ST.ANNE'S COLLEGE OF ENGINEERING AND TECHNOLOGY

(Approved by AICTE, New Delhi. Affiliated to Anna University, Chennai)
(An ISO 9001: 2015 Certified Institution)
ANGUCHETTYPALAYAM, PANRUTI – 607 106.

DEPARTMENT OF MECHANICAL ENGINEERING

ME 8511-KINEMATICS AND DYNAMICS LABORATORY

THIRD YEAR – FIFTH SEMESTER

PREPARED BY

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DEPARTMENT OF MECHANICAL ENGINEERING

III YEAR / V SEMESTER

ME 8511–KINEMATICS AND DYNAMICS LABORATORY

COMPLETED ON:

EXP NO: 1 **VERIFICATION OF GRASHOF'S LAW USING**
DATE: **SLIDER CRANK MECHANISM**

AIM To verify Grashof's law using slider crank mechanism.

APPARATUS REQUIRED Slider crank mechanism set up and scale.

PROCEDURE

A slider crank mechanism is a modification of the basic four bar chain. It is usually found in reciprocating engine. Rotary motion can be converted into reciprocating motion and vice versa. In the given apparatus, the link 1 and 2, link 2 and 3, link 3 and 4 form a sliding pair. The link 1 corresponds to the frame which is fixed. The link 2 corresponds to the connecting rod and link 4 corresponds to the cross bead. As the crank rotates the cross head reciprocates in the guides and thus the piston reciprocates inside the cylinder.

TABULATION

S.No	Angular displacement (Degree)	linear displacement (mm)	S.No	Angular displacement (Degree)	linear displacement (mm)
1	**30**		**7**	**210**	
2	**60**		**8**	**240**	
3	**90**		**9**	**270**	
4	**120**		**10**	**300**	
5	**150**		**11**	**330**	
6	**180**		**12**	**360**	

GRAPH Angular Displacement Vs Linear Displacement

RESULT

Thus the Grashof's law has been verified using slider crank mechanism.

EXP NO: 2 **VERIFICATION OF GRASHOF'S LAW USING**
DATE : **CRANK ROCKER MECHANISM**

AIM To verify Grashof's law using crank Rocker mechanism.

APPARATUS REQUIRED Crank rocker mechanism set up and scale

PROCEDURE

- A crank rocker mechanism is a mechanism of a beam engine which consists of four links.
- Rotary motion can be converted into reciprocating motion and vice versa.
- In the given apparatus, when the crank rotates about the fixed centre, the lever oscillates about the fixed centre D.
- The end of the lever is connected to a piston rod which reciprocates due to the vibration of the crank.

TABULATION

S.No	Angular displacement (Degree)	linear displacement (mm)	S.No	Angular displacement (Degree)	linear displacement (mm)
1	**30**		**7**	**210**	
2	**60**		**8**	**240**	
3	**90**		**9**	**270**	
4	**120**		**10**	**300**	
5	**150**		**11**	**330**	
6	**180**		**12**	**360**	

GRAPH Angular Displacement Vs Linear Displacement

RESULT

Thus the Grashof's law has been verified using crank Rocker mechanism.

EXP NO: 3 **DETERMINATION OF MOMENT OF INERTIA OF**

DATE: **FLYWHEEL TURNTABLE APPARATUS**

AIM To find the moment of inertia of a circular fly wheel turn table apparatus.

APPARATUS REQUIRED **1.** Stop - clock and Turn table apparatus

FORMULAE USED Energy possessed by turn table i.e. flywheel.

N = rev/time taken in m/s **$\frac{1}{2} I\omega^2 = mgh$**

ω = 2 π N in rad/sec ω- Angular velocity in rad / sec

I = 2mgh/ ω in kgm^2 h - Height of fall m - Weight of falling load

I – Moment of inertia of turn table in kgm^2

THEORY

The apparatus consists of a circular M.S wheel fixed on the cylindrical column. The column is located on a footstep bearing with thrust support as located for specimen. The apparatus consists of a circular M.S flywheel.

PROCEDURE

- Allow the weight to fall through a height (h) this gaining kinetic energy.
- Assuming the law of conservation of energy induced in the flywheel is equal to the energy lost by the falling weight passed by the turn table.
- Thus, I of the turn table can be obtained by sub standing the measured quantity.

TABULATION

S. No	Hanging mass in **kg** (m)	No of revolutions	Speed in **m/s**	Height of falling in **m** **(h)**	Time taken in **sec**	Angular velocity in **rad/sec**	Moment of inertia in kgm^2
1	0.5			0.5			
2	1			0.5			

RESULT Thus, the MOI of the given circular M.S flywheel is obtained.

EXP NO: 4 **DETERMINATION OF RADIUS OF GYRATION BY**

DATE: **USING BI-FILAR SUSPENSION**

AIM To determine the radius of gyration of a body using bi-filar suspension.

APPARATUS 1. String 2. Weight bar 3. Disc 4. Steel rule

PROCEDURE

- Attach the bi-filar suspension strings to the books at top beam of the frame.
- Fix the weights required over the beam of bi-filer.
- Oscillate the system about vertical axis passing through the center of beam.
- Measure the time required for 10 oscillations.
- Repeat the procedure by changing the length of suspension.

CALCULATIONS For bi-filar suspension

$Fn = 1/2\pi \times b/k. \sqrt{g/l}$ Where, Fn = frequency of oscillations

B = distance of string from centre of gravity=0.15m

L = length of strings K = radius of gyrations.

Therefore, $k= b/2\pi.fn\ (g/l)$ ^0.5

Now, Temp = Time for 10 oscillations /10 sec.

