ARUNAI ENGINEERING COLLEGE

VELU NAGAR, TIRUVANNAMALAI -

DEPARTMENT OF MECHANICAL ENGINEERING

ME8511- KINEMATICS AND DYNAMICS LABORATORY

LAB MANUAL

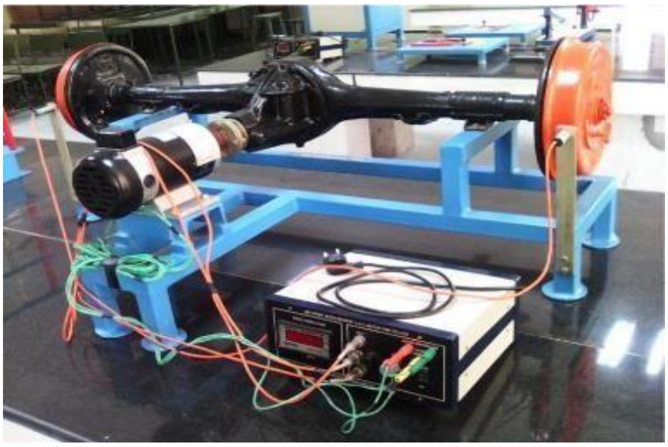
INDEX

STUDY OF GEAR PARAMETERS

Exp.No: **Date:**

Simple Gear Train

A simple gear train uses two gears, which may be of different sizes. If one of these gears is attached to a motor or a crank then it is called the driver gear. The gear that is turned by the driver gear is called the driven gear.



Differential gear mechanism

Idler Gear

When a simple gear train has three meshed gears, the intermediate gear between the driver gear and the driven gear is called an idler gear.

An idler gear does not affect the gear ratio (velocity ratio) between the driver gear and the driven gear.

Compound Gear Train

Compound gear trains involve several pairs of meshing gears. They are used where large speed changes are required or to get different outputs moving at different speeds.
Gear ratios (or velocity ratios, VR) are calculated using the same principle as for simple gear trains, i.e. VR = number of teeth on the driver gear divided by the number of teeth on the driven gear. However, the velocity ratio for each pair of gears must then be multiplied together to calculate the total velocity ratio of the gear train:

Total VR = VR1 x VR2 x VR3 x VR4….

$$\frac{\text{No of teeth on B}}{\text{No of teeth on A}} \times \frac{\text{No of teeth on D}}{\text{No of teeth on C}} = \text{Gear Ratio}$$

EPICYCLIC GEAR TRAIN

Exp.No: **Date:**

Objective:

The fundamental objectives of this study are:

1) Calculate and experimentally observe the angular velocity ratios of gear trains,

2) Experimentally obtain the torque ratios of gear trains,

3) Compute the efficiencies of gear trains.

Apparatus Required:

- Sanderson coupled epicyclic unit,
- Weights,
- Weight hanger.

Theory:

Gear trains of the type shown in Figures 1, 3, 4 and 5 are called epicyclic gear trains or planetary gear trains. In these gear trains, one or more gears are carried on a rotating planet carrier rather than on a shaft that rotates on a fixed axis. Several types of gear trains may be shifted manually to obtain greater or lesser values of speed reduction. The shifting process, however, is difficult to accomplish automatically with gears that rotate about fixed centers. On the other hand, epicyclic gear trains are readily adapted to automatic control. Some epicyclic gear trains are designed to change velocity ratios simply by using electrically or hydraulically operated band brakes to keep one or more of the gears stationary. Other epicyclic gear trains operating with fixed velocity ratios are selected for their compact design and high efficiency.

A simple epicyclic gear train consists of a sun gear (S) in the center, a planet gear (P), a planet carrier or arm (C), and an internal or ring gear (R). The sun gear, ring gear and planet carrier all rotate about the same axis. The planet gear is mounted on a shaft that turns in the bearing in the planet carrier and meshes with both the sun gear and the ring gear. (Figure 1)

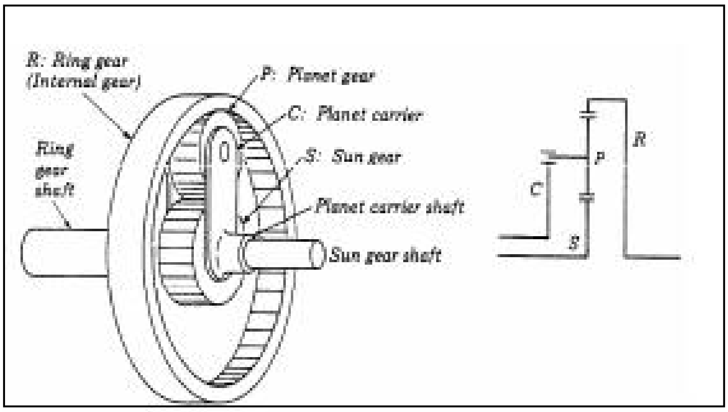


Figure 1. An Epicyclic Gear train and its associated terminology

The general expressions pertaining to the gear train are given below:

Ti = input torque,

θ_i = input angular displacement,

T_o = output torque,

θ_o = output angular displacement, then

Angular Velocity ratio R_v = $\frac{\theta i}{\theta c}$

Input work = T_i x θ_i

Output work = T_o x θ_o, and

Efficiency (h) = $\frac{\text{Outputvork}}{\text{InputWork}} = \frac{T_o\theta_o}{T_i\theta_i}$

If the input shaft tends to rotate in the direction of Ti, the gear train is balanced if

$T_i\,\theta_i = M_{Friction} + T_o$ x θ_o

If there is no friction loss, the efficiency should be 1.0 and the output work equals to input work. The corresponding T_o is ideal and is equal to T_i / R_v

$T_o = T_i / R_v$

WhereT_o is the ideal output torque.

With the existence of $M_{Friction}$, The actual output torque is smaller. The smallest Toto balance the gear train will be used to determine the system's efficiency. The efficiency can only be determined experimentally.

Calculation of velocity ratio can be performed by either the formula method or the tabular method. This ratio can also be experimentally obtained.

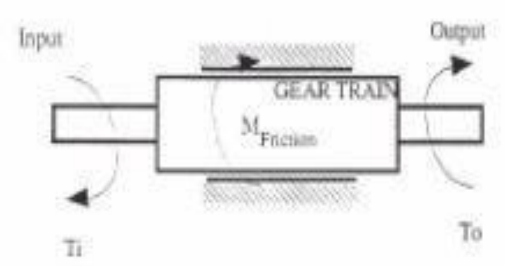


Figure 2. Schematic of a Gear Train and its balanced works.

Description of the Apparatus:

The Sanderson Coupled Epicyclic Unit consists of two standard epicyclic gear trains (Schematic Figs. 4 and 5) for laboratory demonstration of gear system similar to ones used in automotive applications. Pulleys fitted with protractors are attached to the input and output shaft so that torque and velocity ratios may be determined. Torques can be applied to the shaft by adding weights on ropes wrapped on the pulleys. Bearings are used in the entire unit to minimized friction losses. The Sanderson Coupled Epicyclic Unit can be operated in one of two modes by holding one of the ring gears stationary. The actual figure of the unit is shown in Fig. 3.

