DEPARTMENT OF
MECHANICAL ENGINEERING

LABORATORY MANUAL

THEORY OF MACHINES

(CSMSS/ENGG/MECH/TOMLAB)

Mechanical Engineering Lab – II Group B (BTMCL406)

Mechanical Engineering Lab – III Group B (BTMCL507)

CSMSS
CHH. SHAHU COLLEGE OF ENGINEERING

Vision and Mission of the Institute

Vision:

To be an institution of repute through multidisciplinary educational approach to develop the next generation competent technocrats for industry and society.

Mission:

M1:	Developing student centric educational practices for curriculum delivery and assessment.
M2:	Imparting entrepreneurial and employability skills among students through value-based and skill-based training in collaboration with industry and academia.
M3:	Inculcating social and professional values among students through awareness and outreach activities.
M4:	Providing an environment for innovation and research through various interdisciplinary activities.

CSMSS CHH. SHAHU COLLEGE OF ENGINEERING DTE CODE 2533 AURANGABAD

PRINCIPAL

Principal
C.S.M.S.S. Chh. Shahu College of Engineering
Kanchanwadi, Aurangabad.

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Department of Mechanical Engineering

Vision and Mission of the Department

Vision

To Be a Centre of Repute for Preparing Engineering Students as Professionals in the Services, Domain Leadership, Research, and Good Citizens.

Mission

M1: Developing the continuous improvement process for academic practices to strengthen the academic base of students.

M2: Developing the research culture through various efforts that can inspire students for research.

M3: Developing the students for social skills like team building, leadership and social values.

M4: Upgrading the above activities through communication with internal and external stakeholders (Students, Parents, Alumni, Employers, Experts, Society people, etc.).

Dr. R. H. Shinde,
HOD,
Mechanical Engg. Department

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Department of Mechanical Engineering

Program Outcomes

PO	PROGRAM OUTCOMES (POs)
PO 1	**Engineering knowledge:** Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems.
PO 2	**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.
PO 3	**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.
PO 4	**Conduct investigations of complex problems:** Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.
PO 5	**Modern tool usage:** Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modeling to complex engineering activities with an understanding of the limitations.
PO 6	**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.
PO 7	**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.
PO 8	**Ethics:** Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.
PO 9	**Individual and team work:** Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.
PO 10	**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.
PO 11	**Project management and Finance:** Demonstrate knowledge and understanding of the engineering and management principles and apply these to ones own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.
PO 12	**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.

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Department of Mechanical Engineering

Program Specific Outcomes (PSOs)

Graduates will be able to,

PSO 1: Interpret, analyze, and provide solutions to real-life mechanical engineering and interdisciplinary problems for realistic outcomes.

PSO 2: Apply the knowledge of ethical and management principles required to work in a team as well as to lead a team.

PSO 3: Respond to the demand of society by engaging in a lifelong learning approach.

Program Educational Objectives (PEOs)

Graduates will,

PEO 1: Achieve professional skills by applying their mechanical engineering expertise and communication in the field of industries or their own start-ups.

PEO 2: Achieve higher qualifications/skills through further studies.

PEO 3: Become responsible citizens by contributing significant roles among society.

Dr. R. H. Shinde,
HOD,
Mechanical Engg. Department

Course Name: MECHANICAL ENGINEERING LAB - II Group B THEORY OF MACHINES LAB -I

MECHANICAL ENGINEERING LAB - III Group B THEORY OF MACHINES LAB -II

Course Code: BTMCL406 & BTMCL507

Examination Scheme:

1. **Internal Assessment:** 60 Marks (CA1 = 30 Marks and CA2 = 30 Marks)
2. **External Assessment:** 40 Marks

Course Outcomes:

After successfully completion the course, the students will be able to:		**Blooms Level**
CO1	Demonstrate the ability to apply theoretical knowledge to practical scenarios through hands-on experimentation	Apply
CO2	execute well-structured experiments, including selecting appropriate methodologies, equipment, and controls to achieve reliable and reproducible results.	Apply
CO3	Analyze experimental data, identify sources of error, and propose improvements to procedures, fostering critical thinking and troubleshooting skills.	Analyze
CO4	Use appropriate tools and techniques to collect, process, and interpret experimental data, effectively communicating findings through reports, charts, and presentations.	Analyze

LIST OF EXPERIMENT

PREPARED BY:	CHECKED BY:	VERIFIED BY:	APPROVED BY:
Mr. V. Y. Gosavi Course Teacher	Mr. V. Y. Gosavi Lab In-charge	Department Review Committee	Dr. R. H. Shinde HOD

EXPERIMENT 1: CORIOLIS COMPONENT OF ACCELERATION

Aim:

To study and understand the Coriolis component of acceleration using a hydraulic analogy.

Apparatus:

- Coriolis component of acceleration appratus
- Water
- Power supply

Theory:

When a point on one link is sliding along another rotating link, such as in quick return motion mechanism, then the coriolis component of the acceleration must be calculated. Consider a link OA and a slider B as shown in Fig. 1 (a). The slider B moves along the link OA. The point C is the coincident point on the link OA.

Let

ω = Angular velocity of the link OA at time t sec.

v = Velocity of the slider B along the link OA at time t sec.

ω.r = Velocity of the slider B with respect to O (perpendicular to the link OA) at time t sec

$(\omega + \delta\omega)$, $(v + \delta v)$ and $(\omega + \delta\omega)(r + \delta r)$ = Corresponding values at time $(t + \delta t)$ seconds

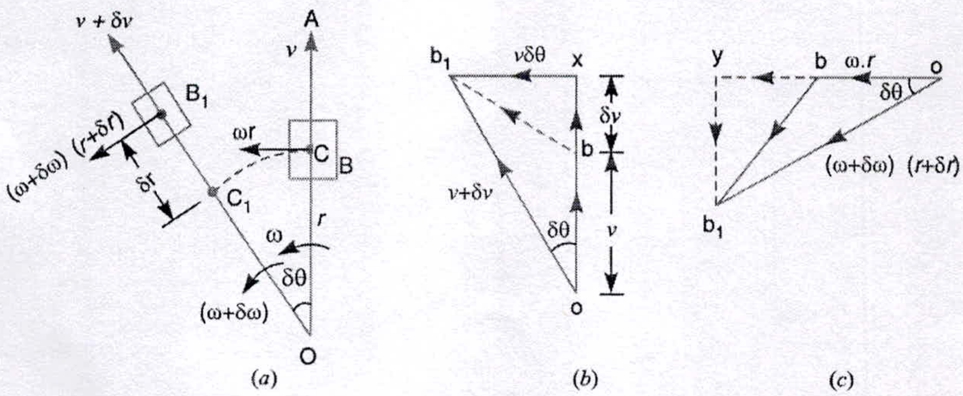


Fig 1: Coriolis component of acceleration

This tangential component of acceleration of the slider B with respect to the coincident point C on the link is known as coriolis component of acceleration and is always perpendicualr to the link.

Coriolis component of the acceleration of B with respect of C

$$a_{BC}^{c} = a_{BC}^{t} = 2\omega.v$$

where

ω = Angular velocity of the link OA,

v = Velocity of slider B with respect to coincident point C

To maintain this acceleration long enough for measurement to be taken the conventional slider mechanism is replaced by two streams of water flowing radially outward from an inverted (T) shaped tube, which is rotated about vertical axis so that the water is passing along the tube is subjected to coriollis component of acceleration.

Hydaulic analogy: consider short column of the filuid of length δr at a distance r from the axis of rotation of the tube. If the velocity of fluid realvive to the tube is v and the angular velocity of the tube is ω , the coriollis component of the column is 2 v ω in a direction perpendicular to and in the plane of rotation of the tube.

Experimental Procedure:

1. Check the bypass valve it must be fully open state.
2. Check the position of dimmer-stat, it must be at zero position.
3. Switch on main switch and with the help of dimmerstat increase the speed of motor up to some certain speed. Now start the water pump and with the help of bypass valve adjust the water level constant (any level) in the vertical header tube.
4. Take the readings of ammeter and voltmeter for actual torque, rotameter for LPH (liters per hour) and RPM for speed of the shaft.
5. Now switch off the water pump and by dimmerstat adjust the speed of the shaft to its previous value and take the reading on the ammeter and voltmeter for frictional torque.
6. Repeat the procedure by varying the speed of the shaft and take the reading.

Observations:

1. Length of rotating arms (Pipe) = 600 mm
2. Diameter of outlet tube (Pipe) = 8 mm
3. Pipe cross sectional area = 5.03×10−5 m^2
4. For torque calculation of the motor take reading of Volatage and Current directly from ammeter and voltmeter.

Observation Table:

Sr. No.	Spped (N) rpm	Voltage (V) volts	Current (I) Amps	Power (P) Watts	Flow rate LPG	Coriollis component (CA) m/s^2
1						
2						
3						

Calculations:

1. $$P = V \times I$$

2. $$T = \frac{P \times 60}{2\pi \times N}$$

3. $$CA_{\text{actual}} = \frac{2 \times g \times T}{W \times a \times l^2}$$

4. $$CA_{\text{theoretical}} = 2 \times v \times w$$

Where

P: Power (in Watts, W)

V: Voltage (in Volts, V)

I: Current (in Amperes, A)

T: Torque (in Newton-meters, Nm)

N: Rotational speed (in revolutions per minute, RPM)

g: Acceleration due to gravity (9.8 m/s2).

W= Density of water (1000 kg/m3)

a = Pipe cross sectional area (in m^2)

l = Length of Pipe (mm)

Conclusion:

These two values of coriollis coponent were found to agree reasonably well and perticularly at low angular velocities.

EXPERIMENT 2: CAM PROFILE ANALYSIS

Aim:

To study cam profile using graphical method by drawing follower displacement vs angle of cam rotation curve

Apparatus:

- Motorised Cam analysis appratus
- Dial gaude

Theory:

A cam is a rotating machine element which gives reciprocating or oscillating motion to another element known as follower. The cam and the follower have a line contact and constitute a higher pair. The cams are usually rotated at uniform speed by a shaft, but the follower motion is pre determined and will be according to the shape of the cam. Figure 1 shows different follower types as per surface in contact with cam.

