Mission and vision of the Department

Vision of Mechanical Department

To establish the state of the art learning center in Mechanical Engineering which will impart global competence, enterprising skills, professional attitude and human values in the student.

Mission of Mechanical Department

1. To impart quality technical education to the students.
2. To develop comprehensive competence in the students through various modes of learning.
3. To enable students for higher studies and competitive examinations.
4. To facilitate students and industry professionals for continuous improvement and innovation.

Program Educational Objectives:

[1] Use core competence acquired in various areas of Mechanical Engineering to solve techno-managerial issues for creating innovative products that lead to better livelihoods & economy of resources.

[2] To establish themselves as effective collaborators and innovators to address technical, managerial and social challenges.

[3] To equip students for their professional development through lifelong learning and career advancement along with organizational growth.

[4] Serve as a driving force for proactive change in industry, society and nation.

PROGRAM   SPECIFIC OUTCOMES

Student should have

1) An ability to work professionally in mechanical systems including design, analysis, production, measurement and quality control.

2) An ability to work on diverse disciplinary tasks including manufacturing, materials, thermal, automobile, robotics, mechatronics, engineering software tools, automation and computational fluid dynamics.

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EXPERIMENT NO. :-01

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TITLE: - To generate involute tooth profile with help of rack on gear blank

AIM: - Generation of involute tooth profile using involute gear blank

PROCEDURE:-

1) Paper cut in circular form is fixed on circular disc having graduation on it.

2) Set the lower rack scale to zero position

3) By pencil mark out position of rack on paper

4) Rotate the disc by one graduation & again draw rack on paper.

5) Follow same procedure for number of time

6) Finally we will get tooth profile on paper.
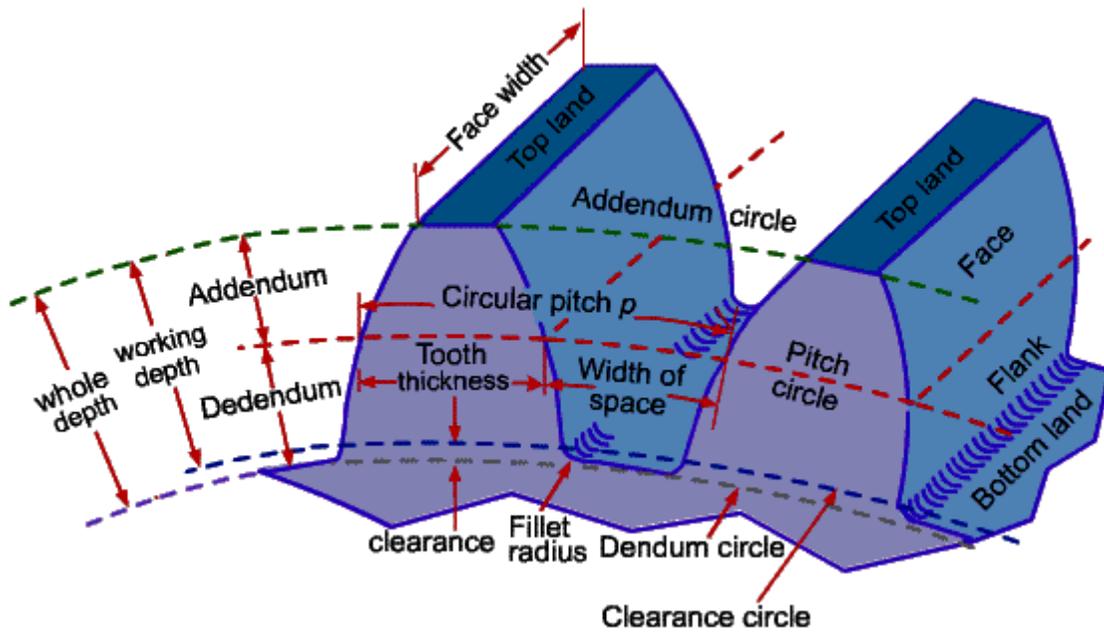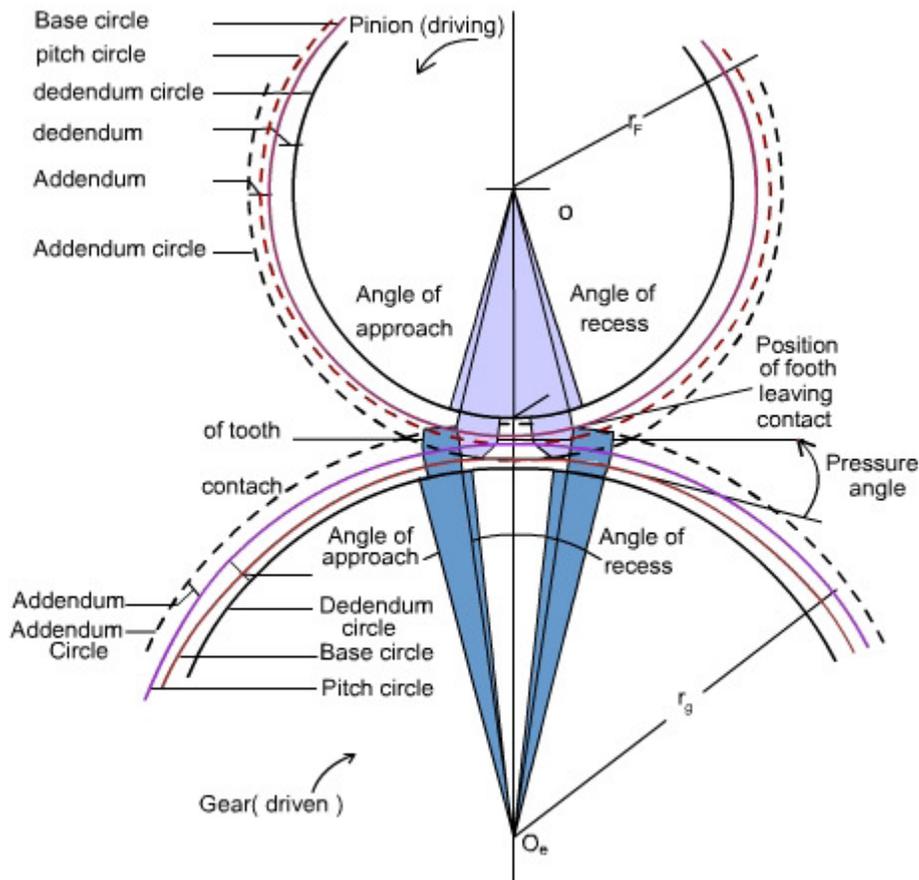
- Pitch surface: The surface of the imaginary rolling cylinder (cone, etc.) that replaces the toothed gear.
- Pitch surface: The surface of the imaginary rolling cylinder (cone, etc.) that replaces the toothed gear.
- Pitch circle: A normal section of the pitch surface.
- Addendum circle: A circle bounding the ends of the teeth, in a normal section of the gear.
- Dedendum circle or Root circle: The circle bounding the spaces between the teeth, in a normal section of the gear.
- Addendum: The radial distance between the pitch circle and the addendum circle.
- Dedendum: The radial distance between the pitch circle and the root circle.
- Clearance: The difference between the Dedendum of one gear and the addendum of the mating gear.
- Face of a tooth: That part of the tooth surface lying outside the pitch surface.
- Flank of a tooth: The part of the tooth surface lying inside the pitch surface.
- Top land: The top surface of a gear tooth.
- Bottom land: The bottom surface of the tooth space.