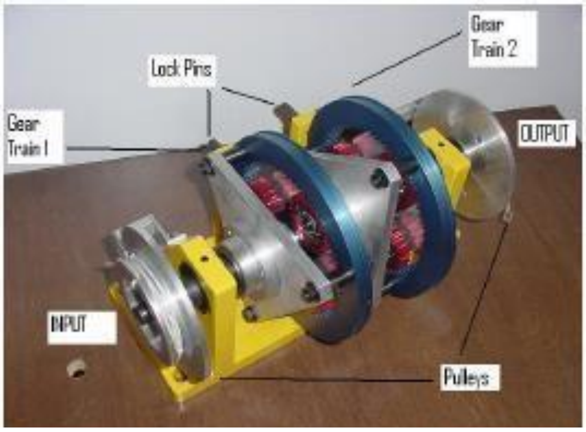


Figure 3. The Sanderson Coupled Epicyclic Unit

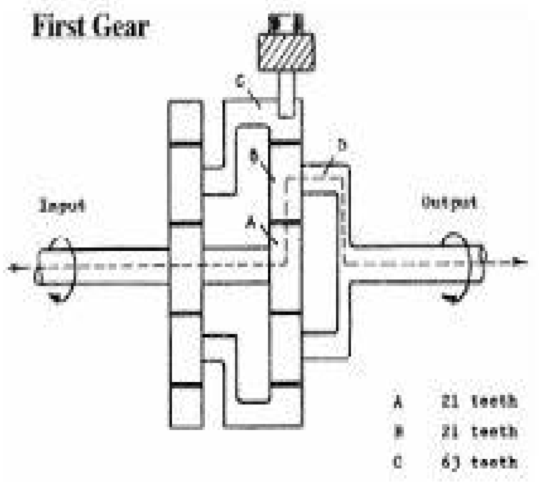


Figure 4. Schematic of the first epicyclic gear train

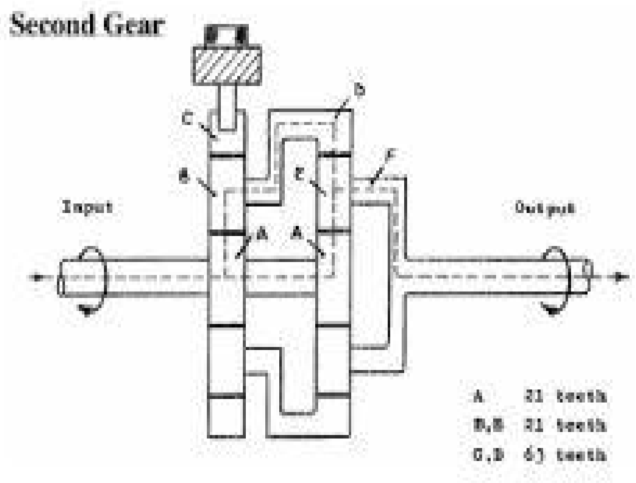


Figure 5. Schematic of the second epicyclic gear train

Experimental

In this experiment the gear ratio is initially calculated using the formula involving the number of teeth. The angular velocity ratios and the torque ratios of the two gear units are calculated experimentally. The procedure is described below:

- ❖ To calculate the angular velocity of the 1^{st} gear unit, lock the 2^{nd} gear unit using the lock pin.
- ❖ Rotate the input shaft one full turn clockwise (as viewed from the input shaft end) and observe the sense and magnitude of rotation of the output shaft (as viewed from in input shaft end). Repeat the procedure for the 2^{nd} gear unit with the 1^{st} unit locked. From this $\frac{\theta i}{\theta o}$ is calculated where θ_i is the rotation of input shaft (360^O) and θ_O is the rotation read at the output shaft.
- ❖ To estimate the torque ratios lock one of the gear units and hang a weight of 250gm at the input shaft.
- ❖ Start adding weights at the output shaft till equilibrium is achieved. Repeat the procedure for the other gear unit.
- ❖ Record the weight at the output shaft for calculation purposes.

Fig-6. Prototype Different Type of Gear Train

Results:

Obtain the following results for this experiment.

1. Theoretical gear ratio (show calculations, using the number of teeth, for both gear trains).
2. Angular velocity ratio $R_v = \frac{\theta i}{\theta o}$ which is equal to the experimental angular rotation ratio.
3. Torque ratio defined by To / Ti .
4. Efficiency.

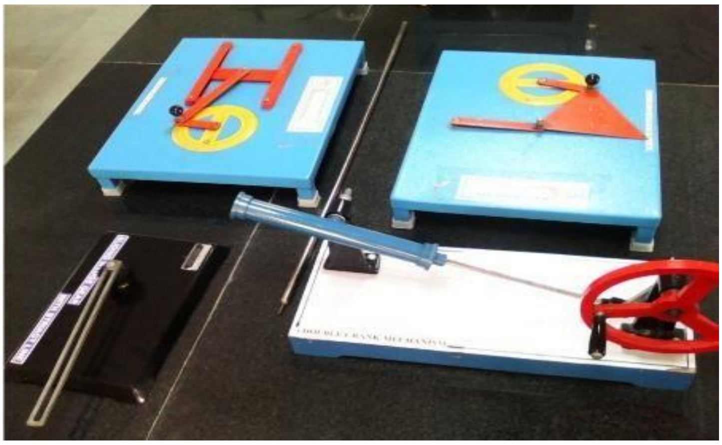
KINEMATICS OF MECHANISM

Exp.No: **Date:**

Four-Bar Linkage

A four-bar linkage or simply a 4-bar or four-bar is the simplest movable linkage. It consists of four rigid bodies (called bars or links), each attached to two others by single joints or pivots to form a closed loop.

Four-bars are simple mechanisms common in mechanical engineering machine design and fall under the study of kinematics.

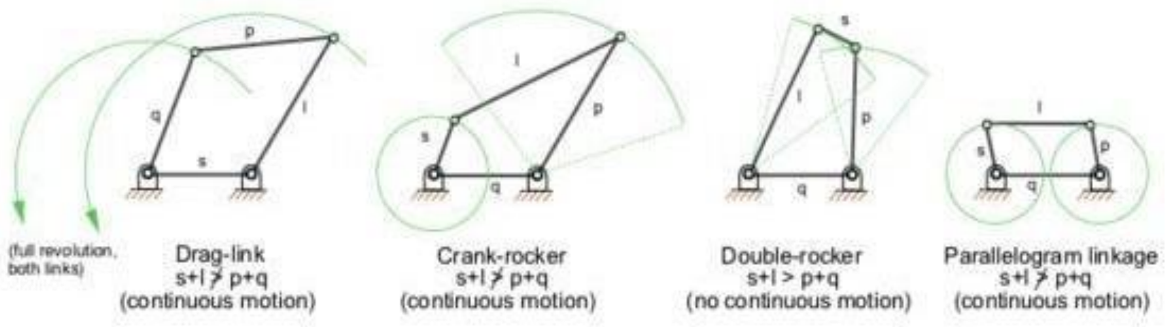


Prototypes of Inversions of Slider Crank Mechanism

If each joint has one rotational degree of freedom (i.e., it is a pivot), then the mechanism is usually planar, and the 4-bar is determinate if the positions of any two bodies are known (although there may be two solutions). One body typically does not move (called the ground link, fixed link, or the frame), so the position of only one other body is needed to find all positions. The two links connected to the ground link are called grounded links. The remaining link, not directly connected to the ground link, is called the coupler link. In terms of mechanical action, one of the grounded links is selected to be the input link, i.e., the link to which an external force is applied to rotate it. The second grounded link is called the follower link, since its motion is completely determined by the motion of the input link.

Planar four-bar linkages perform a wide variety of motions with a few simple parts. They were also popular in the past due to the ease of calculations, prior to computers, compared to more complicated mechanisms.

Grashof's law is applied to pinned linkages and states; The sum of the shortest and longest link of a planar four-bar linkage cannot be greater than the sum of remaining two links if there is to be continuous relative motion between the links. Below are the possible types of pinned, four-bar linkages;



Types of four-bar linkages, *s* = shortest link, *ℓ* = longest link

Double crank or crank-crank

- It has the shortest link of the four bar mechanism configured as the fixed link or the frame.
- If one of the pivoted links is rotated continuously, the other pivoted link will also rotate continuously.
- If double crank mechanism is also called a drag link mechanism

Crank rocker

- It has the shortest link of the four bar mechanism configured adjacent to the frame.
- If this shortest link is continuously rotated the output link will oscillate between limits. Thus the shortest link is called the crank; the output link is called the rocker.

Double rocker

The double rocker or rocker-rocker, it has the link opposite the shortest link of the four bar mechanism configured as frame. In this configuration neither link connected to the frame will be able to complete a full revolution. Thus, both input and output links are constrained to oscillate between limits and called rocker. However, the coupler is able to complete a full revolution.

Slider crank mechanism

When one of the pairs of a four bar chain is replaced by a sliding pair, it becomes a single slider crank chain or simply a slider crank chain. It is also possible to replace two sliding pairs of a four bar chain to get a double slider crank chain

In a slider may be passing through the fixed pivoted O or may be displaced. The distance e between the fixed pivot O and the straight line path of the slider is called the off-set and the chain so formed an off-set slider crank chain.

