduate studies either in advanced studies or as employees in engineering industries.
PEO 2	Graduates of the program will have successful technical and professional career.
PEO 3	Graduates of the program will continue to learn to adopt constantly evolving technology.
PEO 4	Graduates will demonstrate sensitivity towards societal issues.

Program Outcomes

POs	Particulars	Description
PO1	**Engineering knowledge**	Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems
PO2	**Problem analysis**	Identify, formulate, review research literature, and analyze complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.
PO3	**Design/development of solutions**	Design solutions for complex engineering problems and design system components or processes that meet the specified needs with appropriate consideration for the public health and safety, and the cultural, societal, and environmental considerations.
PO4	**Conduct investigations of complex problems:**	Use research-based knowledge and search methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.
PO5	**Modern tool usage**	Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modelling to complex engineering activities with an understanding of the limitations.
PO6	**The engineer and society**	Apply reasoning informed by the contextual knowledge to assess societal, health, safety, legal and cultural issues and the consequent responsibilities relevant to the professional engineering practice.
PO7	**Environment and sustainability**	Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the

		knowledge of, and need for sustainable development.
PO8	**Ethics**	Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.
PO9	**Individual and team work**	Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.
PO10	**Communication**	Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.
PO11	**Project management and finance**	Demonstrate knowledge and understanding of the engineering and management principles and apply these to one's own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.
PO12	**Life-long learning**	Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

Program Specific Outcomes:

PSO 1	Ability to design & analyze components & systems for mechanical performance
PSO 2	Ability to apply and solve the problems of heat power and thermal systems
PSO 3	Ability to solve real life problems with the exposure to manufacturing industries

Course Objective:

1. To understand classification and types of mechanisms.
2. To understand kinematics and dynamics of various machines.
3. To understand functions and types of various machine elements.
4. To understand balancing of various unbalanced forces in machines.

Course Outcomes:

CO	Code	Statement
CO 1	MED 305.1	Understand the principles of kinematic pairs, chains and their classification, degree of freedom, inversions and other mechanisms.
CO 2	MED 305.2	Analyze the planar mechanisms for the position, velocity and acceleration.
CO 3	MED 305.3	Synthesize planar four bar and slider crank mechanisms for specified kinematic conditions.
CO 4	MED 305.4	Design cams and followers for specified motion profiles.
CO 5	MED 305.5	Understand the working principle of Flywheel, Governor, Brakes and Dynamometer.
CO 6	MED 305.6	Determine the unbalanced forces in various types of rotating and reciprocating masses and engines.

University Syllabus:

Dr. Babasaheb Ambedkar Marathwada University, Aurangabad
(Faculty of Science & Technology)
Syllabus of T. Y. B. Tech. (Mechanical)

Course Code: MED305	**Course: Theory of Machines**
Teaching Scheme: 04 Hrs/week	**Class Test: 20 marks**
Theory: 04 Hrs/week	**Theory Examination (Duration): 04Hrs**
Credits: 04	**Theory Examination (Marks): 80**

Objectives	1. To understand classification and types of mechanisms. 2. To understand kinematics and dynamics of various machines. 3. To understand functions and types of various machine elements. **4.** To understand balancing of various unbalanced forces in machines.
Unit: I	**Mechanisms and Inversions**: Rigid body, Mechanism and Machine, Kinematic Link, Kinematic Pair and their Classification, Degrees of Freedom, Kinematic Chain, Linkage, Mechanism and Structure, Gruebler's Criterion for degrees of freedom, Inversions of Four Bar mechanism, Slider-Crank chain mechanism, Kinematic inversions, Double slider-crank chain mechanism. **04 Hrs**
Unit: 2	**Velocity Analysis**: Velocity analysis of mechanisms (having maximum six links) using relative velocity method and Instantaneous centre method, Kennedy's theorem, Determination of linear and angular velocities and their directions. **06 Hrs**
Unit: 3	**Acceleration analysis**: Acceleration analysis of mechanisms. Problems involving Corioli's component of acceleration. Determination of linear and angular component of acceleration using graphical and analytical method

	Klein's construction and Ritterhaus construction method for simple engine mechanisms. **08 Hrs**
Unit: 4	**Classification of cams and followers**: Types of cams and followers, Terminology and definitions. Displacement diagrams of follower with Uniform velocity, simple harmonic motion, uniform acceleration and retardation and cycloidal motions. Construction of cam profile using these motions. Determination of velocity and acceleration. **06 Hrs**
Unit: 5	**Flywheel, Governors, Brakes and Dynamometer:** Turning moment diagram, fluctuation of energy in flywheel. Principle, working and types of governor like Watts, Porter, Proell and Hartnell governor. Functions and types of Brakes. Types of dynamometers. Absorption type dynamometers like Prony brake rope brake and transmission type dynamometer like belt transmission, epicyclic gear train and torsion dynamometer. **06 Hrs**
Unit: 6	**Balancing**: Balancing of rotating masses acting in one or more planes. Static and dynamic balancing. Balancing of reciprocating engines. Primary and secondary forces and couples acting on single cylinder and double cylinder engines. Balancing of in line, radial, V-engine. **06 Hrs**

Text Books / References Books	**Sr. No.**	**Title**	**Author**	**Publication**
	1	Theory of Machines	S. S. Ratan	Tata McGraw Hill Education

	2	Theory of Machines	T. Beven	Pearson Education India
	3	Theory of Machines	Balaney	Khanna Publications.
	4	Theory of Machines	Joseph E Shigley	John Uicker McGraw Hill
	5	Text book Theory of Machines	R.K.Bansal	Laxmi publications

List of Practical (Any 10 practicals to be conducted)

1. Study of kinematics, pairs, various simple mechanisms and their inversions.
2. Solution of two problems on velocity analysis by instantaneous center method.
3. Solution of two problems on velocity analysis by relative velocity method.
4. Solution of two problems on acceleration analysis by relative velocity method.
5. Study of various types of brakes and dynamometers.
6. Solution of two problems on balancing.
7. Study of gyroscopic effect and finding moment of inertia of gyroscopic disc.
8. Determine radius of gyration of a given bar using bifilar and trifilar suspension.
9. Plotting controlling force diagram for porter and Hartnell governor.
10. Solution of two problems on Cams.
11. Study of Whirling of Shaft
12. Study of Static and dynamic balancing machine

Lab Instructions

1. Student should wear college ID-card and must carry record and observation.

2. Take signature of lab in charge after completion of observation and record.

3. If any equipment fails in the experiment report it to the supervisor immediately.

4. Students should come to the lab with thorough theoretical knowledge.

5. Don't touch the equipment without instructions from lab supervisor.

6. Don't crowd around the experiment and behave in-disciplinary.

7. Students should carry their own stationary and required things.

8. Using the mobile phone in the laboratory is strictly prohibited.

EXPERIMENT NO. 1

Aim: - Study of kinematics, pairs, various simple mechanisms and their inversions.

Theory: - Introduction

A machine consists of a number of parts or bodies. In this chapter, we shall study the mechanisms of the various parts or bodies from which the machine is assembled. This is done by making one of the parts as fixed, and the relative motion of other parts is determined with respect to the fixed parts

Kinematics Link or Element:-

Each part of a machine, which moves relative to some other parts, is known as Kinematic link or element. A link may consist of several parts, which are rigidly fastened together, so that they do not move relative to one another. A link or element needs not to be a rigid body, but it must be a resistant body. A body is said to be a resistant body if it is capable of transmitting the required forces with negligible deformation. Thus a link should have the following two characteristics:

1. It should have relative motion.
2. It must be a resistant body.

Types of Links:-

In order to transmit motion, the driver and the follower may be connected by the following three types of links:

1. ***Rigid link***. A Rigid link is one, which does not undergo any deformation while transmitting motion. Strictly speaking, rigid links do not exist. However, as the

deformation of a connecting rod, crank etc. of a reciprocating steam engine is not appreciable, they can be considered as rigid links.

2. ***Flexible link***. A flexible link is one, which is partly deformed in a manner not to affect the transmission of motion. For example, belt, ropes, chains and wires are flexible links and transmit tensile forces only.

3. ***Fluid link***. A fluid link is one, which is formed by having a fluid in a receptacle, and the motion is transmitted through the fluid by pressure or compression only, as in the case of hydraulic presses, jacks and brakes.

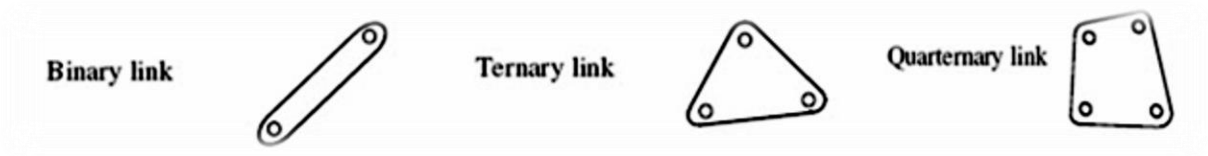


Fig.1.1 Different type of Links

Structure:-

It is an assemblage of a number of resistant bodies (known as members) having no relative motion between them and meant for carrying loads having straining action. A railway bridge, a roof, truss, machine frames etc.' are the example of a structure.

Kinematics Pair:-

The two links or elements of a machine, when in contact with each other, are said to form a pair. If the relative motion between them is completely or successfully constrained (i.e. in a definite direction), the pair is known as Kinematics pair.

Classification of Kinematics Pairs:-

The kinematics pairs may be classified according to the following consideration:

1. According to the type of relative motion between the elements. The kinematics pairs according to type of relative motion between the elements may be classified as discuss below:
   (a) *Sliding pair*. When the two elements of a pair are connected in such a way that one can only slide relative to the other, the pair is known as sliding pair. The piston and cylinder, cross-head and guides of a reciprocating steam engine, ram and its guides in shaper, tail stock on the lathe bed etc. are the example of a sliding pair. A little consideration will show, that a sliding pair has a completely constrained motion.
   (b) *Turning point*. When the two elements of a pair are connected in such a way that one can only turn or revolve about a fixed axis of another link, the pair is known as turning pair. A shaft with collars at both ends fitted into a circular hole, the crankshaft in a journal bearing in an engine, lathe spindle supported in head stock, cycle wheels turning over their axles etc. are the examples of a turning pair. A turning pair also has a completely constrained motion.
   (c) *Rolling pair*. When the two elements of a pair are connected in such a way that one rolls over another fixed link, the pair is known as rolling pair. Ball and other bearings are examples of rolling pair.
   (d) *Screw pair*. When the two elements of a pair are connected in such a way that one element can turn about the other by screw threads, the pair is known as screw pair. The lead screw of a lathe with nut, and bolt with a nut are examples of a screw pair.
   (e) *Spherical pair*. When the two elements of a pair are connected in such a way that one element (with spherical shape) turns or swivels about the other fixed element, the pair formed is called a spherical pair. The ball and socket joint, attachment of a car mirror, pen stand etc., are the example of a spherical pair.

2. **According to the types of contact between the elements**:- The kinematics pairs according to the type of contact between the elements may be classified as discussed below:
   (a) *Lower pair*. When the two elements of a pair have a surface contact when relative motion takes place and the surface of one-element slides over the surface of the other, the pair formed is known as lower pair.
   (b) *Higher pair*. When the two element of a pair have a line or point contact when relative motion takes place and the motion between the two elements is partly turning and partly sliding, then the pair is known as higher pair. A pair of friction discs, toothed gearing, belt and rope drives; ball and roller bearings and are the examples of higher pair.

3. **According to the type of closure.**:-The kinematics pairs according to the types of closure between the elements may be classified as discussed below:
   (a) ***Self-closed pair.*** When the two elements of a pair are connected together mechanically in such a way that only required kind of relative motion occurs, it is then known as self-closed pair. The lower pairs are self-closed pair.
   (b) ***Force-closed pair***. When the two elements of a pair are not connected mechanically but are kept in contact by the action of external forces, the pair is said to be a force-closed pair. The cam and follower is an example of force closed pair, as it is kept in contact by the forces exerted by spring and gravity.