- Circular thickness (tooth thickness): The thickness of the tooth measured on the pitch circle. It is the length of an arc and not the length of a straight line.

- Tooth space: The space between successive teeth.
- Width of space: The distance between adjacent teeth measured on the pitch circle.
- Backlash: The difference between the tooth thickness of one gear and the tooth space of the mating gear.
- Circular pitch p: The width of a tooth and a space, measured on the pitch circle. It is equal to the pitch circumference divided by the number of teeth. If,

  p - circular pitch P - diametral pitch
  z - number of teeth d - pitch diameter
- Diametral pitch P: The number of teeth of a gear per unit pitch diameter. The diametral pitch is hence the number of teeth divided by the pitch diameter.
$$P = z/d$$
- The product of the diametral pitch and the circular pitch equals π.
$$pP = \pi$$

- Pinion: The smallest of any pair of mating gears.
- Gear: The largest of the pair is called simply the gear.

- Velocity ratio i: The ratio of rotational speed of the driving gear n_1 (or input gear) to the rotational speed of the driven gear n_2 (or output gear). If z_1 & z_2, d_1 & d_2, r_1 & r_2, and ω_1 & ω_2 are corresponding number of teeth, pitch diameters, radii and angular velocities of pinion and the gear, then

- Base circle: An imaginary circle used in involute gearing to generate the involutes that form the tooth profiles.

- Pitch point: The point of tangency of the pitch circles of a pair of mating gears.
- Common tangent: The line tangent to the pitch circle at the pitch point.
- Line of action: A line normal to a pair of mating tooth profiles at their point of contact.
- Path of contact: The path traced by the contact point of a pair of tooth profiles.
- Pressure angle φ: The angle between the common normal at the point of tooth contact and the common tangent to the pitch circles.
- Pressure angle is also the angle between the line of action and the common tangent

EXPERIMENT NO.02

Title :- Study Of Interference & Under Cutting

Theory :- To study of Interference & Under Cutting in toothed gearing.

"The Phenomenon when the tip of the tooth under cuts the root on it's mating gear is Known as interference."

The radius of addendum of circle of pinion is increased on O_1N the point of the contract L will come or move from L to N when this radius is further increased L will be inside the base of circle of the Wheel. The tip of the tooth will be under cut the tooth of the wheel and remove part of the involutes profile of the tooth on the wheel this effect is known as Interference and occur when the teeth are begin to cut.

If the radius of addendum circle of the wheel increases beyond O_2N. Then the tip of tooth of the wheel will cause interference with the tooth on the pinion. The points M & N are called as interference points.

The interference may be avoided if point of contact between two teeth is always as a involutes profile and both teeth in other word the interference may be prevented if the addendum circle of the two mating gear cut the curve in tangent to the base circle between the point of tangents.

