Government college of Engineering

Department of Mechanical Engineering

LAB Manual

Mechanism and Machines

Mechanical 4th Semester

PME4I101

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INSTRUCTIONS TO STUDENTS

1. Be prompt in arriving to the laboratory and always come well prepared for the experiment.
2. Be careful while working on the equipment's operated with high voltage power supply.
3. Work quietly and carefully. Give equal opportunity to all your fellow students to work on the instruments.
4. Every student should have his/her individual copy of the Mechanisms & Machines Practical Book.
5. Every student has to prepare the notebooks specifically reserved for the Mechanisms & Machines Practical work "Mechanisms & Machines laboratory Book".
6. Every student has to necessarily bring his/her Mechanisms & Machines Practical Book and laboratory book when he/she comes to the laboratory to perform the experiment.
7. Record your observations honestly. Never makeup reading or to doctor them to get a better fit on the graph or to produce the correct result. Display all your observations on the graph (If applicable).
8. All the observations have to be neatly recorded in the Mechanisms & Machines laboratory Book and verified by the instructor before leaving the laboratory.
9. If some of the readings appear to be wrong then repeat the set of observations carefully.
10. After verification of the recorded observations, do the calculation in the Mechanisms & Machines laboratory Book and produce the desired results and get them verified by the instructor.
11. Never forget to mention the units of the observed quantities in the observation table. After calculations, represent the results with appropriate units.
12. Calculate the percentage error in the results obtained by you if the standard results are available and also try to point out the sources of errors in the experiment
13. Do not forget to get the information of your next allotment (the experiment which is to be performed by you in the next laboratory session) before leaving the laboratory from the Technical Assistant.
14. Calculate the percentage in the results obtained by you if the standard results are available and also try to point out the sources of errors in the experiment.
15. Finally record the verified observations along with the calculation and results in the Mechanisms & Machines Practical Book.
16. Do not forget to get the information of your next allotment (the experiment which is to be performed by you in the next laboratory session) before leaving the laboratory from the Technical Assistant.
17. The grades for the Mechanisms & Machines course work will be awarded based on your performance in the laboratory, regularity, recording of experiments in the Mechanisms & Machines Practical Book. lab quiz, regular viva-voce and end-term examination.

CERTIFICATE

This is to certify that

Mr./Ms.....

with enrollment no. ..... during the academic year

.....

Date of Submission: ..... Staff In charge: .....

Head of Department: .....

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Experiment No.1

Title: Determination of gyroscopic couple using gyroscopic test rig.

Objective:

1. To understand the basic idea of the working principle of the gyroscope and its applications.
2. Experimental verification of gyroscopic effect ($C = I\omega\omega_p$).

Outcome: After successfully completion of practical the student will be able to know the importance of requirement of the gyroscope and its function as well as applications.

Theory:

The gyroscope is a massive rotor that is fixed in light supporting rings called gimbals. The gimbals consist of frictionless bearings that isolate the central rotor from outside torques. The axle of the spinning wheel defines the spin axis.

The rotor possesses three degrees of rotational freedom and spins about an axis. It acquires extraordinary stability of balance at high speeds as it maintains the high speed rotation axis of its central rotor.

<u>Working Principle</u>

When the gyroscope is applied with external torques or rotations about the given axis, the orientation can be measured by a precession phenomenon. When an object rotating about an axis is applied with external torque along a direction perpendicular to the rotational axis, the precession occurs.

This rotation about the spin axis is detected and information on this rotation is delivered to a motor or other device that applies torque in an opposite direction thereby cancelling the precession and maintaining its orientation.

The precession can also be prevented using two gyroscopes that are arranged perpendicular to each other. The rotation rate can be measured by the pulsation of counteracting torque at constant time intervals.

Technical data:

(1) Rotor diameter: - 245 mm
(2) Rotor thickness: - 10 mm
(3) Moment of inertia of the disc coupling and motor rotor about central axis: - 0.02986 Nm-sec^2
(5) Distance of bolt of weight pan from disc center: - 0.155 m
(6) Motor max speed: - 6000rpm

Observation table:

ω_p= angular velocity of precession
$\omega = \frac{2\pi N}{60}$ = Angular velocity
C = gyroscopic couple
F = Force applied = _____
L = Distance of bolt of weight pan from disc centre: - 155 mm

S. No.	Angle of rotation ($\boldsymbol{\theta}$)	Time taken for rotation	Average time	$\boldsymbol{\omega = \frac{\theta}{t}}$ ($\boldsymbol{\theta}$in rad.)	N (rpm)	$\boldsymbol{\omega}$ (rad/sec)	$\boldsymbol{C_{th} = I\omega\omega_p}$	$\boldsymbol{C_{exp} = FL}$
1								
2								
3								
4								

Calculate the Percentage of deviation between theoretically and experimentally calculated gyroscopic couple.

$\%\, deviation = \frac{C_{th} - C_{exp}}{C_{th}} \times 100$ = _____

Conclusion:

_____
_____
_____
_____

Marks Obtained	Signature by faculty	Date of checking

Experiment No.2

Title: Performance characteristics of gravity and spring -controlled governor.

Objective:

1. To understand the basic idea of the working principle of the gravity governor and its applications.
2. To know function of governor, types of governor and plotting the characteristic curves for different gravity-controlled governor.

Outcome: - After successfully completion of practical the student will be able to know the importance of requirement of the governor and its function as well as applications.

Theory:

The function of a governor is to regulate the mean speed of an engine, when there is variation in the load e.g. when the load on an engine increases, its speed decreases, therefore it becomes necessary to increase the supply of working fluid and when the load on the engine decreases, its speed increases and thus less working fluid is required. The governor automatically controls the supply of working fluid to the engine with the varying load conditions and keeps the mean speed within certain limits.

The governors may, broadly, be classified as:

1. Energy conservation governor and
2. Energy dissipation governor.

The energy conservation type of governor may further be classified as follows:

1. Centrifugal governor and
2. Inertia governor

The centrifugal governors may further be classified as follows:

1. Pendulum type: - Watt governor
2. Loaded type:
   i. Dead weight governor (gravity governor): - Porter governor and Proell governor
   ii. Spring controlled governors: -Hartnell governor, Hartung governor, Wilson-Hartnell governor and Pickering governor

<u>Watt Governor:</u>

The simplest form of a centrifugal governor is a Watt governor. It is basically a conical pendulum with links attached to a sleeve of negligible mass. The arms of the governor may be connected to the spindle in the following three ways:

(a) The pivot may be on the spindle axis.
(b) The pivot may be offset from the spindle axis and the arms when produced intersect at O.
(c) The pivot may be offset, but the arms crosses the axis at O.