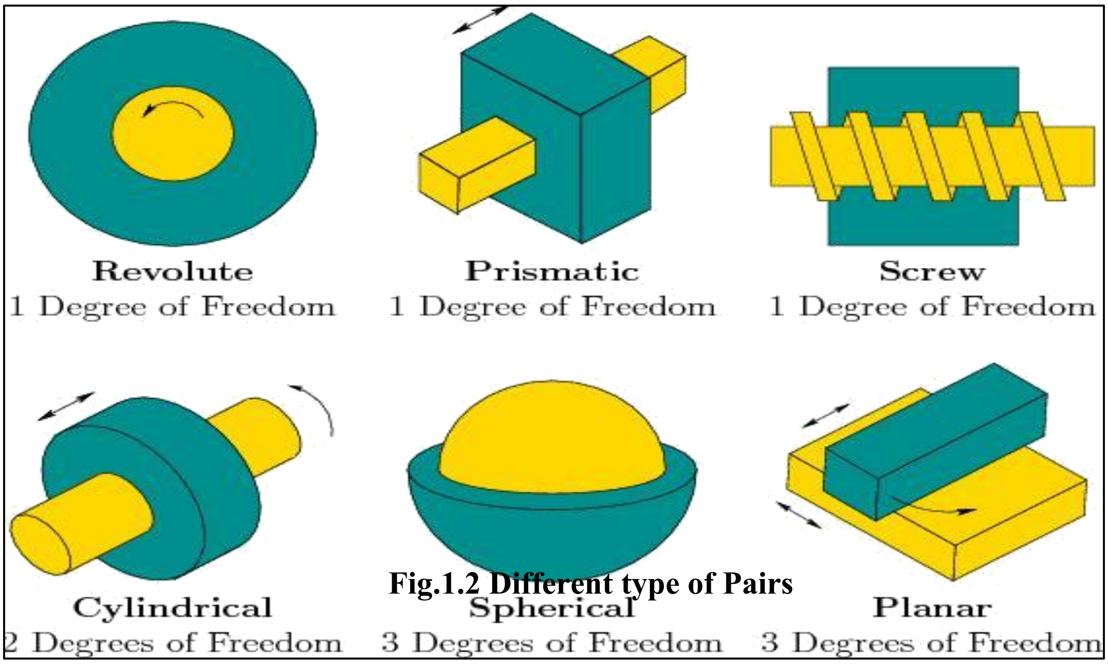


Fig.1.2 Different type of Pairs

Kinematics Chain:-

When the kinematics pairs are coupled in such a way that the last links is joined to the first link to transmit definite motion (i.e. completely or successfully constrained motion),it is called a kinematics chain. In other words, a kinematics chain may be defined as a combination of kinematics pairs, joined in such a way that each link forms a part of two pairs and the relative motion between the links or elements is completely or successfully constrained. For example, the crankshaft of an engine forms a kinematics pair with the bearing which are fixed in a pair, the connecting rod with the crank forms a second kinematics pair, the piston with the connecting rod forms a third pair and the piston with the cylinder forms a fourth pair. The total combination of these links is a kinematics chain.

If each link is assumed to form two pairs with two adjacent links, then the relation between the number of pairs

(p) forming a kinematic chain and the number of links (l) May be expressed in the form of an equation:

$$L = 2p - 4$$

Since in a kinematic chain each link forms a part of two pairs, therefore there will be as many links as the number of pairs.

Another relation between the number of links (L) and the number of joints (J) which constitute a kinematic chain is given by the expression:

$$J=\frac{3L-2}{2}$$

Mechanism: When one of the links of a kinematics chain is fixed, the chain is known as mechanism. It may be used for transmitting or transforming motion e.g. engine indicators, typewriter etc. A mechanism with four links is known as simple mechanism and the mechanism with more than four links as compound mechanism. When a mechanism is required to transmit power or to do some particular type of work, it then becomes a machine. In such cases, the various links or elements have to be designed to withstand the forces (both static and kinetic) safely. A little consideration will show that a mechanism may be regarded as a machine in which each part is reduced to the simplest form to transmit the required motion.

Four Bar Mechanism : A four bar link mechanism or linkage is the most fundamental of the plane kinematics linkages. It is a much preferred mechanical device for the mechanization and control of motion due to its simplicity and versatility. Basically it consists of four rigid links which are connected in the form of a quadrilateral by four pin joints. A link that makes complete revolutions is the crank, the link opposite to the fixed link is the coupler and the fourth link a lever or rocker if oscillates or an another crank, if rotate.

By fixing the link:

- Shortest Link Fixed
- Link opposite to Shortest Link fixed

I. Four Bar Chain Mechanism

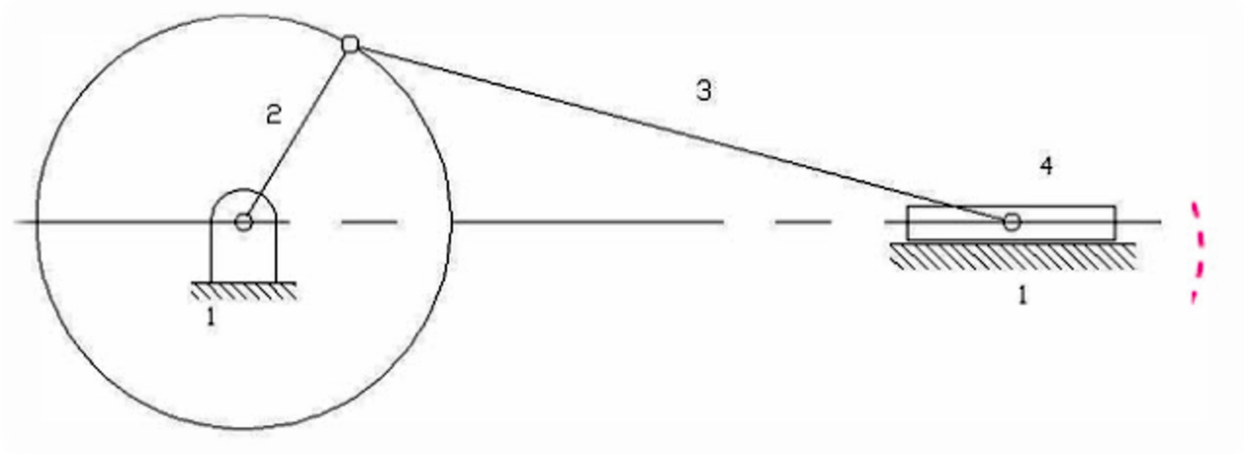


Fig.1.3 Slider Crank Mechanism

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.

II. 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: -

1. **First inversion (**i.e; Reciprocating engine and compressor) – this inversion is obtained when link 1 is fixed and links2 and 4 are made the crank and the slider respectively.

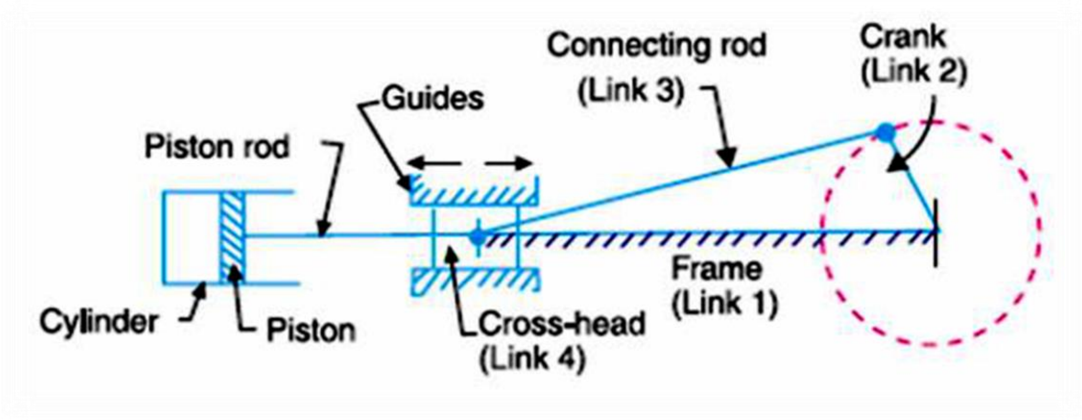


Fig.1.4 Reciprocating Engine

2. **Second inversion** (i.e., Whitworth quick return mechanism and Rotary engine) – fixing of link 2 of a slider – crank chain.

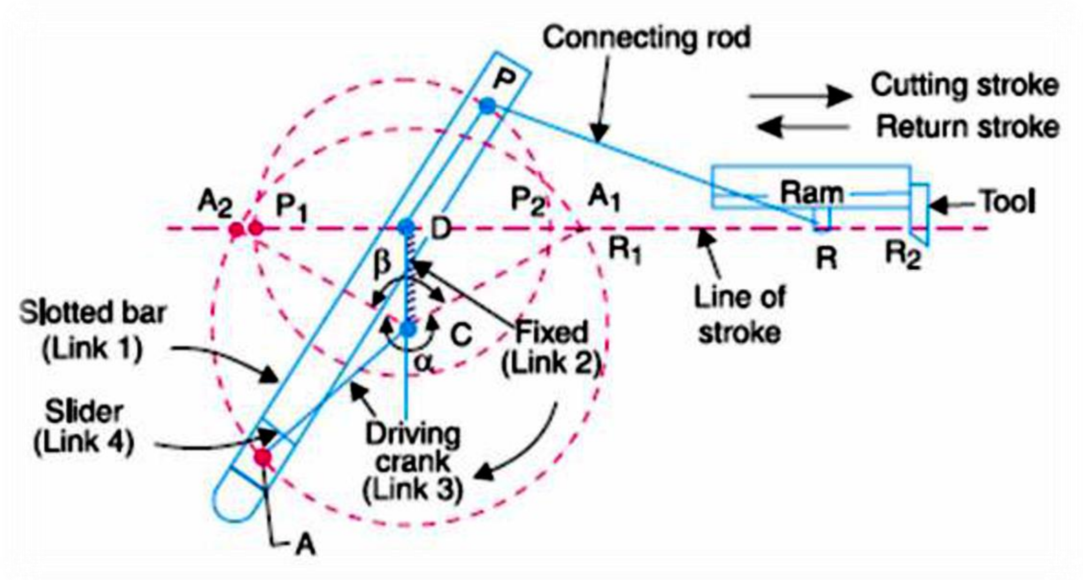


Fig.1.5 Whitworth quick return mechanism and Rotary engine

3. **Third inversion** (i.e., Oscillating cylinder engine and crank & slotted – lever mechanism)- By fixing link 3 of the slider crank mechanism.

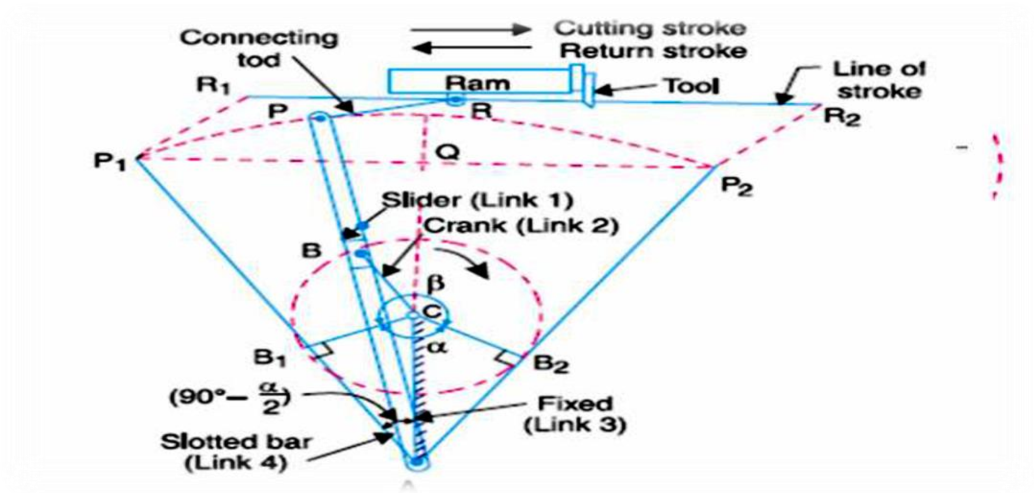


Fig.1.6 Oscillating cylinder engine and crank & slotted – lever mechanism

4. **Fourth inversion** (**Rotary engine mechanism or Gnome Engine**) – if link 4 of the slider crank mechanism is fixed, the fourth inversion is obtained.