Fn = 1/temp thus, value of ‘k’ can be determined.

TABULATION

S. NO	*LENGTH OF STRING in m (L)*	*WEIGHT In kg (W)*	*TIME FOR 10 OSCILLATION’S ‘t’ SEC*		*RADIUS OF GYRATION (m)*	*FREQUENCY OF OSCILLATION (Fn)*
			10sec	*1sec*		
1						
2						
3						

RESULT

The value of radius of gyration is found using bi-filer suspension is 1.20m.

TABULATION

S. NO	*END POSITION*	*TIME FOR 'n' OSCILLATIONS 'T' SEC*	*PERIODIC TIME (tp) In sec*	*MOMENT OF INERTIA*	*MEAN MOMENT OF INERTIA*
1					
2					
3					

OBSERVATION

	CONNECTING ROD 1	*CONNECTING ROD 2*
L		
M		
D1		
D2		
n		

EXP NO : 5 **DETERMINATION OF MOMENT OF INERTIA BY**
DATE : **OSCILLATION FLYWHEEL AND CONNECTING ROD**

AIM

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

APPARATUS REQUIRED Stop Watch and Vernier Scale

PROCEDURE

1. Measure the centre to centre distance of connecting rod. Also measure inner dial of both side connecting rod and measure the weight of connecting rod.
2. Attach small end of the connecting rod to the shaft. Give oscillation to the connecting rod.
3. Measure the time for five oscillation and calculate the time period (tp_1). Remove the connecting rod from the shaft and again attach the big end of the connecting rod to the shaft.
4. Again measure the time for five oscillation and calculate the periodic time (tp_2). Calculate the moment of inertia of connecting rod.
5. Repeat the procedure for the times and take mean of it. Attach flywheel to the other side of the shaft and repeat the same procedure as above and see the effect of it on the oscillations of the connecting rod.

FORMULA

Moment of inertia = mk^2 $k^2 = h(Le - h)$ m = mass of the connecting rod

k = radius of gyration Le = Equivalent length of simple pendulum

Therefore $k^2 = h1(L1 - h1) = h2(L2 - h2)$

L1 = Length of equivalent simple pendulum from the top of small end.

$L1 = g(tp_1 / 2\pi)^2$ $L2 = g(tp_2 / 2\pi)^2$ $h = (D_1 / 2) + L + (D_2 / 2)$

RESULT

Thus the moment of inertia was determine by oscillation of flywheel and connecting rod ----------------kgm^2

EXP NO : 6 **DETERMINATION OF NATURAL FREQUENCY**

DATE : **BY USING COMPOUND PENDULUM METHOD**

AIM

To find the natural frequency of oscillations of the composite body 2 mm MS bar 50 cm long at CG of the frequency body.

APPARATUS REQUIRED Stop Watch and Vernier Scale

PROCEDURE

1. Initially the given bar is set at O position and then moved to 2 cm to calculate 10 Oscillations of time.
2. The MS bar is then set to 4 cm to calculate the 10 oscillations of time.
3. The distance from the point of suspension to measure the centre of gravity.
4. The same procedure is repeated suspending the pendulum and time is noted for Particular number of oscillation.

FORMULA Frequency of Oscillation = 1/ Time period x 60 (Hertz)

TABULATION

S.NO	Deflection in **mm**	Time taken for 10 Oscillation in **sec**	Time taken for one Oscillation in **sec**	Frequency of oscillation in **Hz**
1	10			
2	20			
3	30			
4	40			

RESULT

The Frequency of oscillation of compound pendulum is found out.

EXP NO: 7 DETERMINATIONS OF GYROSCOPIC COUPLE

DATE:

AIM To verify the gyroscopic couple of the given motorized gyroscope experimentally.

APPARATUS REQUIRED 1. Motorized Gyroscopic set up 2. Tachometer 3. Regulator and Stop watch

PROCEDURE

Balance the rotor along the horizontal plane. The motor is started and by rotating the dimmer stat the required speed is attained. Weight is added to the pan and stopwatch is started. The time is noted in seconds regularly for precession. The procedure 3 & 4 are repeated for 45 & 60 degrees by increasing weights.

SPECIFICATIONS

Diameter of rotor (D) = 0.3 m, Weight of rotor (W_{r}) = 6.13 Kg

Distance from weight pan to rotor (L) = 0.194 m

FORMULA

Actual Gyroscopic couple $T_{act} = I\varpi\varpi_p$ in N-m

Where moment of Inertia of the disc $I = \frac{W_R}{g} \times \frac{D^2}{8}$ in Kg-m^2

Angular velocity of the disc $\varpi = \frac{2\pi N}{60}$ in rad /s

Angular velocity of Precession of Yoke $\varpi_p = \frac{d\theta}{dt}$ in rad / s