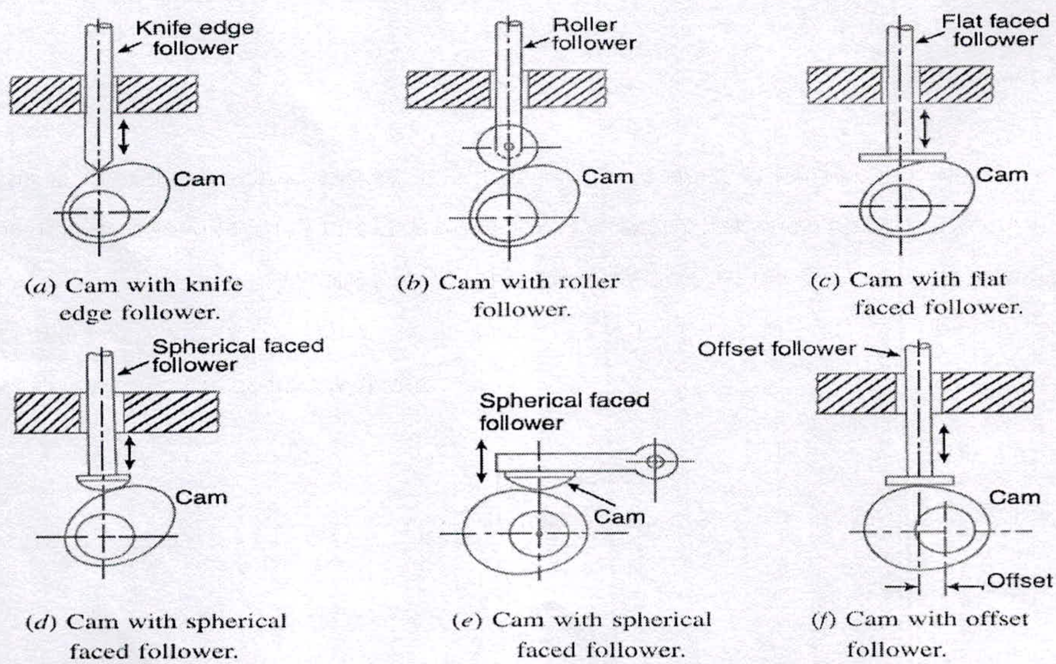


Fig 1: Types of Followers

1. Knife edge follower: When the contacting end of the follower has a sharp knife edge, it is called a knife edge follower, as shown in Fig. 1 (a). The sliding motion takes place between the contacting surfaces (i.e. the knife edge and the cam surface). It is seldom used in practice because the small area of contacting surface results in excessive wear. In knife edge followers, a considerable side thrust exists between the follower and the guide.

2. Roller follower: When the contacting end of the follower is a roller, it is called a roller follower, as shown in Fig. 1 (b). Since the rolling motion takes place between the contacting surfaces (i.e. the roller and the cam), therefore the rate of wear is greatly reduced. In roller followers also the side thrust exists between the follower and the guide. The roller followers are extensively used where more space is available such as in stationary gas and oil engines and aircraft engines.

3. Flat faced or mushroom follower: When the contacting end of the follower is a perfectly flat face, it is called a flat-faced follower, as shown in Fig. 1 (c). It may be noted that the side thrust between the follower and the guide is much reduced in case of flat faced followers. The only side thrust is due to friction between the contact surfaces of the follower and the cam.

4. Spherical faced follower: When the contacting end of the follower is of spherical shape, it is called a spherical faced follower, as shown in Fig. 1 (d). It may be noted that when a flat-faced follower is used in automobile engines, high surface stresses are produced. In order to minimise these stresses, the flat end of the follower is machined to a spherical shape.

Cam Profile Breakdown:

The cam profile typically consists of the following phases:

Phase	Description	Follower Motion
Rise	The cam lifts the follower away from its rest position.	Follower moves upward (displacement).
Dwell	The cam continues to rotate, but the follower remains stationary.	Follower remains in place (zero displacement).
Return	The cam moves the follower back to its rest position.	Follower moves downward (displacement).

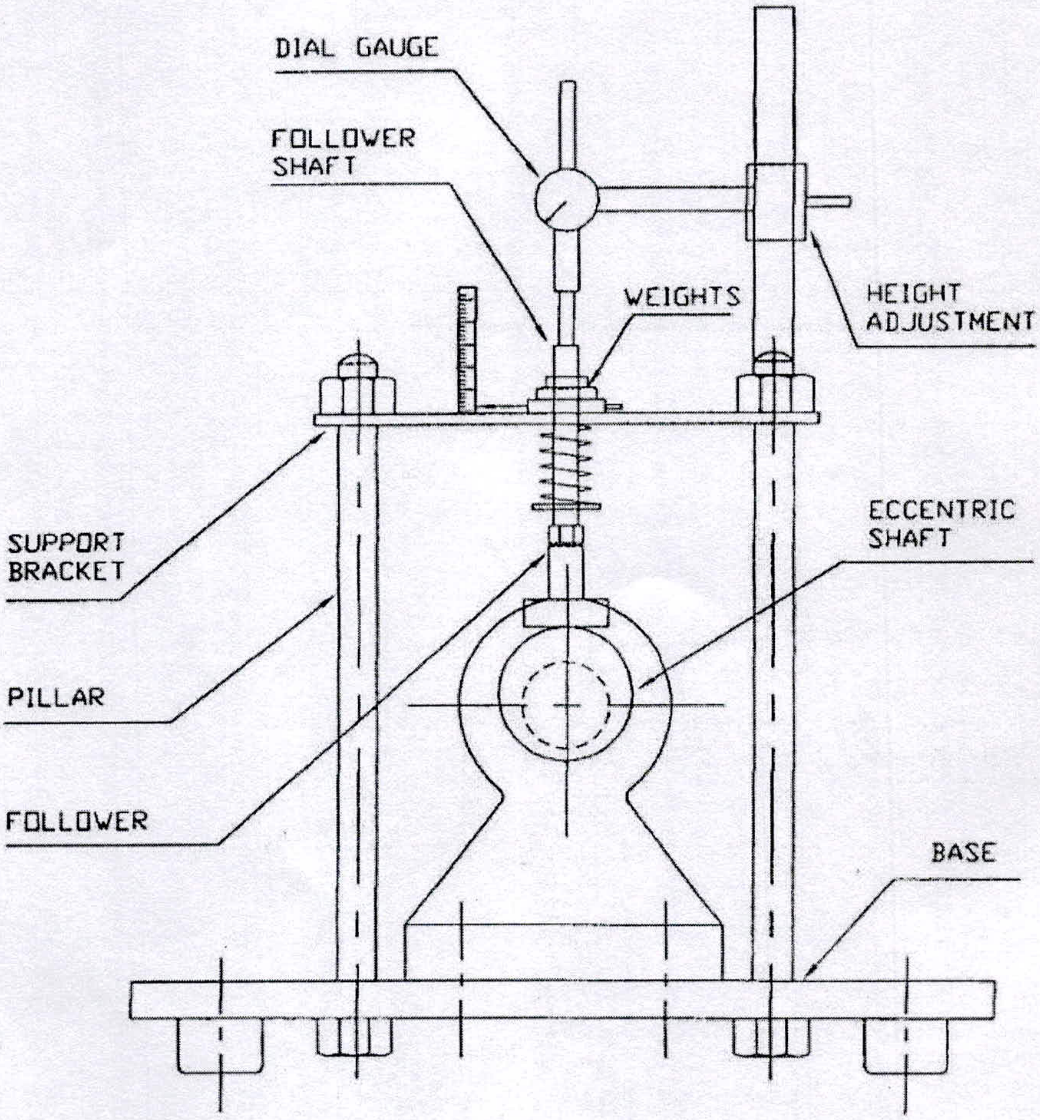


Fig 2: Cam analysis experimental set-up

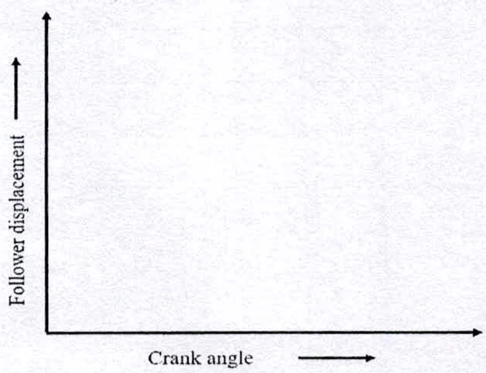
Experimental Procedure:

1. Mount dial guage and check the dial to show zero without any error.
2. Give required initial compression to the spring. In order that initial compression is not lost during operation, the check nut is to be tightened against spring seat.
3. Rotate the cam at different angle by using handle and note down the dial gauge reading.
4. Repeat the experiment by rotating cam with different angle
5. Plot the cam profile by using dial gauge reading and angle of cam rotation.

Observation table:

Sr. No.	Cam angular displacement (Θ) Degrees	Follower vertical height (h) mm
1		
2		
3		
4		
5		

Graph: cam rotation vs follower displacement



Conclusion:

- The graph shows that, the cam profile typically consists of three phases, rise dwell and return that depend on the cam position

EXPERIMENT 3: CAM JUMP ANALYSIS

Aim:

To study cam jump phenomenon to Find out Jump speed for different weights.

Apparatus:

- Motorised Cam analysis appratus
- Weights

Theory:

Cam Jump Phenomenon

The cam jump phenomenon refers to a situation in cam and follower mechanisms where the follower unexpectedly "skips" or "jumps" over a portion of its intended motion. The Jump phenomenon occurs in case of cam operating under the action of compression spring load. This is transient conditions that occur only with high speed, highly flexible cam-follower systems. With jump the cam and the follower separate owing to excessively unbalanced forces exceeding the spring force during the period of negative acceleration. This is undesirable since the fundamental function of the cam-follower system; the constraint and control of follower motion are not maintained. Also related are the short life of the cam flank surface, high noise, vibrations and poor action.

Causes of Jump:

1. High Speed and Inertia: At high speeds, the inertia of the follower may become large enough to overcome the contact force, causing the follower to lose contact with the cam and jump over parts of the cam's profile.
2. Inadequate Contact Force: Insufficient contact force between the cam and follower can occur due to poor design, wear, or improper preload in the follower spring. This can result in the follower losing contact with the cam during part of its motion.
3. Sharp Cam Profile Transitions: Abrupt changes in velocity or acceleration in the cam profile (e.g., sharp edges or sudden profile changes) can induce forces that cause the follower to lose contact or jump.