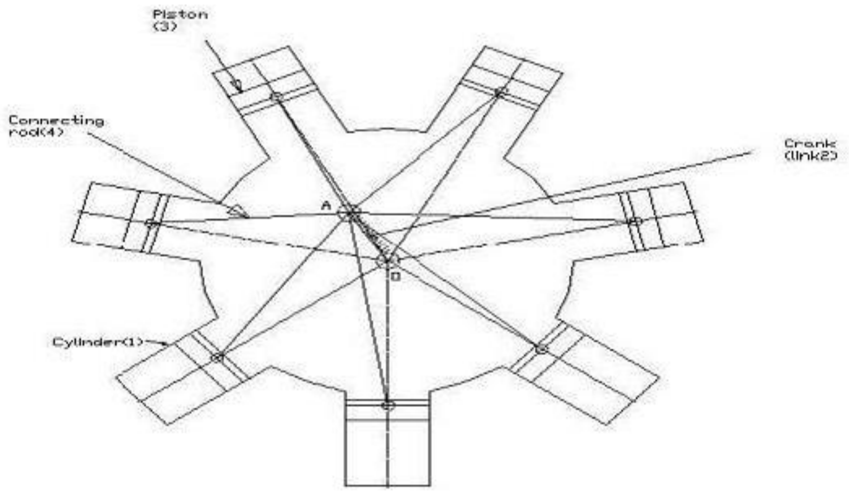


Fig.1.7 Gnome Engine

Rotary engine mechanism or gnome engine is another application of third inversion. It is a rotary cylinder V – type internal combustion engine used as an aero – engine

III. Double-Slider Crank-Chain:

A four-bar chain having two turning and two sliding pairs such that two pairs of the same kind are adjacent is known as a double-slider-crank chain. The following are its inversions:

1. **First inversion** (i.e., Elliptical trammel)

   This is an instrument for drawing ellipses. Here the slotted link is fixed. The sliding block P and Q in vertical and horizontal slots respectively. The end R generates an ellipse with the displacement of sliders P and Q.

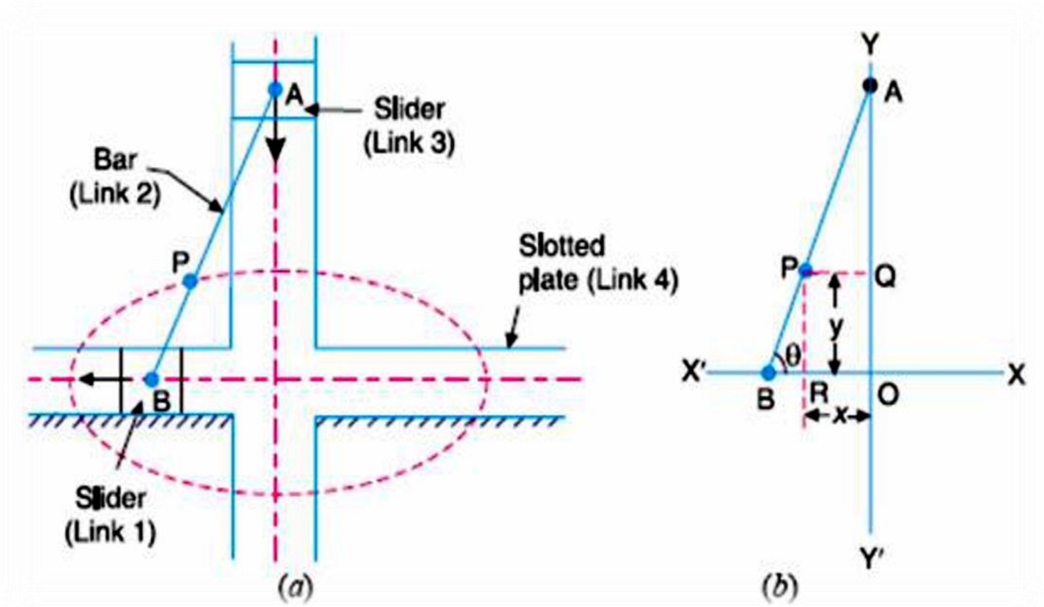


Fig.1.8 Elliptical trammel

2. **Second inversion** (i.e., Scotch yoke)

   This mechanism is used to convert rotary motion in to reciprocating motion. The inversion is obtained by fixing either the link 1 or link 3. Link I is fixed. In this mechanism when the link 2 rotates about B as centre, the link 4 reciprocates. The fixed link 1 guides the frame.

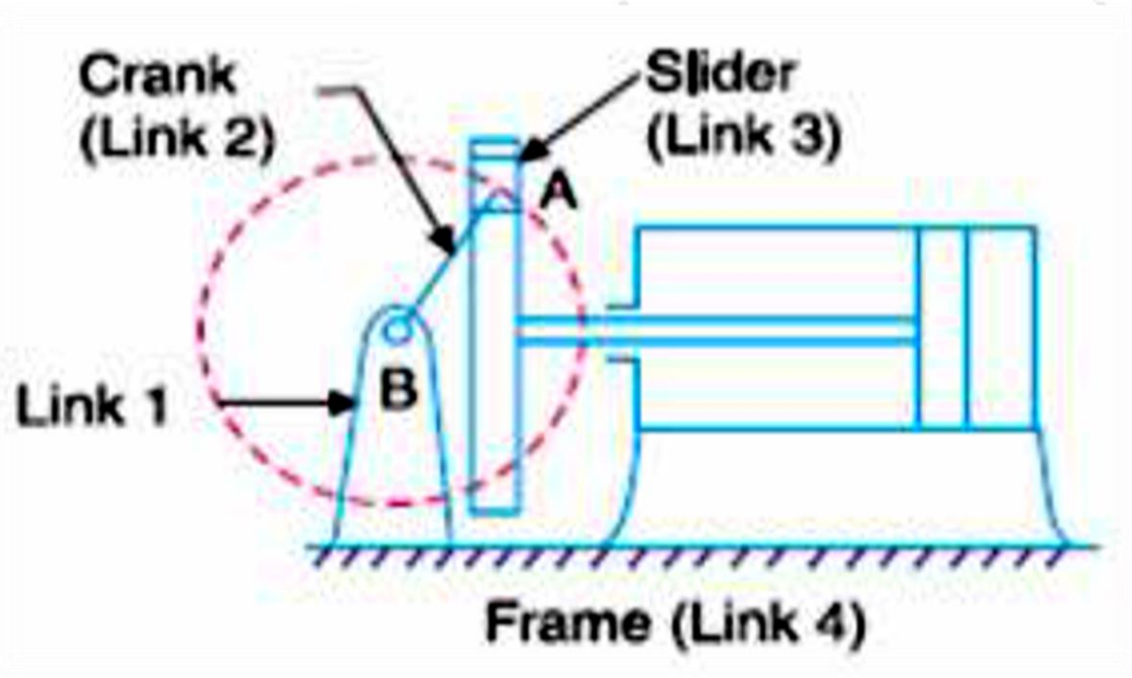


Fig.1.9 Scotch yoke

1. **Third inversion** (i.e., Actual Oldham's coupling)

   The third inversion of obtained by fixing the link connecting the 2 blocks P & Q. If one block is turning through an angle, the frame and the other block will also turn through the same angle. It is shown in the figure below.

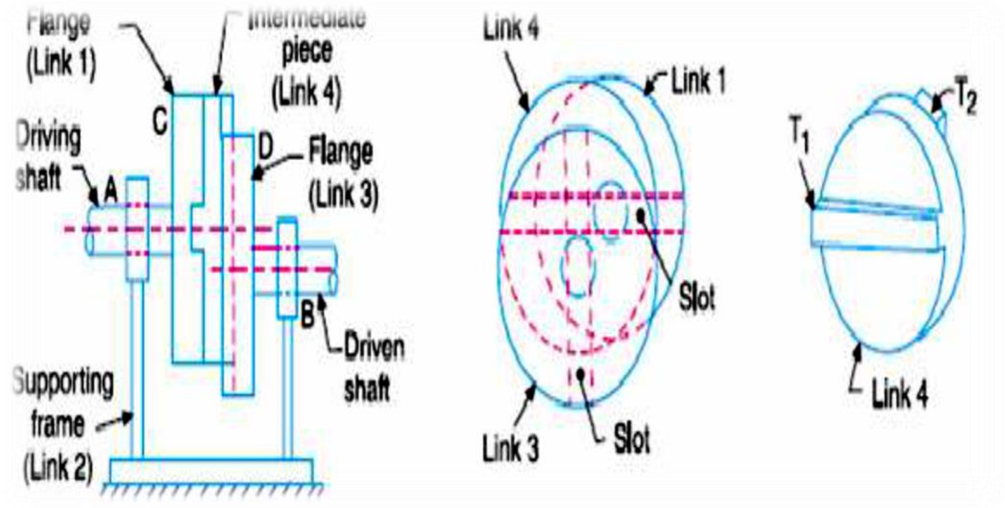


Fig.1.10 Oldham's coupling

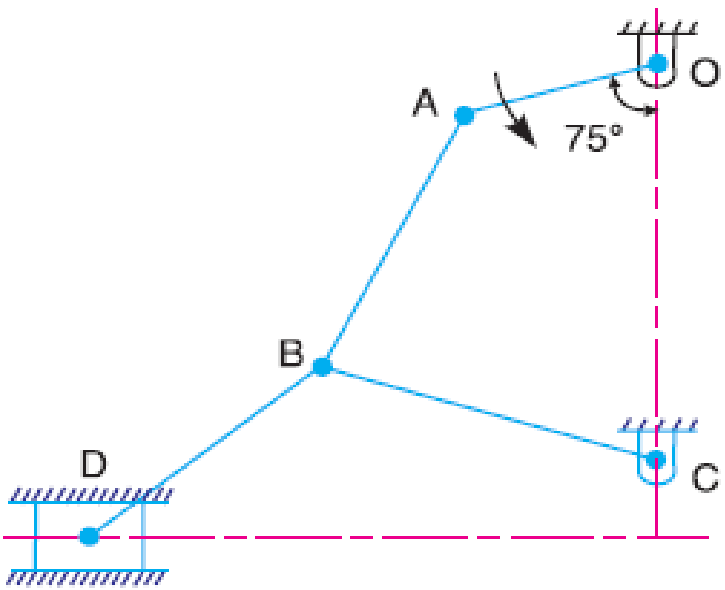
Viva-Questions:

1. Explain different kinds of kinematic pairs giving example for each one of them.
2. Explain the term kinematic link. Give the classification of kinematic link.
3. In what way a mechanism differ from a machine?
4. What is the significance of degrees of freedom of a kinematic chain when it functions as a mechanism? Give examples.

EXPERIMENT NO. 2

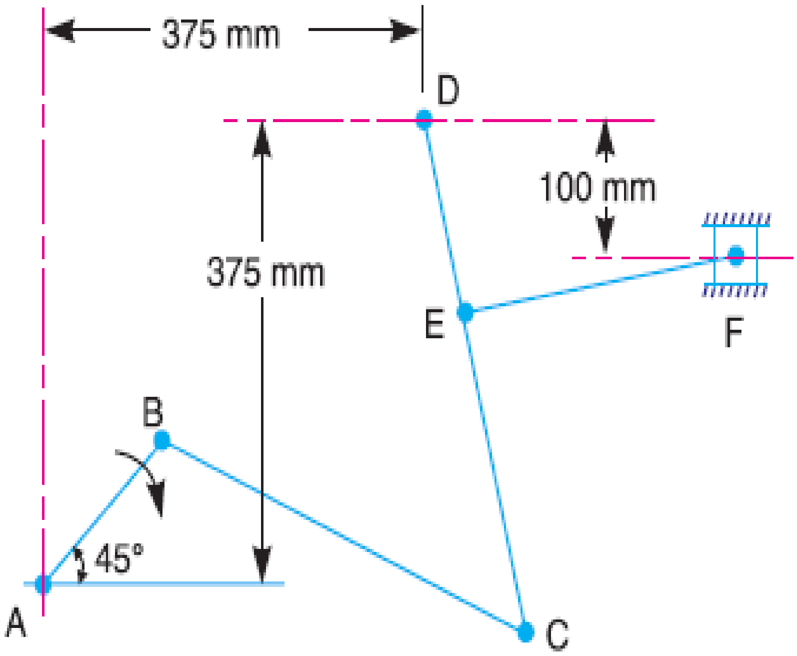
Aim: - Solution of two problems on velocity analysis by relative velocity method.

Q.1 In Fig. the angular velocity of the crank OA is 600 r.p.m. Determine the linear velocity of the slider D and the angular velocity of the link BD, when the crank is inclined at an angle of 75° to the vertical. The dimensions of various links are: OA = 28 mm; AB = 44 mm; BC 49 mm; and BD = 46 mm. The centre distance between the centres of rotation O and C is 65 mm. The path of travel of the slider is 11 mm below the fixed point C. The slider moves along a horizontal path and OC is vertical.



Q.2 The mechanism, as shown in Fig. 7.11, has the dimensions of various links as follows: AB = DE = 150 mm; BC = CD = 450 mm; EF = 375 mm. The crank AB makes an angle of 45° with the horizontal and rotates about A in the clockwise direction at a uniform speed of 120 r.p.m. The lever DC oscillates about the fixed point D, which is connected to AB by the coupler BC. The block F moves in the horizontal guides, being driven by the link EF. Determine: 1.

velocity of the block F, 2. angular velocity of DC, and 3. rubbing speed at the pin C which is 50 mm in diameter.