<u>Porter Governor:</u>

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

Proell Governor:

The porter governor is known as a Proell governor if the two balls (masses) are fixed on the upward extensions of the lower links which are in the form of bent links.

Hartnell Governor

A Hartnell governor is a spring-loaded governor. It consists of two bell crank levers pivoted at the point to the frame. The frame is attached to the governor spindle and therefore rotates with it. Each lever carries a ball at the end of the vertical arm and a roller at the end of the horizontal arm. A helical spring in compression provides equal downward forces on the two rollers through a collar on the sleeve. The spring force be adjusted by screwing at nut up or down on the sleeve.

Observation:

- Mass of the ball (m) = _____kg.
- Weight of the ball (w) = _____N.
- Height of the governor (h) = _____m.
- Maximum equilibrium speed (N1) = _____ r.p.m.
- Minimum equilibrium speed (N2) = _____ r.p.m.
- Frictional force (F) = _____ N.
- Mean equilibrium speed (N) = $(N_1 + N_2)/2$ = _____ r.p.m
- Mass of the central load = _____ kg.
- Weight of the central load (W) = _____ N.
- Angle of inclination of the arm to the vertical (α) = _____
- Angle of inclination of the link to the vertical (β) = _____

Equilibrium radius:

Radius at which $F_c = F_r$

Where F_c= centrifugal force (outward) = $m\omega^2 r$

F_r= restoring force (inward) = $\frac{ar}{bh} mg\left[1+\frac{(M \pm f)(1+k)}{2m}\right]$

Where:

$k = \frac{\tan\beta}{\tan\alpha}$

f =frictional force

For Watt Governor: - $a = b;$ $M = 0;$ $f = 0;$ $Fr = mg\frac{r}{h}$

For Porter Governor: a=b; $Fr = \frac{r}{h} mg\left[1+\frac{(M \pm f)(1+k)}{2m}\right]$

For Proell Governor: $a \neq b$ $Fr = \frac{ar}{bh} mg\left[1+\frac{(M \pm f)(1+k)}{2m}\right]$

At equilibrium $F_c = F_r$

$\therefore \quad m\omega^2 r = Fr \quad and \quad \omega = \frac{2\pi N}{60}$

$$\therefore \qquad N^2 = \frac{895a}{h}\left[1 + \frac{(M \pm f)(1+k)}{2m}\right]$$

For Watt Governor - $N^2 = \frac{895}{h}$

For porter Governor - $N^2 = \frac{895}{h}\left[1 + \frac{(M \pm f)(1+k)}{2m}\right]$

For Proell Governor - $N^2 = \frac{895a}{h}\left[1 + \frac{(M \pm f)(1+k)}{2m}\right]$

Observation Table:

X =

Y =

r_o=

S. No.	N (rpm)	Lift 'h' (m)	Mass (kg)	$F = m\omega^2 r$
1				
2				
3				

Observation:

- Mass of each ball (m) = _____ kg.
- Mass of the sleeve (M)= _____ N.
- Minimum radius of rotation (r_1) = _____ m.
- Maximum radius of rotation (r_2) = _____ m.
- Angular speed of the governor at minimum radius (ω_1) = _____ rad/s.
- Angular speed of the governor at maximum radius (ω_2) = _____ rad/s.
- Spring force exerted on the sleeve at ω_1 (S_1) = _____ N.
- Spring force exerted on the sleeve at ω_2 (S_2) = _____N.
- Length of the vertical or ball arm of the lever (X) = _____ m.
- Length of the horizontal or sleeve arm of the lever (Y) = _____ m.
- Distance of fulcrum 'O' from the governor axis or the radius of rotation when the governor isin mid-position (r) = _____ m.
- Minimum equilibrium speed (N_1) = _____r.p.m.
- Minimum equilibrium speed (N_2) = _____r.p.m.
- Frictional force (F) = _____ N.
- Angle of inclination of the arm to the vertical (α) = _____
- Angle of inclination of the link to the vertical (β) = _____

Equilibrium radius:
For spring controlled Hartnell Governor:

$$s = Spring\,stiffness$$

$$\frac{S_2 - S_1}{h} = s \qquad and \qquad \frac{h}{Y} = \frac{r_2 - r_1}{X}$$

$$\therefore h = (r_2 - r_1)\frac{Y}{X}$$

$$\left(\frac{S_2 - S_1}{2}\right)Y = (Fc_2 - Fc_1)X$$

$$\therefore \frac{S_2 - S_1}{2} = (Fc_2 - Fc_1)\frac{X}{Y}$$

$$\therefore hs = 2(Fc_2 - Fc_1)\frac{X}{Y}$$

$$\therefore (r_2 - r_1)\frac{Y}{X}S = 2(Fc_2 - Fc_1)\frac{X}{Y}$$

$$\therefore S = 2\left(\frac{Fc_2 - Fc_1}{r_2 - r_1}\right)\left(\frac{X}{Y}\right)^2$$

Procedure:

To find stiffness ‘s’ of Hartnell governor:

- Measure X, Y and radius $r_{o.}$
- Run the governor and gradually increase speed, after some time the sleeve starts lifting.
- Stop increase of speed and note the lift of the sleeve and speed.
- Again, increase speed until it starts lifting and stop increase of speed. Note the lift and speed.
- Repeat the experiment four times.
- Calculate the radius at different speed from equation:

$$\frac{r - r_0}{X} = \frac{Lift}{Y}$$

Observation Table: -

$X =$

$Y =$

$r_o =$

S. No.	N (rpm)	Lift 'h' (m)	$r = \frac{X}{Y} lift + r_o$	Mass (kg)	$F = m\omega^2 r$	Spring constant
1						
2						
3						

Conclusion: - 'S' is almost constants.

Marks Obtained	Signature by faculty	Date of checking

Experiment No.3

Title: Determination of critical speed of rotating shaft

OBJECTIVES:

1) Verification of Whirling theory.
2) Verification of Dunkerley's Equation.

OUTCOMES:

As an outcome of this experiment, students will be able to,
1) Understand the concept of critical speed of shaft.
2) Understand why high speed engine is operated above critical speed?

THEORY:

For any rotating shaft, a certain speed exists at which violent instability occurs. The shaft suffers excessive deflection and bows, a phenomenon known as whirling. If this critical speed of whirling is maintained (called First Critical speed), then the resulting amplitude becomes sufficient to cause buckling and failure. However, if the speed is rapidly increased before such effects occur, then the shaft is seen to re-stabilize and run true again until another specific speed is encountered where a double bow is produced as shown in Figure-10.1. The second speed is called "Second Critical". Whirling speed depends primarily on the stiffness of the shaft and mass distribution when the shaft is loaded, the whirling speed will be shifted due to the effect of the new mass. Dunkerley's set the equation that relates the overall whirling frequency with critical frequencies introduced by the shaft and load individually. This equation is valid for any number of loads.