$MP = r \sin \phi$; $PN = R \sin \phi$

Maximum length of contact $= (R + r) \sin \phi$

Are of contact $= (R + r) \sin \phi / \cos \phi = (R + r) \tan \phi$
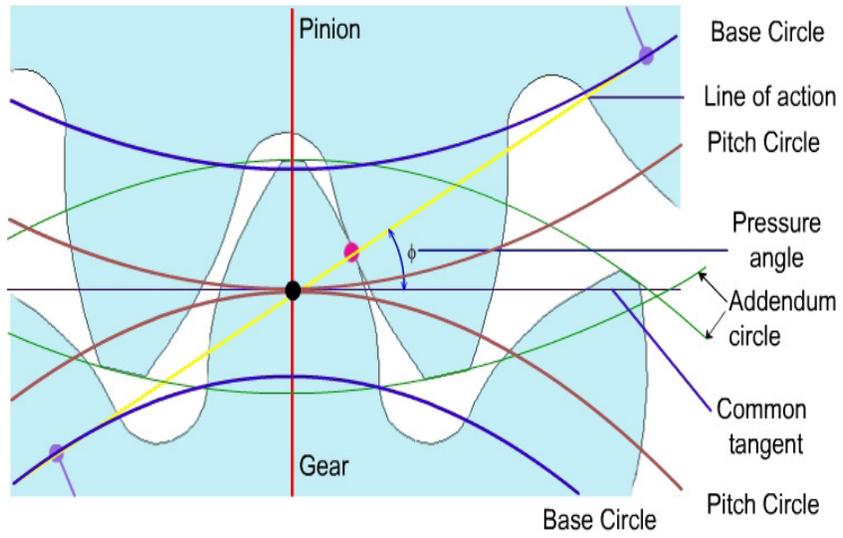
Gear Meshing
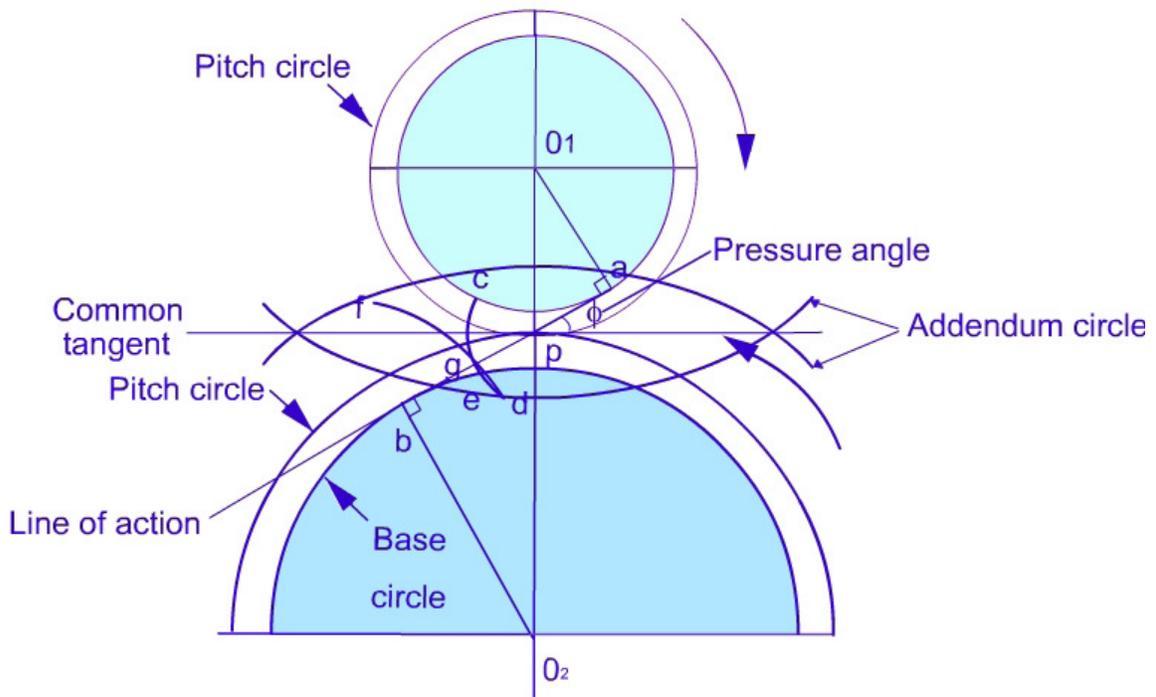
INVOLUTE GEAR TOOTH PROFILE

Minimum number of teeth on pinion in order to avoid interference

In order to avoid interference the addendum circle for both mating gears must cut the common tangent to the base circle between point of tangency.

Let t = Number of teeth on pinion

   T = Number of teeth on wheel.

   M = Module of the teeth.

   R = pitch circle radius of pinion.

   G = Gear ratio = T/t = R/r

   ϕ = Pressure angle or obliquity angle.

For $O_1N\,P$,

$$=(O_1N)^2 = (O_1P)^2 + (PN)^2 - 2\,O_1P\,PN\cos O_1NP$$

$$= r^2 + R^2 - 2r.\,R\sin\phi\ = \qquad \cos(90 + \phi)\quad [PN = O_1P\sin\phi = R\sin\phi]$$

$$= r^2 + R^2 + \sin^2\phi + 2\,r\,R\sin^2\phi\ ]$$

$$= r^2\,[1 + R^2\sin^2\phi\,/\,r^2 + 2R\sin^2\phi/r]$$

$$= r^2\,[1 + R/r\,(R/r + 2)\sin^2\phi\ ]$$

Limiting radius of the pinion addendum circle

$$O_1N = r\sqrt{1 + R/r(R/r + 2)\sin^2\phi}$$

Let ApM = addendum of pinion where Aap is fraction by which the standard addendum of one module for pinion should be multiplied in order to avoid interference.

Addendum of pinion = $O_1\,N - O_1\,P$

$$ApM = mt/2\sqrt{1 + T/t(T/t + 2)\sin^2\phi} - mt/2$$

$$= mt/2\{\sqrt{1 + T/t(T/t + 2)\sin^2\phi} - 1\}$$

$$t = \frac{Ap}{\sqrt{1 + T/t(T/t + 2)\sin^2\phi} - 1}$$

The equation gives the minimum number of teeth required on the Wheel in order to avoid interference.

Shift in center distance — new pitch point

pitch radius of pinion — new larger pitch radius of period — base circle is unchaged