Viva-Questions:

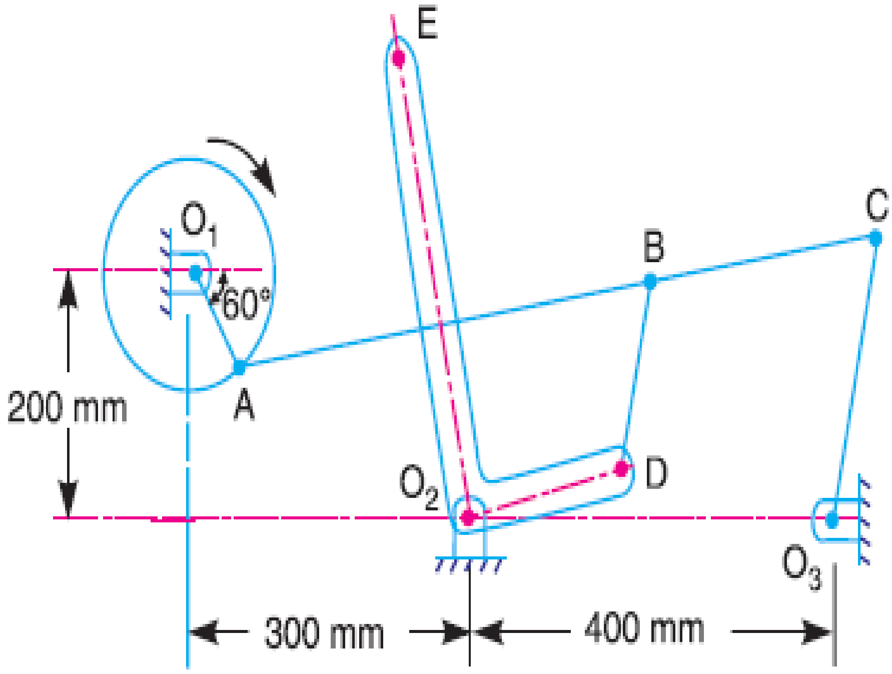
1. Describe the method to find the velocity of a point on a link whose direction (or path) is known and the velocity of some other point on the same link in magnitude and direction is given.

2. Explain how the velocities of a slider and the connecting rod are obtained in a slider crank mechanism.

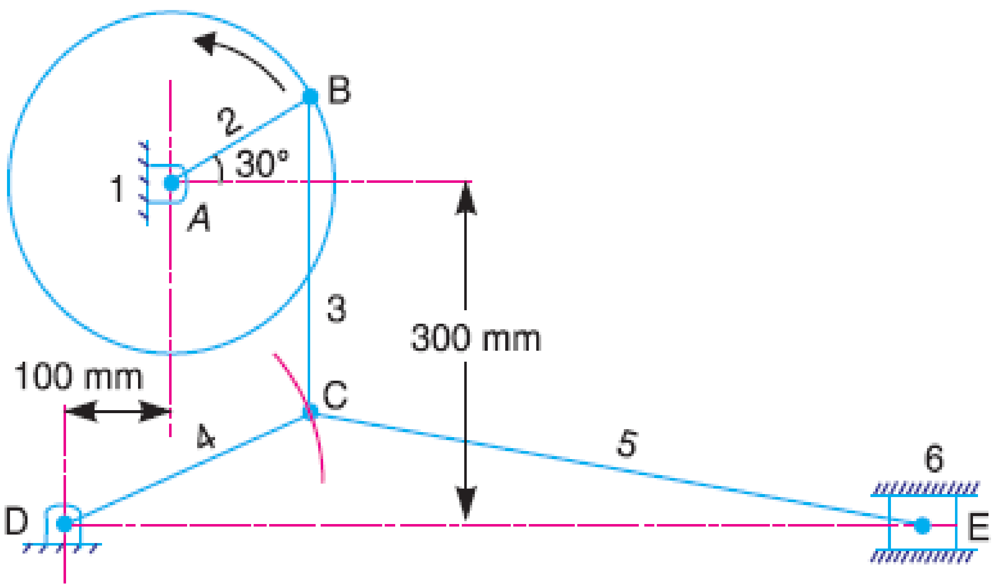
EXPERIMENT NO. 3

Aim: - Solution of two problems on velocity analysis by instantaneous center method.

Q.1 The mechanism of a wrapping machine, as shown in Fig. 6.18, has the following dimensions: O1A = 100 mm; AC = 700 mm; BC = 200 mm; O3C = 200 mm; O2E = 400 mm; O2D = 200 mm and BD = 150 mm. The crank O1A rotates at a uniform speed of 100 rad/s. Find the velocity of the point E of the bell crank lever by instantaneous centre method.



Q.2 Locate all the instantaneous centres of the mechanism as shown in Fig. The lengths of various links are : AB = 150 mm ; BC = 300 mm ; CD = 225 mm ; and CE = 500 mm. When the crank AB rotates in the anticlockwise direction at a uniform speed of 240 r.p.m.; find 1. Velocity of the slider E, and 2. Angular velocity of the links BC and CE.



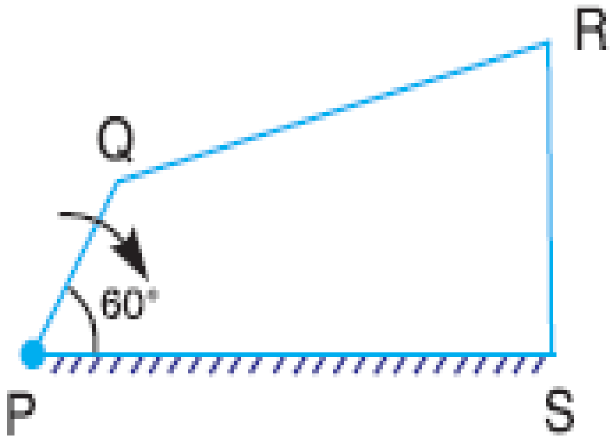
Viva-Questions:

1. How do you know the numbers of I- centers
2. State and Explain Kennedy's Theorem
3. Define rubbing velocity at a pin joint. What will be the rubbing velocity at pin joint when the two links? Move in the same and opposite directions?
4. Explain Space Centrode and Body centrode.

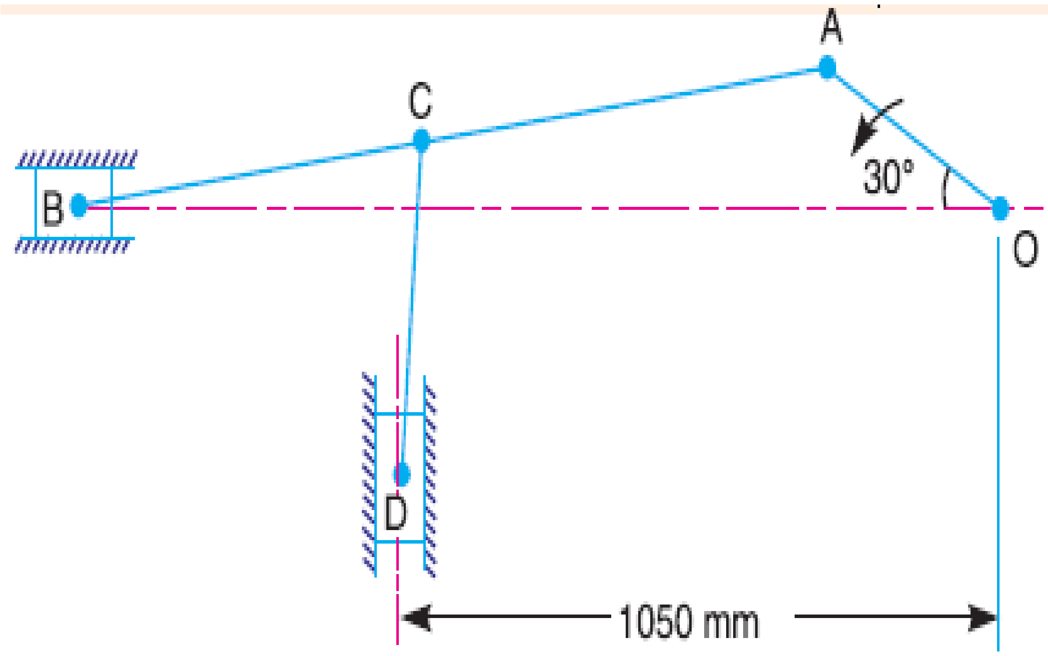
EXPERIMENT NO. 4

Aim: - Solution of two problems on acceleration analysis by relative velocity method.

Q. 1 PQRS is a four bar chain with link PS fixed. The lengths of the links are PQ = 62.5 mm; QR = 175 mm; RS = 112.5 mm; and PS = 200 mm. The crank PQ rotates at 10 rad/s clockwise. Draw the velocity and acceleration diagram when angle QPS = 60° and Q and R lie on the same side of PS. Find the angular velocity and angular acceleration of links QR and RS.



Q.2 In the mechanism, as shown in Fig. 8.12, the crank OA rotates at 20 r.p.m. anticlockwise and gives motion to the sliding blocks B and D. The dimensions of the various links are OA = 300 mm; AB = 1200 mm; BC = 450 mm and CD = 450 mm.



Viva-Questions:

1. What is Coriolis acceleration Component? In which Cases does it occur? How Is it determined
2. Explain the Procedure to Construct Klein's Construction to determine the velocity and acceleration of Slider – Crank Mechanism

EXPERIMENT NO. 5

Aim: - To determine the radius of gyration of a given bar by using bifilar suspension system.

Objectives:-

THEORY: The bifilar suspension is used to determine the moment of inertia of a body about an axis passing through its centre of gravity. The body is suspended by two parallel cords of length "L", at a distance "d" apart. If the mass of the body is "M", then the tension in either cord is Mg/2. If the system is now displaced through Q small angle θ at its central axis, then an angular displacement $\emptyset$ will be produced at the supports (see figure below).

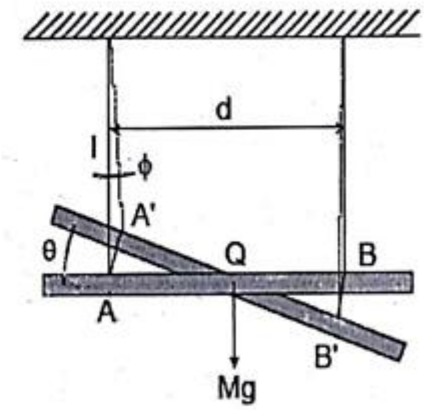


Fig.5.1 Bifilar suspension

Procedure:

Suspend the beam by wires and adjust it to some suitable length l. Measure the distance between the threads "d" accurately, before displacing the beam through some angle. Measure time for l0 oscillations, from which the periodic time can be calculated. Repeat the procedure three times. Change the length of the wires l and time a further 20 swings. The periodic times should be calculated for four such lengths.

Observation Table:

Sr. No	Length of the wire	Time for 20 Oscillations			Mean 't' sec	Tp = $t/20$
		tl	t2	t3		

CALCULATIONS:

1. Determination of $K_{Theoretical}$

$$K_{Theoretical} = L/2 * \sqrt{3}$$

2. Determination of $K_{Experimental}$

$$K_{Theoretical} = \frac{tp*q}{2*\pi} * \sqrt{g/l}$$

RESULT:

CONCLUSION:

Aim: To determine radius of gyration of a given disc using Trifilar suspension system

Description: Experimental setup consists of a uniform circular section disc, suspended from support by three chords. When the body is twisted about its axis through a small angle θ and then released, it will oscillate with simple harmonic motion.

Experimental Procedure:

1. Suspend the given disc from the rigid support; adjust length of each chord equal and convenient.
2. Note the suspension length of each chord.
3. Allow the disc to oscillate about vertical axis.
4. Measure time for 20 Oscillations.
5. Repeat the same procedure for varying length of chords.

Observation Table:

1. Diameter of the Disc =
2. Distance 'a' =

Sr. No	Length of the wire	Time for 20 Oscillations			Mean 't' sec	Tp = $t/20$
		tl	t2	t3		

Calculations:

1. Determination of $K_{Theoretical}$

$$K_{Theoretical} = D/4$$

2. Determination of $K_{Experimental}$

$$K = \frac{tp*q}{2*\pi} * \sqrt{g/l}$$

Result:

Conclusion:

Viva-Questions:

EXPERIMENT NO. 6

Aim: - Study of gyroscopic effect and finding moment of inertia of gyroscopic disc.

