JAIPUR INSTITUTE OF TECHNOLOGY-GROUP OF INSTITUTIONS

DEPARTMENT OF MECHANICAL ENGINEERING

MECHANICAL VIBRATION LAB MANUAL

VI Semester B.E. Mechanical Engineering

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Name:______________________________________

Roll No:___________ **Sem**___________ **Yr**_________

Subject Name ______________________________

Subject Code ______________________________

DEPARTMENT OF MECHANICAL ENGINEERING

VISION OF THE DEPARTMENT

Department of mechanical engineering is committed to prepare graduates, post graduates and research scholars by providing them the best outcome based teaching-learning experience and scholarship enriched with professional ethics.

MISSION OF THE DEPARTMENT

M-1: Prepare globally acceptable graduates, post graduates and research scholars for their lifelong learning in Mechanical Engineering, Maintenance Engineering and Engineering Management.

M-2: Develop futuristic perspective in Research towards Science, Mechanical Engineering Maintenance Engineering and Engineering Management.

M-3: Establish collaborations with Industrial and Research organizations to form strategic and meaningful partnerships.

PROGRAM SPECIFIC OUTCOMES (PSOs)

PSO1 Apply modern tools and skills in design and manufacturing to solve real world problems.

PSO2 Apply managerial concepts and principles of management and drive global economic growth.

PSO3 Apply thermal, fluid and materials fundamental knowledge and solve problem concerning environmental issues.

PROGRAM EDUCATIONAL OBJECTIVES (PEOS)

PEO1: To apply industrial manufacturing design system tools and necessary skills in the field of mechanical engineering in solving problems of the society.

PEO2: To apply principles of management and managerial concepts to enhance global economic growth.

PEO3: To apply thermal, fluid and materials engineering concepts in solving problems concerning environmental pollution and fossil fuel depletion and work towards alternatives.

PROGRAM OUTCOMES (POS)

PO1 **Engineering knowledge**: Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems.

PO2 **Problem analysis**: Identify, formulate, review research literature, and analyze complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.

PO3 **Design/development of solutions**: Design solutions for complex engineering problems and design system components or processes that meet the specified needs with appropriate consideration for the public health and safety, and the cultural, societal, and environmental considerations.

PO4 **Conduct investigations of complex problems**: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.

PO5 **Modern tool usage**: Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modeling to complex engineering activities with an understanding of the limitations.

PO6 **The engineer and society**: Apply reasoning informed by the contextual knowledge to assess societal, health, safety, legal and cultural issues and the consequent responsibilities relevant to the professional engineering practice.

PO7 **Environment and sustainability**: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.

PO8 **Ethics**: Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.

PO9 **Individual and team work**: Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.

PO10 **Communication**: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.

PO11 **Project management and finance**: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one‟s own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.

PO12 **Life-long learning**: Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

DESIGN LABORATORY

COURSE OBJECTIVES:

1. To demonstrate the concepts discussed in Design of Machine Elements, Mechanical Vibrations & Dynamics of Machines courses.
2. To visualize and understand the development of stresses in structural members and experimental determination of stresses in members utilizing the optical method of reflected photo-elasticity.

COURSE CONTENT

Part-A

1. Determination of natural frequency of a spring mass system.
2. Determination of natural frequency logarithmic decrement, damping ratio and damping Co-efficient in a single degree of freedom vibrating systems (longitudinal and torsional)
3. Determination of critical speed of rotating shaft.
4. Balancing of rotating masses.

Part-B

5. Determination of fringe constant of Photo-elastic material using Circular disk subjected diametric compression, Pure bending specimen (four point bending)
6. Determination of equilibrium speed, sensitiveness, power and effort of Porter/Hartnell Governor.
7. Determination of pressure distribution in Journal bearing
8. Experiments on Gyroscope (Demonstration only)

COURSE OUTCOMES:

Upon completion of this course, students should be able to:

CO1 To practically relate to concepts discussed in Design of Machine Elements, Mechanical Vibrations & Dynamics of Machines courses.

CO2 To measure strain in various machine elements using strain gauges and determine strain induced in a structural member using the principle of photo-elasticity.

CONTENTS

Experiment #1

SPRING MASS SYSTEM

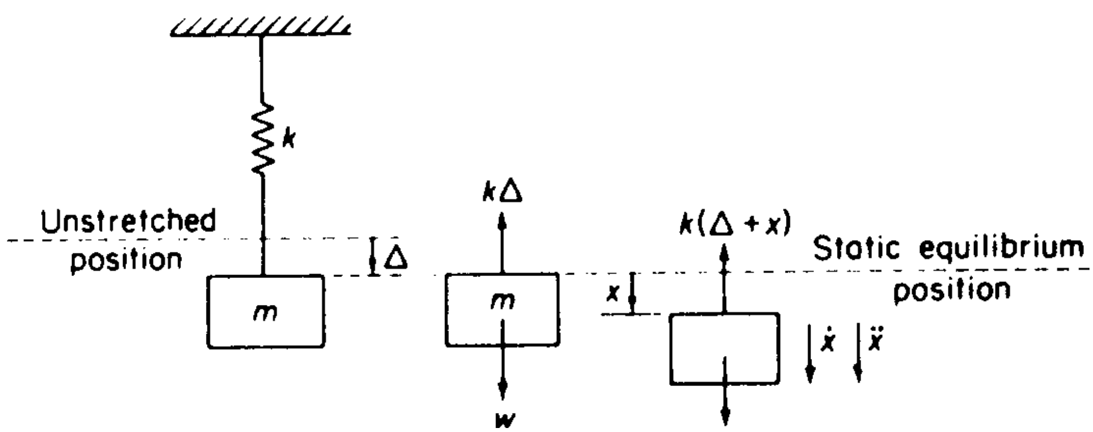
Aim:

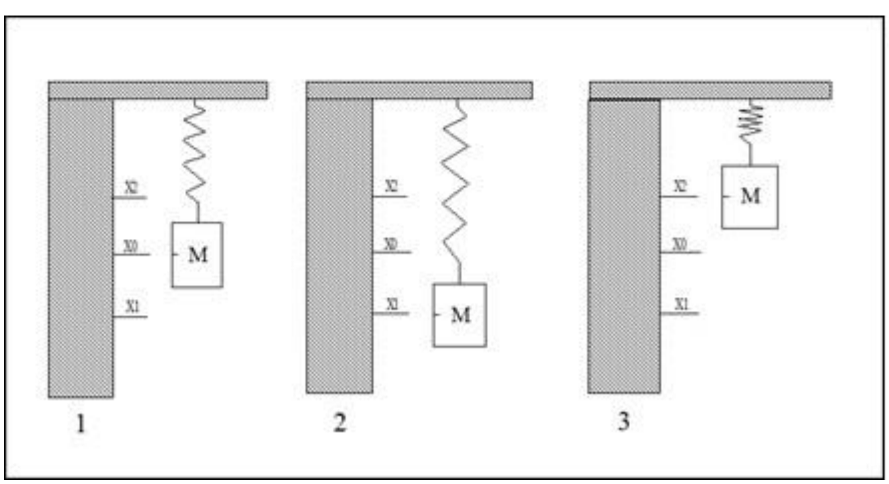
1. To determine the spring constant of the given spring.
2. To determine the natural frequency and compare the same with the Theoretical frequency of:

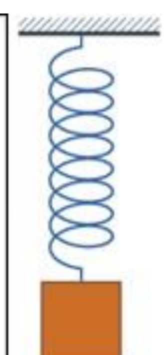
   a. Springs in parallel
   b. Spring Mass System
   c. Spring in Series

Apparatus: Springs, Rigid Frame, Scale, Stop Watch, Pan and Weights.

Theory: Students should write about static equilibrium position, natural frequency, derive expression for natural frequency for free vibrating body, derive expression for springs in series and parallel.







Equation of Motion: Natural Frequency

Above figure shows a simple undamped spring-mass system, which is assumed to move only along the vertical direction. It has one degree of freedom (DOF), because its motion is described by a single coordinate x.

When placed into motion, oscillation will take place at the natural frequency f_n which is a property of the system. We now examine some of the basic concepts associated with the free vibration of systems with one degree of freedom.

Newton's second law is the first basis for examining the motion of the system. As shown in Figure. The deformation of the spring in the static equilibrium position is Δ, and the spring force $k\Delta$ is equal to the gravitational force w acting on mass „m‟

$$\mathrm{k}\Delta = w \ = mg$$

By measuring the displacement x from the static equilibrium position, the forces acting on „m' are $k\ \Delta + x$ and „w'. With „x' chosen to be positive in the downward direction, all quantities - force, velocity, and acceleration are also positive in the downward direction.

We now apply Newton's second law of motion to the mass m:

$$mx \qquad F \ = w - k(\Delta + x)$$

and because $k\Delta = w$, we obtain :

$$mx \ = \ -kx \quad \text{where } \omega_n^2 \ = \frac{\lambda}{m}$$

$$x \quad + \omega_n^2$$

and we conclude that the motion is harmonic. A homogeneous second order linear differential equation has the following general solution:

$$\boldsymbol{x = A \sin \omega_n t + \ B \cos \omega_n t}$$

where „A‟ and „B' are the two necessary constants. These constants are evaluated from initial conditions x 0 and$x$ 0 ,

$$\boldsymbol{x = \frac{x\ 0}{\omega_n} \sin \omega_n t + x\ 0 \cos \omega_n\ t}$$

The natural period of the oscillation is established from, $\omega_n t = 2\pi$ or

$$t = 2\pi\sqrt{\frac{m}{k}}$$

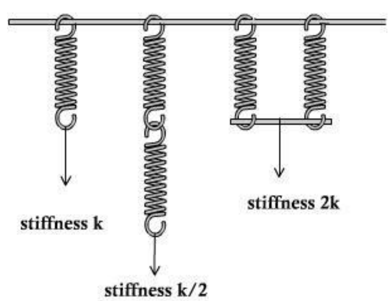
and the natural frequency is

$$f_n = \frac{1}{t} = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

These quantities can be expressed in terms of the static deflection „Δ‟ by observing , $k\Delta = mg$. Thus natural frequency can be expressed in terms of the static deflection „Δ‟ as

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\Delta}}$$

Note that τ, f_n and ω_n, depend only on the mass and stiffness of the system, which are properties of the system.



Spring in Series $$\frac{1}{k_{eq}} = \frac{1}{k_1} + \frac{1}{k_2}$$

Springs in parallel $$k_{eq} = k_1 + k_2$$

Procedure:

1. To determine the spring stiffness:

a. Take the initial length of the spring.
b. Fix the spring to the frame and the pan to the other end of the spring.
c. The spring stretches due to pan weight. Take the final length.
d. Add weights to the pan and take the final length of the spring. corresponding to the weights added and tabulate the results.
e. The above procedure is repeated for other springs.