pitch point shift position

presure angle φ=20° — new pitch point

new presure angle φ=23° — new pitch circles

pitch radius of gear — base circle is unchanged

line of action (common noraml) is tangent to both base circles — new larger pitch radius of gear
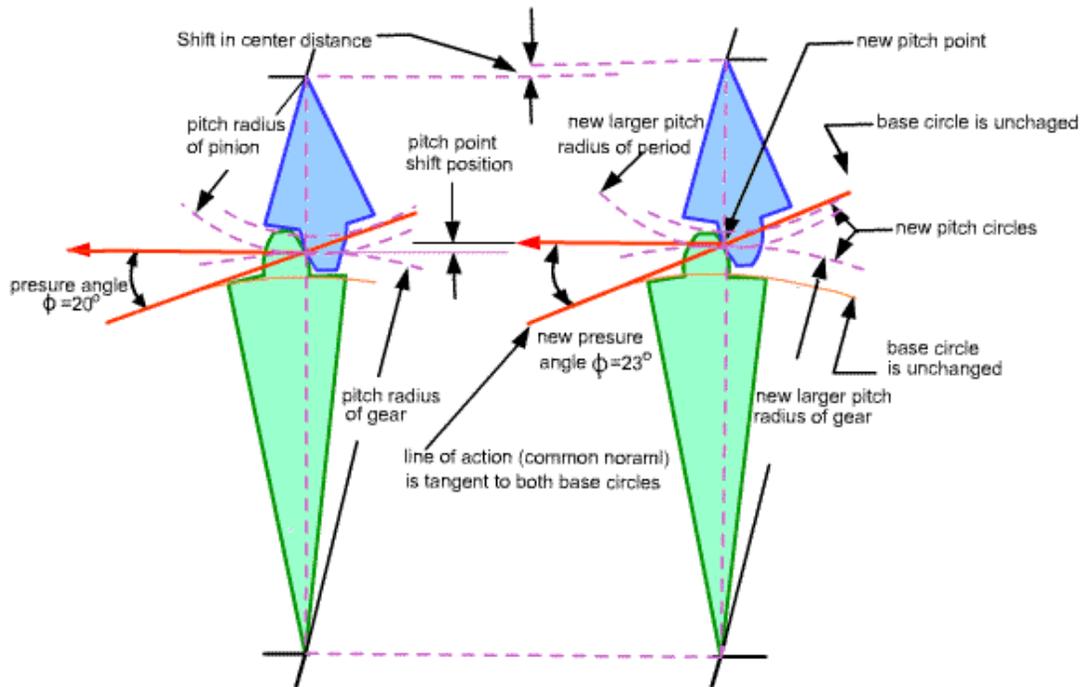
Minimum number of teeth on wheel in order to avoid interference

Let T = Minimum number of teeth required on wheel to avoid interference.

Aw = Addendum of wheel, where A_w is fraction which the standard addendum for wheel should be multiplied.

Let   t = Number of teeth on pinion

T = Number of teeth on wheel.

M = Module of the teeth.

R = pitch circle radius of pinion.

G = Gear ratio = T/t = R/r

φ = Pressure angle or obliquity angle.

For $O_2M\ P$,

$=(O_2M)^2 = (O_2P)^2 + (PM)^2 - 2\ O_2P\ PM\ \cos O_2MP$

$= R^2 + r^2 - 2r.\ R \sin \phi \times \cos (90 + \phi)$ $[PM = O_2P \sin \phi = r \sin \phi]$

$= R^2 + r^2 + \sin^2\phi + 2\ R\ r\ \sin^2\ \phi\ ]$

$= R^2\ [1 + r^2\sin^2\phi\ /\ R^2 + 2r\ \sin^2\phi/R]$

$= R^2\ [1 + r/R\ (r/R + 2)\ \sin^2\phi\ ]$

Limiting radius of the wheel addendum circle

$$O_2M = R\sqrt{1 + r/R(r/R+2)\sin^2 \phi}$$

$$= MT/2\sqrt{1 + t/T(t/T+2)\sin^2 \phi}$$

Addendum of Wheel = $O_2M - O_2P$

$$Aw = MT/2\sqrt{1+t/T(t/T+2)\sin^2\phi} - MT/2$$

$$Aw = MT/2\{\sqrt{1+t/T(t/T+2)\sin^2\phi} - 1\}$$

$$T = \frac{2Aw}{\sqrt{1+t/T(t/T+2)\sin^2\phi} - 1}$$

$$T = \frac{2Aw}{\sqrt{1+1/G(1/G+2)\sin^2\phi} - 1}$$

Undercutting

　　　　Figure shows a pinion. A pinion of its addendum falls inside the base circle. The profile of the tooth inside the base circle. If the addendum of the matting gear is more than the limiting value, it interferes with addendum of the pinion and the two gears are located.

　　　　Now, if instead of the gear mating with pinions a cutting rack having similar teeth is used to cut the teeth in the pinion. The portion of the pinion tooth, which would have interfered with the gear, will be removed as shown in figure.

　　　　A gear having it's material removed in this manner is said to be undercut and have process, undercutting. However, when the actual gear meshes with the undercut pinion no interference occurs.

　　　　Undercutting will not be take place if the teeth are designed to avoid interference.

Experiment No.03

Title:- Study Of Governor

The function of a Governor is totally different to that of a Flywheel. The former is required to control the mean speed of a engine over a period of time as opposed to a flywheel which is used to limit the almost inevitable fluctuations in speed, which occur during one cycle. A good example of this is the single cylinder Four stroke engine. There is only one power stroke to two revolutions of the crankshaft. Without a flywheel, the speed would either fluctuate, within one cycle, by an unacceptable amount or the engine would not work at all since the energy stored in the flywheel is required to carry the engine through to the next Power Stroke.
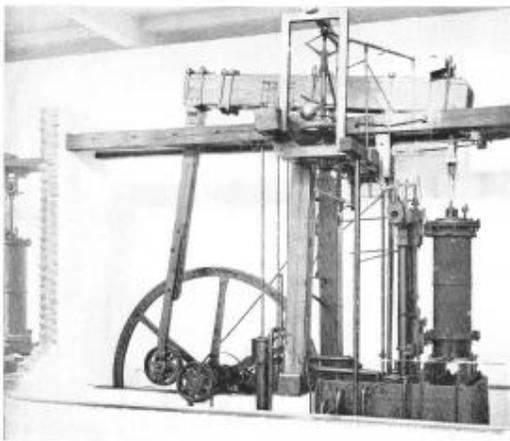
A Boulton and Watt Rotative Engine made in 1797.

The Watt type Governor can be seen at the top of the picture.