The spinning body exerts a torque or a couple in such a direction which tends to make the axis of spin coincides with that of the precession. To study the phenomenon of forced precession following procedure is adopted.

1. Balance initial horizontal position of rotor.
2. Start the motor and adjust the voltage to get the constant speed.
3. Press the yoke frame about the vertical axis by applying the necessary force by hand in the clockwise direction viewed from the top.
4. It will be observed that rotor frame swing about the horizontal axis so that the motor side moves upwards.
5. Rotating the yoke axis in the opposite direction causes the rotor frame to move in the opposite direction.

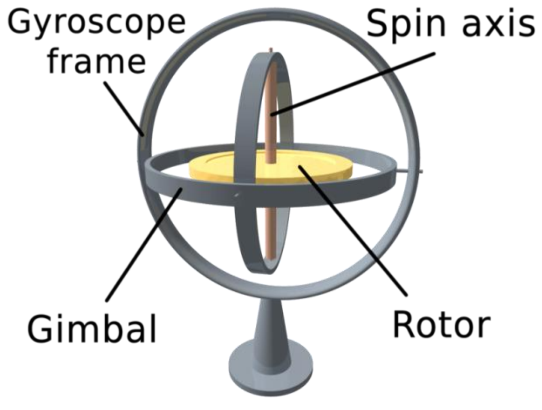


Fig.6.1 Gyroscope

The spinning body process in such a way that to make the axis of spin to coincide with that of the applied couple.

The direction is verified by following the procedure given below and using the apparatus as well as the relation for the magnitude of the couple.

1. Balance the rotor in the horizontal plane.

2. Start the motor and adjust the speed with the help of voltage regulation. The speed is measured using a tachometer.

3. Put weights on the side opposite to the motor.

4. The yoke start processing.

5. Note down the direction of precession.

6. Verify this direction.

7. Measures this direction.

8. Verify the relation C = $I \ X \ \omega \ X \omega p$

Observations:

1. Direction of spin axis: CLOCKWISE/ANTICLOCKWISE

2. Direction of forced precession: DOWNWORD

3. Direction of couple acting on the frame: CLOCKWISE/ANTICLOCKWISE

1. Mass of rotor (m) : 3 kg

2. Thickness of rotor : 19 mm

3. Rotor diameter : 220mm

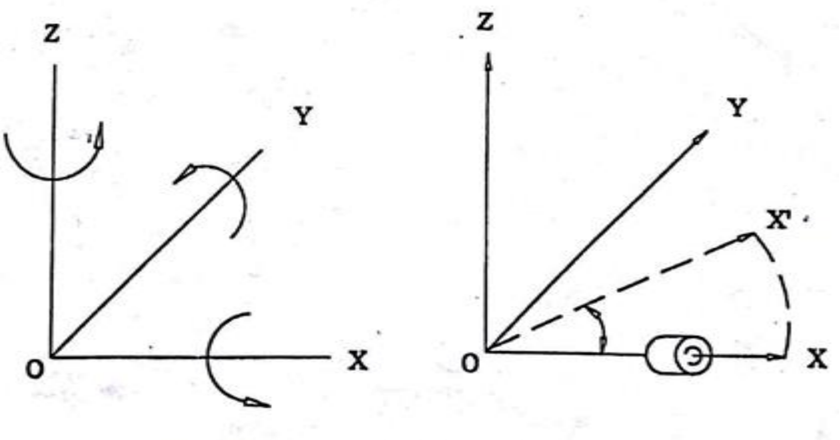
4. Moment arm (r') : 200mm

5. Motor power : 120w

6. Speed of motor : 0-200 rpm

Theory:

The angular velocity is a vector quantity and change in its magnitude can be caused by acceleration. To create this angular acceleration a torque or couple is required. To keep this angular velocity constant in magnitude due to the angular acceleration caused by the couple the spinning mass of the gyroscope undergoes a change called the angle of precision. This causes the gyroscope couple to incline to a certain degree so that it can be retain its angular velocity. This angle of precession for different torques and couple can be analyzed by this experiment.



OX - Axis of spin, OY - Axis of couple, OZ - Axis of precession

Observation Table:

Sr. No.	Weight (kg)	Spin Speed Spin, N (rpm)	Angle turned angle, θ (degrees)	Time, t (s)	Direction of rotation
1.					
2.					
3.					

Calculation:

1. Moment of Inertia (I) = $Kg/m2$

2. ω = 2Π N / 60 = rad/s

3. ωp = θ/ $(t\ X\ \frac{\Pi}{180})$ = rad/s

4. Gyroscopic Couple C = $I\ X\ \omega\ X\ \omega p$ = $N - m$

5. Applied Torque (T) = ωr = $N - m$

Result Table:

Sr. No.	Spin Velocity ω (rad/s)	Precessional velocity ωp	Gyroscopic Couple C $(N-m)$ $(rad/s$	Applied torque T $(N-m)$

Conclusions:

a) Comments are to be written based on the observations of direction observed.

b) The values tabulated in the result table are to be compared (i.e. the values of C & T are compared) and comments on the variation is to be written.

c) Different case where the gyroscopic couple is observed is to be mentioned.

Viva-Questions:

EXPERIMENT NO. 7

Aim: - Solution of two problems on Cams.

Q.1 A cam is to be designed for a knife edge follower with the following data: **1.** Cam lift = 40 mm during 90° of cam rotation with simple harmonic motion. **2.** Dwell for the next 30°. **3.** During the next 60° of cam rotation, the follower returns to its original position with simple harmonic motion. **4.** Dwell during the remaining 180°.

Draw the profile of the cam when

(a) The line of stroke of the follower passes through the axis of the cam shaft, and

(b) The line of stroke is offset 20 mm from the axis of the cam shaft.

The radius of the base circle of the cam is 40 mm. Determine the maximum velocity and Acceleration of the follower during its ascent and descent, if the cam rotates at 240 r.p.m.

Q.2 A cam, with a minimum radius of 25 mm, rotating clockwise at a uniform speed is to be designed to give a roller follower, at the end of a valve rod, motion described below:

1. To raise the valve through 50 mm during 120° rotation of the cam;
2. To keep the valve fully raised through next 30°;
3. To lower the valve during next 60°; and
4. To keep the valve closed during rest of the revolution i.e., 150°;

The diameter of the roller is 20 mm and the diameter of the cam shaft is 25 mm.

Draw the profile of the cam when (a) the line of stroke of the valve rod passes through the axis of the cam shaft, and (b) the line of the stroke is offset 15 mm from the axis of the cam shaft. The displacement of the valve, while being raised and lowered, is to take place with simple harmonic motion. Determine the maximum acceleration of the valve rod when the cam shaft rotates at 100 r.p.m. Draw the displacement, the velocity and the acceleration diagrams for one complete revolution of the cam.

Viva-Questions:

1. Explain with sketches the different types of cams and followers
2. State the Pitch curve of cam
3. Classify the Cam according to shape of follower
4. Explain with sketches the different types of cams and followers

EXPERIMENT NO. 8

Aim: To study the different types of brakes.

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

Theory: 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

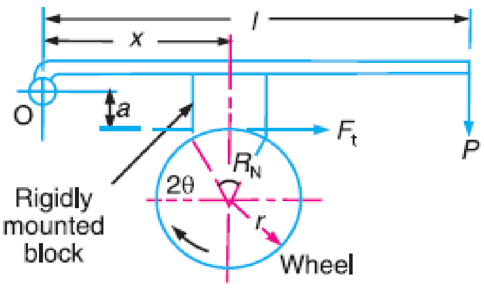
Fr = radial force applied on the drum

Rn = normal reaction on the block (= Fr)

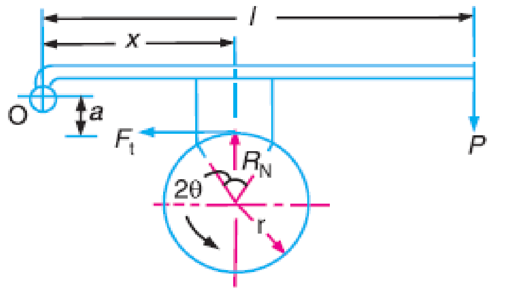
F = Force applied at the lever end

Ff = frictional force = μ Rn

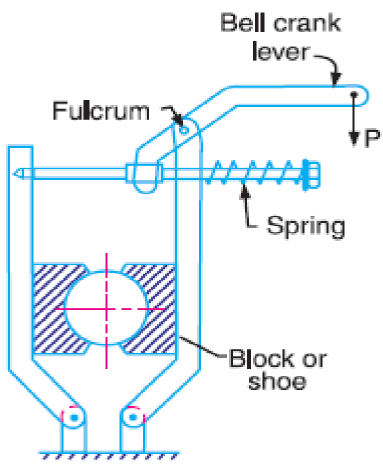
Assuming that the normal reaction Rn and the frictional force Ff act at the mid-point of the block.



(*a*) Clockwise rotation of brake wheel.



(*b*) Anticlockwise rotation of brake wheel.



Double block or shoe brake.

Fig.8.1 Different type of Brakes

Breaking torque on the drum ==frictional force × radius

Or TB = μ Rn × r

To obtain Rn, 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 the block it is in the same direction. Taking moments about the pivot O

F × a - Rn × b + μ Rn × c = 0

Rn = Fa / b - μc

Also $F = Rn \left\{\frac{(b+\mu)c}{a}\right\}$ When b = μc,

$F = 0,$

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, is known as a self-locking brake.

As the moment of the force F F, Ff aids in applying the brake. Such a brake is known as self If the rotation of the drum is reversed, i.e. it is made clockwise

$F = Rn \left\{\frac{(b+\mu)c}{a}\right\}$ This 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 = $Rn\ \frac{a}{b}$ Which is valid for clockwise as well as for counter-clockwise rotation.

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

$F = Rn \left\{\frac{(b+\mu)c}{a}\right\}$ For counter-Clockwise rotation

and

$F = Rn \left\{\frac{(b+\mu)c}{a}\right\}$ For clockwise rotation

In case the pivot is provided on the same side of applied force and the block as shown in figure 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 angle of contact is more than 40^{0}. The normal pressure is less at the ends than the center, In that case, μ given by

$$\mu' = \mu \left\{\frac{4\sin\frac{\theta}{2}}{\theta + Sin\theta}\right\}$$

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 = (T1 – T2)r Where r is the effective radius of the drum. The ratio of the tight and the slack side tensions is given by T1/T2 = eμθ 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. 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

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

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 -T1a + T2b = 0 Or F = As T1 > T2 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 Then 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 F is applied in the downward direction when a > b. 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 clockwise: clockwise rotation of the drum, the tight and the slack sides of the band will be as shown Considering the forces acting on the lever and taking moments about the pivot, T1a + T2b l and a > b under all conditions, the effectiveness of the brake will depend upon the force In this case, the right and the slack sides are reversed, i.e. T2 becomes greater th be tightened on the drum. This is possible if If the force applied is not as above, the band is further loosened on the drum, which means no braking clockwise rotation of the drum, the tight and the slack sides of the band will be as shown and a > b under all conditions, the effectiveness of the brake will depend upon the force becomes greater than T1. 17 T1 < T2 and a > b. The brake will be effective as long as T1a is greater than T2b. Or T2b. < T1a Or T2 < a T1 b i.e. as long as the ratio of T2 to T1 is less than the ratio a/b. When T2 ≥ a, F is zero or negative i.e. the brake becomes self-locking as no T1 b 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.

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

(iii) 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 - T1a = 0 Or F = T1 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.

(iv) 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 [(T2/T1) < (a/b)]. For case

(ii), when a < b and F is upwards, F required is less when the rotation is counter clockwise (assuming that [(T2/T1) < (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 - T1a - T2a F

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, Tension on the slack side. Tension on the tight side after one block. Tension on the tight side after two block Tension on the tight side after n blocks Coefficient of friction Normal reaction on the block. The forces on one block of the brake are shown in fig. 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.

The band brake may be lined with blocks of wood or other material, as shown in **Figure 8.2.** (*a*). The friction between the blocks and the drum provides braking action. Let there are '*n*' number of blocks, each subtending an angle 2θ at the centre and the drum rotates in anticlockwise direction.