Sl NO.	Weight in Kgs	Initial Length in mm	Final Length in mm	Static deflection in mm	Spring stiffness in N/mm	Average spring stiffness in N /mm
1.						
2.						
3.						

$$k\Delta = w = mg$$

$$\boldsymbol{k} = \frac{\boldsymbol{mg}}{\Delta} \ \textbf{N/ mm}$$

2. To determine the natural frequency of spring mass system and compare it with the theoretical frequency:

Procedure:

a. Fix the spring to the rigid frame and attach pan to it.
b. Stretch the pan down and release it. The system oscillates. Note down the time taken for 10 oscillations.
c. Repeat the above step with different weights on the pan and tabulate the results.

Sl No.	Weight in Kgs	Times taken for 10 oscillations in sec	Frequency = n/t CPS or HZ	Theoretical frequency in CPS or HZ	Error= theoretical frequency- actual frequency

K = spring stiffness used in the experiment in N /mm

$$f_n = \frac{1}{T} = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \text{ in cps or Hz}$$

3. To determine the natural frequency of spring mass system for springs in series and compare it with the theoretical frequency:

Procedure:

a. Fix the spring to the rigid frame. Attach the second spring to the first spring and attach pan to it.
b. Stretch the pan down and release it. The system oscillates. Note down the time taken for 10 oscillations.
c. Repeat the above step with different weights on the pan and tabulate the results

Sl No.	Weight in Kgs	Times taken for 10 oscillations in sec	Frequency = n/t CPS or HZ	Theoretical frequency in CPS or HZ	Error = theoretical frequency- actual frequency

Where k_e is the equivalent spring stiffness in N /mm

$$f_n = \frac{1}{T} = \frac{1}{2\pi}\sqrt{\frac{k_e}{m}} \text{ in CP or Hz}$$

4. To determine the natural frequency of spring mass system for springs in Parallel and compare it with the theoretical frequency:

Procedure:

a. Fix the spring to the rigid frame. Attach the second spring parallel to the first spring. Insert a rod and attach pan at the center of the rod.
b. Stretch the pan down and release it. The system oscillates. Note down the time taken for 10 oscillations.
c. Repeat the above step with different weights on the pan and tabulate the results.

Sl No.	Weight in Kgs	Times taken for 10 oscillations in sec	Frequency = n/t CPS or HZ	Theoretical frequency in CPS or HZ	Error=Theoretical frequency- actual frequency

Specimen Calculation:

Natural frequency, (theoretical) $\omega_n = \sqrt{\frac{k_e}{m}}$ rad /sec

$$f_n = \frac{1}{T} = \frac{1}{2\pi}\sqrt{\frac{k_e}{m}}$$

Inference of the Results:

Experiment #2

TORSIONAL VISCOUS DAMPER SYSTEM

Aim: To determine the natural frequency of the damped system, logarithmic decrement, damping ratio and damping factor for different depth of immersion of the disc in a viscous medium (OIL).

Apparatus: Torsional system with a rod and disc attached to the rigid frame, an oil bath and weights to increase the depth of immersion.

Theory: Students have to write about damping and its effect on the system, different types of damping, derive expression for logarithmic decrement and establish a relation between logarithmic decrement and damping ratio.

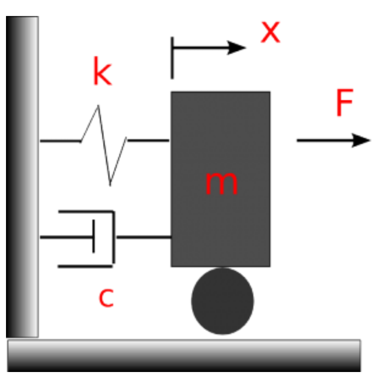
Viscous Damping

It is encountered by bodies moving at moderate speed through liquid. This type of damping leads to a resisting force proportional to the velocity. The damping force

$$F_d \propto \frac{dx}{dt}$$

$$F_d = cx$$

„c" is the constant of proportionality and is called viscous damping Co-efficient. With the dimension of N-S/m



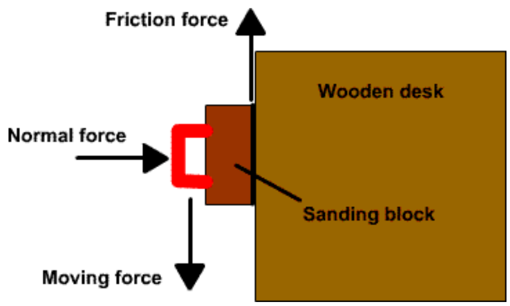
Coulomb Damping

This type of damping arises from sliding of dry surfaces. The friction force is nearly constant and depends upon the nature of sliding surface and normal pressure between them as expressed by the equation of kinetic friction