4. High Friction or Sticking: If the follower is subject to high friction or binding at certain points of contact, the sudden release of friction can cause the follower to jump as it overcomes the resistance.
5. Improper Cam Design: A cam profile with rapid acceleration and deceleration can result in an excessive force that causes the follower to momentarily lose contact.

Effects of Jump:

1. Erratic Motion: Jump can result in an unpredictable follower motion, leading to inconsistent performance.
2. Loss of Synchronization: In systems like internal combustion engines, where precise timing is crucial, cam jump can cause the system to go out of synchronization, leading to misfiring or valve timing issues.
3. Increased Wear: Loss of contact between the cam and follower can cause uneven wear on the surfaces, resulting in premature failure of components.

Crossover shock Phenomenon

Crossover shock refers to the sudden impact or force experienced by the follower as it transitions from one phase of motion to another, particularly when it crosses over from one cam lobe or section to another. This shock typically occurs when the follower changes direction or when it moves from a region of acceleration to deceleration (or vice versa), often resulting in a sudden change in velocity and impact forces.

Causes of Crossover Shock:

1. Abrupt Change in Acceleration: Crossover shock often occurs when the cam profile has sharp transitions in acceleration or velocity, causing a sudden change in direction for the follower. If the cam's motion does not have a smooth or gradual transition between lobes, the follower may experience a "shock" as it suddenly changes direction.
2. Cam Profile Discontinuities: When the cam profile has discontinuities or irregularities (e.g., sudden edges or sharp transitions), the follower may experience a sudden impact as it moves from one lobe to another.

3. Follower Stiffness and Spring Force: If the follower's spring or damping system is too stiff or poorly tuned, it may not absorb the impact when transitioning from one section of the cam to another, causing a crossover shock.
4. Follower Inertia: In high-speed systems, the follower's inertia can cause it to overshoot its intended position during transitions, leading to a crossover shock.

Effects of Crossover Shock:

1. Increased Wear: Crossover shock can cause significant wear on both the cam and follower surfaces due to the sudden impact and high forces involved. This wear can lead to reduced system life and inefficient operation.
2. Noise and Vibration: The sudden shock may lead to undesirable noise and vibrations in the system, which could be detrimental to both performance and durability.
3. Reduced Performance: Crossover shock can cause the system to behave unpredictably, reducing the overall performance, precision, and efficiency of the mechanism.
4. Damage to Components: In extreme cases, the shock can cause physical damage to the cam, follower, or other associated components, leading to failure or malfunction.

Observation:

To observe the phenomenon of jump by naked eye. When jump occurs the follower pounds on the cam surface giving a good thumping sound.

Upward inertia force = Downward retaining force

$$\frac{W}{g}\,\omega^2 r = W + S$$

This is the equilibrium of force equation when the jump will just start.

W = follower weight (Assembly)
S = spring force
ω = angular velocity of cam.
r = distance according to the geometry of cam.

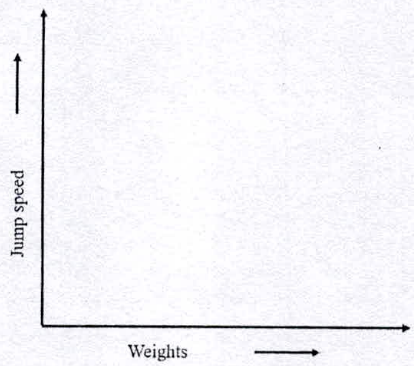
To study the effect of follower assembly weight on the jump speed when the spring force is kept constant. To study this effect, keep the initial spring compression at a certain level and observe jump speed for different follower weights by adding them successively and plot the graph of follower weights Vs. Jump speed.

$$\omega = \frac{(W + s)g}{w\,r}$$

Experimental Procedure:

1. Give required initial compression to the spring. In order that initial compression is not lost during operation, the check nut is to be tightened against spring seat.
2. Check the knob of dimmer stat, it should be at zero position.
3. Switch on main switch and with the help of dimmerstat increase the speed of motor up to some certain speed. Now observe the cam for jump phenomenon.
4. Take the readings of weight and speed at which jump occurs.
5. Repeat the procedure by varying the weight and take the reading for corresponding speed.
6. Finally plot graph of weight and jump speed.

Graph: Weight vs Jump Speed



Conclusion:

- The graph shows that, as the follower weight increases the jump speed goes on decreases.
- Jump and crossover shock are two significant phenomena in cam and follower mechanisms that can cause irregular motion, wear, and damage to components. By understanding the causes and effects of these phenomena, and implementing design improvements such as smooth cam profiles, proper speed control, and adequate spring or damping systems, the likelihood of these issues can be reduced.

EXPERIMENT 4: VERIFICATION OF PRINCIPLE OF GYROSCOPE

Aim:

To determine theoretical and practical gyroscopic couple using motorized gyroscope.

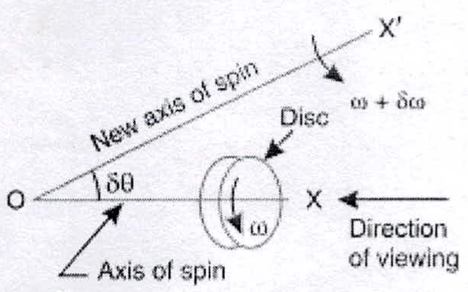
Apparatus:

- motorized gyroscope
- weights
- tachometer
- power supply

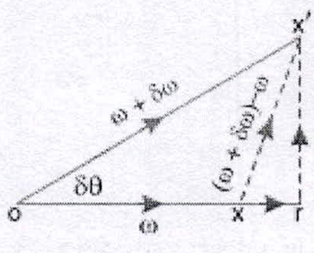
Theory:

A gyroscope is a device used to measure or maintain orientation and angular velocity. It operates on the principle of angular momentum, which states that a spinning object tends to maintain its axis of rotation unless acted upon by an external torque. Gyroscopes are widely used in navigation systems, aerospace engineering, robotics, smartphones, and more.

Consider a disc, as shown in Fig. (*a*), revolving or spinning about the axis *OX* (known as axis of spin) in anticlockwise when seen from the front, with an angular velocity ω in a plane at right angles to the paper. After a short interval of time d*t*, let the disc be spinning about the new axis of spin *OX*' (at an angle $d\theta$) with an angular velocity ($\omega + d\omega$). Using the right hand screw rule, initial angular velocity of the disc (ω) is represented by vector *ox*; and the final angular velocity of the disc ($\omega + d\omega$) is represented by vector *ox*' as shown in Fig. (*b*).

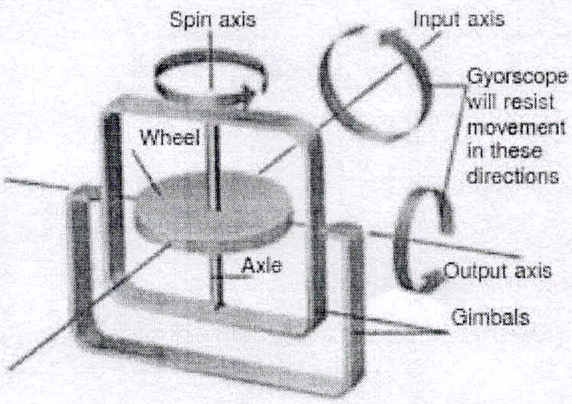


(*a*)

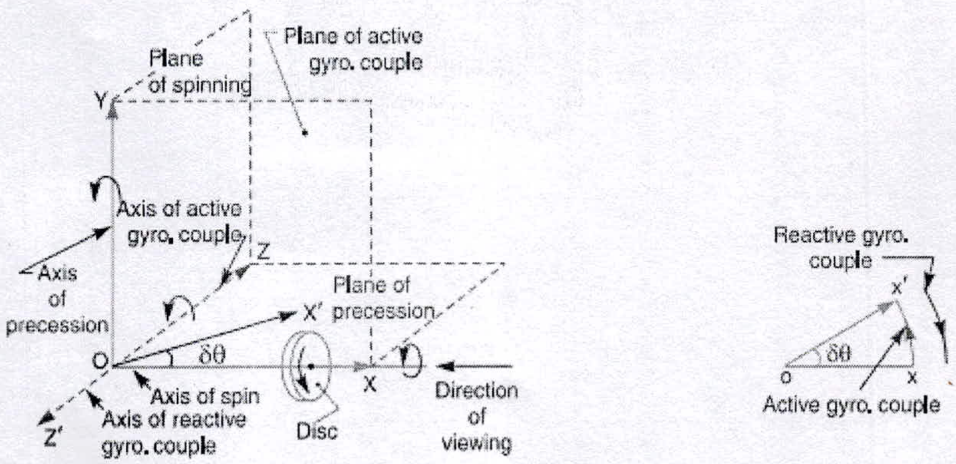


(*b*)

The vector *xx*' represents the change of angular velocity in time d *t i.e.* the angular acceleration of the disc. This may be resolved into two components, one parallel to *ox* and the other perpendicular to *ox.* This angular velocity of the axis of spin (i.e. dθ /dt) is known as angular velocity of precession and is denoted by ω. The axis, about which the axis of spin is to turn, is known as axis of precession. The angular motion of the axis of spin about the axis of precession is known as precessional angular motion. The axis of precession is perpendicular to the plane in which the axis of spin is going to rotate.



When the axis of spin itself moves with angular velocity ω, the disc is subjected to *reactive couple* whose magnitude is same (*i.e. I.* ω.ωP) but opposite in direction to that of active couple. This reactive couple to which the disc is subjected when the axis of spin rotates about the axis of precession is known as *reactive gyroscopic couple.* The axis of the reactive gyroscopic couple is represented by *OZ*' in Fig. (c)



- Effect of the Gyroscopic Couple on an Aeroplane

Procedure:

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 precessing.
5. Note down the direction of precession.
6. Verify this direction
7. Measure the velocity of precession using the pointer provided the yoke and stop watch.
8. Verify the relation C = I X ω. X ω.P.

Observation:

1. Moment of Inertia of Disc I = ___________Kgm^2.
2. Distance of point of load application from Center L = ___________ m.