A change in load on a engine will almost certainly lead to a change in speed and the **Governor** is required to alter the supply of energy to the engine to bring the speed back to its original value. This is achieved by connecting the rotating parts of the governor, through suitable levers, to a sleeve on its axis of rotation. Any change in the speed causes a change in the position of the rotating parts and consequently to the sleeve and this movement actuates the fuel supply valve (This includes Compressed air; Steam or water) of the particular engine or turbine. This function is of particular importance in A.C. electric generators since it is important to maintain the correct number of cycles per second from the generator who's load may change rapidly and unpredictably.

Generally Governors can be classified as either **Centrifugal** or **Inertia**

Centrifugal Governors

The effort of the governor is obtained from the change in centrifugal force on (usually) two rotating masses known as Balls, when an increase or decrease in governor speed occurs. Two main types can be distinguished.

Dead-weight Governors:

The radius of the ball path is controlled by lever and weights the latter being usually attached to the sleeve as in the (a)"Watt" (b)"Porter" or (c) "Proell" governors.

(a)   (b)   (c)
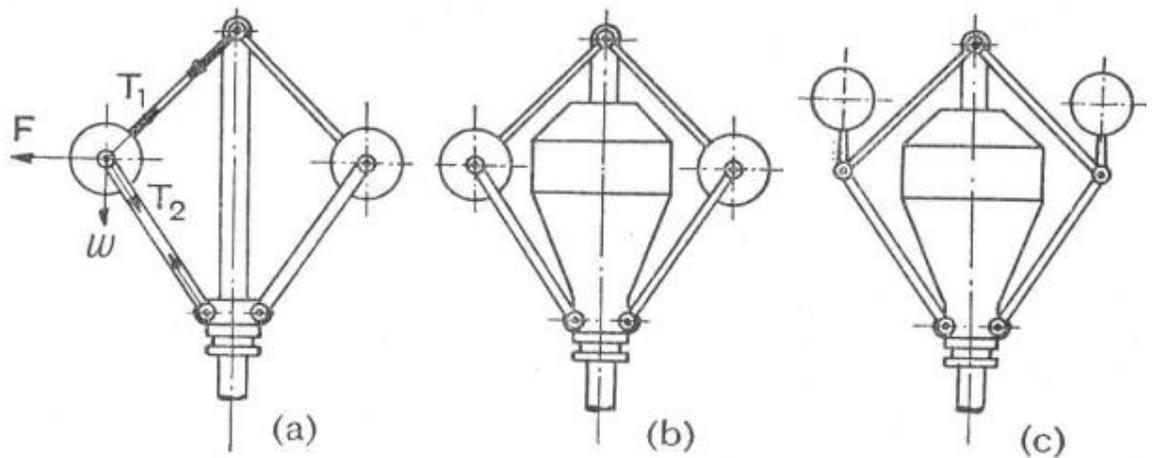
The equilibrium position can usually be determined by considering the forces acting on one ball and the arm or arms to which it is attached. (See Examples 1,2 and 5)

Spring-loaded Governors.:

The balls are controlled by springs acting on them or the sleeve. Three examples are shown in the diagram.
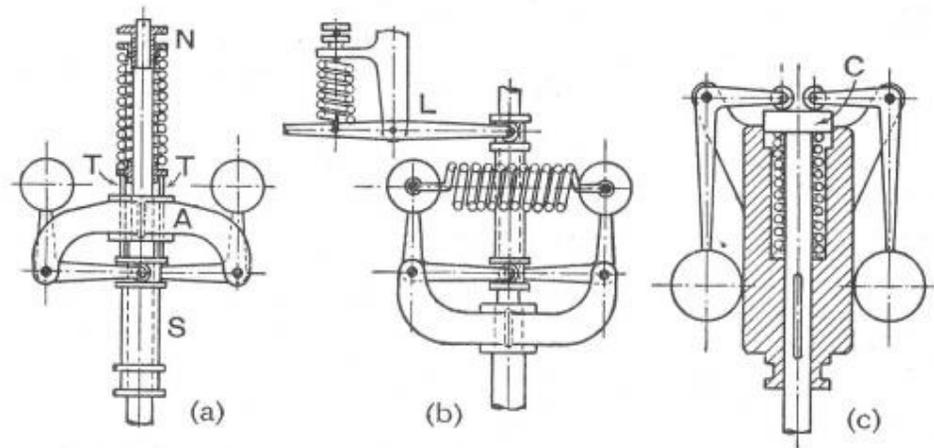
The Hartnell governor (a) is a well known example of this type. Each ball is attached at one end of a bell-crank lever and at the other end to the actuating sleeve. The method of analysis is to take moments about the pivot of the lever (See Examples 6,7 and 8)

Inertia Governors

Inertia Governors work on a different principle. The governor balls are arranged so that the inertia forces caused by angular acceleration or retardation of the governor shaft tend to alter their positions. The amount of the displacement of the balls is controlled by springs and the governor mechanism to alter the supply of energy to the engine.

The advantage of this type of Governor is that the positions of the balls are affected by the rate of change of speed of the governor shaft. Consequently a more rapid response to a change of load is obtained, since the action of the governor is due to acceleration and not to a finite change of speed. The advantage is offset, however by the practical difficulty of arranging for a complete balance of the revolving parts of the governor. For this reason centrifugal governors are much more frequently used.

EXPERIMENT NO.04
Title: - To determine moment of inertia of uniform
rod by using -
- **A. Compound pendulum.**
- **B. Bifilar suspention**.

(A) **Compound Pendulum Method**

Aim: - To determine the mass moment of inertia of a uniform rod by

Using compound pendulum method.

Discription Of Set Up: - the compound pendulum consist of a steel

Bar. The bar is supported by knife-edge. It is possible to change the

Length of suspended pendulum by supporting the bar in different

Holes.