$$F = \mu N$$

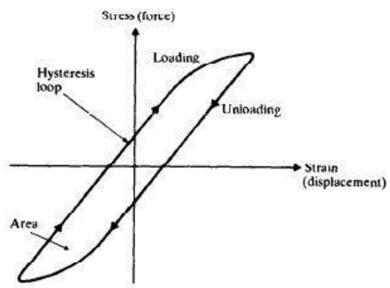
Where μ = co-efficient of friction

N = normal force



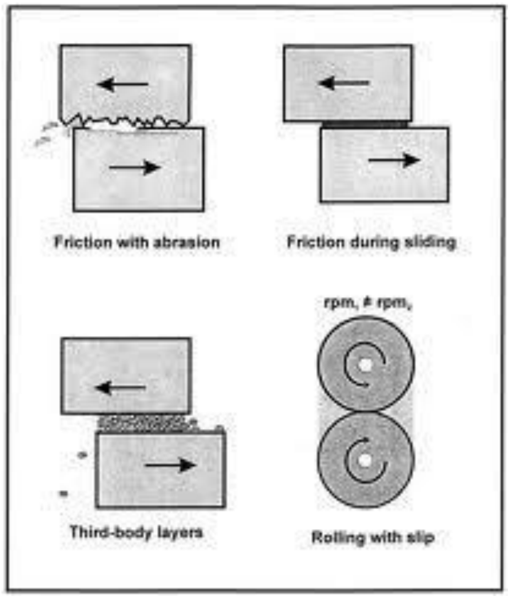
Solid or structural Damping

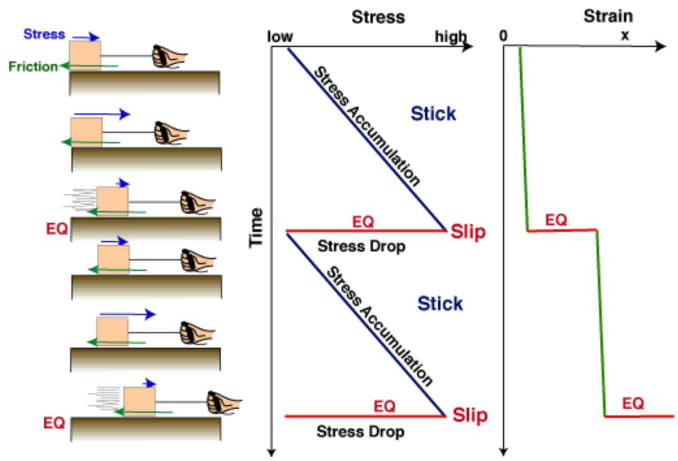
This is due to internal friction within the material itself. Experiment indicates that the solid damping differs from viscous damping in that it is independent of frequency and proportional to maximum stress of vibration cycle.



Slip or Intrefacial damping

Energy of vibration is dissipated by microscopic slip on the interfaces of machine parts in contact under fluctuating loads. Microscopic slip also occurs on the interface of the machine elements having various types of joints. This type is essentially of a linear type.


Friction with abrasion
Friction during sliding
rpm, ≠ rpm,
Third-body layers
Rolling with slip


Stress
Friction
EQ
EQ
Stress
low
high
Time
Stress Accumulation
Stick
EQ
Slip
Stress Drop
Stress Accumulation
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Slip
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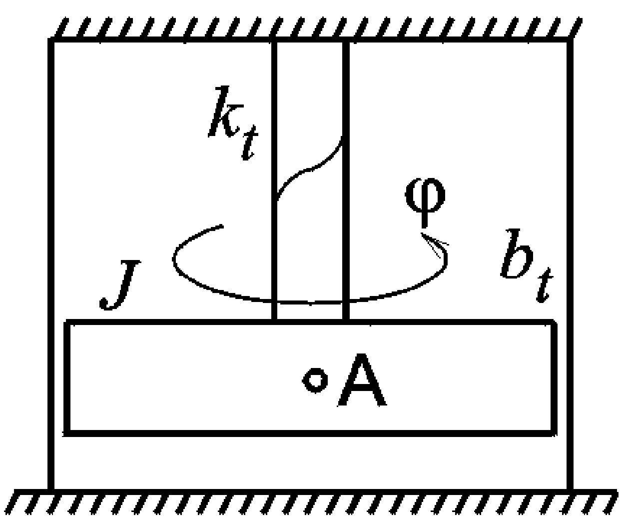
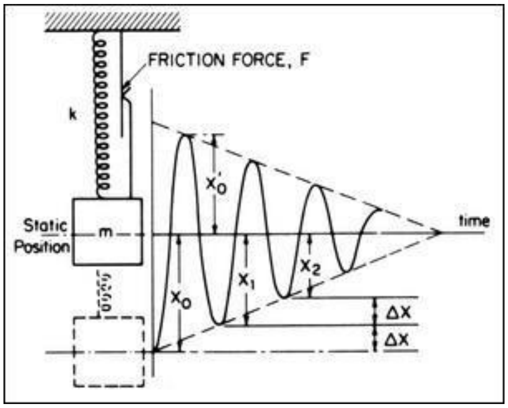
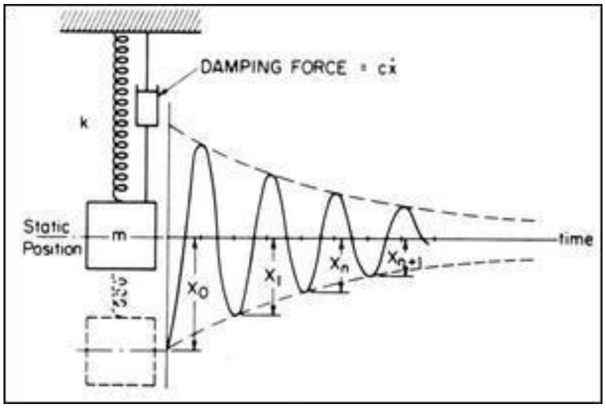
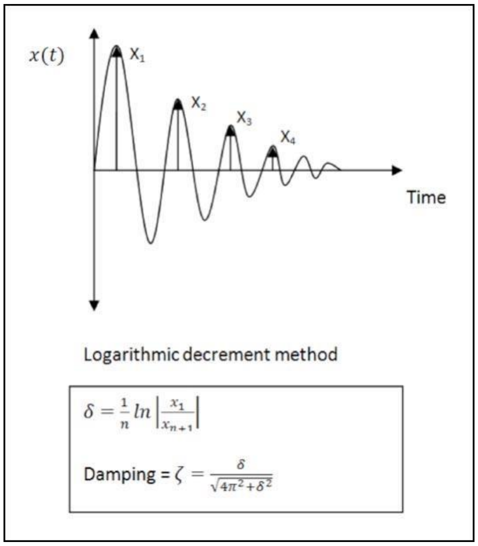
Strain
0
x
EQ
EQ


k_t
φ
J
b_t
A







Observation:

1. Diameter of the rod = mm
2. Diameter of the disc = mm
3. Length of the rod = mm
4. Shear modulus of rigidity for the rod material = GPa

Procedure:

i. Keep the torsional system such that the disc is not immersed in the oil. (case of no damping)
ii. Wrap a white paper around the drum to note down the signature of the vibration produced by the torsional system.
iii. Rotate the drum and release it. The system vibrates take the signature of the vibration on the white paper.

iv. Raise the oil drum so as to dip the disc of the torsional system in the oil to produce viscous damping. This is done by placing weights below the drum. Twist the drum and take down the signature on the white paper.
v. Repeat the procedure for three depths of immersion and tabulate the results.
vi. Plot a graph of depth of immersion V/S damping coefficient.

Sl No.	**Depth of Immersion in mm**	θ_0 **mm**	θ_1 **mm**	**Static deflection δ in mm**	ζ	**Natural frequency of the damped system ω_d in rad /sec**	**Damping Co-efficient 'C' N-sec/m**
1.							
2.							
3.							
4.							

Specimen Calculation: logarithmic decrement = $\boldsymbol{\delta = ln \frac{\theta_0}{\theta_1}}$

$$\zeta = \frac{\delta}{\sqrt{\delta^2 + 4\pi^2}}, \quad \zeta = \frac{C}{C_c}, \text{ where } C_c = 2\sqrt{mk} \text{ N- sec/m.}$$

Therefore, $C = C_c \zeta$ N-sec/m.

To determine ω_d

$$\omega_n = \sqrt{\frac{K_t}{J}} \quad \textbf{where } K_t = \frac{GL}{\theta} \quad \textbf{N-m-sec/rad}$$

and J = polar Moment of Inertia $= \frac{\pi d^4}{32}$

Draw a graph of **depth of Immersion V/S damping co efficient C**.

Inference of result:

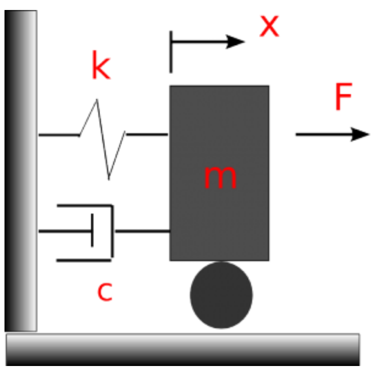
Experiment #3

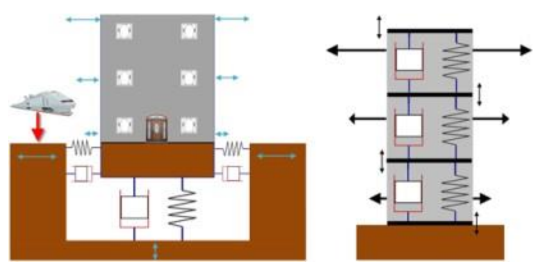
SPRING MASS DAMPER SYSTEM

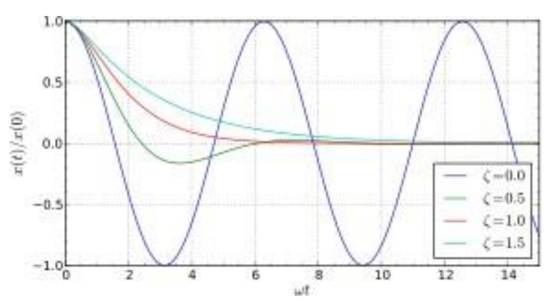
Aim: To determine the natural frequency of the damped system, logarithmic decrement, damping ratio and damping factor.

Apparatus: spring mass damper system and a rotating drum to note down the signature.

Theory: student should obtain the differential equation for a spring mass system and the solution for under damped case, critically damped case and over damped case. He should represent the same with a sketch.







Observation:

Spring stiffness = N/mm

Rotation speed of the drum = rps

Procedure:

i. Wrap a white paper around the drum to note down the signature of the vibration produced by the spring mass system.

ii. Place a weight on the pan and release the system to vibrate. Take the signature on the rotating drum.
iii. Repeat the procedure for three weights and tabulate the results.
iv. Plot a graph of depth of immersion V/S damping coefficient.

Sl No.	Mass in kg	Wave length λ in m	θ_0 mm	θ_1 mm	Static deflection δ in mm	ζ	Natural frequency of the damped system ω_d in rad /sec	Damping Co-efficient 'C' N-sec/m
1.								
2.								
3.								
4.								

Specimen Calculation: $f_d = \frac{V}{\lambda}$ **cps or Hz.**

where V = velocity in m/sec and λ is in m

logarithmic decrement = $\delta = \ln \frac{\theta_0}{\theta_1}$

$\zeta = \frac{\delta}{\sqrt{\delta^2+4\pi^2}}$, $\zeta = \frac{C}{C_c}$ where $C_c = 2\sqrt{mk}$ N- sec/m

Therefore, $C = C_c\zeta$ N-sec/m.