Observation Table:

Sr No	Speed (RPM)	ω (rad/sec)	Weight W (kg)	dθ (degree)	dt (sec)	ω_P = dθ/ dt	Theoretical Torque = W x L (N-m)	Gyroscopic Couple = I*ω* ω_P (N-m)
1								
2								

Calculation:

1. Theoretical Torque = T = W*L

= mg*L (N-m)

Were,

L = distance of point of application of load from the centre of disc = 0.19 m

W = Weight in pan in kg

2. Gyroscopic couple = C = $I*\omega*\omega_P$

$$= mK^2 *\omega* \omega_P \text{ (N-m)}$$

Were,

m = mass of the disc = 5.1 kg

k = radius of gyration = $R/\sqrt{2}$ m

R = radius of the disc = 0.145 m

ω = angular velocity of disc = $2\pi n/60$ rad/sec

ω_P = Velocity of precession = $\frac{d\theta}{dt} \times \frac{\pi}{180}$ rad /sec

Conclusion:

Theoretical Torque = T = ____________N-m.

Gyroscopic Couple C = ____________N-m.

Thus, the theoretical and practical values confirm the principle of gyroscope.

EXPERIMENT 5: WATT GOVERNOR

Aim:

To study characteristics of watt governor.

Apparatus:

- Universal governor appratus
- tachometer
- power supply

Theory:

The function of a governor is to maintain the mean speed of a machine/prime mover, by regulating the input to the machine/prime mover automatically, when the variation of speed occurs due to fluctuation in the load. The simplest form of a centrifugal governor is a watt governor. It is basically a conical pendulum with links attached to a sleeve of negligible mass.

Procedure:

Mount the watt governor mechanism on the drive unit of the governor apparatus. Vary the governor spindle speed by adjusting the dimmer. The speed can be determined by the hand tachometer. Increase the speed of the governor spindle gradually by adjusting the dimmer and note down the speed at which the sleeve just begins to move up. Take four or five sets of readings by increasing the governor speed in steps and note down the corresponding sleeve displacement within the range of the governor and tabulate the observations.

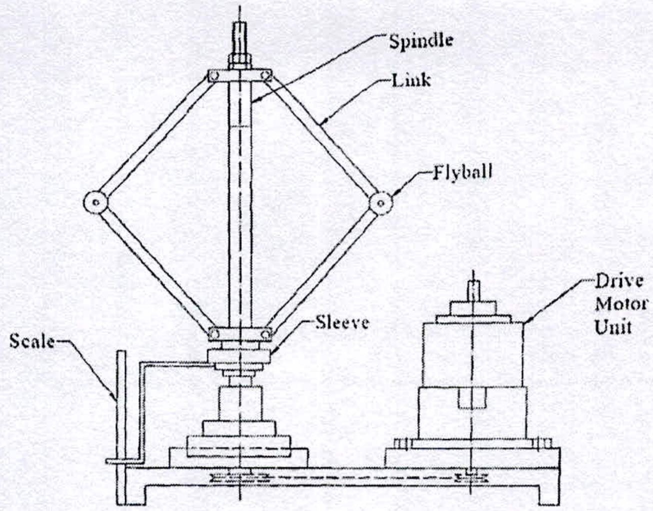
Specifications:

Length of each link l= 125 mm

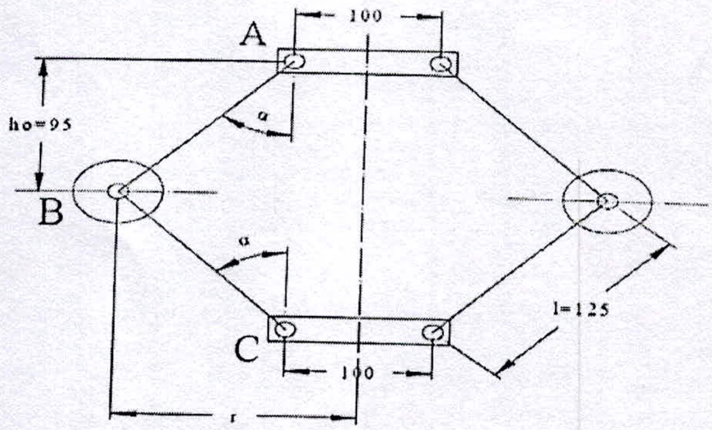
Initial height of governor (h0) = 110 mm

Initial radius of rotation (r0) = 120 mm

Mass of each ball (m) = 0.400 kg



Experimental set-up for watt governor



Watt governor configuration

Observation table:

SR.NO	Speed in RPM	Sleeve Displacement (X) in	
		mm	m

Calculations:

Sensitiveness = (max equal speed - Min equal speed) / Mean equal speed

Mean equal speed = (Max equal speed + Min equal speed) /2

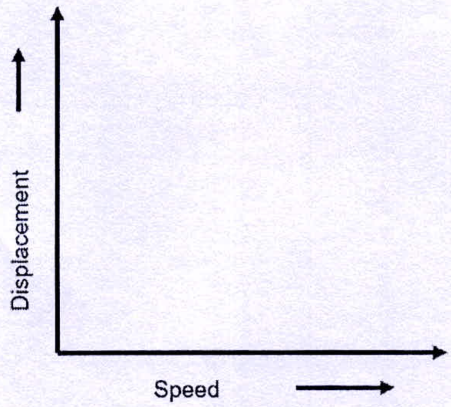
Range of speed = N2 - N1

N1 = Min equal speed in Rpm (Initial displacement of the sleeve)

N2 = Max equal speed in Rpm (Maximum displacement of the sleeve)

Graph:

Speed vs Displacement



Conclusion:

EXPERIMENT 6: PORTER GOVERNOR

Aim:

To study characteristics of Porter governor.

Apparatus:

- Universal governor appratus
- tachometer
- power supply

Theory:

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 increase the speed of revolution required to enable the balls to rise to any to any predetermined level.

Procedure:

Mount the porter governor mechanism on the drive unit of the governor apparatus. Vary the governor spindle speed by adjusting the dimmer. The speed can be determined by the hand tachometer. Increase the speed of the governor spindle gradually by adjusting the dimmer and note down the speed at which the sleeve just begins to move up. Take four or five sets of readings by increasing the governor speed in steps and note down the corresponding sleeve displacement within the range of the governor and tabulate the observations.

Specifications:

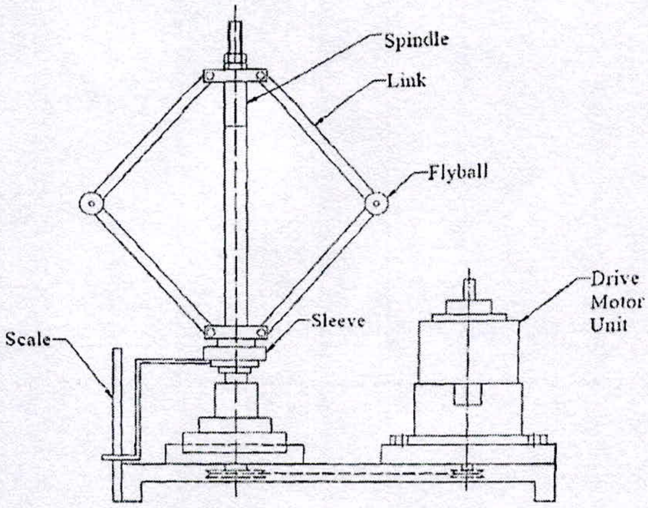
Length of each link l= 125 mm

Initial height of governor (h0) = 110 mm

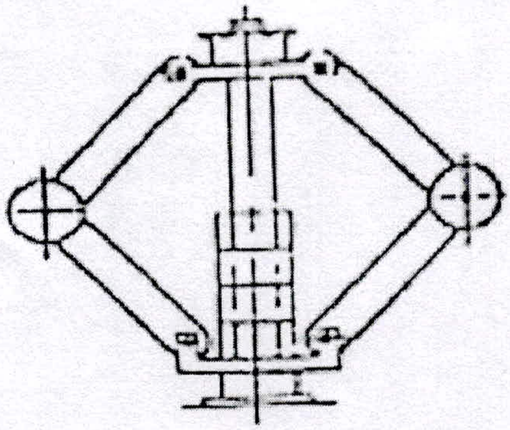
Initial radius of rotation (r0) = 120 mm

Mass of each ball (m) = 0.400 kg

Mass of each sleeve (M) = 0.500 kg



Experimental set-up for porter governor



Porter governor configuration

Observation table:

SR.NO	Speed in RPM	Sleeve Displacement (X) in	
		mm	m

Calculations:

Sensitiveness = (max equal speed - Min equal speed) / Mean equal speed

Mean equal speed = (Max equal speed + Min equal speed) /2

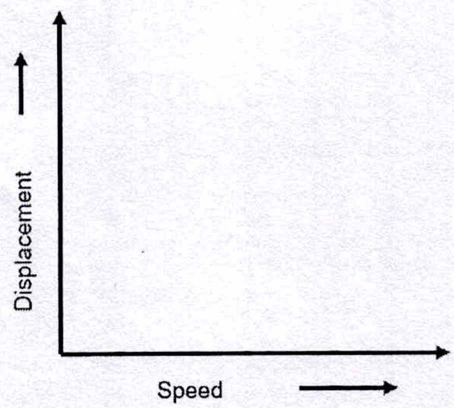
Range of speed = N2 - N1

N1 = Min equal speed in Rpm (Initial displacement of the sleeve)

N2 = Max equal speed in Rpm (Maximum displacement of the sleeve)

Graph:

Speed vs Displacement



Conclusion:

EXPERIMENT 7: PROELL GOVERNOR

Aim:

To study characteristics of Proell governor.

Apparatus:

- Universal governor appratus
- tachometer
- power supply

Theory:

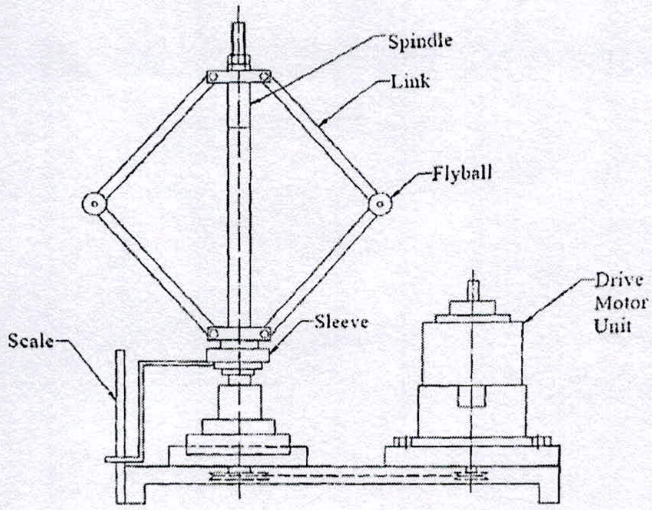
The function of a governor is to maintain the mean speed of a machine/prime mover, by regulating the input to the machine/prime mover automatically, when the variation of speed occurs due to fluctuation in the load. The porter governor is known as a proell governor if the two ball (masses) are fixed on the upward extensions of the lower links which are in the form of bent links.