KINEMATICS OF SINGLE AND DOUBLE UNIVERSAL JOINT

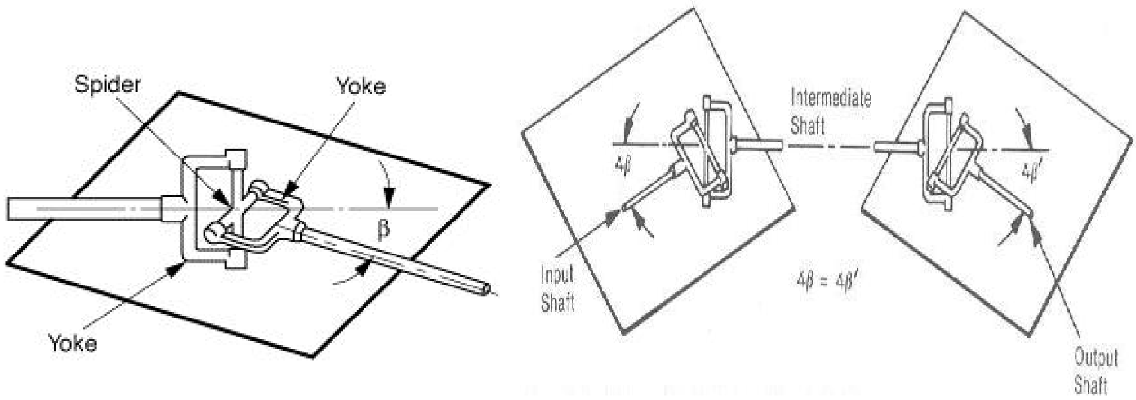
Exp.No: **Date:**

Aim

To study the kinematics of single and double universal joints.

Theory

A universal joint is used to connect two shafts, which are intersecting at small angle. The end of each shaft is forked to U type and each fork provided two bearings for the arms of a cross. The arms of the cross are perpendicular to each other. The motion is transmitted from the driving shaft to driven shaft through a cross. The inclination of the two shafts may be constant, but in a actual practice it varies, when the motion is transmitted. The main application of the universal joint is found in the transmission from the gear box to the differential or back axle of the automobiles. It is also used for transmission of power to different spindles of multiple drilling machines. It is also used as a knee joint in milling machines.



Kinematics of Single And Double Universal Joint

FLYWHEEL AND CONNECTING ROD

Exp.No: **Date:**

Aim:

To determine the moment of inertia by oscillation of flywheel and connecting rod.

Procedure:

- Measure the center to center distance of connecting rod, also measure inner diameter of both side of connecting rod.
- Measure the weight of connecting rod and flywheel.
- Attach small end of connecting rod of shaft.
- Give oscillation of connecting rod.
- Measure time taken for 5 oscillations and calculate time period $t_{p1.}$
- Remove the connecting rod from the shaft and again attach the big end of the connecting rod of shaft.
- Again measure time taken for 5 oscillations and calculate time period $t_{p2.}$
- Calculate moment of inertia of connecting rod.
- Repeat the procedure and take mean $t_{p.}$
- Attach flywheel to other side of the shaft and repeat the same procedure as above.

L – Center to center distance of connecting rod (0.228m)

M – Weight of connecting rod (3.2kh)

D_1 – Diameter of small end of connecting rod (0.062m)

D_2 -- Diameter of big end of connecting rod (0.33m)

N – Number of oscillations

Tabular column:

End position	Time taken for n oscillations t (sec)	Periodic time tp (sec)	Moment of inertia

$$I = mk^2$$

$$L_1 = g(tp_1/2\pi)^2$$

$$L_2 = g(tp_2/2\pi)^2$$

$$K^2 = h(L - h)$$

$$h_1+h_2 = d_1/2 + d_2/2$$

$$L = (L_1 + L_2)/2$$

Result:

Thus the moment of inertia by oscillation of flywheel and connecting rod was found.

MULTI DEGREE FREEDOM SUSPENSION SYSTEM

a) BIFILAR SUSPENSION

Exp. No: **Date:**

Aim:

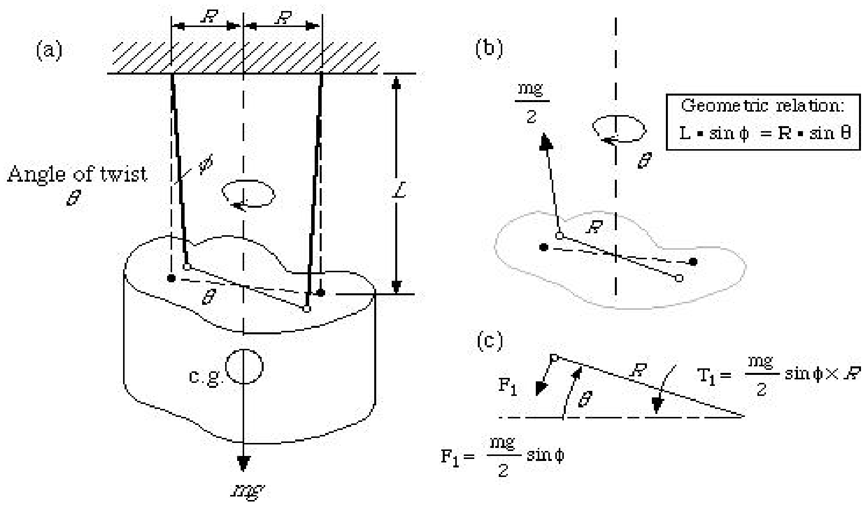
To determine the radius of gyration of given bar by using bifilar suspension and periodic time experimentally and compare it with the theoretical values.

Apparatus Required:

1) Vibration lab machine
2) Measuring tape
3) Weights
4) Stop watch
5) Bar

Description of the setup:

A uniform rectangular section bar is suspended from the pendulum support frame by two parallel cords. Top ends of the cords pass through the two small chucks fitted at the top. Other ends are secured in the bifilar bar. It is possible to adjust the length of the cord by loosing the chucks. The suspension may also be used to determine the radius of gyration of any body. In this case the body under investigation is bolted to the centre. Radius of gyration of the combined bar and body is then determined.



(a) Bifilar Suspension

Procedure:

- Suspend the bar from chuck and adjust the length of the cord 'L' conveniently. Note the suspension length of each cord must be the same.
- Allow the bar to oscillate about the vertical axis passing through the
- centre and measure the periodic time T by knowing the time for say n = 10 oscillations.
- Repeat the experiment by mounting the weights at equal distance from the centre (D / 2 as shown).

Observation:

Distance between two cords, 2a = cm

Distance from centre to cord, a = , cm

Length of the bar,L = cm

Mass added = g

Tabulation:

S. No	Weight added m (kg)	Length of String L (m)	Time taken for N osc. T sec	Natural frequency Fn (Hz)	Radius of gyration K (mm)

Formulae:

Experimental periodic time, T exp = Tm / n , sec

Where,

Tm = mean time taken for n oscillations

n = number of oscillations = 10

Natural frequency $f_n = 1/T$ Hz

Radius of gyration = $K = (T_b/2\pi) / \sqrt{g/l}$ (mm)

b- Distance of string from center of gravity = 10×10^{-2} mm

L- Length of string

N- Number of oscillations

t- Time taken for N oscillations

Experimental periodic time,

$T_{exp} = [(2\pi K_{exp}) / a] [\sqrt{L / g}]$, sec

Where,

K_{exp} = experimental radius of gyration, cm

a = distance from centre to cord, cm

L = suspension length, cm

Theoretical periodic time,

$T_{theo} = [(2\pi K_{theo}) / a] [\sqrt{L / g}]$, sec

Theoretical radius of gyration,

$K_{theo} = l / (2 \sqrt{3})$, cm

Where,

l = length of the bar , cm

Result:

Radius of gyration of given bar:

i) Experimentally, Kexpm = , cm

ii) Theoretically, Ktheo = , cm

The periodic time of the given bar is determined experimentally and verified with the theoretical values.