Inference of result:

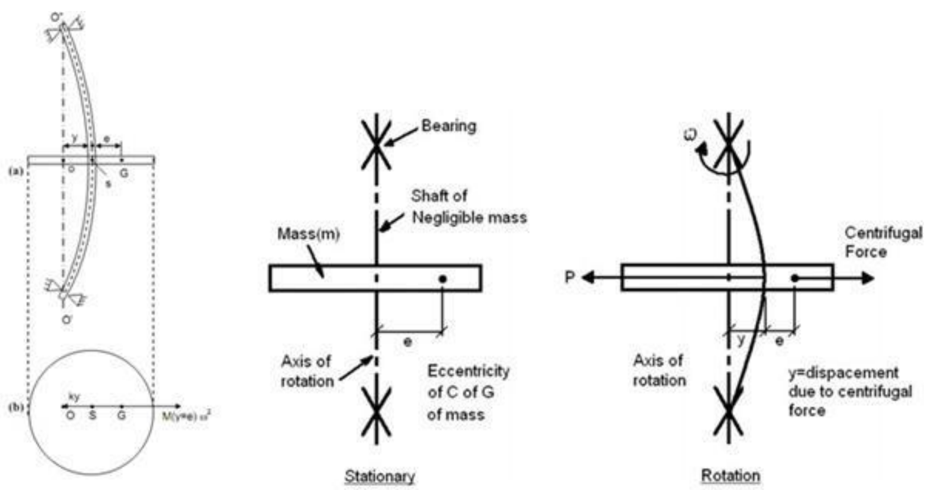
Experiment #4

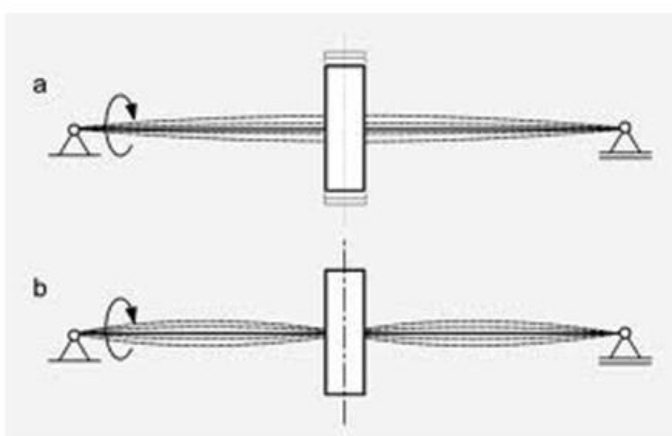
CRITICAL SPEED OF SHAFT OR WHIRLING OF SHAFT

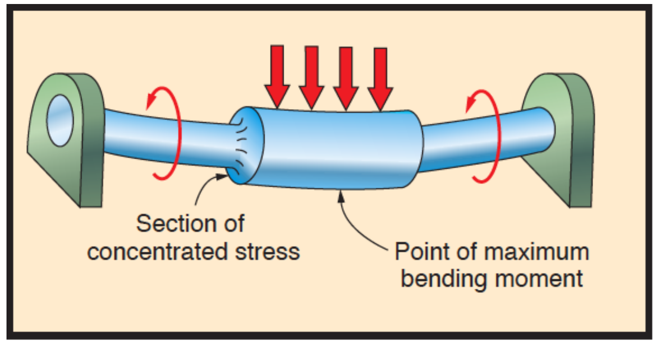
Aim: To determine the critical speed of the shaft.

Apparatus: Disc mounted at the center of a rotating shaft, the shaft is supported in bearings, a motor is coupled to the shaft and the motor is connected to the variac for speed adjustment. A stroboscope is provided to note down the speed of the shaft.

Theory: students are expected to write about critical speed of shat or whirling of shaft or whipping of shaft and it importance. The student should derive an expression for critical speed of shat and discuss the four conditions in critical speed of shaft.







Shafts and axles often have stepped geometry to accommodate gears and pulleys and to restrict axial displacement. These sudden changes in cross section, as well as features like notches and holes, are local stress intensifiers and potential trouble spots for fatigue.

$$\frac{X}{e} = \frac{\frac{m\omega^2}{M}}{\sqrt{\left(\frac{k}{M} - \omega^2\right)^2 + \left(\frac{c}{M}\omega\right)^2}}$$

$$\frac{X}{e}\frac{M}{m} = \frac{\left(\frac{\omega}{\omega_n}\right)^2}{\sqrt{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(\frac{2\zeta\omega}{\omega_n}\right)^2}}$$

Since, $\frac{c}{m} = 2\zeta\omega_n$ and $\omega_n = \sqrt{\frac{k}{m}}$

$$\tan\phi = \frac{2\zeta\frac{\omega}{\omega_n}}{1 - \left(\frac{\omega}{\omega_n}\right)^2}$$

$$\frac{c\omega}{k - m\omega^2}$$

Critical (Whirling) Speed of Shafts

Introduction

For a rotating shaft there is a speed at which, for any small initial deflection, the centripetal force is equal to the elastic restoring force. At this point the deflection increases greatly and the shaft is said to “whirl”. Below and above this speed this effect is very much reduced. This critical (whirling speed) is dependent on the shaft dimensions, the shaft material and the shaft loads. The critical speed is the same as the frequency of traverse vibrations.

The critical speed „N_c′of a shaft is simply

$$N_c = \frac{\sqrt{\frac{k}{m}}}{2\pi}$$

Where m = the mass of the shaft assumed concentrated at single point.
„k‟ is the stiffness of the shaft to traverse vibrations

For a horizontal shaft this can be expressed as

$$N_c = \frac{\sqrt{\frac{g}{y}}}{2\pi}$$

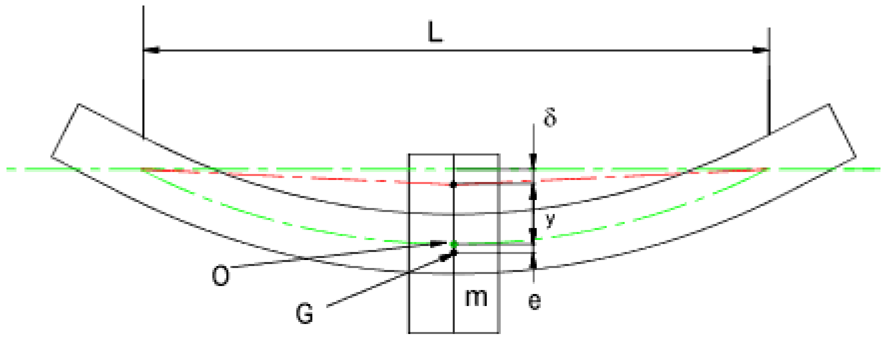
Where y = the static deflection at the location of the concentrated mass.