Theoretical Gyroscopic couple $T_{theo} = WL$ in Nm

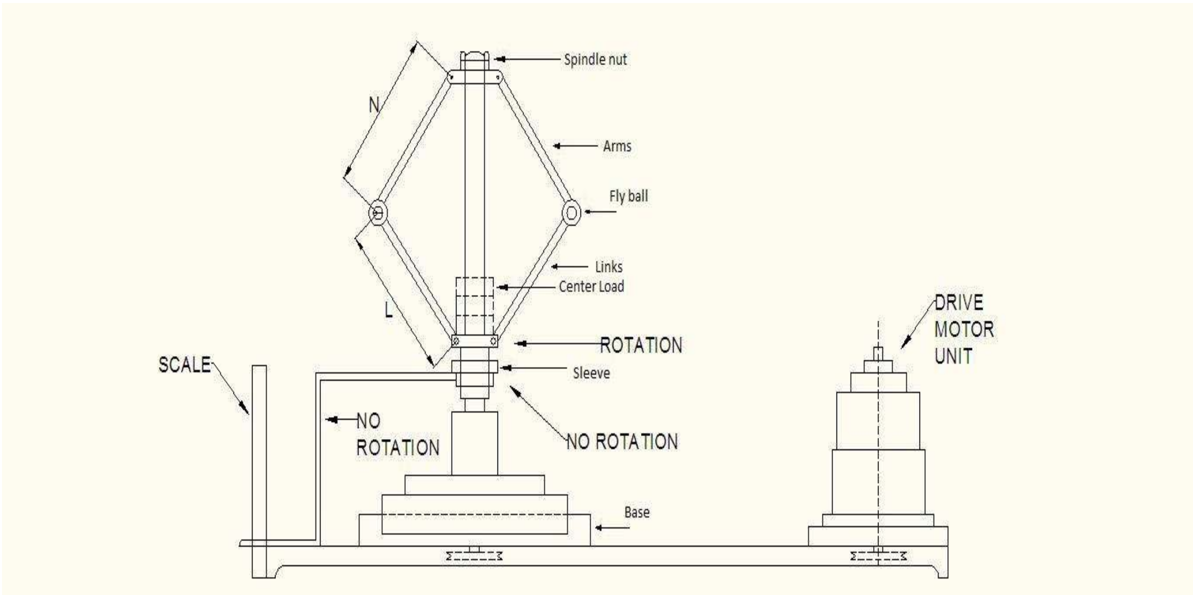
TABULATION

S. No	Speed N (rpm)	Weight W (Kg)	Angle of precession dθ (rad)		Time taken for 30 degree (secs)	Angular velocity of disc ω (rad /s)	Angular Velocity of Precession of Yoke ω_P in rad /s	Gyroscopic Couple in Nm	
			deg	rad				T_{theo} In Nm	T_{act} In Nm
1		0.5	30						
2		0.5	30						
3		0.5	30						

RESULT

The gyroscopic couple of the given motorized gyroscope through various angle of precession is verified experimentally.

PORTER GOVERNOR



EXP NO: 8 CHARACTERISTICS OF PORTER GOVERNOR

DATE:

AIM To determine the effort, sensitivity and to draw the characteristics curves of the Porter Governor.

APPARATUS REQUIRED

1. Tachometer 2. Measuring tape 3.Porter arm set up 4.Sleeve weights.

PROCEDURE

The control unit is switched on and the speed control knob is slowly rotated to increase the governor speed. When center sleeve aligns with 1cm in graduated scale, the speed of spindle is recorded. The governor speed is increased in steps to give the corresponding steps in the scale and the speeds are noted down. The radius of rotation for corresponding sleeve displacement is measured directly. ***Then the following graphs are drawn:*** 1.Force Vs Radius of rotation 2. Speed of rotation Vs sleeve displacement 3. sleeve displacement Vs Radius of rotation

SPECIFICATION Mass of each ball (m) = 0.35 kg

Mass of the sleeve (m) = 2.25 kg

FORMULA

Force $F=m\omega^2 r$ in N; where m is the weight of each ball in N, ω is the angular velocity in rad/s is given by $\omega = \frac{2\pi N}{60}$ and BE (r) = $\sqrt{AB^2 - (AE - X \div 2)^2}$ in (AB = 0.235m & AE = 0.2m) r is the radius of rotation in m. Mean speed $N = (N_{max} + N_{min} \div 2)$ in rpm.

TABULATION

S. NO	Sleeve displacement in m (x)	Speed of rotation in rpm (N)			Radius of rotation in m (r)	Force in N (F)	Sensitivity (s)	Effort (e)
		Nmin	Nmax	Mean speed				
1	0.02							
2	0.04							
3	0.06							

RESULT The characteristics curves of the Porter governor are drawn.

EXP NO : 9 CHARACTERICTICS OF HARTNELL GOVERNOR

DATE :

AIM To draw the characteristics curves of the Hartnell governor.

APPARATUS REQUIRED

1. Tachometer 2. Measuring tape 3.Hartnell arm set up 4. Sleeve weights

PROCEDURE

The control unit is switched on and the speed control knob is slowly rotated to increase the governor speed. When center sleeve aligns with 1cm in graduated scale, the speed of spindle is recorded. The governor speed is increased in steps to give the corresponding steps in the scale and the speeds are noted down. The radius of rotation for corresponding sleeve displacement is measured directly. Then the following graphs are drawn: **Force Vs Radius of rotation Speed Vs Displacement.**

SPECIFICATION

Weight of each ball m = 0.35 kg and w = 0.35*9.81

FORMULA

X=0.169m, Centrifugal Force Fc =$m\omega^2 r$ in N k=1600N/m r =0.180m,
Y=0.150m, where m is the weight of each ball in N. ω is the angular velocity in rad/s is given by $\omega = \frac{2\pi N}{60}$ and r is the radius of rotation r = (H*X ÷ Y) + r in m, spring force s =H * k Frictional force Fr = (2xFc ÷ y) – (w + s) in N

TABULATION

S.No	Speed in rpm (N)	Lift in m (H)	Radius of rotation in m	Centrifugal force in N (Fc)	Spring force in m (s)	Frictional force in N
1		0.02				
2		0.04				
3		0.06				

RESULT The characteristics curves of the Hartnell governor are drawn.

EXP NO : 10 **CAM ANALYSIS**

DATE :

AIM To analyses the working of the given cam and followed by applying.