MULTI DEGREE FREEDOM SUSPENSION SYSTEM

b) COMPOUND PENDULUM

Exp.No: **Date:**

Aim:

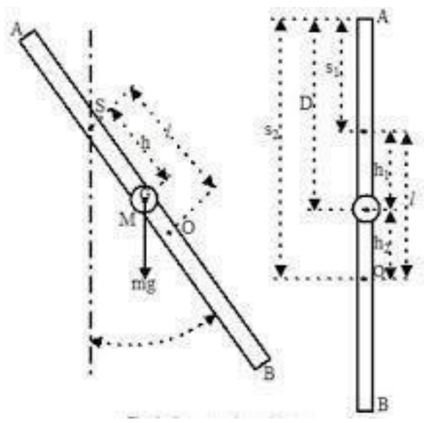
To determine moment of inertia by using compound pendulum and period and radius of gyration of the given steel bar experimentally and compare it with the theoretical values.

Apparatus Required:

1) Steel bar
2) Knife edge support
3) Stop watch and
4) Measuring tape

Description of the setup:

The compound pendulum consists of 100 cm length and 5 mm thick steel bar. The bar is supported by the knife edge. It is possible to change the length of suspended pendulum by supporting the bar in different holes.



(b) Compound Pendulums

Tabulation

S. No	Distance L_1 (mm)	Time taken for 10 oscillation t(sec)-Tmean	Time taken for N osc. T sec	Natural frequency Fn (Hz)	Radius of gyration K (mm)

Procedure:

- Support the steel bar in any one of the holes.
- Note the length of suspended pendulum to measure.
- Allow the bar to oscillate and determine Texp by knowing the time taken for n = 10 oscillations.
- Repeat the experiment with different length of suspension.

Observation:

Length of the steel bar, L = , cm

Number of holes =

Distance between two holes = , cm

Mass of the steel bar = 1.575 , kgf

Formulae:

Experimental periodic time, $T_{exp} = t / n$, sec

Where,

t = time taken for n oscillations

n = number of oscillations = 10

Experimental time period = $T = 2\pi\sqrt{(K^2 + L_1^2)/gL_1}$

K – Radius of gyration.

$K = \sqrt{gL_1T^2 / 4\pi^2L_1^2}$

Theoretical radius of gyration,

$K_t = L/\sqrt{2}$

L – Total length of rod

L_1 – Distance from point of suspension to center of gravity of rod

Natural frequency $F_n = 1/T$ Hz

Moment of Inertia = $I_m = mK^2$ (kgm^2)

Result:

Radius of gyration of given bar:

i) Experimentally, K_{exp} = cm

ii) Theoretically, K_{theo} = cm

The periodic time of the given bar is determined experimentally and verified with the theoretical values

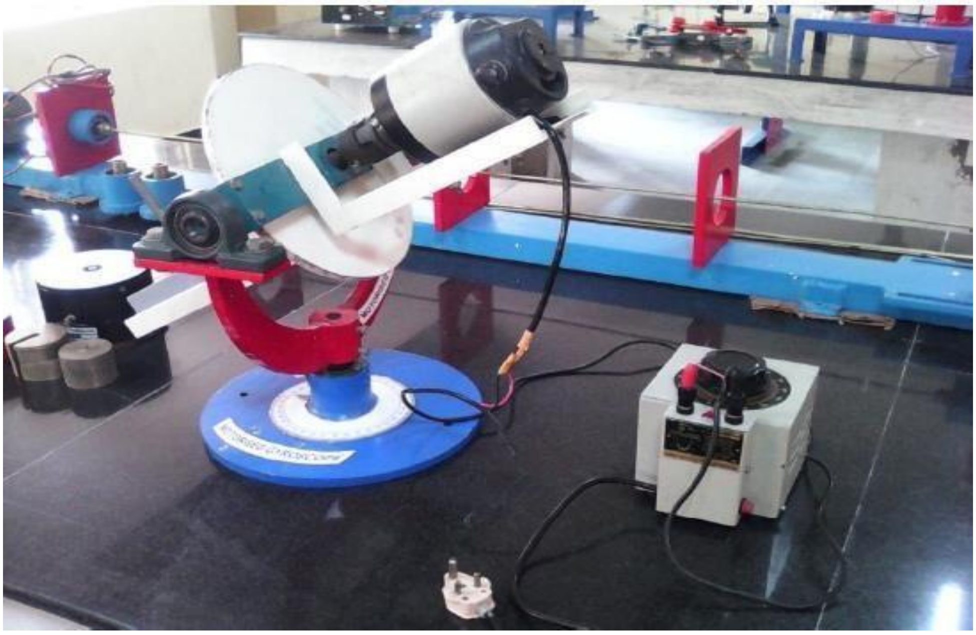
GYROSCOPIC COUPLE

Exp. No: **Date: Aim:**

To determine the active and reactive gyroscopic couples and compare them.

Apparatus required:

Gyroscope, tachometer, or stroboscope, variable voltage transformer, rotating disc with a light reflecting sticker for stroboscope speed measurement.



Procedure:

- The disc as made to rotate at a constant speed at a specific time using variable voltage transformer.
- The speed of the (N) disc is measured using a tachometer or a stroboscope.
- A weight/mass is added on the extending platform attached to the disc.
- This causes an active gyroscopic couple and the whole assembly (rotating disc, rotor and weight platform with weight) is standing to move in a perpendicular plane to that of plane of rotating of disc. This is called gyroscopic motion.
- The time taken (t) to traverse a specific angular displacement ($\varphi = 45°$) is noted.

Formula used:

Mass moment of Inertia of the disc, $I = md^2/8$, $kg - m^2$,

m – Mass of the disc

d – Diameter of the disc.

Angular velocity of the disc, $\omega = 2\pi N/60$, rad/s,

N – Speed of the disc in rpm

Angular velocity of precession, $\omega_p = (\varphi/t) \times (\pi/180)$ rad/s

Reactive gyroscopic couple, $C_r = I.\ \omega_p.\ \omega$ Nm

active gyroscopic couple, $C_a = W \times L$.

W – weight added = mg, N and

L – distance between centers of weight to center plane of disc.

Tabulation

Constant Load Method

S.No	Speed of disc. N, rpm	Weight added m, kg	Time taken for 60° precision t, sec	Active couple $C_a = W \times L$ Nm	Reactive couple $C_r = I.\ \omega_p.\ \omega$ Nm

Constant Speed Method

S.No	Speed of disc. N, rpm	Weight added m, kg	Time taken for 60° precision t, sec	Active couple $C_a = W \times L$ Nm	Reactive couple $C_r = I.\ \omega_p.\ \omega$ Nm

Graph:

1. Active couple Vs. Reactive coupe
2. Weight added Vs. Reactive couple

Result:

Thus the active and reactive gyroscopic couples were determined and compared.

UNDAMPED FREE VIBRATION OF SPRING MASS SYSTEM

Exp. No: **Date:**

Aim:

To determine stiffness of the given helical spring, period and frequency of undamped free vibration (longitudinal vibration) of spring mass system experimentally and compare it with the theoretical values.

Apparatus Required:

1) Helical spring
2) Platform
3) Weights
4) Measuring tape and
5) Stop watch

Description of the setup:

It consists of an open coil helical spring of which one end is fixed to the screw rod and a platform to the other end. This platform is used to add weights and a lock nut is also provided to clamp the weights added.

Procedure:

- Fix one end of the helical spring to the upper screw rod.
- Measure the free length of the spring.
- Attach the other end to the platform and add some weight.
- Note down the deflection.
- Stretch the spring through some distance and release.
- Observe the time taken for n = 10 oscillations.
- Repeat the steps from 3 to 6 for other known weights.

Observation:

Length of the spring before loading = , m

Formulae:

Stiffness of the spring, K exp = Load / deflection = W / X, N/m

Deflection, X = (Length of the spring after loading – length of the spring before loading) ,m

Experimental period of vibration , T exp = tm / n , sec

Where,

tm = mean time taken for n oscillations

n = number of oscillations = 10

Experimental Natural Frequency = F_n (exp) = 1/t

Theoretical Natural Frequency F_n (theo) = $(1/2\pi)/\sqrt{g/\delta}$

Stiffness K = Load / Deflection

Tabulation:

S.NO	Load added, W (kg)	Load added, W (N)	Length of the spring after loading (m)	Deflection, X (m)	Stiffness, K exp = W / X (N / m)	Time taken for n = 20 oscillations (sec) t_1	t_2	t_3	t_4	t_5	T_m	Period of vibration (sec) T exp	T the	Frequency of vibration (Hz) F exp	F theo

Graph:

Deflection Vs Load added

Result:

Stiffness of the spring,

i) Experimentally, K expm = N/m

ii) Graphically = N/m

The period and frequency of undamped free vibration (longitudinal vibration) of spring mass system are determined experimentally and verified with the theoretical values.