Procedure: -

1. Support the rod in any of the one hole.
2. Note the length of suspended pendulum and determine O to G.
3. Allow the bar to oscillate and determine time (t), by knowing the time for 10 oscillations.
4. Repeat same procedure.

Theory: -

From fig.

Angular acceleration $= \alpha = d^2\theta/dt^2 = \theta$.

Torque T= $I_0\alpha = I_0 \theta$.

Restoring couple = mgx.

The magnitude of torque and restoring couple is same but there direction are opposite.

$I_0\theta = - mgx$.

$I_0\theta + mgx = 0$.

But from fig

$X = L \sin \theta = L\theta$ as θ is very small.

$\theta = \beta \sin \omega t$

$\theta = \beta \omega \cos \omega t$

$\theta = -\beta \omega^2 \sin \omega t$

$\square = -\beta \omega^2 \sin \omega t$

$I_0 \square = - MgL\square\square\square$

$I_0(-\beta \omega^2 \sin \omega t) = - MgL\beta \sin \omega t\square$

$- \omega^2 I_0 = MgL = \quad MgL/ \omega^2$

$\omega = 2\Pi f$ BUT FREQUENCY = No of Oscillation / Time

$I_0 = MgL/ 2\Pi f$

$Ig = Io - ML^2$

Observation -:

1. Mass of rod = m =   gm

2. Length of O to G = L =   cm

3. G = 9.81 M/S^2

Observation Table

Sr,NO	Oscillation	Time

Calculation -:

FREQUENCY = F = No of Oscillation / Time

$$Io = MgL/ (2\Pi f)^2$$

$$Ig = Io - ML^2$$

$$I_G = \quad Kg .M^2$$

Result -: The mass moment of inertia   of uniform rod =   Kg m^2

[B] Bifilar Suspension.

Aim -: To determine mass moment of inertia of uniform rod (Plate) by

   Bifilar suspension method

Description Of Set Up : A uniform rectangular section bar is suspended from pendulum support frame by two parallel cords. Top end of the card passes 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 card by loosening the chucks.

Procedure: 1) Suspended bar from chuck and adjust the length of the cord conveniently. Note the suspension length of the each cord must be same

2) Allow the bar to oscillate about the vertical axis passing through the center and measure the periodic time T by knowing the time for 10 oscillations.

Theory -: Let

T- be the tension in the string.

Arc - $L\theta$

Φ- Angular displacement of the string

θ- Angular displacement of the plate

g- Gravitational acceleration

a- Distance of C.G from string.

I_G – M.M.I of plate about C.G. G

We have $T = Mg/2$

Arc $L\Phi = a\theta$ $\Phi = a\theta/L$

Torque $= I_G\theta$

Restoring couple $= (T\sin\Phi a)2a = T\Phi 2a$ $[\sin\Phi = \Phi]$

$= T(a\theta/L) 2a$

Torque = Restoring couple

$I_G\theta = - T(a\theta/L) 2a$ $\theta = \beta \omega \cos \omega t$

$I_G\theta + T(a\theta/L) 2a = 0$ $\theta = -\beta \omega^2 \sin \omega t$

$- I_G \omega^2 + Mg a^2/L = 0$ $\Box = -\beta \omega^2 \sin \omega t$

$I_G = Mg a^2/L\omega^2$ $[\omega = 2\Pi f]$

Frequency = No. Oscillation / Time Hz

Observations-:

1. Mass of the plate = M=   gm

2. Length of plate = 2a =   cm

3. Gravitational force = $g = 9.81 \, M/S^2$

Observation table -:

Sr. No	Length	Oscillation	Time
1			
2			
4			

Calculation -:

For L =   cm

Frequency = No. Oscillation / Time Hz

=   Hz

$I_G = Mg\, a^2 / L2\Pi\, f$

Result -: The mass moment of inertia of given body by bifilar method   Kg M ^{2}

EXPERIMENT NO.05

Title: - To determine moment of inertia of disc by using -
[a] Trifilar suspention
[b] Single rotor system

[a] Trifilar suspention

Aim – To determine mass moment of inertia of disc   by using trifler suspension
THEORY -: Suspended by three long flexible wire A, B, C as shown in fig when the body is twisted about its axis through a small angle θ and then release will oscillate with simple harmonic motion

For small displacement $r\theta = L\Phi$

$\Phi = r\theta / L$

Since the thee wire are attached symmetrically with respect to the axis there fore the tension in each wire will be one third of weight of body
Tension in each wire = Mg / 3
Components at the tension in each perpendicular to r = Mg sinΦ /.3 = Mg rθ / 3L
Torque applied to each wire to restore the body to its initial equilibrium position
 T =   Mg rθ / 3L * r = Mg $r^2\theta$ / 3L
Total restoring torque applied to three wires T == 3 Mg $r^2\theta$ / 3L = Mg $r^2\theta$ / L ---[1]
We have disturbing torque = I∝ = $MK^2 \propto$
      θ / ∝ = L K^2 / g r^2

Angular displacement
───────────────────   = L K^2 / g r^2
Angular acceleration

Observation :-
1. Mass of disc is =   kg
2. Radius r   =   cm

Observation Table :–

Sr. No.	Length	Oscillation	Time

Calculation :-
1. Frequency = No. of oscillation / time Hz.
2. Mass moment of inertia = mk^2

Frequency = f = r/ 2π k X g/L

Result :-
Mass moment inertia of the disc is _____ kgm^2.

(B) Single Rotor System

Aim :- To determine mass moment inertia of disc

Description Of Set Up :- One end of the shaft is griped in the chuck and heavy flywheel free to rotate in ball bearing is fixed at the other end of the shaft.

The bracket with fixed end of shaft can be clamped at any convenient position along lower beam. Thus length of the shaft can be varied during the expt specially designed chuck are used for clamping end of the shaft the ball bearing support to the flywheel provided negligible damping during expt. The bearing housing is fixed to side member of mainframe.