APPARATUS REQUIRED 1. cam eccentric 2. follower – roller

PROCEDURE

1. Fix the required cam to the cam shaft and the required follower to push rod.
2. Set angular scale at required piston. Adjust the weight and dial gauge.
3. Rotate the cam by hand and move down the dial-gauge reading at every 30° intervals. Remove the dial –gauge and switch on power supply slowly increase motor speed.
4. Repeat the procedure for different weight and spring tension configuration at different cam follower configurations.

TABULATION

S.NO	CAM ANGLE IN DEGREE	F*D	S.NO	CAM ANGLE IN DEGREE	F*D
1	30		7	210	
2	60		8	240	
3	90		9	270	
4	120		10	300	
5	150		11	330	
6	180		12	360	

FORMULA

FOLLOWER DISPLACEMENT (F.D) = (M.S.R × L.C) + S.S.R

RESULT

Thus the analysis of the cam has been performed and the follower displacement has been recorded with respect to the rotation of cam.

EXP NO: 11 SINGLE DEGREE OF FREEDOM SPRING MASS SYSTEM

DATE :

AIM To determine the natural frequency of spring mass system in undamped condition.

INTRODUCTION

Spring mass system is a setup used to determine the experimental frequency. The body whose frequency is to be determined is suspended by a springs. When the body is moved through a small distance along a vertical axis through the centre of gravity, it will be accelerate in a vertical plane. Then by taking the following readings with the single mass system we can determine the frequency of a body.

PROCEDURE

Fix the top bracket at the side of the scale and insert one end of the spring on the hook. At the bottom of the spring fix the other plat form. Note down the reading corresponding to the plat form. Add the weight and observe the change in deflection. With this determine spring stiffness. Add the weight and make the spring to oscillate for 10 times. Note the corresponding time taken for 10 oscillations and calculate time period. From the time period calculate experimental natural frequency. Calculate the damping factor and damping co-efficient

TABULATION **COLUMN FOR OSCILLATIONS**

S. no	Mass added (M) In kg	Deflection (δ) in Cm	Stiffness of spring K=(m *9.81)/ δ in N/cm	S. no	Mass added (M) In kg	No of oscillations	Time Taken in sec
1				1			
2				2			
3				3			

FORMULA(UNDAMPED CONDITION)

1. Natural Frequency = $f_n = 1/t_p$ 2. Theoretical frequency $F_n = (1/2\pi) \times \sqrt{K/M \times 9.81}$

RESULT Thus the natural frequency of spring mass system in undamped condition is determined.

EXP NO : 12 TO DETERMINE NATURAL FREQUENCY OF

DATE : TORSIONAL VIBRATION IN DOUBLE ROTORS SYSTEM

AIM

To determine the period and frequency of Torsional vibration of the double rotor system experimentally and compare it with the theoretical values

APPARATUS REQUIRED

1. Shaft and Spanner 2. Chuck key and Stop Watch 3. Measuring Tape
4. Weights and cross arms

DESCRIPTION OF THE SETUP

Two discs having different mass moment of inertia are clamped one at each end of shaft by means of collet. Mass moment of inertia of any disc can be changed by attaching the cross lever with weights. Both discs are free to oscillate in the ball bearings. This provides negligible damping during experiment

FORMULAE

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

Where, tm = mean time taken for n oscillations n = number of oscillations = 5

Theoretical period of vibration , T theo = 2π { sqrt [$(I_A\ I_B) / Kt(I_A + I_B)$] }, sec

Moment of inertia of disc A, $I_A = m_A\ (D_A^2 / 8)$, Nms^2

Moment of inertia of disc B, $I_B = m_B\ (D_B^2 / 8)$, Nms^2

Torsional stiffness(K_t) = (G Ip) / L in Nm G = modulus of rigidity of the shaft in N/m^2

L = length of the shaft between discs in m. d = shaft diameter in m.

Experimental frequency of vibration, F exp = 1 / T exp , Hz

Theoretical frequency of vibration, F theo = 1 / T theo , Hz

PROCEDURE

1) Fix the discs A and B to the shaft and fit the shaft in bearing.

2) Deflect the discs A and B in opposite directions by hand and release.

3) Note down the time required for n = 5 oscillations.

4) Fit the cross arm to the disc A and attach equal masses to the ends of cross arm and again note down time.

5) Repeat the above procedure with different equal masses attached to the ends of cross arm.

OBSERVATION

Diameter of the disc A , DA = 250 , mm

Diameter of the disc B , DB = 250 , mm

Mass of the disc A, mA = 3.3, kgf

Mass of the disc B, mB = 1.74, kgf

Modulus of rigidity of the shaft, G = 0.35×10^{11}, N/m2

Shaft diameter, d = ----- mm

Length of the shaft between discs, L = ---- m

Mass of the cross arms with bolts and nuts = 0.725

TABULATION

S.No	Length Of Shaft	No. Of Oscillation	Time	T_{th}	T_{exp}	F_{th}	F_{exp}
1.							
2.							
3.							
4.							

RESULT

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

F_{theo} ------------ Hz F_{exp} ------------ Hz .

EXP NO: 13 DETERMINATION OF WHIRLING SPEED OF THE SHAFT
DATE:

AIM

To determine the critical speed for various diameter shafts experimentally and verify it theoretically

APPARATUS REQUIRED

1. Tachometer 2. Spanner 3. Shafts – 3 No's.

PROCEDURE

Start the equipment by switching ON the button. The speed of rotation of the shaft is increased above the I mode of vibration and decreased slowly. The speed at which maximum vibration (I mode) occurs is noted down. The above procedure is repeated for the remaining shafts.