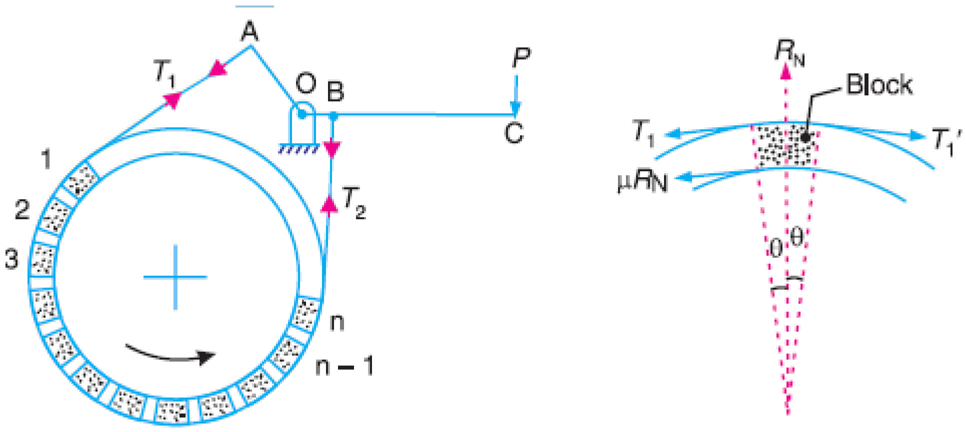


Fig.8.2 Band and Block Brake

Let $T1$ = Tension in the tight side,

$T2$ = Tension in the slack side,

μ = Coefficient of friction between the blocks and drum,

$T1'$ = Tension in the band between the first and second block,

$T2'$, $T3'$ etc. = Tensions in the band between the second and third block, between the third and fourth block etc. Consider one of the blocks (say first block) as shown in Figure, this is in equilibrium under the action of the following forces:

1. Tension in the tight side ($T1$),

2. Tension in the slack side ($T1'$) or tension in the band between the first and second block,

3. Normal reaction of the drum on the block (RN), and

4. The force of friction (μ.RN).

Resolving the forces radially, we have ($T1$ +$T1'$) sinθ=RN ... ***(i)***

Resolving the forces tangentially, we have
($T1$ +$T1'$) cos θ = μ.RN ... ***(ii)***

Dividing equation ***(ii)*** by ***(i)***, we have

$$\frac{(T_1 - T_1')\cos\theta}{(T_1 + T_1')\sin\theta} = \frac{\mu.R_N}{R_N}$$

or

$$(T_1 - T_1') = \mu\tan\theta(T_1 + T_1')$$

∴

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

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

$$\frac{T_1'}{T_2'} = \frac{T_2'}{T_3'} = \frac{T_3'}{T_4'} = \ldots\ldots\frac{T_{n-1}}{T_2} = \frac{1+\mu\tan\theta}{1-\mu\tan\theta}$$

$$\therefore \qquad \frac{T_1}{T_2} = \frac{T_1}{T_1'} \times \frac{T_1'}{T_2'} \times \frac{T_2'}{T_3'} \times \ldots\ldots\ldots \times \frac{T_{n-1}}{T_2} = \left(\frac{1+\mu\tan\theta}{1-\mu\tan\theta}\right)^n$$

Braking torque on the drum of effective radius r_e,

$$T_B = (T_1 - T_2)\, r_e$$
$$= (T_1 - T_2)\, r$$

... [Neglecting thickness of band]

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.

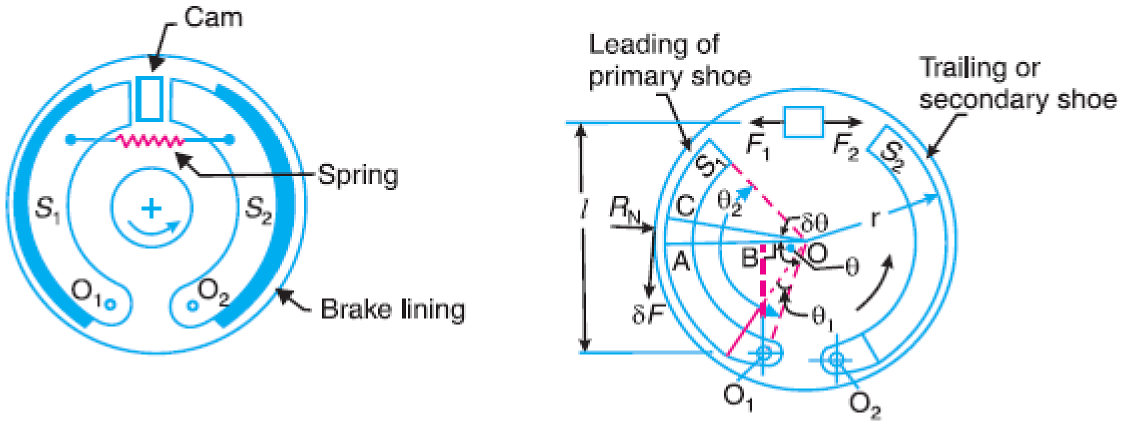


Fig.8.3 Internal Expanding Shoes Brake

Figure: Internal expending shoe brake Figure shows an internal shoe automobile brake 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 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, $\rho \propto l$

We shall now consider the forces acting on such a brake, when the drum rotates in the anticlockwise direction as shown in Fig. 19.25. It may be noted that for the anticlockwise direction, the left-hand shoe is known as *leading* or *primary shoe* while the right-hand shoe is known as *trailing* or *secondary shoe*.

Let r = Internal radius of the wheel rim,

b = Width of the brake lining,

p_1 = Maximum intensity of normal pressure,

p_N = Normal pressure,

F_1 = Force exerted by the cam on the leading shoe, and

F_2 = Force exerted by the cam on the trailing shoe. Consider a small element of the brake lining AC subtending an angle $\delta\theta$ at the centre. Let OA makes an angle θ with OO_1 as shown in Figure. It is assumed that the pressure distribution on the shoe is nearly uniform, however the friction lining wears out more at the free end. Since the shoe turns about O_1, therefore the rate of wear of the shoe lining at A will be proportional to the radial displacement of that point. The rate of wear of the shoe lining varies directly as the perpendicular distance from O_1 to

OA, *i.e.* *O*1*B*. From the geometry of the figure,

$$O_1B = OO_1 \sin\theta$$

and normal pressure at A,

$$p_N \propto \sin\theta \quad \text{or} \quad p_N = p_1 \sin\theta$$

∴ Normal force acting on the element,

$$\delta R_N = \text{Normal pressure} \times \text{Area of the element}$$

$$= p_N(b.r.\delta\theta) = p_1 \sin\theta(b.r.\delta\theta)$$

and braking or friction force on the element,

$$\delta F = \mu \times \delta R_N = \mu.p_1 \sin\theta(b.r.\delta\theta)$$

∴ Braking torque due to the element about O,

$$\delta T_B = \delta F \times r = \mu.p_1 \sin\theta(b.r.\delta\theta)r = \mu.p_1 b\, r^2(\sin\theta.\delta\theta)$$

and total braking torque about O for whole of one shoe,

$$T_B = \mu\, p_1 b\, r^2 \int_{\theta_1}^{\theta_2} \sin\theta\, d\theta = \mu\, p_1 b\, r^2 \left[-\cos\theta\right]_{\theta_1}^{\theta_2}$$

$$= \mu\, p_1 b r^2 (\cos\theta_1 - \cos\theta_2)$$

Moment of normal force δR_N of the element about the fulcrum O_1,

$$\delta M_N = \delta R_N \times O_1B = \delta R_N (OO_1 \sin\theta)$$

$$= p_1 \sin\theta(b.r.\delta\theta)(OO_1 \sin\theta) = p_1 \sin^2\theta(b.r.\delta\theta)OO_1$$

∴ Total moment of normal forces about the fulcrum O_1,

$$M_N = \int^{\theta_2} p_1 \sin^2\theta(b.r.\delta\theta)OO_1 = p_1.b.r.OO_1 \int^{\theta_2} \sin^2\theta\, d\theta$$

$$= p_1.b.r.OO_1 \int_{\theta_1}^{\theta_2} \frac{1}{2}(1-\cos 2\theta)d\theta \qquad \ldots \left[\because \sin^2\theta = \frac{1}{2}(1-\cos 2\theta)\right]$$

$$= \frac{1}{2} p_1.b.r.OO_1 \left[\theta - \frac{\sin 2\theta}{2}\right]_{\theta_1}^{\theta_2}$$

$$= \frac{1}{2} p_1.b.r.OO_1 \left[\theta_2 - \frac{\sin 2\theta_2}{2} - \theta_1 + \frac{\sin 2\theta_1}{2}\right]$$

$$= \frac{1}{2} p_1.b.r.OO_1 \left[(\theta_2 - \theta_1) + \frac{1}{2}(\sin 2\theta_1 - \sin 2\theta_2)\right]$$

Moment of frictional force δF about the fulcrum O_1,

$$\delta M_F = \delta F \times AB = \delta F(r - OO_1 \cos\theta) \qquad \ldots (\because AB = r - OO_1 \cos\theta)$$

$$= \mu\, p_1 \sin\theta(b.r.\delta\theta)\,(r - OO_1\cos\theta)$$

$$= \mu.p_1\, b.r(r\sin\theta - OO_1 \sin\theta\cos\theta)\delta\theta$$

$$= \mu.p_1.b.r\left(r\sin\theta - \frac{OO_1}{2}\sin 2\theta\right)\delta\theta \qquad \ldots (\because 2\sin\theta\cos\theta = \sin 2\theta)$$

$\therefore$ Total moment of frictional force about the fulcrum O_1,

$$M_F = \mu\, p_1 b\, r \int_{\theta_1}^{\theta_2} \left(r\sin\theta - \frac{OO_1}{2}\sin 2\theta\right) d\theta$$

$$= \mu\, p_1 b\, r \left[-r\cos\theta + \frac{OO_1}{4}\cos 2\theta\right]_{\theta_1}^{\theta_2}$$

$$= \mu\, p_1 b\, r \left[-r\cos\theta_2 + \frac{OO_1}{4}\cos 2\theta_2 + r\cos\theta_1 - \frac{OO_1}{4}\cos 2\theta_1\right]$$

$$= \mu\, p_1 b\, r \left[r(\cos\theta_1 - \cos\theta_2) + \frac{OO_1}{4}(\cos 2\theta_2 - \cos 2\theta_1)\right]$$

Now for leading shoe, taking moments about the fulcrum O_1,

$$F_1 \times l = M_N - M_F$$

and for trailing shoe, taking moments about the fulcrum O_2,

$$F_2 \times l = M_N + M_F$$

Note: If MF > MN, then the brake becomes self -locking.

Note that for the same applied force F on each shoe, Pl n is not equal to Pt n and Pl n > Pt 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

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.

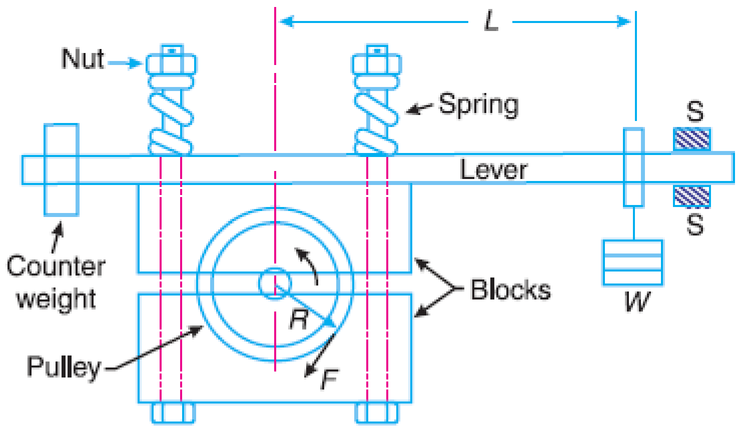


Fig.8.4 Prony Brake Dynamometer

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

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.