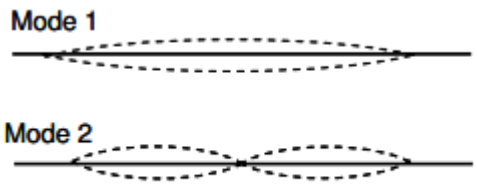


Fig 10.1 Whirling of Shaft

Studying whirling of shaft is of great important due to huge number of applications in various fields. For example, all rotating machinery involve shafts with rotating parts such as rotors in electrical motors, impellers in pumps, blades in turbines …. etc. On the other hand, Dunkerley's Equation is found to be useful not only in studying whirling of loaded shafts, but also in structural analysis and frequency response testing.
The critical frequency for a shaft may be obtained from the fundamental frequency of a beam subjected to a transverse vibration;

$$f = \lambda\sqrt{\frac{EIg}{wL^4}} \qquad (1)$$

Where,
f : critical frequency in Hz
E : Young's modulus

I : Second moment of area of the shaft;
w: Weight per unit length of the shaft
λ: Constant dependent upon the fixing conditions and mode and can be found from the following table;

Type of support	λ_1 (first mode)	λ_2 (second mode)
Simply supported	1.573	6.3
Supported-Fixed	2.459	7.96
Fixed-Fixed	3.75	8.82

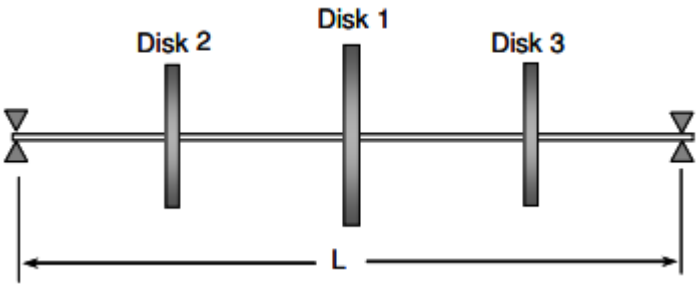


Fig.10.2 Shafts Loaded with three discs

For a shaft loaded with a number of disks as shown in Figure-10.2, the first critical frequency for the system can be found from Dunkerley's Equation as follows;

$$\frac{1}{f^2}=\frac{1}{f_s^2}+\frac{1}{f_1^2}+\frac{1}{f_2^2}+\frac{1}{f_3^2}+.....$$

Where f : critical for the system as a whole

fs : critical speed of the shaft alone (first critical calculated from eq. (1))

f_1, f_2, f_3 : critical speeds due to attaching disk 1, 2 and 3 individually without the effect of other masses.

APPARATUS OF THE EXPERIMENTS:-

This is Whirling of Shafts Apparatus shown in Figure-10.3. The shaft is located in the kinematic coupling and either the fixed or free type end bearing. Several shafts of various lengths and diameters are available.

The kinematic coupling and sliding end bearings have been designed to allow the shaft movement in a longitudinal direction. The sliding end bearing is interchangeable to allow the selection of support type between directionally fixed and free support. A movable part is provided as a part of the kinematic coupling which allows the selection of support type. When this part moved away from the coupling, the resulting support will be directionally free. The shaft is driven by a DC motor capable of providing…….. RPM through the kinematic coupling which possesses double universal joint. The motor speed is controlled by TQ E3 control unit

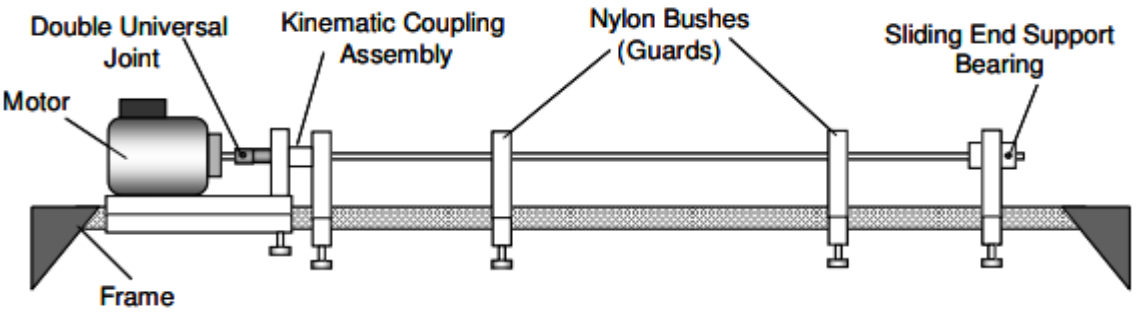


Fig 10.3 Whirling of shaft apparatus

In order to maintain the amplitude of vibration within specific limits, two nylon guards are provided and are adjustable along the length of the apparatus. The sliding end bearing may be moved to enable various shaft lengths to be selected. In addition, four disks are supplied to providing loading to the shaft. These disks can be fitted to the …..mm diameter shaft. Two of them are of equal mass at ….. g, the third has a mass of ….. g. Additionally, a stroboscope is used to measure the rotational speed and also to observe the shaft configuration during whirling.

EXPERIMENTAL PROCEDURE AND CALCULATIONS

Part A: Whirling of shafts without loading

1. Attach a shaft of known diameter and length to the apparatus.
2. Select simply supported configuration by moving out the sliding part of the kinematic coupling and using the free support at the other end. Calculate the theoretical first and second whirling speeds from eq. (1). The density of shaft material is …….. kg/m^3, and Young's modulus is ……..GPa .
3. Switch on the speed control unit and adjust the speed carefully until obtaining the largest amplitude of whirling. Read the speed on the stroboscope and observe the shaft in the first mode, it should contain a single bow. Increase the speed slowly until you obtain the second mode and record the rotational speed. Observe the shaft in the second mode.
4. Change the support type to fixed-supported and then to fixed-fixed and repeat steps 2 and 3.
5. Replace the shaft with another one of different diameter and repeat the above steps. Record the results as in a table as below:

No.	Shaft dia. (mm)	Shaft Length (m)	Simply supported		Supported -fixed		Fixed-fixed	
			T_{theo}	T_{exp}	T_{theo}	T_{exp}	T_{theo}	T_{exp}
1								
2								

CALCULATIONS:

Part B: Whirling of loaded shafts

1. Use the …… mm shaft in simply-supported configuration. Attach the first disk of …….g midway between the two supports.
2. Switch on the speed control unit and adjust the speed carefully until you obtain whirling condition. Record the whirling frequency of the system *f*.
3. Calculate the critical frequency for the first disk alone, f_1 , from the following equation:

$$\frac{1}{f^2} = \frac{1}{f_s^2} + \frac{1}{f_1^2}$$

Where f_s is the whirling frequency for the shaft alone in the simply-supported configuration and can be taken from Part A.
4. Remove disk No. 1 and attach disk No. 2 (……g) at …….L from the motor-side support and repeat the above procedure to calculate f_2 for the second disk alone.
5. Attach disk No. 3 alone at ……. L from the motor-side support and repeat the procedure to calculate f_3 for the third disk alone.
6. Attach all the three disks at the same positions and run the DC motor to find the critical frequency for the combination. Verify that eq. (2) is satisfied. Arrange your reading as in the table below:

RESULT TABLE

No.	Loading	System critical frequency. *f* (as measured)	Shafts critical freq. f_s(from part A)	Disk frequency f_i $i= 1,2,3$
1	Disk 1 alone			
2	Disk 2 alone			
3	Disk 3 alone			
4	All			

CONCLUSIONS:

Marks Obtained	Signature by faculty	Date of checking

Experiment No.4

Title Experiment on static and dynamic balancing apparatus

OBJECTIVES:

1. Understand the causes of unbalance in rotating part of machine
2. Learn the static balancing of single plane system
3. To learn a practical method of dynamic balancing that can be used in the field and appreciate its advantages and limitations.

OUTCOMES:

As an outcome of this study, students will be able to,

1) Differentiate between static and dynamic balancing

2) Understand use balancing in rotating part of machine

1. THEORY:

Balancing is the technique of correcting or eliminating unwanted inertia forces or moments in rotating or reciprocating masses and is achieved by changing the location of the mass centres.

The objectives of balancing an engine are to ensure:

1. That the centre of gravity of the system remains stationery during a complete revolution of the crank shaft and
2. That the couples involved in acceleration of the different moving parts balance each other

1.1 Causes of Unbalance:

In the design of rotating parts of a machine every care is taken to eliminate any out of balance or couple, but there will be always some residual unbalance left in the finished part because of

1. Slight variation in the density of the material or
2. Inaccuracies in the casting or
3. Inaccuracies in machining of the parts.

1.2 Why balancing is so important?

1. A level of unbalance that is acceptable at a low speed is completely unacceptable at a higher speed.
2. As machines get bigger and go faster, the effect of the unbalance is much more severe.
3. The force caused by unbalance increases by the square of the speed.
4. If the speed is doubled, the force quadruples; if the speed is tripled the force increases

1.3 Types of balancing:

a) Static Balancing:

i) Static balancing is a balance of forces due to action of gravity.

ii) A body is said to be in static balance when its centre of gravity is in the axis of rotation.

b) Dynamic balancing:

i) Dynamic balance is a balance due to the action of inertia forces.

ii) A body is said to be in dynamic balance when the resultant moments or couples, which involved in the acceleration of different moving parts is equal to zero.

iii) The conditions of dynamic balance are met; the conditions of static balance are also met.

1.4 IMPORTANT TERMS RELATED TO BALANCING

Rotating centreline:

The rotating centrelines being defined as the axis about which the rotor would rotate if not constrained by its bearings. (Also called the Principle Inertia Axis or PIA).

Geometric centreline: The geometric centreline being the physical centreline of the rotor.

When the two centrelines are coincident, then the rotor will be in a state of balance. When they are apart, the rotor will be unbalanced.

Different types of unbalance can be defined by the relationship between the two centrelines. These include:

Static Unbalance – where the PIA is displaced parallel to the geometric centrelines.

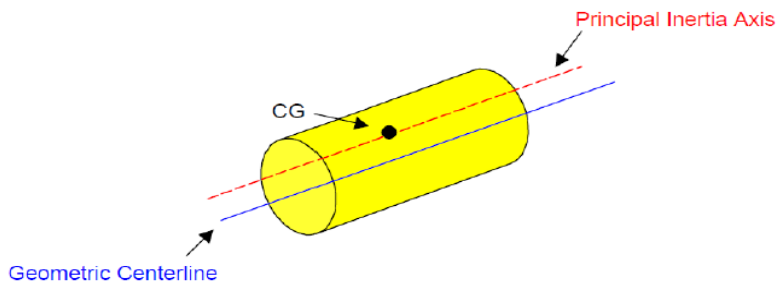


Fig.1.1 static unbalance

Couple Unbalance – where the PIA intersects the geometric centreline at the centre of gravity. (CG)

Dynamic Unbalance – where the PIA and the geometric centreline do not coincide or touch. The most common of these is dynamic unbalance.

1.5 BALANCING OF ROTATING MASSES

When a mass moves along a circular path, it experiences a centripetal acceleration and a force is required to produce it. An equal and opposite force called centrifugal force acts radially outwards and is a disturbing force on the axis of rotation. The magnitude of this remains constant but the direction changes with the rotation of the mass. In a revolving rotor, the centrifugal force remains balanced as long as the centre of the mass of rotor lies on the axis of rotation of the shaft. When this does not happen, there is an eccentricity and an unbalance force is produced. This type of unbalance is common in steam turbine rotors, engine crankshafts, rotors of compressors, centrifugal pumps etc. Balancing involves redistributing the mass which may be carried out by addition or removal of mass from various machine members

Balancing of rotating masses can be of

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

4. Balancing of several masses rotating in different planes

1.6 STATIC BALANCING:

All rotors have some eccentricity. Eccentricity is present when geometrical centre of the rotor and the mass centre do not coincide along their length (Figure 1.2). Examples of rotors are turbines, generator, Compressor or an electric motor. Due to eccentricity, load on bearing will increase. Causes excessive wear and tear, vibration (which causes fatigue failure and faulty operation i.e. in machining or in printing machine). Our aim is to reduce bearing load (unbalanced force). This is achieved by proper.