SPECIFICATION

Density = 7200kg/m²
Constant for simply supported K = 1.27
(Based on end condition, I mode)

Young's Modulus (E) = 200×10^9 N / m²
Length of the shaft (L) = 90 cm
Diameter of the shafts(d) = 4.28 mm, 7.5mm, 12.7 mm

FORMULA

Mass volume of shaft (M) = $\rho \times A \times L$ in kg/m $A = \pi/4 \times d^2$ in m²

Considering the shaft us An UDL (w) = Mg/L in N. $\delta = (WL^4/EI) \times (5/384)$

Critical speed (f) = $29.91/\sqrt{\delta}/1.27$ in rpm $I = \pi/64 \times d^4$ in m^4

TABULATION

S.No	Diameter of shaft in mm	Critical speed in rpm
1		
2		
3		

RESULT The critical speed of the given shafts is experimentally determined and verified theoretical

EXP NO: 14 DYNAMIC BALANCING OF ROTATING MASSES

DATE:

AIM

To determine theoretically the masses to be added in two reference planes to balance the rotating masses in other two planes and to verify experimentally the balancing of the system using dynamic balancing machine.

APPARATUS REQUIRED

1. Dynamic balancing machine 2. Spanner. 3.Measuring tape and masses.

DESCRIPTION

The dynamic balancing machine is a vertically framed structures suspended on two chains which are in turn connected to a main frame. The frame carrying a shaft on two beams at the ends and carrying four adjustable discs A, B, C and D as the four planes, two of which the balancing masses are to be added.

PROCEDURE

- The given problem is graphically represented by a line diagram. In this diagram the distance between the masses of the disc are represented.
- The couple polygon is drawn.
- The force polygon is drawn and the masses are calculated..
- The corresponding distances are calculated.
- Disconnect the drive.

TABULATION

PLANE	MASS (KG)	RADIUS (CM)	CENTRIFUGAL FORCE (KG-M)	DISTANCE FROM REFERENCE PLANE (CM)	COUPLE ($KG\text{-}M^2$)
X					
A					
Y					
B					

RESULT

Thus the given shaft was dynamically balanced with given masses.

EXP NO: 15 TRANSVERSE VIBRATION CANTILEVER BEAM

DATE:

AIM To study the transverse vibrations of a cantilever beam

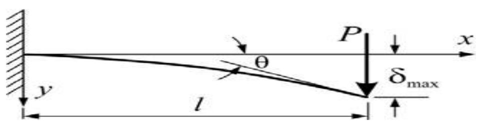
APPARATUS REQUIRED

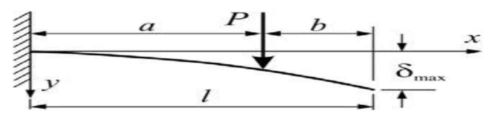
1. Trunnion bearings 2. beams 3. weights

FORMULAE USED

1. Natural frequency = $1/2\pi\sqrt{(g/\delta)}$ Hz **Where**, δ = deflection in m.
2. Theoretical deflection g= acceleration due to gravity in m/s^2

a. concentrated load P at the Free end ***b. concentrated load P at any point***





$\delta_T = Wl^3/3EI$ $\delta_T = [Wa^2 / 6EI] (3l - a)$

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

I= moment of inertia in $mm^4 = bh^3/12$ Mild steel δ = 5.10 mm

3. Experimental stiffness = W/δ N-mm Theoretical stiffness = $W/\delta = 3EI/l^3$ N/mm

PROCEDURE

Fix the beam into the slots of trunnion bearings and tighten. Add the concentrated uniformly distributed. Determine the deflection of the beam for various weights added.

OBSERVATION (Cantilever beam dimensions)

Length=750mm, Breadth= 25mm, Height=4mm

TABULATION

S No	Applied mass m (kg)	Deflection δ (mm)	Theoretical deflection δ_T (mm)	Experimental Stiffness k_e (N/mm)	Theoretical Stiffness k_t (N/mm)	Natural Frequency fn (Hz)

GRAPH 1. Deflection Vs. load (N) 2. Deflection Vs. Natural frequency 3. Load in Vs. natural frequency

RESULT Thus the transverse vibrations of a cantilever beam.

EXP NO: 16 TRANSVERSE VIBRATION - SIMPLY SUPPORTED BEAM

DATE:

AIM

To study the transverse vibrations of a simply supported beam subjected to uniformly distributed load.

APPARATUS REQUIRED

1. Beams 2. Weights 3. Magnetic Stand 4. Dial Gauge.

FORMULAE USED

Defection at the center, $\delta_T = 5wl^4/384EI$ for uniformly distributed load.

$I = bd^3/12$ b = width of the beam, d = depth of the beam, l = length of the beam. **b = 25mm, d = 4mm, l = 1m, Mass of the each weight (m) = 200 gm**

Natural frequency of transverse vibrations, $f_n = 1/2\pi\sqrt{(g/\delta)}$ **Hz**

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

STIFFNESS EXPERIMENTAL

K = load/deflection

$= W/\delta = mg/\delta$ in N/mm

STIFFNESS THEORETICAL

K =384EI/5l^3 for uniformly distributed load

PROCEDURE

Fix the beam into the slots of bearings and tighten. Add the concentrated uniformly distributed. Determine the deflection of the beam for various weights added.

TABULAR COLUMN

S No	Mass added (m) in kg	Experimental Deflection (δ) in m	Theoretical Deflection (δ_T) in m	Theoretical Natural frequency (f_n) in Hz	Experimental Stiffness(K_e) in N/m	Theoretical Stiffness (K_t) in N/m

GRAPH 1. Deflection Vs. load (N) 2. Deflection Vs. Natural frequency 3. Load in Vs. natural frequency

RESULT

Thus the transverse vibrations of a simply supported beam subjected to uniformly distributed load.