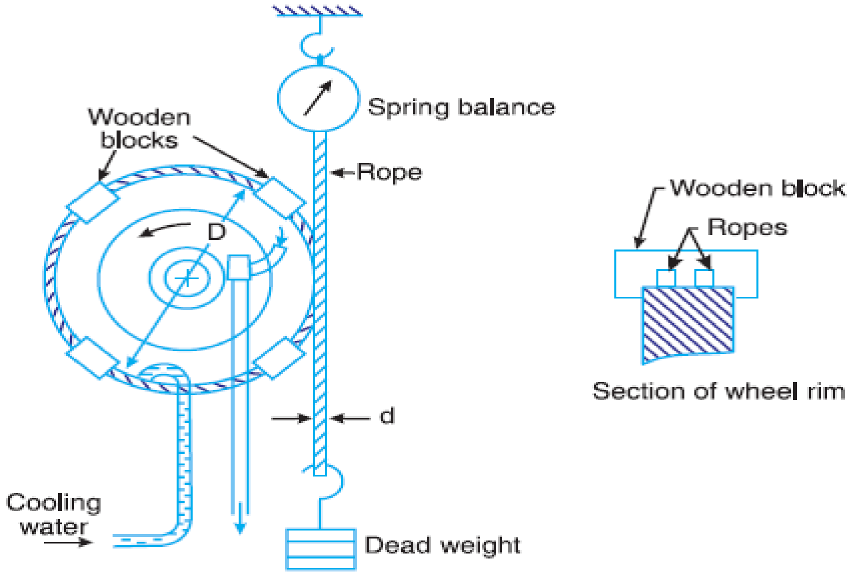


Fig.8.5 Rope Brake Dynamometer

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

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:

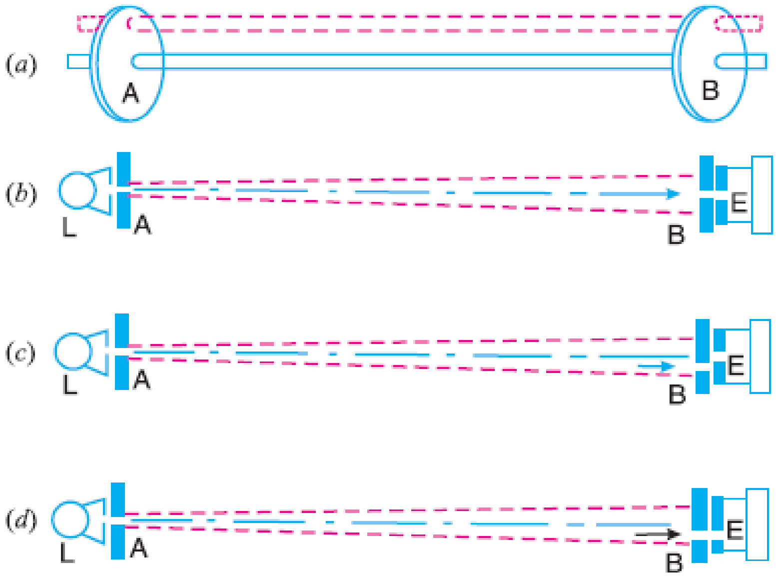


Fig.8.6 Bevis Gibson Flash Light Torsion Dynamometer

T = Ip* C*θ / l

Where Ip = πd4 /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) ii) For power measurement.

Viva-Questions:-

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

EXPERIMENT NO. 9

Aim: To study stability & sensitivity of governor

Apparatus: The porter governor is a modification of a Watt's governor, with central load attached to the sleeve. The load moves up down the central spindle. This additional downward force increases the speed of revolution required to enable the balls to rise to any to predetermined level.

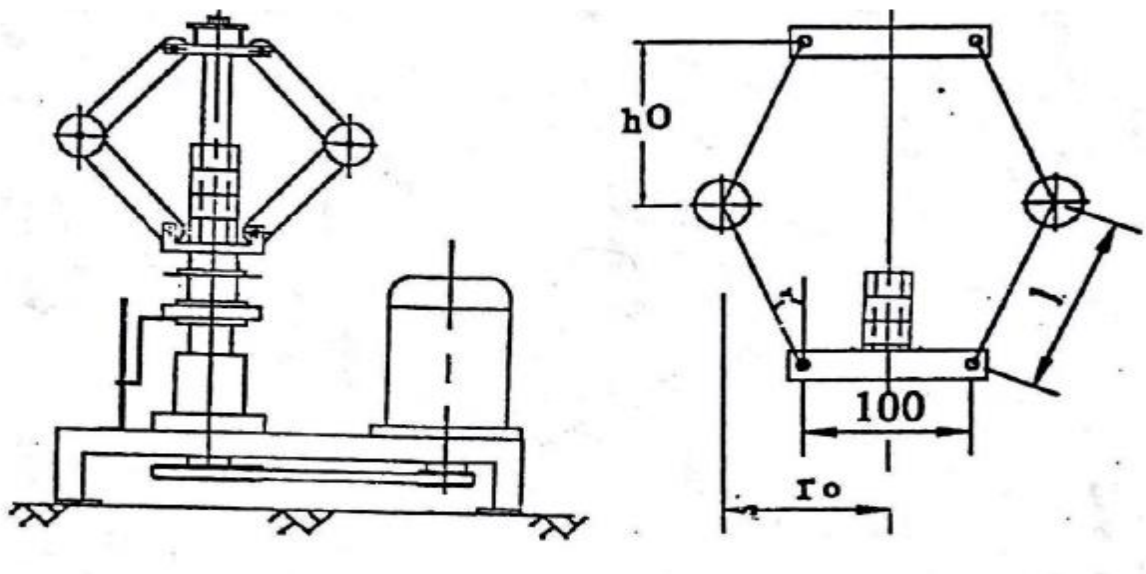
Length of each link (l) : 125 mm.

Initial height of governor (h_o) : 94 mm.

Initial radius of rotation (r_o) : 136 mm.

Weight of each ball (W) : 700 gms.

Procedure: Go on increasing the sped gradually and take the readings of speed of rotation 'N' and corresponding sleeve displacement 'X'. Radius rotation 'r' at any position could be found.



PORTER GOVERNOR

Observation Table:-

Sr. No.	Speed 'N' rpm	Sleeve Displacement 'X' in mm	Height 'h' in mm	Radius of rotation 'r' in mm	Centrifugal Force 'F' in kN
1.					
2.					
3.					
4.					
5.					

Calculations:

1. For height 'h', $h=h_o-(x/2)$

2. $\cos(\alpha) = h/l$

Result:

Conclusion:

Aim: To determine the position of sleeve against controlling force and speed of a Hartnell governor and to plot the characteristics curve of radius of rotation

Apparatus: Hartnell governor, scale, graph sheet.

Theory: The Hartnell governor is shown in figure. It consists of two bell crack levers hinged in the frame. The levers carry balls on the vertical arm and the spherical head contact point at the other end. These spherical head points press against the sleeve that compresses the spring form the bottom. The compression varies with the different position of the sleeve. The initial force in the spring is controlled by the nut F. The speed of rotation can be varied by the electric motor and the voltage regulator.

Vertical length of ball crank lever (a) = 77 mm.

Horizontal length of ball crank lever (b) = 122 mm.

Weights of ball (w) = 700 gms.

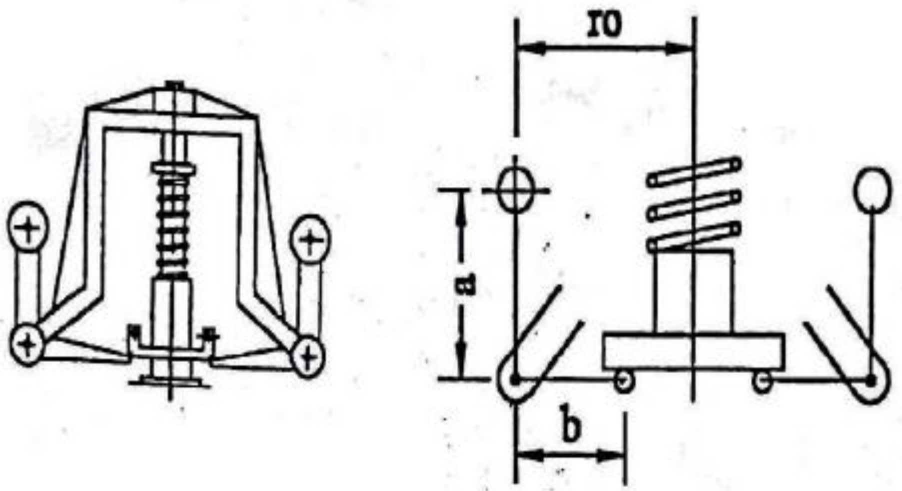
Initial radius of rotation (r_o) = 177.5 mm.

Free height of spring = 100 mm.

Procedure:

1) Measure initial compression of the spring
2) Go on increasing the speed gradually and take the readings of speed of rotation 'N' and corresponding sleeve displacement 'x'. Radius of rotation 'r' at any position could be found.
3) Following graphs can be plotted to study governor characteristics,
   a) Force Vs Radius of rotation

b) Speed Vs Sleeve displacement



Hartnell Governor

Observation Table:-

Sr. No.	Speed 'N' rpm	Sleeve Displacement 'X' in mm	Radius of rotation 'r' in mm	Centrifugal Force 'F' in KN
1.				
2.				
3.				
4.				
5				

Calculations:

1. Radius of rotation 'r' $= ro + X\ (a/b)$

2. Angular speed 'ω' $= 2 * \pi * N/60$

3. Controlling force 'F' $= (W/g) * \omega^2 r$

Where, W = weight of ball in kg and g = 981 cm/s^2

Result:

Conclusion:

Viva-Questions:

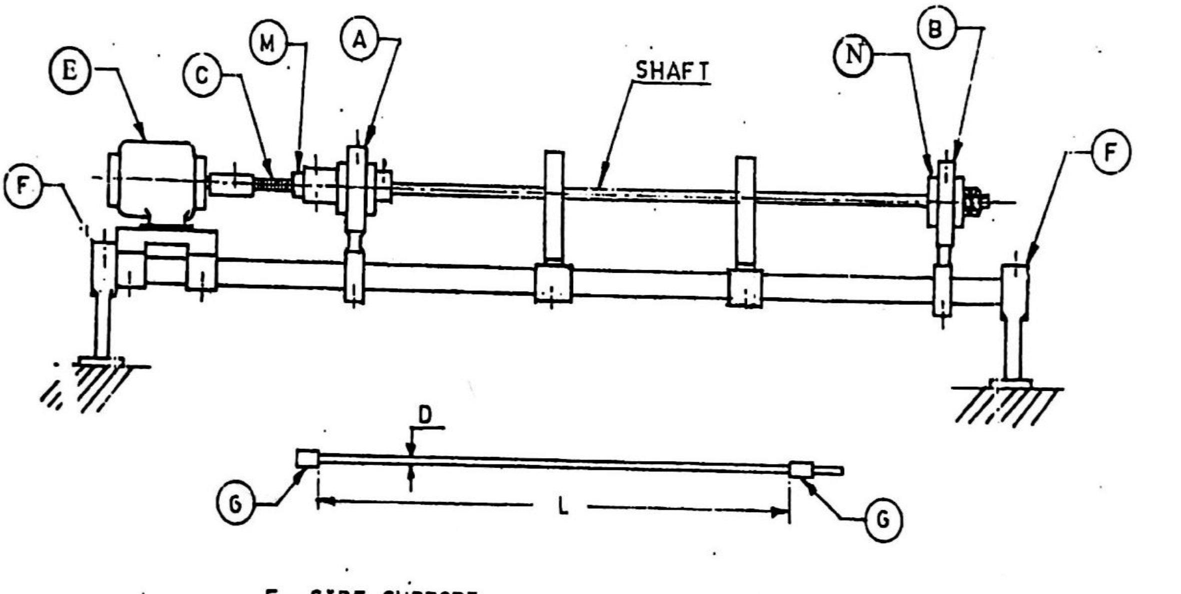
1. Define the following terms

a) Governor b) Sensitiveness of governor c) Isochronism of governor d) Hunting of governor e) Height of governor f) Sleeve Lift g) Maximum and Minimum Equilibrium Speed h) Watt's governor

2. What is function of flywheel and how does it differ from governor

Aim: - To study Whirling of shaft.

Description: This apparatus is developed for the study of whirling phenomenon. The shaft can be tasted for different end condition. The apparatus consists of frame to support its driving motor and fixing and sliding blacks etc. a special design is provided to clear out the effect of bearing of motor spindle from those of testing shafts. The special design features of this equipment are shown in the figure below.



F - SIDE SUPPORT
G - BUSHES AT THE SHAFT ENDS
C - FLEXIBLE SHAFT
MN - BELT BEARING SHAFT

AB - STAND FOR FIXING
M - MOTOR SIDE BEARING BLOCK
N - TAIL SIDE BEARING BLOCK
E - MOTOR

WHIRLING OF SHAFTS APPARATUS

1 **Kinematic Coupling:**

This coupling is specifically designed to eliminate the effects of motor spindle bearing on those of rotating shafts.

2 **Ball Bearing Fix Ends:**

These ends fix the shafts while it rotates. The shaft can be replaced within a short with the help of this unit. The fixing ends provide change of end fixing conditions of the rotating shafts as per the requirement.