Symbols

m	= Mass (kg)	E	=Young‟s Modulus (N/m^2)
N_c	= critical speed (rev/s)	I	= Second Moment of Area (m^4)
g	= acceleration due to gravity (m.s^{-2})	y	= deflection from δ with shaft rotation
O	= centroid location	δ	= static deflection (m)
G	= Centre of Gravity location	ω	= angular velocity of shaft (rads/s)
L	= Length of shaft		

Theory

Consider a rotating horizontal shaft with a central mass (m) which has a centre of gravity (G) slightly away from the geometric centroid (O)



The centrifugal force on the shaft = m ω ^{2}(y + e) and the inward pull exerted by the shaft $y = \frac{48EI}{L^3}$ The more general formulae for the restoring traverse force of the beam is y (K EI / L ^{3}) where K = a constant depending on the position of the mass and the end fixing conditions.

Equating these forces…

$$m\,\omega^2(y+e) = y\,\frac{KEI}{L^3} \quad \text{therefore} \quad y = \frac{e}{\left(\frac{KEI}{m\omega^2 L^3} - 1\right)}$$

When the denominator = 0, that is [KEI / m ω ^{2} L ^{3}] = 1 , the deflection becomes infinite and whirling takes place.

The whirling or critical speed is therefore.

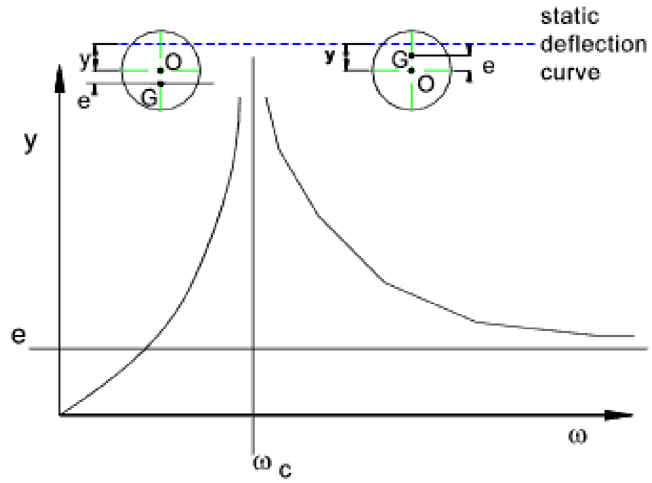
$$\omega_c = \sqrt{\frac{KEI}{mL^3}}$$

For a simply supported beam with a central mass K = 48. See examples below

Substituting $\omega_c^{\,2}$ for KEI /mL^3 in the above equation for y results in the following equation which relates the angular velocity with the deflection.

$$y = \frac{\omega^2 . e}{(\omega_c^2 - \omega^2)}$$

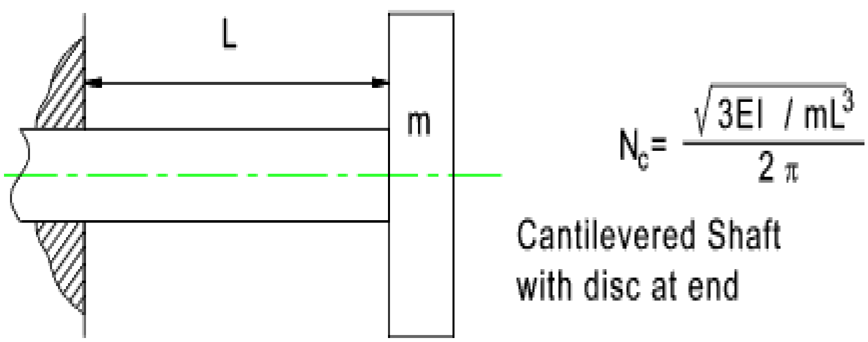
This is plotted below.



This curve shows the deflection of the shaft (from the static deflection position) at any speed ω in terms of the critical speed.
When $\omega < \omega_c$ the deflection y and e have the same sign that is G lies outside of O. When $\omega > \omega_c$ then y and e are of opposite signs and G lies between the centre of the rotating shaft and the static deflection curve. At high speed G will move such that it tends to coincide with the static deflection curve.