Procedure: - 1) Fix the bracket at convenient position along the lower beam.
2) Grip one end of the shaft at the bracket by chuck.
3) Fix the rotor on the other end of the shaft
4) Twist the rotor through some angle and release.
5) Note down the time required for 10 to 20 oscillations

Theory :- d = diameter of the shaft
L = Length of shaft
Kt = Stiffness of rod
W = Wight of disc
Inertia torque = $I\theta$
Restoring torque = $-Kt\ \theta$
Inertial torque and restoring torque both are equal but opposite in direction.
$I\theta = -Kt\theta$
$I\theta + Kt\theta = 0$
$-w^2 + Kt / I = 0$
$w = \sqrt{Kt / I}$
$w = 2\pi f$
Frequency $= 1 / 2\pi \sqrt{Kt / I}$

Observation :-

1) Shaft diameter =d= 0.4 cm
2) Diameter of disc = D = 25 cm
3) Weight of Disc = w = 4.2 kg
4) Modulus of rigidity for shaft G= 0.8×10^6 Kg/cm^2

Observation Table :-

Sr. No.	Length	No. of Oscillation	Time
1			
2			
3			

Calculation :-
Determination of Torsional stiffness $Kt = T/\theta = GI_p / L$
Ip= Polar M.I. of Shaft $= \pi d^4/32$
$F = 1/2\pi \sqrt{Kt / I}$

Result :- Mass moment of inertia of disc =   kg-m^2

Title :- To determine equivalent mass of a spring for a spring mass system.
Description Of Set Up :-
The arrangement is shown in fig. It is designed to study free forced damped and undamped vibrations. It consists of M.S. rectangular beam supported at one end by a trunnion pivoted in ball bearing. The bearing Housing is fixed to the side member of the frame. The other end of the beam is supported by the lower end of helical spring. Upper end of spring is attached to the screw.
The exciter unit can be mounted at any position along the beam additional known weights may be added to the wt. Platform under side the exciter.
PROCEDURE :

1) Support one end of the beam in the slot of turnnion and clamp it by means of screw.
2) Attach the other end of beam to the lower end of spring.
3) Adjust the screw to which the spring is attached such that beam is horizontal in the above position.
4) Weight the exciter assembly along with discs and bearing and weight platform.
5) Clamp the assembly at any convenient position.
6) Measure the distance L1 of the assembly from pivo. Allow system to vibrate freely.
7) Measure the time for any 10 osc and find the periodic time and natural frequency of vibrations.
8) Repeat the experiment by varying L1 and by also putting different weights on the platform.

NOTE : It is necessary to clamp the slotted weights to the platform by means of out so that weights do not fall during vibrations.

Calculations :
First calculate M.I. of the weight around the pivot.

$$I = \frac{w}{g}(L_1)^2$$

where W = weight of exciter assembly along with weight and platform in Kg.
        i.e. W = ('m' + added wt. in platform)
    L_1 = Distance of W from pivot in cm
    K = Stiffness of the spring in Kg/cm.
    L = Distance of spring from pivot = length of beam in cm
    m = Mass of exciter assembly along with wt. platform.
∴   M=   Kg.

$$\therefore\ T(\text{Theoretical}) = 2\pi\sqrt{\frac{Me}{K}}$$

where Me = equivalent mass at the spring = $M[L_1^2 / L^2]$
Result :-

EXPERIMENT NO.07

Title :- To determine the damping coefficient for a spring mass dash pot system.
Description Of The Set Up :-
The arrangement is as shown in the Fig. It is similar to that described for expt. The exciter unit is coupled to D.C. variable speed motor through the flexible shaft. Speed of the m motor can be varied with the dimmer stat provided on the control panel. Speed of rotation can be known from the speed indicator on control panel. It is necessary to connect the damper unit to the exciter. Amplitude record of vibration is to be obtained on the strip-chart recorder. Speed of strip-chart recorder is 33 mm/sec.
Procedure :
1) Arrange the set up as described for expt. No.
2) Connect the exciter to D.C. Motor through flexible shaft.
3) Start the motor and allow the system to vibrate.
4) Wait for 3 to 5 minutes for the amplitude to build for particular forcing frequency.
5) Adjust the position of strip-chart recorder. Take the record of amplitude Vs. time on strip-chart by starting recording motor. Press the recorder platform on the pen gently. Pen should be wet with ink. Avoid excessive pressure to get good record.
6) Take record by changing forcing frequencies.
7) Repeat the experiment for different damping.
   Damping can be changed by adjusting the holes on the Piston of the exciter.
8) Plot the graph of amplitude vs frequency for each damping conditions.

Observation Table : Set 1 with little damping.

Forcing frequency cp.s. β	Amplitude mm.

Prepare similar observation tables for various damping conditions.

Calculations : Plot the graph of amplitude Vs. frequency for each setting. Typical graph is shown in fig.

Result :

Title : Determination Of Gyroscopic Couple.

Theory : Consider a disc with an angular velocity "W" rad/sec about the axis of spin ox in anticlockwise direction when seen from the front as shown in fig. Since the plane in which the disc is rotating in parallel to the plane xoz is a horizontal plane and the axis of spin rotates in a plane parallel to the horizontal plane about an axis oy in other word the axis of spin is said to be rotating or processing about an axis oy at an angular speed W_p rad/sec. This horizontal plane of processing and oy is the axis of precision.