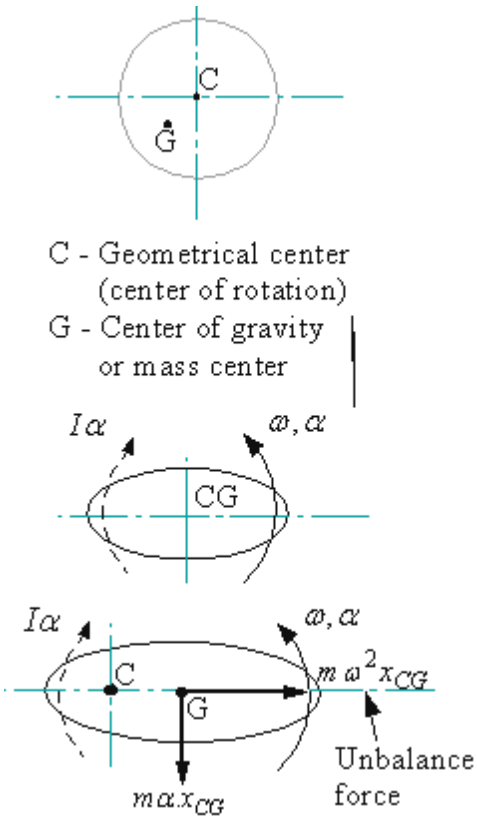


Fig 1.2 Geometrical centre and centre of gravity

Unbalance in single plane:

Such unbalance occurs in gear wheels, grinding wheels, single stage compressor, etc. Unbalance comes from material inhomogeneous , limitations of manufacturing process, mounting and limitation errors etc. The unbalance in the disc is defined as (Figure 1.3)

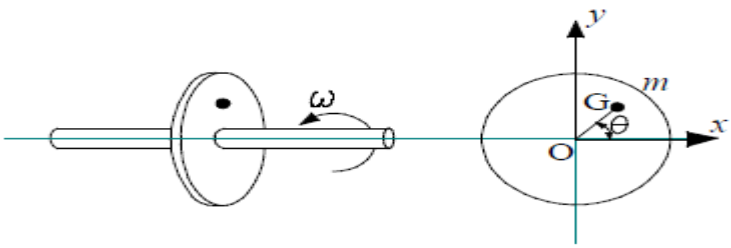


Fig.1.3 Unbalance in single plane

Unbalance $U = m\,e \quad (U \rightarrow Kg.m)$ ---------- (1)

Where, m = is mass of disc (kg), e = is the eccentricity (m).

The unbalance force is given as

$$F = m\omega^2 e \qquad \text{----------} \quad (2)$$

Where, ω = is the angular velocity, rad/sec.

If we want to know correction mass m_c at a radius of r, it will be given by

$$m_c = \frac{e/r}{m} \qquad \text{----------} \quad (3)$$

For $r \to \infty$; we have $m_c = 0$ and it should be placed 180^0 away from unbalance mass 'm'. Such a correction is called balancing of disc, which will eliminate the inertia forces transmitted to foundation or bearing.

In the actual practice location of point G is unknown. The radial direction we can obtain by keeping the rotor on frictionless (knife edge) support and we will allow rotor to rotate freely. Rotor will become stationary with heavy spot (G) vertically downwards. Now we will place a correction mass in at 180^0 to the heavy spot (i.e. at light spot) and again allow rotor to rotate. (i) if heavy spot is again coming vertically downwards means correction mass m to be increased. (ii) if heavy spot comes vertically upward position, means correction mass is more, and it has to be decreased. (iii) if heavy spot is resting at some other position, means rotor is nearly balanced. This can be confirmed by rotating rotor again and finding whether it rests at some indifferent equilibrium position. Such a process is called static balancing of rotor (disc) and it is valid for a rotor with only one disc or balancing is required in single plane only. For single plane static balancing rotor will be dynamically also balanced.

Experiment No.5

Title: Determination of natural frequencies of un-damped as well as damped vibrating systems.

OBJECTIVES:

1. Verification of simple mass-spring system theory
2. Determination of natural frequencies of un-damped as well as damped vibrating systems

OUTCOMES:

As an outcome of this experiment, students will be able to,

1) Determine the spring stiffness of helical spring experimentally

2) Calculate the natural frequency of any single degree of freedom system.

THEORY:

Longitudinal Vibration:

When the particles of the spring moves along the longitudinal axis of spring as shown in Fig 4.1, then the vibrations are known as longitudinal vibration. In this case, spring is elongated and shortened alternately and thus tensile and compressive stresses are induced alternated in the spring

Let us consider a spring-mass system as shown in Fig. 4.1. The system is constrained to move in the vertical direction (longitudinal) only along the axis of the spring. Let k and m be the stiffness of the spring and the mass of the block, respectively. Let x be the position of the mass at any instant from the equilibrium position of the mass and it is assumed that x is positive in the downward direction and negative in the upward direction. In the spring-mass system only one coordinate is enough to describe the position of the mass at any time, and hence, it is single degree-of-freedom system. Here the coordinate is x.

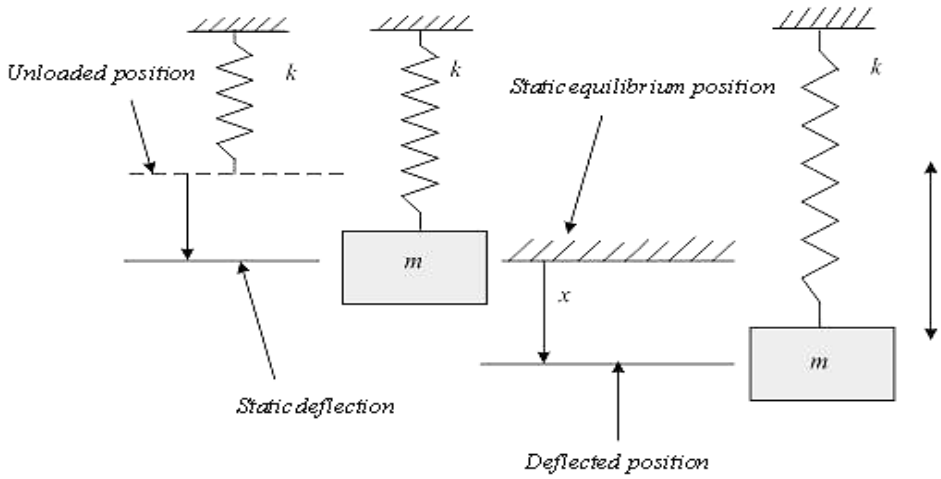


Fig. 4.1 Spring-mass system

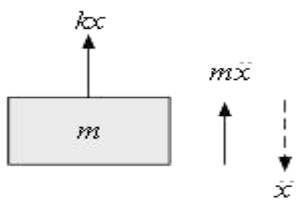


Fig 4.2 Free body Diagram of Mass

The free-body diagram of the mass is shown in Fig.4.2. Applying Alembert's principle, the equation of motion of the mass can be obtained as

$$m\ddot{x} + kx = 0$$

As spring mass system performs simple harmonic motion, on comparing equation of motion with the governing equation of SHM ($\ddot{x} + \omega^2 x = 0$) will get circular natural frequency (ω_n)of single degree of freedom spring mass system as

$$\omega^2 = \omega_n^2 = \frac{k}{m}$$

$$\omega_n = \sqrt{\frac{k}{m}} = \text{Circular Natural Frequency in rad/s}$$

$$f = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k}{m}} = \text{Linear Natural Frequency in cycle/s or Hz}$$

$$\text{Time Period } T = \frac{1}{f} = 2\pi\sqrt{m/k} \quad \text{in sec}$$

DESCRIPTION OF APPARATUS:

One end of the open coil spring is fixed to the screw which engages with the screwed hand wheel. The screw can be adjusted vertically in any convenient position and then clamed to upper supported plane by the means of lock nuts. Lower end of the spring is attached to the platform carrying the dead loads. Thus the design of the system incorporates vertical positioning of the unit.