Procedure:

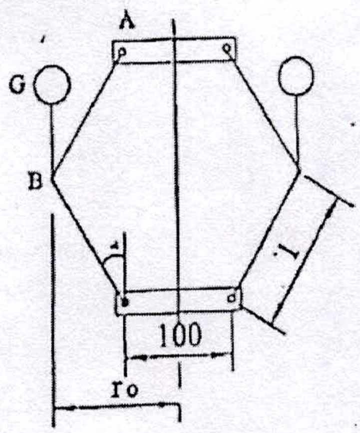
Mount the proell governor mechanism on the drive unit of the governor apparatus. Vary the governor spindle speed by adjusting the dimmer. The speed can be determined by the hand tachometer. Increase the speed of the governor spindle gradually by adjusting the dimmer and note down the speed at which the sleeve just begins to move up. Take four or five sets of readings by increasing the governor speed in steps and note down the corresponding sleeve displacement within the range of the governor and tabulate the observations.

Specifications:

Length of each link l= 125 mm
Initial height of governor (h0) = 110 mm
Initial radius of rotation (r0) = 120 mm
Mass of each ball (m) = 0.400 kg
Mass of each sleeve (M) = 0.500 kg



Experimental set-up for proell governor



Proell governor configuration

Observation table:

SR.NO	Speed in RPM	Sleeve Displacement (X) in	
		mm	m

Calculations:

Sensitiveness = (max equal speed - Min equal speed) / Mean equal speed

Mean equal speed = (Max equal speed + Min equal speed) /2

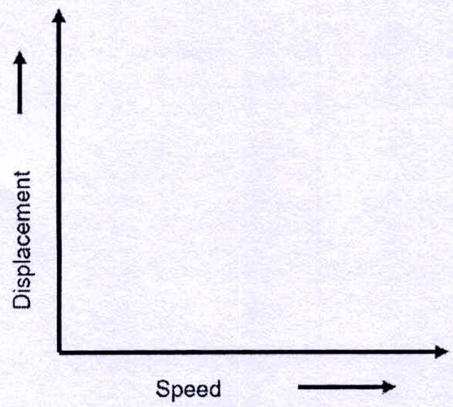
Range of speed = N2 - N1

N1 = Min equal speed in Rpm (Initial displacement of the sleeve)

N2 = Max equal speed in Rpm (Maximum displacement of the sleeve)

Graph:

Speed vs Displacement



Conclusion:

EXPERIMENT 8: HARTNELL GOVERNOR

Aim:

To study characteristics of hartnell governor.

Apparatus:

- Universal governor appratus
- tachometer
- power supply

Theory:

This governor comes under the spring loaded type centrifugal governors. The control of the speed is affected either wholly or in part by means of springs. The centrifugal 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. It consists of two balls of equal mass, which are attached to the arms as shown in fig. These balls are known as governor balls or fly balls. The balls revolve with a spindle, which is driven by the engine through bevel gears. The upper ends of the arms are pivoted to the spindle, so that the balls may rise up or fall down as they revolve about the vertical axis. The arms are connected by the links to a sleeve, which is keyed to the spindle. This sleeve revolves with the spindle but can slide up & down. The balls and the sleeve rise when the spindle speed increases and falls when the speed decreases. The sleeve is connected by a bell crank lever to a throttle valve. The supply of the working fluid decreases when the sleeve rises and increases when it falls.

Procedure:

Mount the hartnell governor mechanism on the drive unit of the governor apparatus. Vary the governor spindle speed by adjusting the dimmer. The speed can be determined by the hand tachometer. Increase the speed of the governor spindle gradually by adjusting the dimmer and note down the speed at which the sleeve just begins to move up. Take four or five sets of readings by increasing the governor speed in steps and note down the corresponding sleeve displacement within the range of the governor and tabulate the observations.

Specifications:

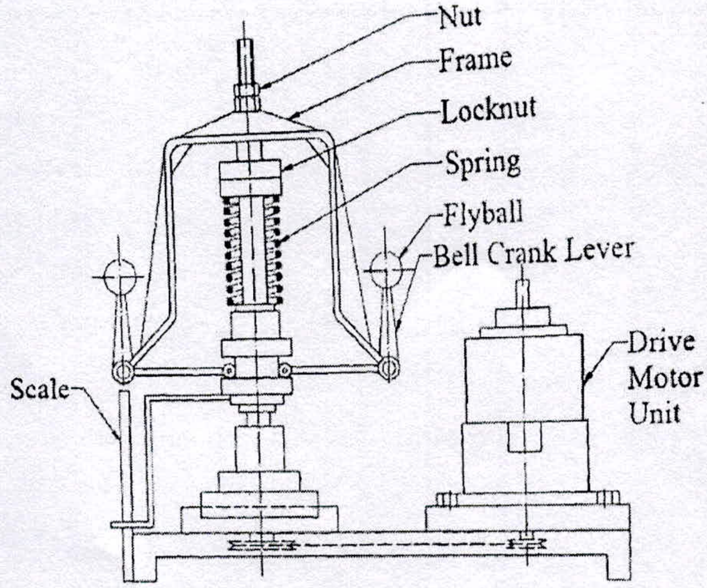
Mass of the fly ball = 0.4 kg

Length of ball arm (a) = 75 mm

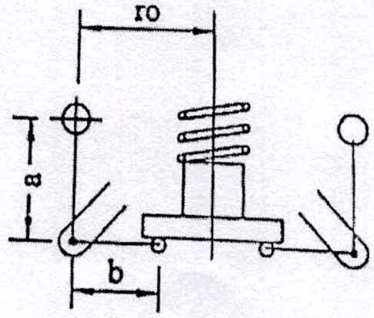
Length of sleeve arm (b) = 125 mm

Initial radius of rotation (r0) = 175 mm

Spring stiffness = 3.2 x 10'1 Kg/mm



Experimental set-up for Hartnell governor



Hartnell governor configuration

Observation table:

SR.NO	Speed in RPM	Sleeve Displacement (X) in	
		mm	m

Calculations:

Sensitiveness = (max equal speed - Min equal speed) / Mean equal speed

Mean equal speed = (Max equal speed + Min equal speed) /2

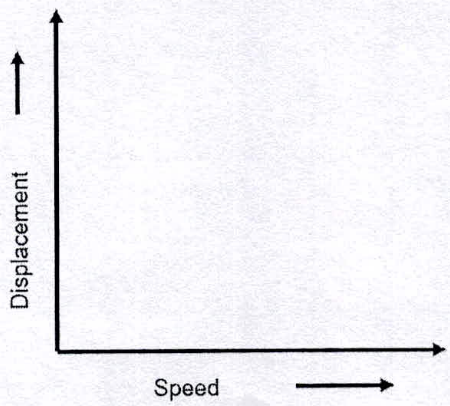
Range of speed = N2 - N1

N1 = Min equal speed in Rpm (Initial displacement of the sleeve)

N2 = Max equal speed in Rpm (Maximum displacement of the sleeve)

Graph:

Speed vs Displacement



Conclusion:

EXPERIMENT 9: FREQUENCY RESPONSE CURVES FOR DIFFERENT AMOUNT OF DAMPING

Aim:

To obtain frequncy response curve for the trsaverse vibrations of cantilever beam.

Apparatus:

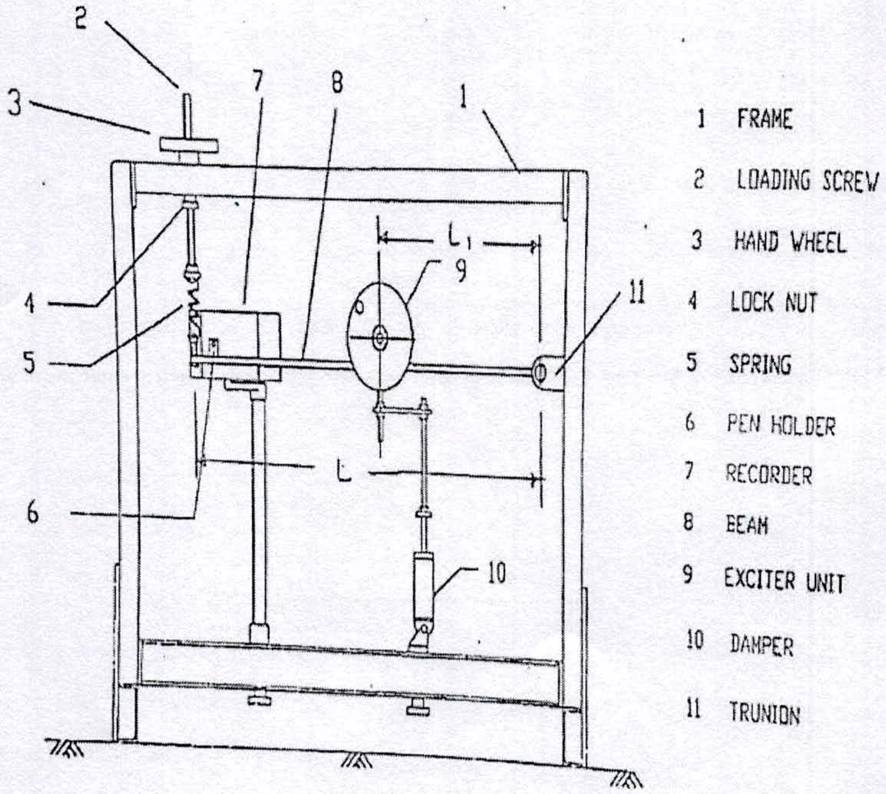
- Universal vibration apparatus
- Power supply

Theory:

The vibration that the system executes under damping system is known as damped vibrations. In general, all the physical systems arc associated with one or the other type of damping. In certain cases, amount of damping may be small in other case large. In damped vibrations there is a reduction in amplitude over every cycle of vibration. This is due to the fact that a certain amount of energy possessed by the vibrating system is always dissipated in overcoming frictional resistances to the motion. The rate at which the amplitude *of* vibration decays depends upon the type and amount of damping in the system. Damped vibrations can be free vibrations or forced vibrations. Shock absorber is an example of damped vibration.