DAMPED FREE VIBRATION OF SPRING MASS SYSTEM

Exp. No: **Date: Aim:**

To calculate the damped natural frequency of spring mass system

Apparatus Required:

1. Weight
2. Spring
3. Stop watch

Procedure:

- Determination of natural frequencies
- Add the weight add and make the spring to oscillate for 10 times
- Note the corresponding time taken for 10 oscillation and calculate the time period
- From the time period calculate natural frequency.

Formulae:

Stiffness of the spring, K exp = Load / deflection = W / X, N/m

Deflection, X = (Length of the spring after loading – length of the spring before loading) ,m

Experimental period of vibration , T exp = tm / n , sec

Where,

tm = mean time taken for n oscillations

n = number of oscillations = 10

Experimental Natural Frequency = F_n (exp) = 1/t

Theoretical Natural Frequency F_n (theo) = $(1/2\pi)/\sqrt{g/\delta}$

Stiffness K = Load / Deflection

Tabulation :

S/No	Weight Added (kg)	Weight added (N)	Deflection (δ)	Stiffness (k)	Theoretical Natural Frequency (Hz)

Result

Stiffness of the spring,

i) Experimentally, K expm = N/m

ii) Graphically = N/m

The period and frequency of damped free vibration (longitudinal vibration) of spring mass system are determined experimentally and verified with the theoretical values.

DETERMINATION OF NATURAL FREQUENCY OF TORSIONAL VIBRATION IN SINGLE ROTOR SYSTEM.

Exp No. **Date:**

Aim:

To determine natural frequency of torsional Vibration in single rotor System

Equipment:

1. Vibration machine,
2. Shaft,
3. Chuck,
4. Stop watch.

Formula Used:

$T_{theo} = 2\pi\sqrt{I/K_t}$

Procedure

- Fix the bracket at convenient position along the tower beam.
- Grip one end of the shaft at bracket by the chuck.
- Fix other end of shaft in the rotor.
- Twist the motor rotor to some angle and then release.
- Note down the time for no. of oscillations.
- Repeat the procedure for different length of shaft.

Observation:

Observations are to taken for mild steel brass shafts.

Observation Table:

S.No.	Length. of Shaft (m)	No. of oscillation (n)	Time s	K N/mm	Tth s	Texp s	Fth Hz	Fexp Hz

Result:

The natural frequency of the torsional vibration in single rotor system is = Hz

Conclusion:

Natural frequency of torsional vibration experimental to theoretical is nearly same.

DETERMINATION OF NATURAL FREQUENCY OF TORSIONAL VIBRATION IN TWO ROTOR SYSTEM

Exp No. **Date:**

Aim:

To determine natural frequency of torsional vibration theoretically experimentally in a two rotor system**.**

Equipment:

1. Shaft,
2. Two rotor disc,
3. Chuck,
4. Stop watch.

Formula Used:

$T_{theo} = 2\pi\sqrt{I/K_t}$

Procedure

- Fix two disc of the shaft and fit the shaft in the bearing.
- Deflect the disc in opposite direction by hand and then release.
- Note down the time required for particular number of oscillations.
- Fit cross arm to one end of the disc and again note down the time.
- Repeat the procedure with different and equal masses attached to the ends of cross arm and note down the time**.**

Observation:

Observations are to taken for copper and steel shafts.

Tabulation:

S./No.	Θ rad	θb rad	No. of Oscillation n	Time s	Texp s	Tth s	Fth $=1/T_{ex}$ Hz	F exp = $1/T_{exp}$ Hz

Result:

The natural frequency of the torsional vibration in two rotor system is = Hz

Conclusion:

It is studied to determine the natural frequency of vibration of the given shaft.

It is necessary to find out the natural frequency, so that during working resonance will be taken care of**.**

CAM PROFILE ANALYSIS

Exp. No: **Date:**

Aim:

To study the profile of given cam analysis system and to draw the displacement diagram for the follower and the cam profile, also study the jump speed characteristics of the cam and follower mechanism.

Apparatus required:

1. Cam analysis system

2. Dial gauge.

Description:

A cam is a machine element such as a cylinder or any other solid with a surface of contact so designed as to give a predetermined motion to another element called the follower. A cam is a rotating body importing oscillating motor to the follower. All cam mechanisms are composed of at least three links viz: 1. Cam 2. Follower and 3. Frame which guides follower and cam.

Specification:

Diameter of base circle = 150mm, lift = 18mm, Diameter of cam shaft = 25mm

Diameter of follower shaft = 20mm, Diameter of roller = 32mm, Dwell period = 180

Type of follower motion = SHM (during ascent and descent)

Procedure:

Cam analysis system consists of cam roller follower, pull rod and guide of pull rod.

- Set the cam at 0° and note down the projected length of the pull rod.
- Rotate the cam through 10° and note down the projected length length of the pull rod above the guide.
- Calculate the lift by subtracting each reading with the initial reading.

Jump – Speed:

1. The cam is run at gradually increasing speeds, and the speed at which the follower jumps off is observed.
2. This jump – speed is observed for different loads on the follower.

Tabulation:

1. Cam profile

S.No	Type of Follower	Angle of rotation	Lift in mm	Lift + base circle radius (mm)
1				
2				
3				
4				
5				
6				
7				
8				
9				
10				
11				
12				