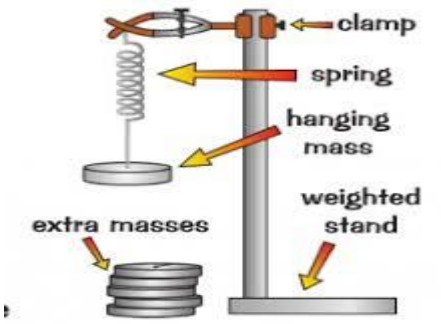


Fig 4.3 Experimental step up spring mass system

PROCEDURE:-

- Fix one end of the helical spring to the upper screw.
- Determine the free length.
- Put some mass on the platform and note down the deflection.
- Stretch the spring through some distance and release.
- Count the time requires for oscillations in seconds, say **n**.
- Determine the actual time period.
- Repeat the procedure for different masses.

OBSERVATIONS:

- ✓ Length of the spring : _____mm
- ✓ Mean diameter of the Spring **D**: _____mm
- ✓ Wire diameter **d :** _____mm
- ✓ Active No. of coil, **n**: _____mm
- ✓ Dead Weight: _____mm

OBSERVATION TABLE NO. 1 : FOR STIFFNESS OF SPRING (K)

Obs. No	Mass **'m'** attached(kg)	Deflection of the spring **'ȣ'** (m)	**K**=mg/ȣ (N/m)	$\mathbf{K}_{mean}$= K_{Exp} (N/m)
1				
2				
3				

4				

OBSERVATION TABLE NO. 2: FOR TIME PERIOD AND NATURAL FREQUENCY

Obs. No	Mass m attached (kg)	No. of Oscillations n	Time required for 'n' oscillation	Average Time (T_{avg}) (sec)	Periodic Time T_{Exp} (sec)	$f_{Exp} = \frac{1}{T_{Exp}}$
1						
2						

CALCULATIONS:

$K = Gd^4/8nD^3$

Where G=Shear Modulus=80* GPa

d= wire diameter

D= Mean coil dia.

n= No. of active turns

RESULTS:

- **FOR SPRING STIFFNESS 'K'**

K_{Exp} (N/m)	$K_{theo} = Gd^4/8nD^3$

- **FOR TIME PERIOD AND NATURAL FREQUENCY**

Obs. No	Mass m attached (kg)	Periodic Time (Exp) T_{Exp} (sec)	$f_{Exp} = \frac{1}{T_{Exp}}$	$f_{Th} = \frac{1}{2\pi}\sqrt{\frac{K_{Th}}{m}}$	Periodic Time (Th.) $T_{Th} = \frac{1}{f_{Th}}$	% Deviation
1						
2						

Damped Vibration

The vibration that the system executes under damping system is known as damped vibrations. In general all the physical systems are associated with one or the other type of damping. In certain cases amount of damping may be small in other case large. In damped vibrations there is a reduction in amplitude over every cycle of vibration. This is due to the fact that a certain amount of energy possessed by the vibrating system is always dissipated in overcoming frictional resistances to the motion. The rate at which the amplitude of vibration decays depends upon the type and amount of damping in the system. Damped vibrations can be free vibrations or forced vibrations. Shock absorber is an example of damped vibration. Mainly the following two aspects are important while studying damped free vibrations: 1. The frequency of damped free vibrations and
2. The rate of decay.

FIGURE:

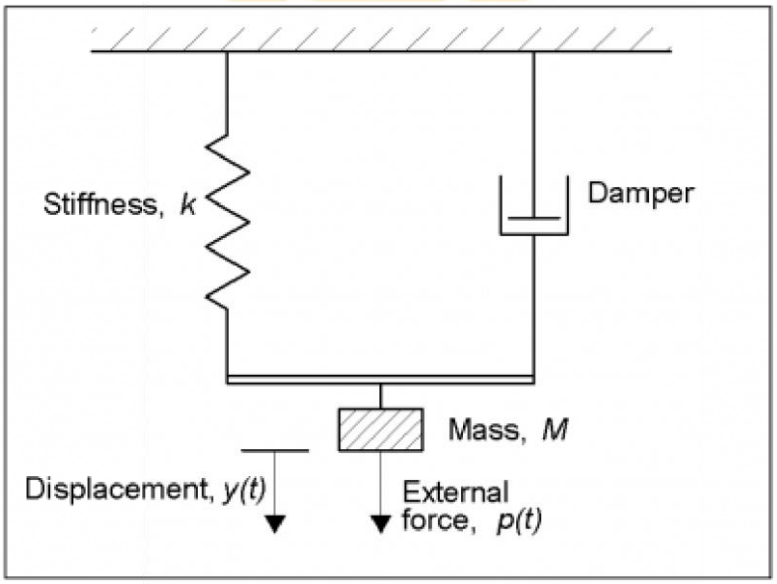


Figure: Damped forced vibration

PROCEDURE:

1. Connect the exciter to D.C. motor.
2. Start the motor and allow the system to vibrate.
3. Wait for 3 to 5 minuets for the amplitude to build for particular forcing frequency.
4. Adjust the position of strip-chart recorder. Take the record of amplitude Vs time on the strip chart.

5. Take record by changing forcing frequency.
6. Repeat the experiment for different damping. Damping can be changed adjusting the position of the exciter.
7. Plot the graph of amplitude Vs frequency for each damping condition.

OBSERVATIONS:

Sr. No.	Number of oscillations, n	Time required for n oscillations, t	Periodic Time, T	Forcing frequency, f = 1/T	Amplitude, mm

CONCLUSION:

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

Marks Obtained	Signature by faculty	Date of checking

Title: Study of Interference & Under Cutting for gear drives.

Objective:

1. Understand the cause of Interference and Undercutting in gear drives.

Outcomes:

After completing this experiment, the student will be able to

1) Understand the probable causes of Interference in gear drives and will be able to design gear drives without any fault

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 undercut 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.