EXP NO: 17 DETERMINATION OF NATURAL FREQUENCY

DATE: USING VIBRATING TABLE

AIM

To find natural frequency of free vibration and forced vibration using vibration table.

APPARATUS REQUIRED

1. Spring 2. Mass 3. Damper 4. Stopwatch 5. Steel rule

FORMULA USED

Forced frequency fn = N / T Hz   N - No of oscillation

T - Time period of 5 oscillation in sec

TECHNICAL SPECIFICATIONS

Mass of the Beam	=	1.150 kg
Total length of the beam (L)	=	950 mm
Mass of the Exciter (M)	=	2 kg
Exciter position from one trunnion end (L1)	=	
Stiffness of the spring	=	180 N /m

PROCEDURE *(FORCED VIBRATION)*

Fit the spring, mass damper in proper position note down the spring stiffness mass of the beam, Length of the beam from one trunnion point and measure the exciter mass. The electrical motor is switched ON, using stop watch note down oscillation time for small jerk. Then repeat the procedure for different length of the beam.

TABULATION *(FORCED VIBRATION)*

S No	Vibration	Exciter position	No. of oscillation	Time period (T) in sec	Frequency (Hz)	Speed (rpm)
1						
2						
3						

RESULT Thus the natural frequency of free and forced vibration using vibrating table was found.

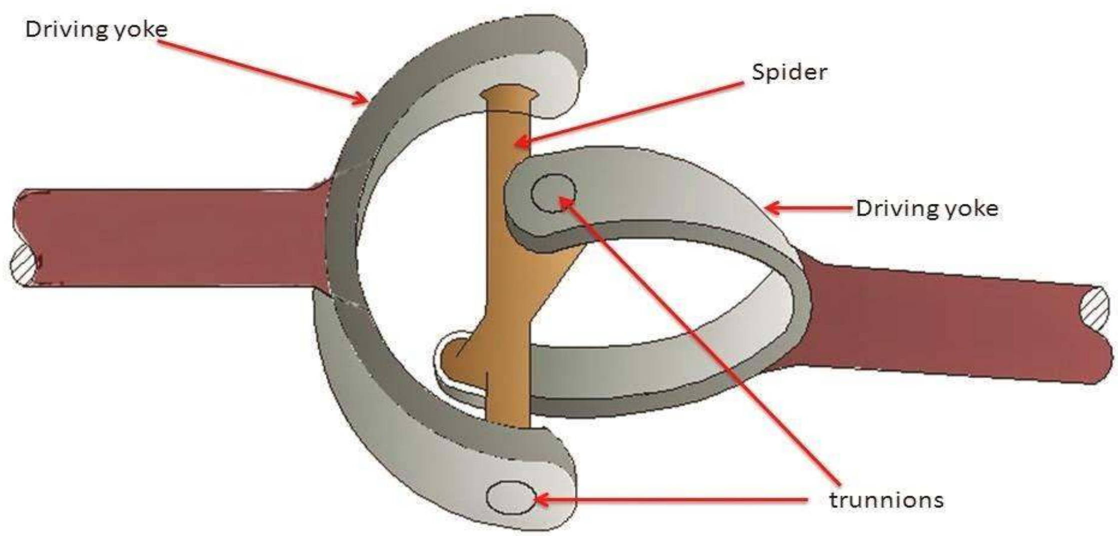
STUDY OF UNIVERSAL JOINT

AIM

The board model of universal joint is useful to demonstrate the working of universal joint used mainly in automobile.

INTRODUCTION

The arrangement of connecting two parts with each other for completing specific motion is called joint. The joint is used to connect two shafts which are intersecting at small angles with each other is called universal joint. It is also called as hooks joint. The common application of universal joint is found in the transmission from the gear box to the differential or back axle of automobile. It is also used for transmission of power to different spindles of multiple drilling machines. It is also used as the knee joint in the milling machines



Universal joint

DESCRIPTION

The apparatus consists of two assemblies one is single universal joint and the other is double universal joint. A u-joint (universal joint) is basically a flexible pivot point that transmits power through rotational motion between two shafts not in a straight line. The u-joint needs to be flexible to compensate for changes in drive line angle due to the constantly changing terrain under the vehicle. The u-joint is considered to be one of the oldest of all flexible couplings. It is commonly known for its use on automobiles and trucks. A universal joint in its simplest form consists of two shaft yokes at right angles to each other and a four point cross which connects the yokes. The cross rides inside the bearing cap assemblies, which are pressed into the yoke eyes. One of the problems inherent in the design of a u-joint is that the angular velocities of the components vary over a single rotation.

RESULT

Thus the working of gear model and gear trains has been studied.

STUDY OF GEAR MODEL AND GEAR TRAINS

AIM

The gear model and gear train is used to demonstrate the function of different types of gears and gears trains.