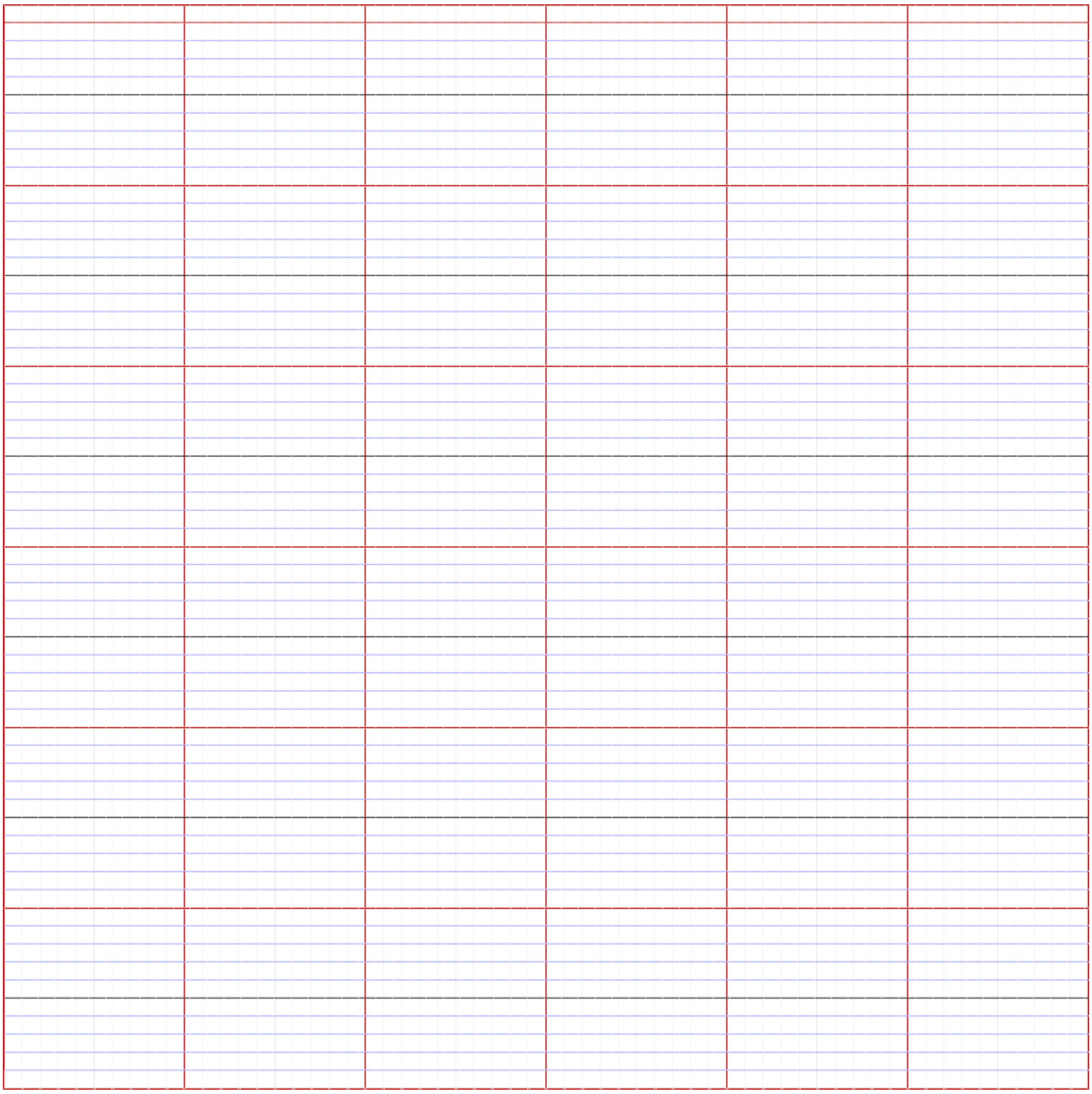
2. Jump - Speed

S.No	Load on the follower, F(N)	Jump – Speed (RPM)
1		
2		
3		
4		
5		
6		
7		
8		
9		
10		
11		
12		

Graph:

Displacement diagram and also the cam profile is drawn using a polar graph chart.

The force Vs Jump – speed curve is drawn.



Result:

Thus the profile of given cam analysis system was studied.

BALANCING OF ROTATING MASSES

Exp. No: **Date:**

Aim:

To balance the given rotor system dynamically with the aid of the force polygon and the couple polygon.

Apparatus required:

1. Rotor system,
2. Weights,
3. Steel rule.

Procedure:

- Fix the unbalanced masses as per the given conditions: radius, angular position and plane of masses.
- Find out the balancing masses and angular positions using force polygon and couple polygon.
- Fix the balancing masses (calculated masses) at the respective radius and angular position.
- Run the system at certain speeds and check that the balancing is done effectively.
- If the rotor system rotates smoothly, without considerable vibrations, means the system is dynamically balanced.

Tabulation

S.NO	Planes of mass	Mass m, kg	Radius r, m	C. force/ω2 mr, kg-m	Distance from ref. plane I, m	Couple / ω2 mrl, kg-m2

Diagrams:

1. Plane of the masses
2. Angular position of the masses
3. Force polygon
4. Couple polygon

Result:

Thus the given rotor system has been dynamically balanced with the aid of force polygon and couple polygon.

BALANCING OF RECIPROCATING MASSES

Exp. No: **Date:**

Aim:

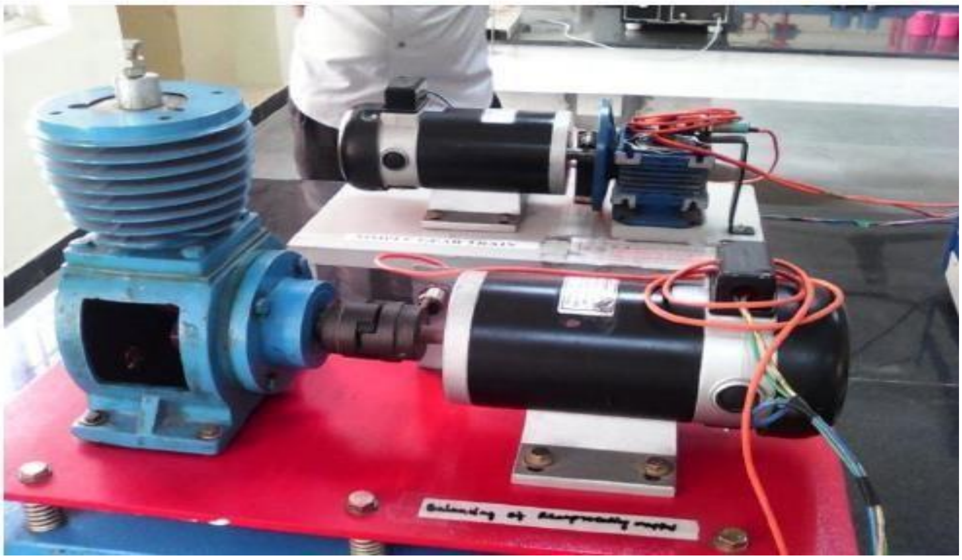
To balance the given rotor system dynamically with the aid of the force polygon and the couple polygon.

Apparatus required:

1. Rotor system,

2. Weights,

3. Steel rule, etc.



Procedure:

- ❖ Initially remove all weights, bolt from the system.
- ❖ Start the motor, give different speeds, observe vibration on the system, note down the speed.
- ❖ Repeat it for different speeds, note them down.
- ❖ Add some weights on piston top, either eccentric or co axial, start the motor, fix at earlier tested speed.
- ❖ If the vibrations are observed, one of the following has to be done to remove the unbalance

– Either remove some of the weights from piston, run at tested speed and observe.
– Add weights in opposite direction of crank, run and observe vibrations at tested speed.
– Combination of both the above.

Tabulation:

S.NO	Crank speed N (rpm)	Mass (gms)			Angular velocity ω
		M1	M2	β= m1+ m2	

β= m1+ m2 N

ω= 2πN/60 rad/sec

Result:

Thus the given rotor system was dynamically balanced with the aid of the force polygon and the couple polygon.

TRANSVERSE VIBRATION OF CANTILEVER BEAM

Exp. No: **Date:**

Aim:

To find the natural frequency of transverse vibration of the cantilever beam.

Apparatus required:

Displacement measuring system (strain gauge) and weights

Description:

Strain gauge is bound on the beam in the form of a bridge. One end of the beam is fixed and the other end is hanging free for keeping the weights to find the natural frequency while applying the load on the beam. This displacement causes strain gauge bridge to give the output in mill – volts. Reading of the digital indicator will be in mm.

Formulae used:

1. Natural frequency = $1/2\pi\sqrt{(g/\delta)}$ Hz

   Where g = acceleration due to gravity in m/s^2 and δ = deflection in m.

2. Theoretical deflection $\delta = WL^3 / 3EL$

   Where W = applied load in Newton, L = length of the beam in mm

   E = young's modules of material in N/mm^2, I = moment of inertia in $mm^4 = bh^3/12$

3. Experimental stiffness = W/δ N – mm and theoretical stiffness = $W/\delta = 3EI/I^3$ N/mm

Procedure:

- Connect the sensors to instrument using connection cable.
- Plug the main cord to 230v/50hz supply.
- Switch on the instrument.
- Keep the switch in the read position and turn the potentiometer till displays reads "0".
- Keep the switch at cal position and turn the potentiometer till display reads 5.
- Keep the switch again in read position and ensure at the display shows "0".
- Apply the load gradually in grams.
- Read the deflection in mm.

Graph:

1. Draw the characteristics curves of load Vs displacement, natural frequency.
2. Draw the characteristic curves of displacement Vs natural frequency.

Observation:

Cantilever beam dimensions:

Length = 30cm,

Breadth = 6.5cm

Height = 0.4cm

Tabulation:

S.NO	Applied mass m (kg)	Deflection δ (mm)	Theoretical deflection δ_T (mm)	Experimental stiffness k (N/mm)	Theoretical stiffness k (N/mm)	Natural frequency f_n (Hz)

Result:

Thus the natural frequency of transverse vibration of the cantilever beam was determined.

TRANSVERSE VIBRATION OF FIXED BEAM

Exp. No: **Date:**

Aim:

To study the transverse vibration of a simply supported beam subjected to central or offset concentrated load or uniformly distributed load.

Apparatus Required:

1. Trunnion bearing,
2. Beams
3. Weights

Procedure:

1. Fix the beam into the slots of trunnion bearings and tighten.
2. Add the concentrated load centrally or offset, or uniformly distributed.
3. Determine the deflection of the beam for various weights added.