Diameter of the shaft

Length of the shaft

End fixing arrangement

Precaution to Be Observed In Experiment:

1. If the revolution at unloaded shafts are gradually increased it will be found that a certain speed will be found that a certain speed will be reached at which the violent instability will occur, the shaft deflecting in a single bow and whirling round like a skipping rope. If the speed is maintained the deflection will become so large that the shaft will be fractured, but if the speed is quickly run through the shaft will become straight again and run true until at another higher speed the same phenomenon will occur, the deflection now however being in a double bow and so on. Such speeds are called as critical speed of whirling.
2. It is advisable to increase the speed of the shaft rapidly and pass through the critical speed first rather than observing the first critical speed which increases 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. If however the shaft speed is taken to maximum first and the slowly reduce higher ends will be observed first and the corresponding speed noted and the by reducing the speed further the next mode

of tower frequency can be observed without any danger of rise in amplitude as speed is being decreased and the inertial forces are smaller in comparison with the bending spring forces hence possibility of buildup dangerous amplitude at response or near response.

3. Thus it can be seen that it is a destructive test shaft and it is observed that the elastic behavior of the material changes a little after testing it for a few time it is advisable therefore, to use fresh shaft sample afterwards.
4. Fix the apparatus firmly on suitable foundation.

Observations:

Result:

Conclusion:

Viva-Questions:

EXPERIMENT NO. 11

Aim: - Solution of two problems on balancing.

Q.1 Four masses m1, m2, m3 and m4 are 200 kg, 300 kg, 240 kg and 260 kg respectively. The corresponding radii of rotation are 0.2 m, 0.15 m, 0.25 m and 0.3 m respectively and the angles between successive masses are 45°, 75° and 135°. Find the position and magnitude of the balance mass required, if its radius of rotation is 0.2 m.

Q.2 An inside cylinder locomotive has its cylinder centre lines 0.7 m apart and has a stroke of 0.6 m. The rotating masses per cylinder are equivalent to 150 kg at the crank pin, and the reciprocating masses per cylinder to 180 kg. The wheel centre lines are 1.5 m apart. The cranks are at right angles. The whole of the rotating and 2/3 of the reciprocating masses are to be balanced by masses placed at a radius of 0.6 m. Find the magnitude and direction of the balancing masses. Find the fluctuation in rail pressure under one wheel, variation of tractive effort and the magnitude of swaying couple at a crank speed of 300 r.p.m.

Viva-Questions:

1. Differentiate between static and dynamic balancing, Primary and secondary balancing.
2. . Explain Hammer Blow
3. Explain reciprocating engine Partially balanced
4. What do you mean by primary & secondary unbalance in reciprocating engines?

EXPERIMENT NO. 12

Aim: To perform the experiment for static balancing on static balancing machine.

Apparatus Used: Static Balancing m/c.

Theory: A system of rotating masses is said to be static balance if the combined mass centre of the system lies on the axis of rotation. Whenever a certain mass is attached to a rotating shaft. It exerts some centrifugal force. Whose effect is to bend the shaft and a produce vibration in it. In order to prevent the effect of centrifugal force. Another mass is attached to the opposite side of the shaft. The process of providing the second mass in order to counteract the effect of the centrifugal force of the first mass, is called balancing of rotating masses.

The following cases are important from the subject point of view:

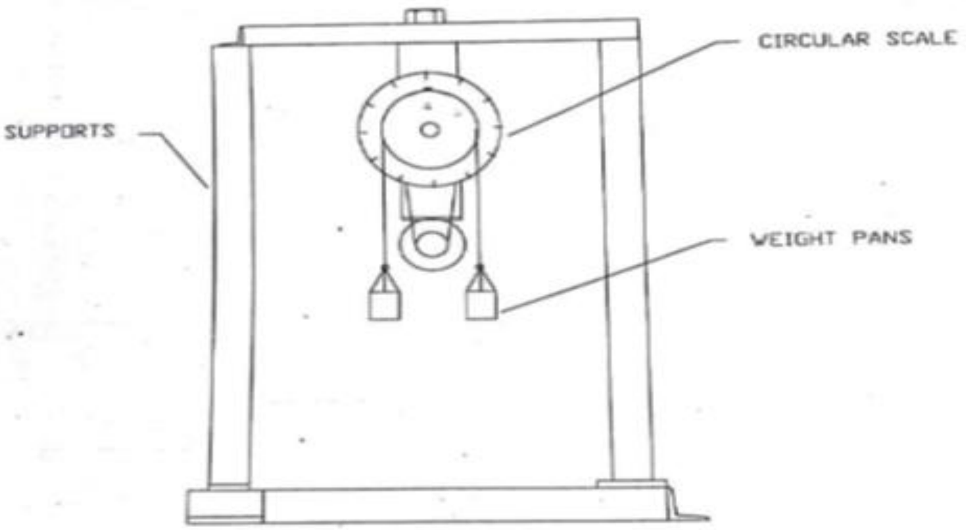
1. Balancing of a single rotating mass by a single mass rotating in the same plane.
2. Balancing of a single rotating mass by two masses rotating in different planes.
3. Balancing of different masses rotating in the same plane.
4. Balancing of different masses rotating in different planes.

Procedure: - Remove the belt. The value of weight for each block is determined by clamping each block in turn on the shaft and with the cord and container system suspended over the protractor disc. The number of steel balls. Which are of equal weight are placed into one of the containers to exactly balance the blocks on the shaft. When the block becomes horizontal. The number of balls N will give value of wt. for the block.

For finding out W, during static balancing proceed as follow:

1. Remove the belt.

2. Screw the combined hook to the pulley with groove. This pulley is diff. than the belt pulley.
3. Attached the cord end of the pans to above combined hook.
4. Attached the block no. 1 to the shaft at any convenient position and in vertical downward direction.
5. Put steel balls in one of the pans till the blocks starts moving up. (upto horizontal position).
6. Number of balls gives the W value of block-1. Repeat this for 2-3 times and find the average no. of balls.
7. Repeat the procedure for other blocks.



STATIC & DYNAMIC BALANCING MACHINE

Set-up for statically balancing of rotory masses

Observation Table:

Sr.no	Plane	Mass (m)kg.	Radius ® m	Cent. force $\div\omega^2$ (m.r) kg-m	Distance From plane x (1)m	Couple $\div\omega^2$ (m.r.l) kg-m^2

Calculation: - The balancing masses and angular positions may be determined graphically as given below:

1. First of all, draw the couple polygon from the data which are calculated in table to same suitable scale. The vector distance represents the balanced couple. The angular position of the balancing mass is obtained by drawing. Parallel to vector distance. By measurement will be find the angle.
2. Then draw the force polygon from the data. Which are calculated in table to some suitable scale. The vector distance represents the balanced force. The angular position of the mass is obtained by drawing. Parallel to vector distance. By measurement will be find the angle in the clockwise direction from mass.

Precautions:

1. Couple should be represented by a vector drawn perpendicular to the plane of the couple.
2. Angular position measure carefully in clockwise direction.

3. Vector diagram should be represented with suitable scale.

Aim: - To perform the experiment for dynamic balancing on dynamic balancing machine.

Apparatus used: - Dynamic balancing m/c.

THEORY: - When several messes rotate in different planes, the centrifugal force. In addition to being out of balance, also form couples. A system of rotating masses is in dynamic balance when there does not exit any resultant centrifugal force as well as resultant couple.

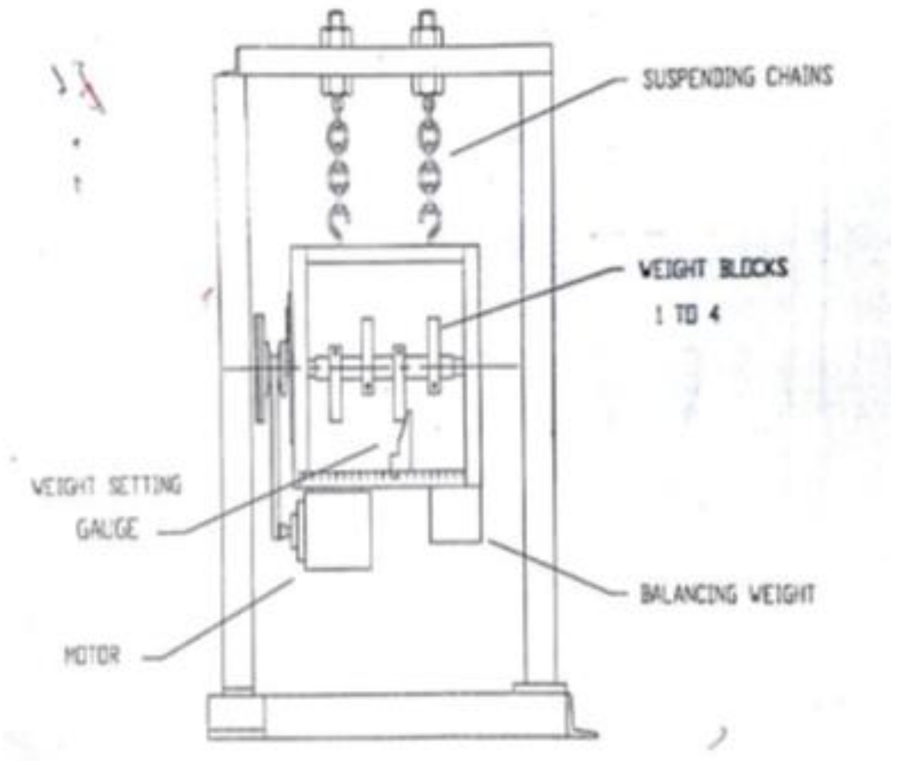
Pivoted-cradle balancing M/C:-

In this type of m/c., the rotor to be balanced is mounted on half-bearing in a rigid carriage and is rotated by a drive motor through a universal joint. Two balancing planes A and B are chosen on the rotor. The cradle is provided with pivots on left and right sides of the rotor which are purposely adjusted to coincide with the two correction planes. Also the pivots can be put in the locked or unlocked position. Thus, if the left pivot is released, the cradle and the specimen are free to oscillate about the locked (right) pivot. At each end of the cradle, adjustable springs and dashpots are provided to have a single degree of freedom system. Usually, their natural frequency is tuned to the motor speed.

Procedure:-

1. First either of the two pivots say left locked so that the readings of the angle of location of the correction in the locked plane as it will have no moment about the fixed pivot.
2. A trial mass at a known radius is then attached to the right hand plane and the amplitude of oscillation of the cradle is noted.
3. magnitude of the trial mass is varied and the exact amount is found by trial and error which reduces the unbalance to almost zero.

4. After obtaining the unbalance in one plane, the cradle is locked in the right hand pivot and released in the left hand pivot. The above procedure is repeated to obtain the exact balancing mass required in that plane.
5. Usually, a large number of test runs are required to determine the exact balance masses in this type



STATIC & DYNAMIC BALANCING MACHINE
Set-up for dynamically balancing of rotary masses

Calculation & Construction:-

Draw a triangle OBE by taking OE= 2 X_1, OB= 2 X_2 and BE= X_4 Mark the mid-point A on OE join AB.

Now, OB= OA + AB

Where, OB= Effect of unbalance mass + effect of the trial mass at 0^0

Thus, AB represents the effect of the attached mass at 0^0. The proof is as follows:

Extend BA to D such that AD=AB. Join OD and DE. Now when the mass m is attached at 180^0 at the same radial distance and speed, the effect must be equal and opposite to the effect at 0^0 i. e. if AB represents the effect of the attached mass at 0^0. AD represents the effect of the attached mass at $180^{0.}$

Since, OD= OA + AD

OD must represent the combined effect of unbalance mass and the effect of the trial mass at 180^0 (X_4).

Now, as the diagonals of the quadrilateral OBED bisect each other at A, it is a parallelogram which means BE is parallel and equal to OD. Thus, BE also represents the combined effect of

Now as OA represents the unbalance, the correction has to be equal and opposite of it or AO. Thus, the correction mass is given by

M_c = m. OA/AB at an angle ɵ from the second reading at 0^0.

For the correction of the unbalance, the mass m_c has to be put in the proper direction relative to AB which may be found by considering the reading X_3.

Draw a circle with A as centre and AB as the radius. As the trial mass as well as the speed of the test run at 90^0 is the same. The magnitude must equal to AB or AD and AC or AC must represent the effect of the trial mass. If OC represents X_3 then angle is opposite to the direction of angle measurement. If OC represents X_3 then angle measurement is in taken in the same direction.

Precaution:-

4. Measure the amplitude carefully.

4.Draw the triangle and parallelogram in correct scale.

4. Vector diagram should be representing with suitable scale.

Conclusion:-

Viva-Questions:

1. Analytical Method for Balancing of Rotating masses.
2. Deduce Expression for Swaying Couple
3. Deduce the Expression for tractive force