INTRODUCTION

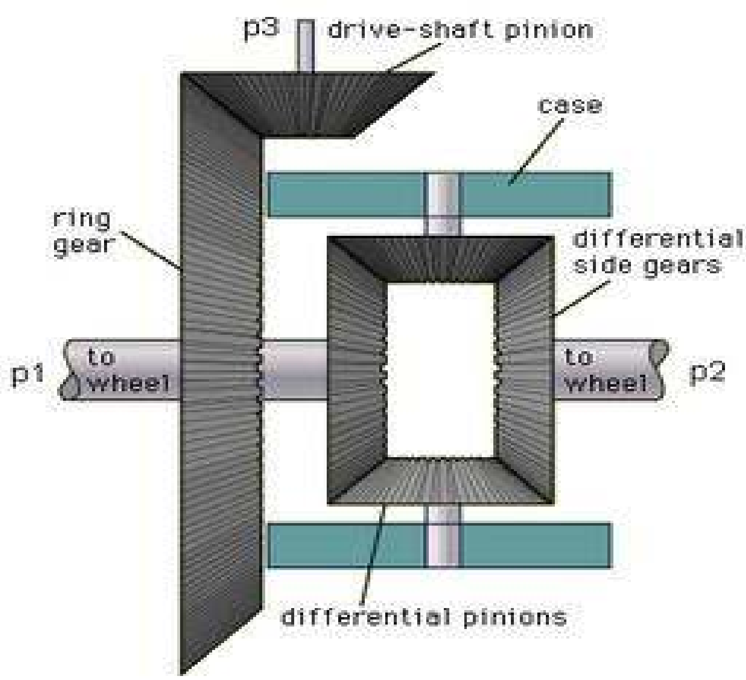
When two or more gears are made to mesh with each other to transmit power from one shaft to another then such combination is called as gear trains. The nature of the gear train used depends upon the velocity ratio required and relative position of the axis of the shaft. The gear train may consists of spur, bevel and helical or spiral gears.

DESCRIPTION

The board model consists of following types of gear trains

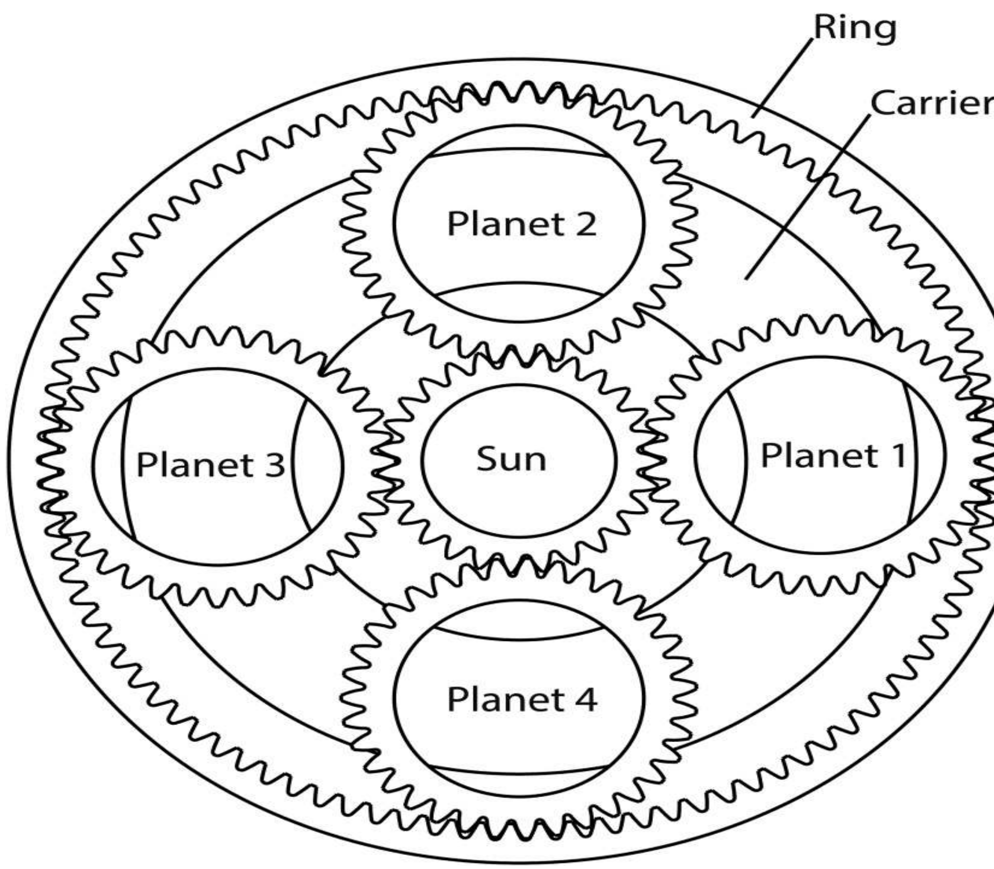
1) Simple gear train
2) Compound gear train
3) Differential gear train
4) Epicyclic gear train

DIFFERENTIAL GEAR TRAIN



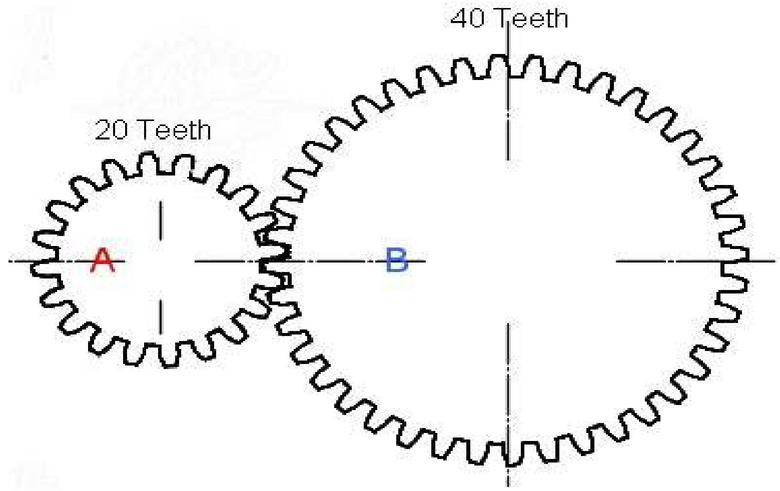
When the axes of the first gear and the last gear are coaxial then the gear train is known as reverted gear train. In this board model differential gear assembly is mounted on board. In which all the gears are bevel type. Two shaft can rotates in exactly opposite in direction with the handle provided on it. The motion from one shaft to another shaft is transmitted through four small equal level gears known as pinions. These pinions can rotate freely on the crossed arm. The reverted gear train is used in automotive transmission lathe back gear assembly in industrial speed reducers