Procedure:

1. Connect the exciter to D.C. motor.
2. Start the motor and allow the system to vibrate.
3. Wait for 3 to 5 minutes for the amplitude to build for particular forcing frequency.
4. Adjust the position of strip-chart recorder. Take the record of amplitude Vs time on the strip-chart.
5. Take record by changing forcing frequency.
6. Repeat the experiment for different damping. Damping can be changed adjusting the position of the exciter.
7. Plot the graph of amplitude Vs frequency for each damping condition



Forced vibration of equivalent spring mass system

Observation table:

Sr. No	L1 (m)	RPM	Number of oscillations, n	Time required for n oscillations, t (sec)	Periodic Time, Texpt (sec)	Forcing frequency, f ⁿexpt = 1/Texpt, (Hz)	Amplitude, (m)

Conclusion:

- From the graph it can be observed that the amplitude of vibration decreases with time.
- Amplitude of vibration is less with damped system as compared to undamped system.

EXPERIMENT 10: BIFILAR SUSPENSION

Aim:

To determine mass moment of inertia of uniform rod by using bifilar suspension.

Apparatus:

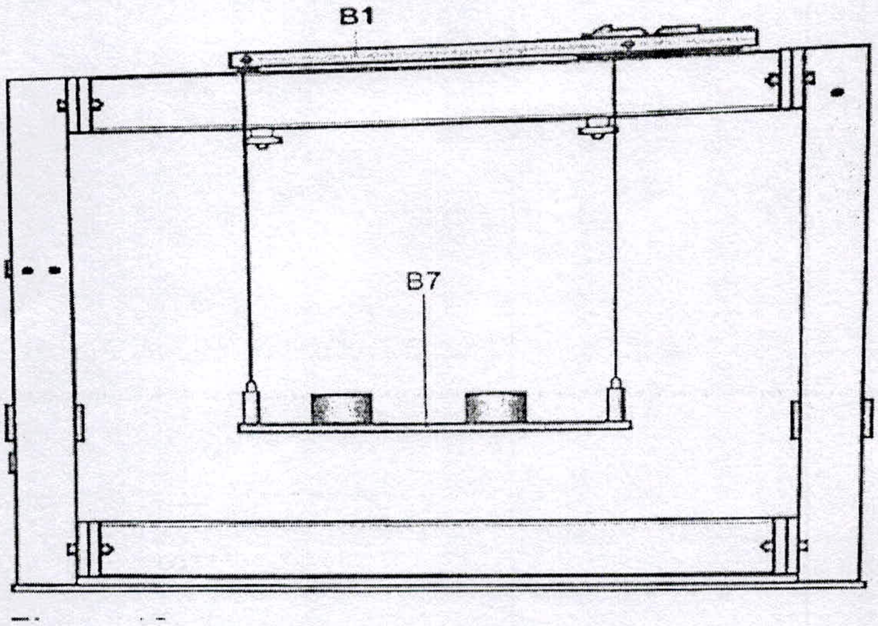
Figure shows the apparatus and consists of a uniform rectangular bar suspended by fine wires from the small chucks as used in experiment. Drawing the two wires through the chucks and tightening alters the lengths of the suspension. The bar is drilled at regular intervals along its length so that two 1.2 kg masses may be pegged at varying points along it.

Theory:

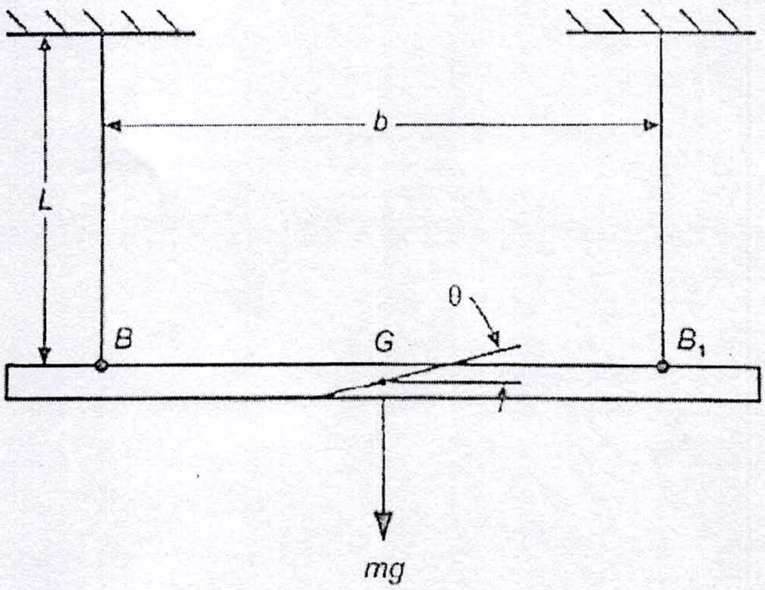
The bifilar suspension can determine the moment of inertia about an axis by suspending two parallel cords of equal length through the mass centre of bodies, as shown in fig. angular displacement of the body of the body about the vertical axis through the mass centre G is by angle Θ, which is sensibly small.

Procedure:

- With the bar is suspended by the wires, adjust length L to a convenient size, and measure the distance between the wires, b.
- displaces the bar through a small angle and measure the time taken for 20 complete oscillations.
- From this, calculate the periodic time.
- Adjust the length of the wires, L, and measure the time taken for a further 20 swings.
- Increase the inertia of the body by placing two masses symmetrically on either side of the centreline distance x apart, and repeating the procedure for various values of L and the distance between the masses.
- Calculate the radius of gyration k of the system as previously outlined.



Experimental set-up of bi-filar suspension



Bi-filar suspension configuration

Observation table:

SR. NO	LENGTH OF WIRE L (m)	DISTANCE BETWEEN TWO WIRE b (m)	NO. OF OSCILLATION n	TIME FOR n OSCILLATION t IN SEC	MASS M (Kg)	PERIODIC TIME $T_{EXP} = t/n$

Calculations:

$$T_{EXPT} = 2\pi . \frac{K}{b} . \sqrt{\frac{L}{g}}$$

$$K_{THOE} = \frac{L}{2\sqrt{3}}$$

Conclusion:

EXPERIMENT 11: SINGLE ROTOR SYSTEM

Aim:

To determine mass moment of inertia of disc by using single rotor system

Apparatus:

Universal vibration appratus with rotor disc

Theory:

When the particles of the shaft or disc move in a circle about the axis of the shaft, then the vibrations are known as torsional vibrations. The shaft is twisted and untwisted alternatively and the torsional shear stresses are induced in the shaft. Since there is no damping in the system these are undamped vibrations. Also, there is no external force is acting on the body after giving an initial angular displacement then the body is said to be under free or natural vibrations. Hence the given system is an undamped free torsional vibratory system

Specifications:

Shaft diameter, d = 3 mm

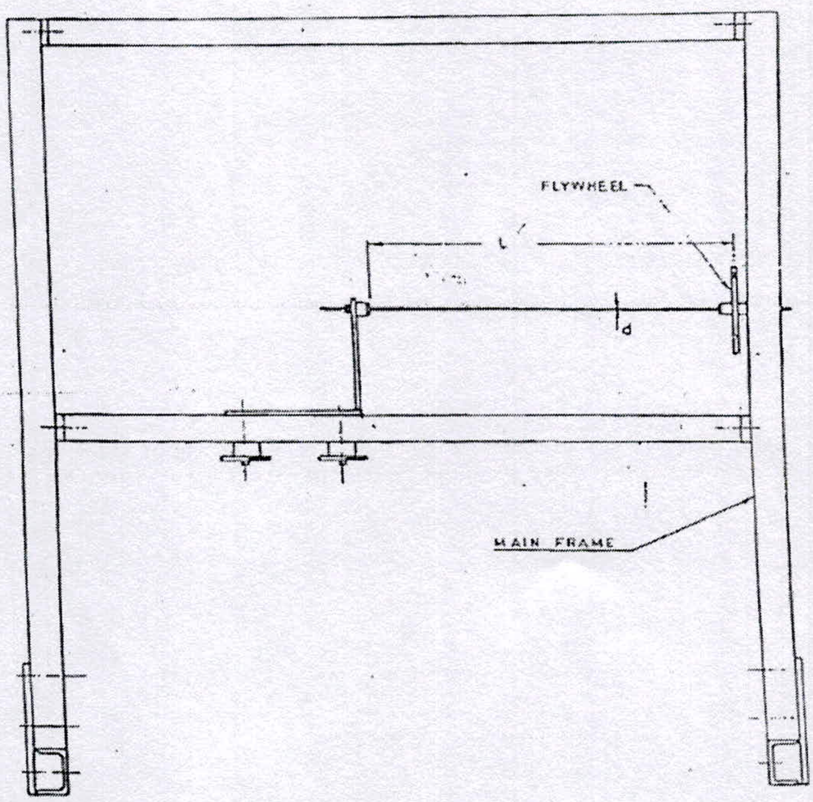
Diameter of disc, D = 225 mm

Weight of the disc, W = 2.795 kg

Modulus of rigidity for shaft, $G = 7.848 * 10^{10}\ N/m^2$

Procedure:

- Fix the brackets at convenient position along the lower beam.
- Grip one end of the shaft at the bracket by chuck.
- Fix the rotor on the other end of the shaft.
- Twist the rotor through some angle and release.
- Note down the time required for 10 to 20 oscillations.
- Repeat the procedure for different length of the shaft.