Let I = Mass moment of the Inertia of the disc about ox
W = Angular velocity of the disc
Angular Momentum of the disc = Iw

Since the angular Momentum is a vector quantify, therefore it may be represented by vector ox as shown in fig. B the axis of spin ox is also rotating anticlockwise when seen from the top about the axis oy let the axis ox is turned in the plane xoz through a small angle so radians to the position ox in the time in sec. Assuming the angular velocity w to be constant time angular momentum will now be constant the angular momentum will now be represented by vector ox
Change of momentum = ox' – ox = ox. dθ = I w dθ
Rate of change of angular momentum = Iw X dθ /dt
Since, the Rate of change of angular momentum will result by the application of couple to the disc, therefore the couple applied to the disc causing precession.
C = Limit Iw X dθ /dt
  st $\longrightarrow$ 0
C = I w X dθ / dt
C = I w w_p w_p = dθ /dt
Where w_p = Angular velocity of precession of the axis of the spin ox the speed of the rotation fo the axis of the spin about the axis of precession oy.
In si unit the unit of couple (c) N.M. Where I is kg-m^2.

Observation :-
1) Mass of rotor = 7.1 Kg.
2) Diameter of rotor = 300mm = 0.3m
3) Distance of weight pan = 21cm =0.21
4) Thickness of disc = 11mm

Observation Table :

Sr. No.	Speed rpm	Weight on pan K.g.	dθ radi	dt sec	Tact	Tth

Calculation :-

$$T_{act} = Iww_p$$

Moment of Inertia of disc $= I = Mr^2$

Angular velocity $= w = 2\pi n / 60$

Précis ional axis velocity $w_p = d\theta /dt$

$T_{th} = w * L$

Conclusion :- Verify the relationship $T = Iww_p$ actually and theoretically.

EXERIMENT NO.08

Title :- To Obtain Experimentally
        A)   **Frequency Response Curves**
        B)   **Transmissibility Curves**

A)   **Frequency Response Curves:-**
        The equation of motion can be written as mx + cx + kx = F sin wt
This is the second order linear differential equation with constant coefficent. The general solution of the above equation is

$$x = A_1 e^{-\zeta\omega_n t} \cos(\sqrt{1-r^2/\varepsilon^2}\,\omega_n t + \phi_1) + \frac{X_s \sin(\omega t - \phi)}{\sqrt{[1-(\omega/\omega_n)^2]^2 + [(2\varepsilon\omega/\omega_n)^2]}}$$

TOTAL RESPONSE :- The first part of equation vanishes with time while the second part remains into existence. The amplitude remains constant due to second part and it is called steady vibration. The vibration because of first part is called transient and it occurs at the damped natural frequency of the system.
        The complete solution of equation is the superposition of transient and steady state vibrations which is shown in fig.

Characteristic Curves
        The ration w/w_n is called the frequency ration where w is the frequency of excitation. Similarly, A/X_s is known as magnification factor or amplitude ration.
        A curve between frequency ratio and magnification factor is known as frequency response curve. Similarly a curve between phase angle and frequency ratio is known as phase-frequency response curve. Both curves which are drawn with the help of equations as shown in fig.
        The following points are noted from these equations and figures

1.   At zero frequency magnification is unity and damping does not have any effect on it.
2.   Damping reduces the magnification factor for all values of frequency.
3.   The maximum value of amplitude occurs a little towards left of resonant frequency.
4.   At resonant frequency the phase angle is 90^0.
5.   The phase angle increases for decreasing value of damping above resonance.
6.   The amplitude of vibration is infinite at resonant frequency and zero damping factor.
7.   The amplitude ratio is below unity for all values of damping which are more than 0.70.
8.   The variation in phase angle is because of damping. Without damping it is either 180^0 or 0.

B) Transmissibility Curves

Let us consider fig. For analysis. The spring and damping forces are kx and cx respectively.

The amplitude of vibration is A, so the maximum value of these forces will be *kA* and cwA respectively. The forces are perpendicular to each other because of phase difference of 90^0 so their resultant F_T is given as

$$F_T = \sqrt{(kA)^2 + (c\omega A)^2} = A\sqrt{(k^2 + c^2\omega^2)}$$

In the above equation F_T is the force transmitted to the foundation. The disturbing force is F. The ratio of F_T to F is called transmissibility. So it can be expressed mathematically, as

$$T.R. = F_T / F$$

From equation we have

$$F = Ak\sqrt{[1-(\omega/\omega_n)^2]^2 + [2\varepsilon\omega/\omega_n]^2}$$

Substituting the values of F and F_T in equation, we have

$$T.R. = \frac{A\sqrt{k^2 + c^2\omega^2}}{Ak\sqrt{[1-(\omega/\omega_n)^2]^2 + [2\varepsilon\omega/\omega_n]^2}}$$

$$= \frac{\sqrt{(1+c^2\omega^2/k^2}}{\sqrt{[1-(\omega/\omega_n)^2]^2 + [2\varepsilon\omega/\omega_n]^2}}$$

$$T.R. = \frac{F_T}{F} = \frac{\sqrt{1+(2\varepsilon\omega/\omega_n)^2}}{\sqrt{[1-(\omega/\omega_n)^2]^2 + [2\varepsilon\omega/\omega_n]^2}}$$

If $F_T/F = 0.25$, then the system is said to have 25% transmissibility.

The following informations are available from equation :

1. The area of vibration isolation starts when transmissibility is less than unity and $w/w_n > \sqrt{2}$. Thus when the frequency w of exciting force is given, the isolation mounts can be designed such that $w/w_n > \sqrt{2}$. If w is small the value of w_n should be small and the value of natural frequency $(\omega_n = \sqrt{\frac{k}{m}})$ results from high value of mass or low value of stiffness. Thus if the disturbing frequency w is low the isolation can be accomplished by adding certain amount of mass to the system.

2. Damping is important specially at resonance otherwise TR will be excessively large.

3. In the vibration isolation region $w/w_n > \sqrt{2}$, so equation in absence of damping can be written as

$$T.R. = \frac{1}{(\omega/\omega_n)^2 - 1}$$

4. Unity value of T.R. occurs at two points where w/w_n is zero and $\sqrt{2}$ for all values of damping.

5. Under ideal operating conditions, T.R. must be zero. Hence, w/w_n should be as large possible. Thus, the natural frequency of the system should be small i.e. weak spring with fairly heavy mass.