EPICYCLIC GEAR TRAIN



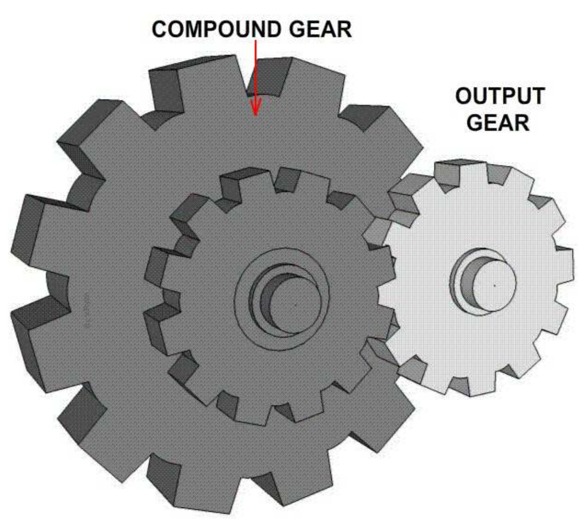
In the Epicyclic gear train there is relative motion between two or more of the axes of the wheels constituting the train. the wheels are usually carried on arm or link pivoted about a fixed center and itself capable of rotating . it is also called as planetary gear train . Epicyclic gear train is divided into two parts, simple Epicyclic gear train and compound Epicyclic gear train.

SIMPLE GEAR TRAIN



A gear train is a power transmission system consisting of gears and shafts only. Gear trains have four functions. Some gear trains increase or decrease the rotational power or speed of a shaft in a mechanism. A gear train is two or more gear working together by meshing their teeth and turning each other in a system to generate power and speed. It reduces speed and increases torque. To create large gear ratio, gears are connected together to form gear trains. The main limitation of a simple gear train is that the maximum speed change ratio is 10:1. For larger ratio, large size of gear trains is required; this may result in an imbalance of strength and wear capacities of the end gears.

COMPOUND GEAR TRAIN

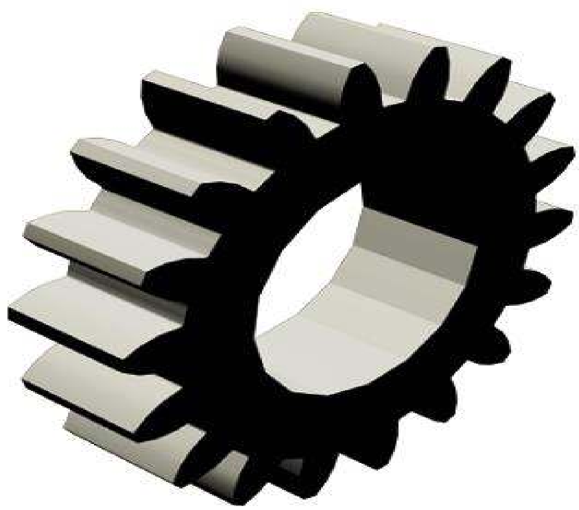


Compound gear trains have two or more pairs of gears in mesh, so that they rotate together. This compound gear train has gears on three shafts. The gear on the input shaft meshes with a larger gear on a counter-shaft or cluster gear. The counter-shaft has a smaller gear formed on it, in mesh with the output shaft gear. The motion of the input is transferred through the large gear, along the counter-shaft to the smaller gear, to the output. The output turns in the same direction as the input, but at a reduced ratio, depending on the relative sizes of the gears.

CLASSIFICATION OF GEARS

SPUR GEAR

Spur gears or straight-cut gears are the simplest type of gear. They consist of a cylinder or disk with the teeth projecting radially, and although they are not straight-sided in form, the edge of each tooth is straight and aligned parallel to the axis of rotation. These gears can be meshed together correctly only if they are fitted to parallel shafts.



HELICAL GEAR

Helical or "dry fixed" gears offer a refinement over spur gears. The leading edges of the teeth are not parallel to the axis of rotation, but are set at an angle. The former refers to when the shafts are parallel to each other; this is the most common orientation. In the latter, the shafts are non-parallel, and in this configuration the gears are sometimes known as "skew gears". The angled teeth

engage more gradually than do spur gear teeth, causing them to run more smoothly and quietly.

BEVEL GEAR

A bevel gear is shaped like a right circular cone with most of its tip cut off. When two bevel gears mesh, their imaginary vertices must occupy the same point. Their shaft axes also intersect at this point, forming an arbitrary non-straight angle between the shafts. The angle between the shafts can be anything except zero or 180 degrees. Bevel gears with equal numbers of teeth and shaft axes at 90 degrees are called miter gears.

WORM GEAR

Worm gears resemble screws. A worm gear is usually meshed with a spur gear or a helical gear, which is called the gear, wheel, or worm wheel. Worm-

and-gear sets are a simple and compact way to achieve a high torque, low speed gear ratio. For example, helical gears are normally limited to gear ratios of less than 10:1 while worm-and-gear sets vary from 10:1 to 500:1. A disadvantage is the potential for considerable sliding action, leading to low efficiency. Worm gears can be considered a species of helical gear, but its helix angle is usually somewhat large (close to 90 degrees) and its body is usually fairly long in the axial direction; and it is these attributes which give it screw like qualities.

RESULT

Thus the working of gear model and gear trains has been studied.