Experimental set-up of single rotor system

Observation table:

Sr. No	Length of shaft L(m)	NO of oscillations, n	Time for n oscillations, (sec)	Periodic time, TTHOE, (sec)	Natural frequency (Hz) fnTHOE	Periodic time, TEXPT = t/n, (sec)	Natural frequency f^n**expt** =1/T, (Hz)

Calculations:

Polar moment of inertia of shaft $(I_P) = \Pi * d^4 / 32$

Moment of inertia of disc, $I = mk^2$

Torsional stiffness K_t

$$K_t = \frac{(G * I_p)}{L}$$

Periodic time, T (theoretically)

$$T = 2\pi \sqrt{\frac{I}{Kt}}$$

Periodic time, T_{EXPT}

$T = t/n$

Frequency, $f_{n_{expt}} = 1 / T_{EXPT}$

Frequency, $f_{n_{THOE}}, = 1/ T_{THOE}$

Result table:

Sr. No.	Length of shaft, L (m)	Theoretical Values		Experimental values	
		T (sec) THOE	f nTheo (hz)	T EXPT = t/n (sec)	f ⁿexpt = n/t (hz)

Conclusion:

1. The natural frequency of undamped free torsional vibration (THOE)
2. The natural frequency of undamped free torsional vibration (EXPT)

EXPERIMENT 12: COMPOUND PENDULUM

Aim:

To determine mass moment of inertia of disc by using compound pendulum

Apparatus:

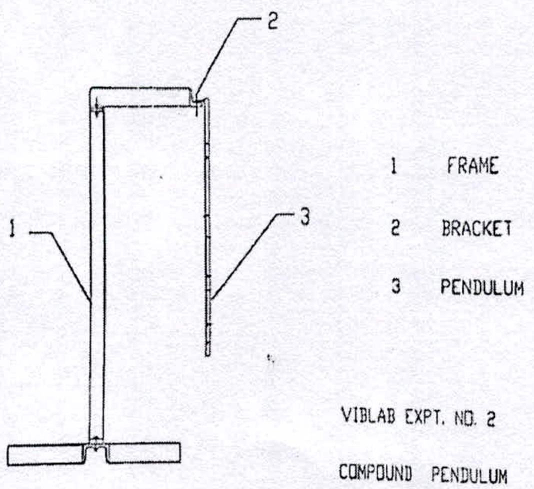
Universal vibration appratus with pendulum set-up

Theory:

The compound pendulum consists of steel bar. The bar is supported in the hole by the knife-edge. It is possible to change the length of suspended pendulum by supporting the bar in different holes.

Procedure:

- Support the rod in any one of the holes.
- Note the length of suspended pendulum and determine OG.
- Allow the bar to oscillate & determine T by knowing the time for say 10 oscillations.
- Repeat the experiment with different length of suspension.
- Complete the observation table given below.



Experimental set-up of compound pendulum

Observation table:

Sr. No	Length of pendulum L in mm	Distance of C.G in mm	No. of oscillation n	Time for n no. of oscillation t in sec	Periodic time T EXPT= t/n (sec)

Calculations:

$$T_{EXPT} = 2\pi\sqrt{\frac{K^2+(C.G)^2}{g \times C.G}}$$

$$K_{THOE} = \frac{L}{2\sqrt{3}}$$

Result table:

Sr. No	KEXPT (m)	KTHOE (m)

Conclusion:

EXPERIMENT 13: UNDAMPED FREE VIBRATION

Aim:

To study undamped free vibration of equivent spring mass system

Apparatus:

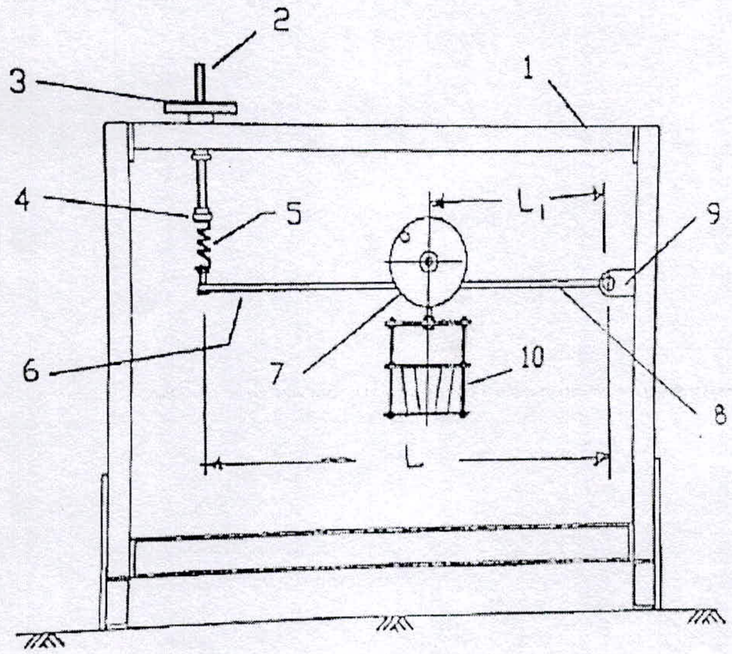
Universal vibration appratus with spring mass system

Theory:

The arrangement shown in figure is designed to study free, force, damped and undamped vibration system. It consists of MS rectangular beam supported at one end by a turning on pivoted in ball bearing. The bearing housing is fixed to side member of the frame. The other end of beam is supported by the lower end of helical spring. Upper end of helical spring is attached to screw.

Procedure:

- Support one end of the beam in the slot of trunnion and clamp it by means of screw.
- Attach the other end of the beam to the lower end of the spring.
- Set the beam in the horizontal position.
- Measure the distance LI of the assembly from pivot.
- Allow the system to vibrate.
- Measure the time for say 10 oscillations and find the periodic time and natural • frequency of vibration.
- Repeat the experiment by varying L.



MAIN FRAME
2 LOADING SCREW
3 HAND WHEEL
4 LOCK NUT
5 SPRING
6 BEAM
7 EXCITER
8 BEAM
8 TRUNION
10 WEIGHT PAN

Experimental set-up of undamped free vibration

Observation table:

Sr. No	LI (m)	m(kg)	NO of oscillations, n	Time for n oscillations t (sec)	Periodic time, $T_{EXPT} = t/n$ (sec)	Periodic time, T_{THOE} (sec)

Calculations:

Periodic time T (theoretical)

$$T = 2\pi \sqrt{\frac{m_e}{K}}$$

Where mass equivalent mass at the spring (m_e) = m (L_1^2 / L^2)

K = stiffness of the spring 0.3 kg/mm = 0.3 x 10^3 kg/m

m = (W + w)

w = weight attached to exciter assembly

W = weight of exciter assembly = 9.14 kg

L_1 =distance of W from pivot = -------- m

L is distance of spring from pivot 0.9 m

Result table:

SR. NO	Natural frequency *Fn Exp* =1/T_{Expt}, (Hz)	Natural frequency f^nTHOE =1/T_{thoe} (Hz)

Conclusion:

EXPERIMENT 14: LOGITUDINAL VIBRATION OF HELICAL SPRING

Aim:

To study logitudinal vibration of helical spring and to determine frequence of vibration

Apparatus:

Universal vibration appratus with helical spring with mass

Theory:

A helical spring, deflecting as a result of applied force, confirms to Hooke's law (deflection proportional to deflecting force). The graph of force against deflection is a straight line as shown in figure 1.

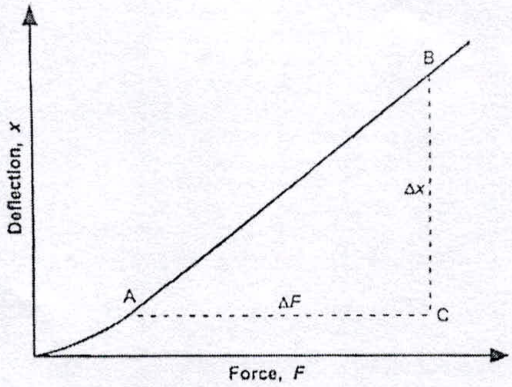


Figure1: Graph of Force vs deflection

Slope of the line $\frac{\Delta x}{\Delta F}$ is the 'deflection coefficient' in metres per newton. The reciprocal of this is the stiffness of the spring and is the force required to produce unit deflection.

Figure 2 shows the required set-up for the experiment. Suspend helical springs supplied from the upper adjustable assembly and clamp to the top member of the portal frame. To the lower end of the spring is bolted a rod and integral platform onto which 0.4 kg masses may be added. Measuring tape can be used to measure deflection, and thereby the stiffness, of a given spring.

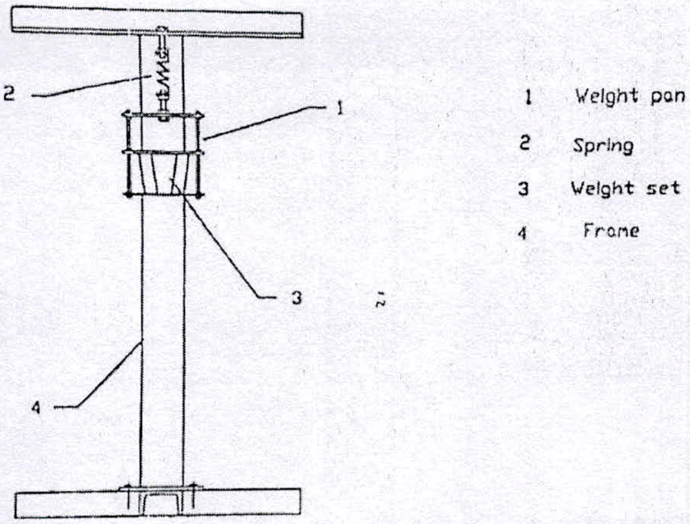


Figure 2: Experimental set-up of experiment on longitudinal vibrations of helical spring

Procedure:

Part A:

Fix the specimen spring to the portal frame, with the loading weight pan form suspended underneath. Using the measuring tape measure the length of the spring with the weight pan unloaded. Add weights in increments, taking note of the extension in Table, until reaching a suitable maximum load. Remove the weights, again noting the length at each increment, as the system is unloaded. From these values determine the mean value of extension for the spring.

Observation table part A:

Sr. No	Mass (kg)	Deflection x		Mean x (mm)
		Loading	Unloading	

Part B:

Add masses to the platform in varying increments; pull down on the platform and release to produce vertical vibrations in the system. For each increment of weight note the time taken for 20 complete oscillations in Table, and from this calculates the periodic time, T.

Observation table part B:

Sr. No	Mass m (kg)	No of oscillations n	Time for n of oscillations t in sec	Periodic time $T_{EXP} = t/n$

Calculations:

$$T_{THOE} = 2\pi\sqrt{\frac{M}{k}}$$

Result table:

SR. NO	T_{Thoe}	T_{Exp}

Conclusion:

EXPERIMENT 15: LOGITUDINAL DECREMENT

Aim:

To study logitudinal decrement and to determine cpefficient of damping

Apparatus:

Universal vibration appratus

Theory:

The figure shows general arrangement for experiment. It consists of long elastic shaft gripped at upper end by chuck in bracket. The bracket is clamped to upper beam of main frame. A heavy flywheel clamped to lower end of shaft suspended from bracket, this drum is immersed in oil which provides damping. Rotor can be taken up and down for varying depth of immersion of damping drum; depth of immersion can be read on scale.