Formulae used:

Deflection at the center $\delta_T = WL^3/48EI$ for central concentrated load.
Deflection at the load point $\delta_T = Wa^2b^2/3EI$ for offset concentrated load.
Deflection at the center, $\delta_T = 5WL^4/384EI$ for uniformly distributed load.
Where $I = bd^3/12$, b = width of the beam, d = depth of the beam. L = length of the beam.
Natural frequency of transverse vibration, $f_n = 1/1/2\pi\sqrt{(g/\delta)}$ Hz
Where g = acceleration due to gravity in m/s^2 and δ = deflection in m.

Observation: b =

d =

I =

E =

Tabulation

S.NO	Mass added m, kg	Experimental deflection δ, m	Theoretical deflection δ_T, m	Theoretical nat. freq. f_n, Hz	Experimental stiffness K, N/m	Theoretical stiffness K, N/m

Graphs:

1. Deflection Vs. load (N) from this get stiffness (graph)
2. Deflection Vs. Natural frequency
3. Load in N Vs. Natural frequency

Stiffness experimental, K = load/deflection = $W/\delta = mg/\delta$ N/mm

Stiffness theoretical, K = $W/\delta T = 48EI/I^3$ for center load,

$= 3EI/a^2b^2$ for offset load

$= 384EI/5I^3$ for uniformly distributed load,

Result:

Thus the transverse vibrations of a simply supported beam subjected to central or offset concentrated load or uniformly distributed load was observed.

VIBRATING TABLE SETUP

Exp. No: **Date:**

Aim:

To determine the transmissibility of forced vibrations and to analyze all types of vibrations with its frequency and amplitude.

Specification:

Mass of beam = 7.2 kg

Stiffness of spring = 180 N/m

Procedure:

- Attach the vibrating recorder at suitable position with the pen hidden slightly pressing the paper.
- Attach the damp unit to the stud.
- Start the motor and set required speed and start the recorder motor.
- The vibrations are recorded to the vibration recorder.
- At the resonance speed, the amplitude of the vibration may be recorded as merged over one another.
- Hold the system and max speed little more than the reasonable speed.
- Analyze the recorder frequency and amplitude for both damped and undamped forced vibrations.

Transmissibility = F_{TR}/F

$$= \frac{\text{max force of bars}}{\text{Max impressed force}}$$

$F_{TR} = S \times X_{max}$

S = 180 N/m

$F = mr\omega^2$

Result:

Thus the transmissibility of forced vibrations was determined.

TRIFILAR SUSPENSION

Exp. No: **Date:**

Aim:

To determine the radius of gyration of the circular plate and hence its mass moment of inertia.

Apparatus required:

1. Main frame, chucks 6mm diameter,
2. Circular plate,
3. Strings,
4. Stop watch.

Procedure:

1. Hang the plate from chucks with three strings of equal lengths at equal angular intervals (120° each)
2. Give the plate a small twist about its polar axis.
3. Measure the time taken for 5 or 10 oscillations.
4. Repeat the experiment by changing the length of strings and adding weights.

Formulae used:

Time period, T = t/N, Natural frequency, fn = I/T Hz

Radius of gyration, K = (bT/2JI) √(g/l) m.

Where
b – distance of a string from center of gravity of the plate, L –

Length of the string from chuck to plate surface,

Moment of inertia of the plate only, $Ip = (R^2 \times W_1) / (4\pi^2 fn^2 \times l)$

Moment of inertia with weight added, $I_t = R^2 \times (W_1 + W) / (4\pi^2 fn^2 \times l)$

Where

R – Radius of the circular plate
W_1 – Weight of the circular plate = m_1g in N, m_1 = 3.5 kg
W – Weight of the added masses – mg in N
Moment of inertia of weight, $I_w = I_t – I_p$

Observations:

Type of suspension:____________, No. of oscillations_________________,

Radius of circular plate, R -____________, mass of the plate, m_1____________kg

Tabulation

S. NO	Length of string l, m	Added mass m, kg	Time for N oscillations t, sec	Time period T, sec	Radius of gyration, K, m	Natural frequency fn, Hz	M.I of weight I_w, kgm

Graphs:

Weight added Vs Radius of Gyration

Weight added Vs Moment of inertia

Result:

Thus the radius of gyration of the plate and moment of inertia of the weights were determined and tabulated.

DETERMINATION OF WHIRLING SPEED OF SHAFTS

Exp. No: **Date:**

Aim:

To determine the whirling speed for various diameter shafts experimentally and compare it with the theoretical values
.

Apparatus Required:

1) Shaft – 3 nos.
2) Digital tachometer
3) Chuck key
4) AC voltage regulator

Description of the setup:

The apparatus is used to study the whirling phenomenon of shafts. This consists of a frame in which the driving motor and fixing blocks are fixed. A special design is provided to clear out the effects of bearings of motor spindle from those of testing shafts.

Procedure:

- The shaft is to be mounted with the end condition as simply supported.
- The speed of rotation of the shaft is gradually increased.
- When the shaft vibrates violent in fundamental mode (I mode), the speed is noted down.
- The above procedure is repeated for the remaining shafts.

Observation:

Young's modulus, E (for steel) = 2.06 x 1011 , N / m^2

Young's modulus, E (for copper) = 1.23 x 1011 , N / m^2

Length of the shaft, L = , m

Shaft 1 (steel) Shaft 2 (copper) Shaft 3(steel)

m1 = 0.0584 kg m^2 = 0.16496 kg m^3 = 0.16051 kg

d1 = 0.0031 m, d2 = 0.00484 m, d3 = 0.00511 m

L_1 = m, L_2 = m, L_3 = m

Formulae:

Theoretical whirling speed, Nctheo = {0.4985 / [sqrt (δs / 1.27)] } x 60 , rpm Static deflection due to mass of the shaft (UDL), δs = (5wL4) / (384 EI) Where,

w = weight of the shaft per metre , N/m

L = Length of the shaft, m

E = Young's modulus for the shaft material, N/m2

I = Mass moment of inertia of the shaft

= (π / 64) d^4 , m4

Tabulation:

S.No.	Diameter of shaft (m)	Mass moment of inertia of the shaft , I (m4) x 10-12	Weight of the shaft per m, w (N/m)	Whirling speed (rpm)	
				Nc exp	Ne theo
1.					
2.					
3.					

Result:

The whirling speed for various diameter shafts are determined experimentally and verified with the theoretical values.

GOVERNORS (SPEED CONTROL DEVICE)

Function of Governors

The function of a governor is to governor is to automatically maintain the speed of an engine with in specified limits whenever there is a variation load.

Thus the governor keeps the speed of the engine within certain limits by regulating the fuel supply as per load requirements.

Types of Governors

Main types of governors is

- **SIMPLE CONICAL GOVERNOR (or) WATT GOVERNOR**
- **PORTER GOVERNOR**
- **PROELL GOVERNOR**
- **HARTNELL GOVERNOR**

DETERMINATION OF RANGE SENSITIVITY AND GOVERNOR EFFORT OF WATT GOVERNOR

Exp. No: **Date:**

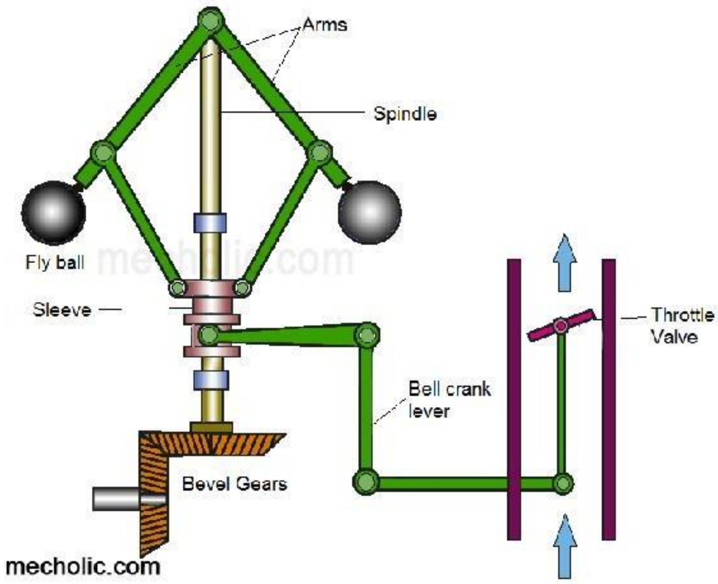
Aim:

To determine the range sensitivity and governor effect of a watt governor.