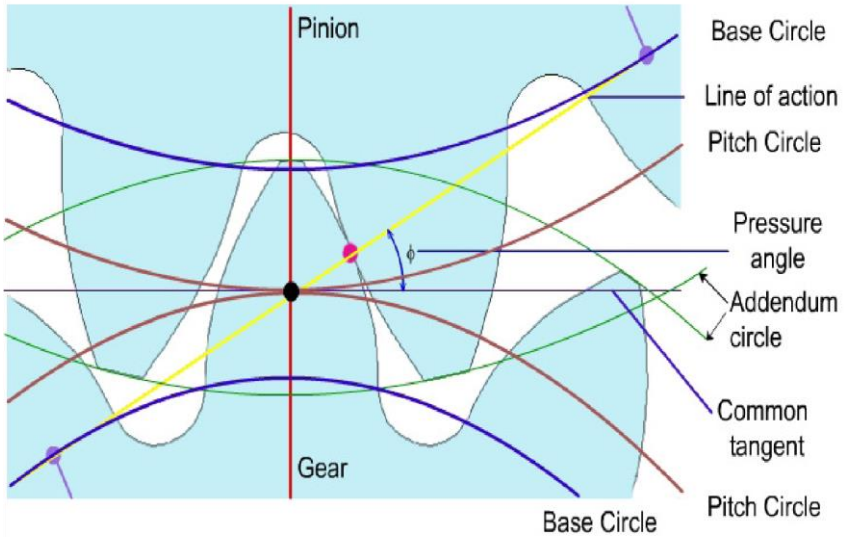
If the radius of addendum circle of the wheel increases beyond O_2N. Then the tip of tooth of the wheel will cause interference with the tooth on the pinion. The points M & N are called as interference points.

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.

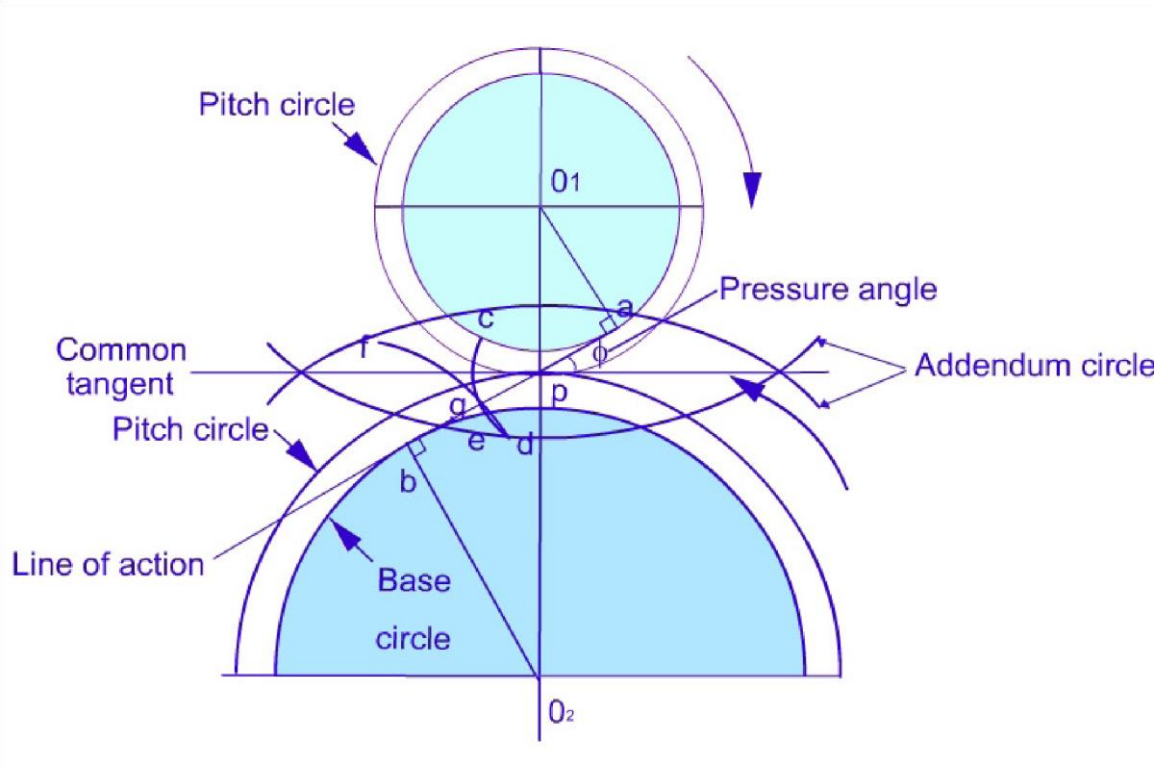
$MP = r \sin \varphi$; $PN = R \sin \varphi$

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

Are of contact = (R+ r) sin φ / cos φ = (R +r) tan φ



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 $O_1N\,P$,

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

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

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

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

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

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.

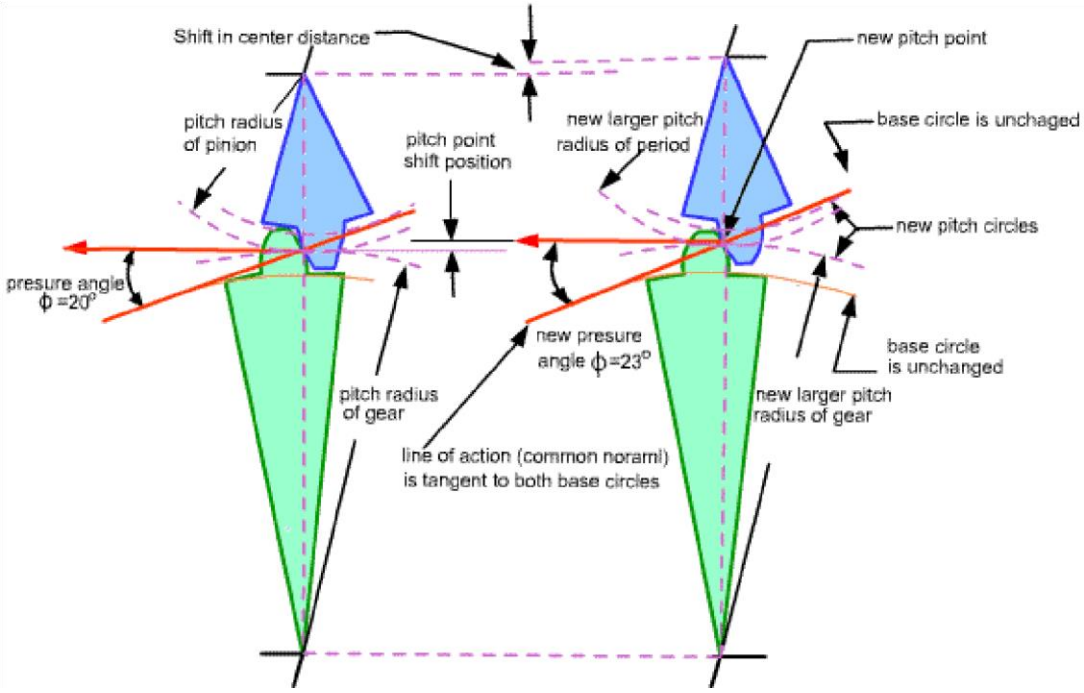
Addendum of pinion = $O_1\,N - O_1\,P$

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

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

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

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



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.