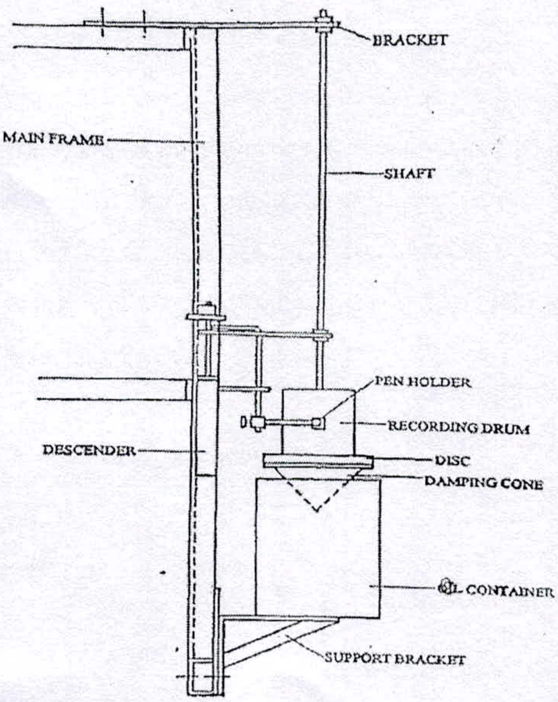


Figure: Experimental set-up of experiment on logarithmic decrement

Procedure:

- With no oil in the container allow the flywheel to oscillate and measure the time for some oscillation.
- Put thin mineral oil in the drum and note the depth of immersion.
- Put the sketching pen in the bracket.
- Allow the flywheel to vibrate.
- Allow the pen to descend and see that it is in contact with the paper.
- Measure the time for some oscillations by means of stop watch.
- Determine amplitude (Xn) at any position and amplitude (Xn+i) after cycle.

Observation table:

SR.NO	Damping medium	X_n (mm)	X_{n+}i (mm)

Calculations:

Determining Damping ratio:

Determining Logarithmic decrement

$$\delta = \ln\left(\frac{Xn}{Xn + 1}\right)$$

Where,

X_n = Amplitude of vibration of the n^{th} cycle

X_n+i= Amplitude of vibration of the $(n+l)^{th}$ cycle

$$\delta = \frac{2\pi\xi}{\sqrt{1-\xi^2}}$$

$$\delta^2 = \frac{4\pi^2\xi^2}{1-\xi^2}$$

$$1-\xi^2 = \frac{4\pi^2\xi^2}{\delta^2}$$

$$\text{We get } \xi = -----$$

Result:

Response curve for air and water

Conclusion:

Damping coefficient for air and water

EXPERIMENT 16: GENERATION OF GEAR TOOTH PROFILE

Aim: To generate gear tooth profile and to study the effects under cutting and rack shift using models

Theory:

Conjugate of Teeth: If the shape of one tooth profile is arbitrarily chosen and another tooth is designed to satisfy the law of gearing, then the second tooth is said to be conjugate to the first. The conjugate teeth are not in common use because of difficulty in manufacture, and cost of production. Therefore, in actual practice following are the two types of teeth commonly used:

1) Cycloidal teeth
2) Involute teeth

1] Cycloidal tooth profile: A cycloid is the curve traced by a point on the circumference of a circle which rolls without slipping on a fixed straight line. The cycloidal profile consists of two curves:

A) Epicycloidal curve
B) Hypocycloid curve

A) Epicycloidal curve: When a circle rolls without slipping on the outside of a fixed circle, the curve traced by a point on the circumference of a circle is known as epi-cycloid. Generally the face is made of epicycloid curve.

B) Hypocycloid curve: if a circle rolls without slipping on the inside of a fixed circle, then the curve traced by a point on the circumference of a circle is called hypo-cycloid. Generally the flank is made of hypocycloid curve.

The cycloidal teeth of a gear may be constructed as shown in Figure 1. The circle C is rolled without slipping on the outside of the pitch circle and the point P on the circle C traces epi-cycloid PA, which represents the face of the cycloidal tooth. The circle D is rolled on the inside of pitch circle and the point P on the circle D traces hypo-cycloid PB, which represents the flank of the tooth profile. The profile BPA is one side of the cycloidal tooth.

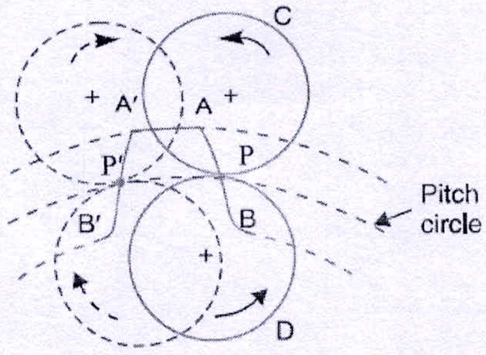


Figure 1: Construction of cycloidal teeth of a gear.

2] Involute tooth profile: It is a curve traced by a thread end when the thread end is unwound from the cylinder. The cylinder diameter is the base circle diameter & the circle is called base circle. Involute profile always exists above its base circle.

GEAR TERMINOLOGY

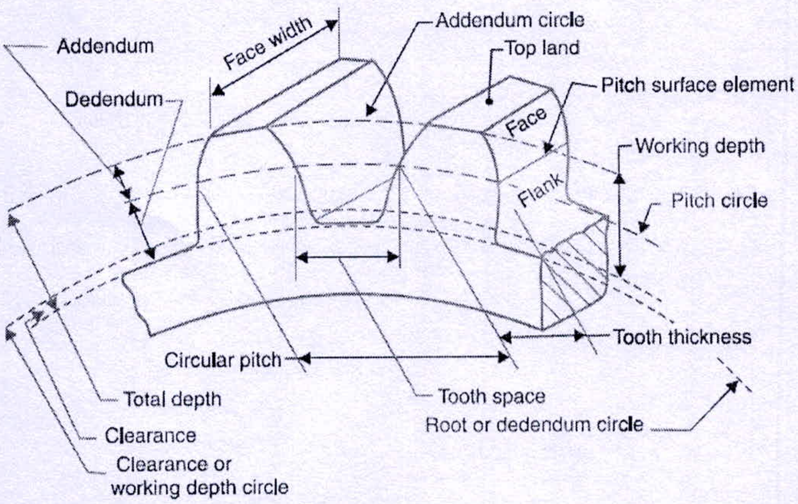


Figure 2: Terms used in gears

1. Pitch circle: It is an imaginary circle which by pure rolling action, would give the same motion as the actual gear.

2. Pitch circle diameter: It is the diameter of the pitch circle. The size of the gear is usually specified by the pitch circle diameter. It is also known as pitch diameter.

3. Pitch point: It is a common point of contact between two pitch circles.

4. Pressure angle or angle of obliquity: It is the angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point.

5. Addendum: It is the radial distance of a tooth from the pitch circle to the top of the tooth.

6. Dedendum: It is the radial distance of a tooth from the pitch circle to the bottom of the tooth.

7. Addendum circle: It is the circle drawn through the top of the teeth and is concentric with the pitch circle.

8. Dedendum circle: It is the circle drawn through the bottom of the teeth. It is also called root circle.

9. Circular pitch: It is the distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth.

$$p_c = \pi D/T$$

10. Diametral pitch: It is the ratio of number of teeth to the pitch circle diameter in millimeters.

$$p_d = \frac{T}{D} = \frac{\pi}{p_c}$$

11. Module: It is the ratio of the pitch circle diameter in millimeters to the number of teeth.

$$m = D/T$$

12. Clearance: It is the radial distance from the top of the tooth to the bottom of the tooth, in a meshing gear. A circle passing through the top of the meshing gear is known as clearance circle.

13. Total depth: It is the radial distance between the addendum and the dedendum circles of a gear. It is equal to the sum of the addendum and dedendum.

14. Working depth: It is the radial distance from the addendum circle to the clearance circle. It is equal to the sum of the addendum of the two meshing gears.

15. Tooth thickness: It is the width of the tooth measured along the pitch circle.

16. Backlash: It is the difference between the tooth space and the tooth thickness, as measured along the pitch circle. Theoretically, the backlash should be zero, but in actual practice some backlash must be allowed to prevent jamming of the teeth due to tooth errors and thermal expansion.

STEPWISE PROCEDURE FOR GENERATION OF INVOLUTE TOOTH PROFILE BY USING RACK CUTTER ARRANGEMENT:

1. Paper cut on in circular form is fixed on the circular disc having graduations on it.
2. Set the lower rack scale to zero position.
3. Mark out the position of the rack on the paper by pencil.
4. Rotate the disc by one graduation and again draw the rack on the paper.
5. Follow the same procedure for number of time
6. Finally stop the procedure when the involute profile becomes clearly visible on paper.

Observation:

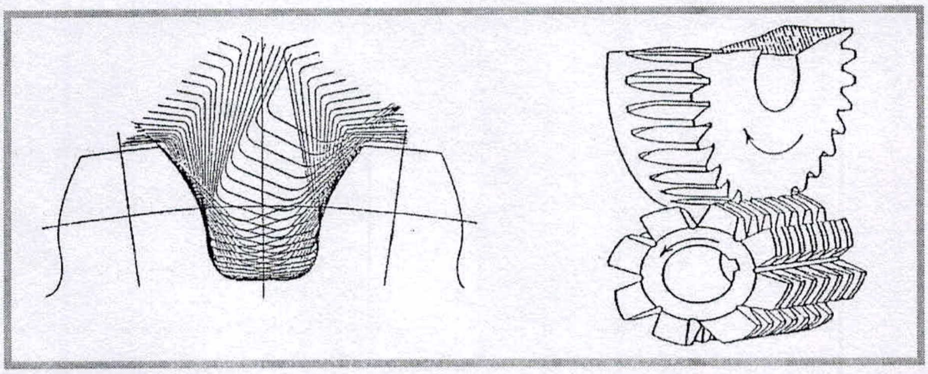


Figure 3: Generation of gear tooth profile

Conclusion:

Thus, by using above mentioned process we can generate the involute tooth profile on blank. For different module we have to use different rack cutters.