Apparatus Required:

1. Tachometer

2. Governor Arrangement.



Watt Governor

Technical loading data:

r_o= mm, h_o = mm , L= mm , W= kg.

Procedure:

- Mount the required governor over the spindle and tighten nut over the top
- Switch on the electricity supply the arms slowly rotate. When speed increases the displacement also increases.
- Note down the speed in different sleeve length.

Tabulation

S.No	Displacement X (mm)	Speed, N (rpm)	Angular velocity ω (rad/s)	Height of Governor h (mm)	Angle of inclination of the arm α (degree)	Range of Sensitivity r (mm)	Governor Effect F (kg)

Formula Used:

Angular Velocity = 2πN/ 60 rad/sec

Height of Governor (h)= ho-(x/2) (ho given value)

Angle of inclination of the arm α= $\cos^{-1}$ (h/L) h (given value)

Range of sensitivity = r= 50 + L sin α (mm)

Governor Effort = F = W/9.81×ω^2 × r (kg)

Graph:

Result

Thus the range sensitivity and governor effort is successfully calculated for watt governor.

DETERMINATION OF RANGE SENSITIVITY AND GOVERNOR EFFORT OF PORTER GOVERNOR

Exp. No: **Date:**

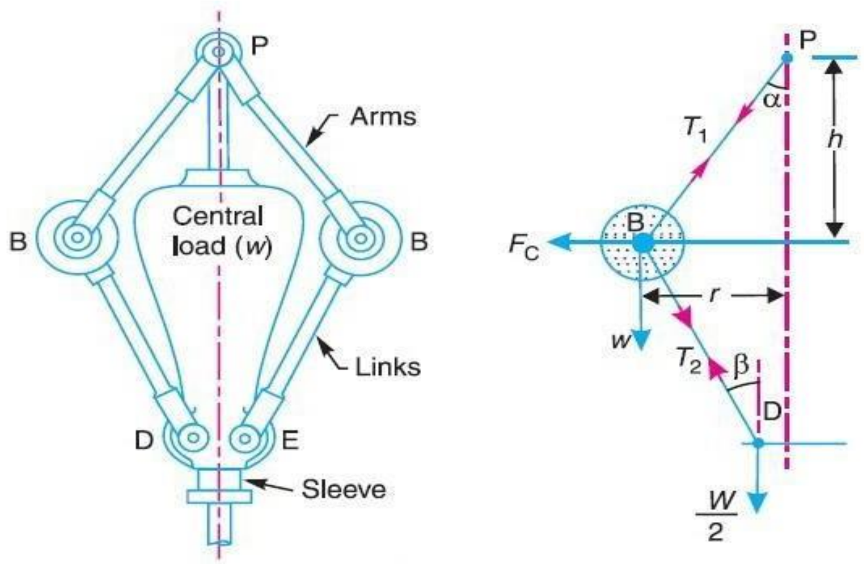
Aim:

To determine the range sensitivity and governor effect of a porter governor.

Apparatus Required:

1. Tachometer

2. Governor Arrangement.



Porter governor.

Technical loading data:

r_o= mm, h_o = mm , L= mm , W= kg.

Procedure:

- Mount the required governor over the spindle and tighten nut over the top
- Switch on the electricity supply the arms slowly rotate. When speed increases the displacement also increases.
- Note down the speed in different sleeve length.

Tabulation

S.No	Displacement X (mm)	Speed, N (rpm)	Angular velocity ω (rad/s)	Height of Governor h (mm)	Angle of inclination of the arm α (degree)	Range of Sensitivity r (mm)	Governor Effect F (kg)

Formula Used:

Angular Velocity = $2\pi N/60$ rad/sec

Height of Governor (h)= ho-(x/2) (ho given value)

Angle of inclination of the arm $\alpha = \cos^{-1}(h/L)$ h (given value)

Range of sensitivity = $r = 50 + L \sin \alpha$ (mm)

Governor Effort = $F = W/9.81 \times \omega^2 \times r$ (kg)

Graph:

Result:

Thus the range sensitivity and governor effort is successfully calculated for porter governor.

DETERMINATION OF RANGE SENSITIVITY AND GOVERNOR EFFORT OF PROELL GOVERNOR

Exp. No: **Date:**

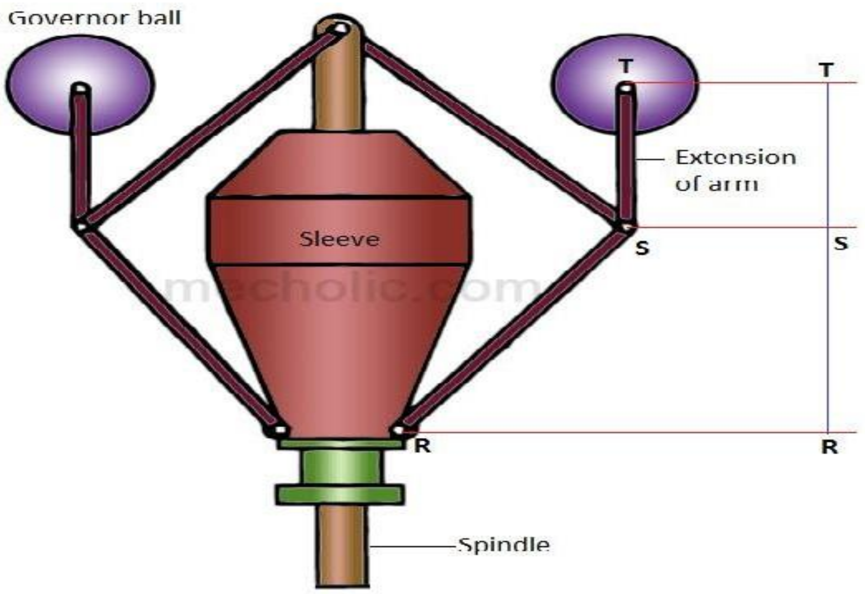
Aim:

To determine the range sensitivity and governor effect of a proell governor.

Apparatus Required:

1. Tachometer

2. Governor Arrangement.



Proell Governor

Technical loading data:

r_o= mm, h_o = mm , L= mm , W= kg.

Procedure:

- ❖ Mount the required governor over the spindle and tighten nut over the top
- ❖ Switch on the electricity supply the arms slowly rotate. When speed increases the displacement also increases.
- ❖ Note down the speed in different sleeve length.

Tabulation

S.No	Displacement X (mm)	Speed, N (rpm)	Angular velocity ω (rad/s)	Height of Governor h (mm)	Angle of inclination of the arm α (degree)	Range of Sensitivity r (mm)	Governor Effect F (kg)

Formula Used:

Angular Velocity = $2\pi N/ 60$ rad/sec

Height of Governor (h)= ho-(x/2) (ho given value)

Angle of inclination of the arm $\alpha = \cos^{-1}$ (h/L) h (given value)

Range of sensitivity = r= 50 + L sin α (mm)

Governor Effort = F = $W/9.81 \times \omega^2 \times r$ (kg)

Graph:

Result:

Thus the range sensitivity and governor effort is successfully calculated for proell governor.

Tabulation

S.No	Displacement X (mm)	Speed, N (rpm)	Angular velocity ω (rad/s)	Height of Governor h (mm)	Angle of inclination of the arm α (degree)	Range of Sensitivity r (mm)	Governor Effect F (kg)

Formula Used:

Angular Velocity = 2πN/ 60 rad/sec

Height of Governor (h)= ho-(x/2) (ho given value)

Angle of inclination of the arm α= $\cos^{-1}$ (h/L) h (given value)

Range of sensitivity = r= 50 + L sin α (mm)

Governor Effort = F = W/9.81×ω^2 × r (kg)

Graph:

Result:

Thus the range sensitivity and governor effort is successfully calculated for hartnell governor.