Marks Obtained	Signature by faculty	Date of checking

Title : Experiment on Cam Analysis Apparatus.

INTRODUCTION

The Cam Analysis Machine(CAM) has been designed to enable students; to observe the dynamic behaviour of cam followers under various operating conditions.

Cam are mounted on the end of a shaft driven by a variable speed motor. A high voltage stylus pen attached to the follower traces the follower lift on teledeltos paper mounted on a drum. Follower bounce may be investigated; for the followers of different weights and springs of various rates.

OBJECTIVE:

The fundamental objectives of this experiment are:

1) Observation of the effect of cam profile on the cam dynamics,
2) To study the displacement, velocity, and acceleration profile of cam
3) To identify the factors which may improve the cam dynamics.

Description of Apparatus

The CAM consists of a set of cams, a set of springs, a pen and drum recording system to record cam displacement diagram and other components. A D.C. shunt wound geared motor is directly coupled by a flexible coupling to an extension shaft, on which is mounted a flywheel to reduce the fluctuation in speed caused by the variable torque required to lift the follower. The cam is mounted on a taper on the end of the shaft, and secured on the shaft by a nut. The followers are of the flat face or roller type, and are mounted on the end of a vertical bar. A drilling in top end of the bar accommodates a steel ball, which is retained in position by a second bar carrying a spring and optional masses. The presence of the steel ball ensures that only an axial force is transmitted to the upper bar. The top end of the spring is supported by a crossbar mounted on two vertical pillars on the base plate. A pen traces a record of the follower amplitude on paper. The paper is fastened to a cylindrical drum driven by a timing belt from the camshaft. The camshaft speed may be measured by means of a Stroboscope.

Dynamic forces on follower and spring

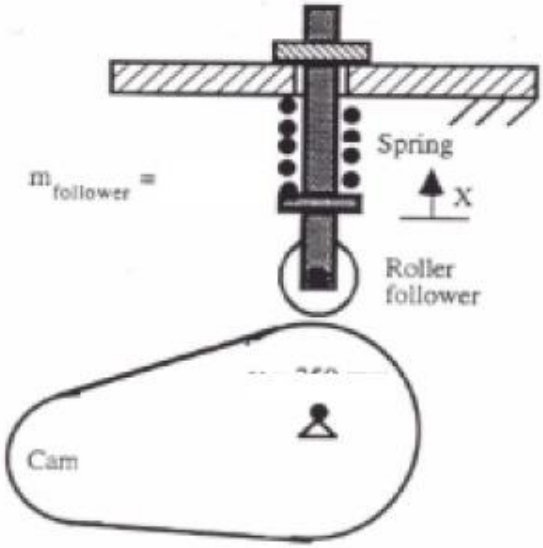
The continuous contact between the follower and the cam is achieved by the downward forces of the spring and follower weight. On the other hand, the force along the axis of the follower exerted by the cam is the driving force of the follower. For the cam system shown in Figure below. If m is the effective mass of the follower, then the driving force required to accelerate the follower:

$$\sum F = mf * Af$$

Let S be the follower spring stiffness and xo the initial compression of the spring. When the follower mass is driven through a distance X the spring force is S(X + xo). The equation of motion is:

$$F - m_f * g - S(X + xo) = m_f * Af$$

where: F = contact force, mf = mass of the follower, A_f = acceleration of the follower.



A_f is a function of the spring stiffness S, the initial spring compression x_0 and the effective mass mf. When the cam contacts the follower the acceleration of the follower, A_f, is equal to acceleration, A, of the cam along the axis of the follower. This A can be derived from the cam's displacement diagram and is a function of the angular velocity of the cam. During a rise phase, A should be larger than or equal to A_f to keep the contact and during a fall phase, A should be smaller than or equal to A_f for the same reason.

At a point of follower bounce, the contact force F is equal to zero. The corresponding cam velocity is called bounce velocity.

Then $- m_f * g - S(X + x_0) = m_f * Af$

To prevent bounce, the F must be larger than zero:

That is

$$A \geq A_f = -\frac{\left[m_f g + S(X + x_0)\right]}{m_f}$$

$$\frac{\left[m_f g + S(X + x_0)\right]}{m_f} \geq A$$

also $A = k*\omega^2$,where k is dictated by the contour of the cam.

Therefore, the condition to prevent bounce is:

$$\frac{\left[m_f g + S(X + x_0)\right]}{m_f} \geq k\omega^2$$

EXPERIMENTAL PROCEDURE:

Before actually beginning the experimental procedure the following precautions are to be followed. Make sure that the nut securing the cam on the shaft is tight. Then revolve the cam through one or two revolutions, by means of the flywheel, to ensure that the spring pre-tension is not excessive. Then switch on the mains electrical supply and rotate the control knob slowly clockwise. The motor will then begin to rotate. Should the motor stall - this may occur if the follower is in contact with the leading flank of the cam when starting - return the control knob to zero and rotate the camshaft by hand in the opposite direction to that of the motor, until the follower is in contact with the other flank. Now rotate the control knob clockwise and the camshaft should revolve. The initial displacement xo has been adjusted to be zero.

WARNING:BEFORE REVERSING THE DIRECTION OF ROTATION OF THE MOTOR ENSURE THAT THE MAINS A.C. SUPPLY IS SWITCHED OFF, OTHERWISE THE MOTOR FUSES WILL BE BLOWN.

Perform the following the steps for the experiment.

1. Insert the pen into the penholder and allow it to gently touch the paper wrapped around the drum.
2. Make the electrical connections between the apparatus, E3MKII Speed controller and the AC power supply.
3. Fix the cam in the machine and rotate the flywheel to record the displacement diagram for the cam.
4. Release the pen and turn on the controller with the speed knob at minimum.
5. Gradually increase the cam speed until the bounce speed is captured. The bouncing speed is at which the cam begins to chatter and a loud banging noise is heard.
6. Measure the bouncing speed using a tachometer.
7. Repeat the same for the other cam.

Changing the Cam Follower and Spring

Remove the front Perspex cover, and withdraw the cam by unscrewing the nut on the shaft, and relieving the follower pressure on the Cam. Unfasten the two knurled knobs securing the top cross member to the two vertical pillars, and remove cross member and spring assembly. The spring may then be removed by unfastening the two locking screws and a different spring substituted. The follower was then be removed by slackening off the Alien screw securing the guide bar. Assembly is the reverse sequence of the above operation.

Marks Obtained	Signature by faculty	Date of checking