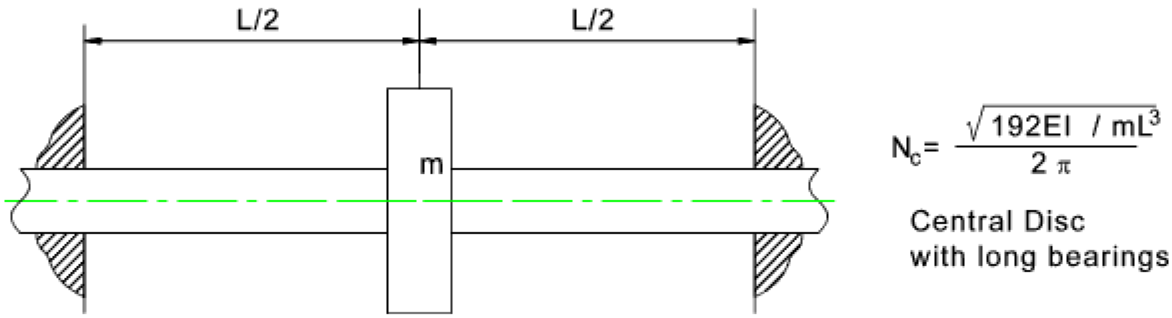
Cantilever rotating mass

Mass of shaft neglected



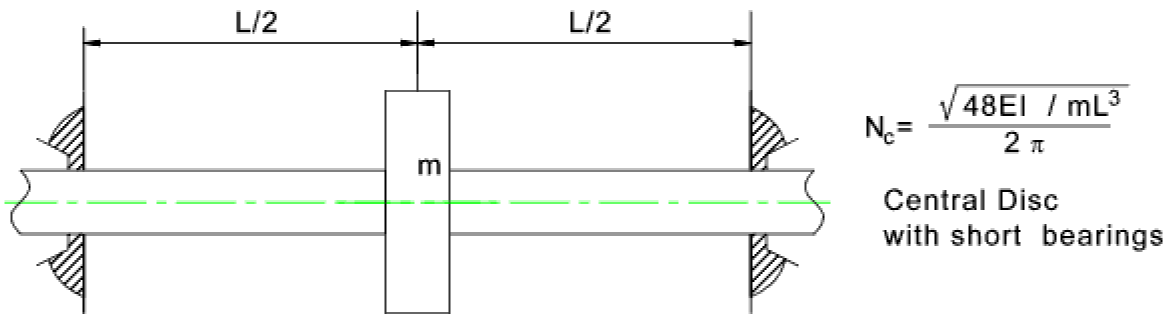
Central rotating mass- Long Bearings

Mass of shaft neglected



Central rotating mass – Short Bearings

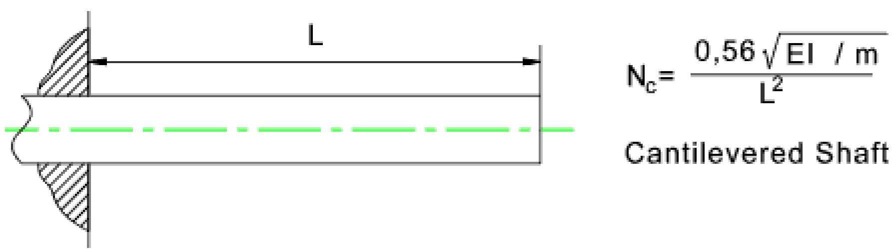
Mass of shaft neglected



Non-Central rotating mass – Short Bearings

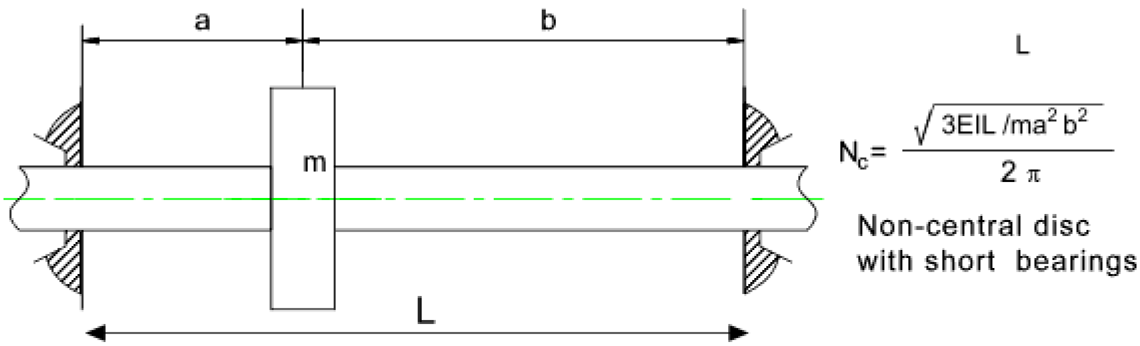
Mass of shaft neglected

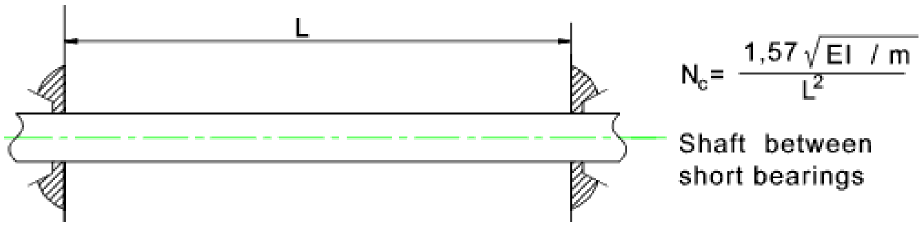
Cantilevered Shaft



m = mass /unit length

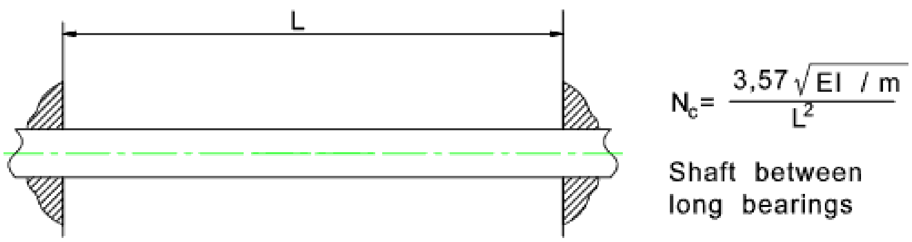
Shaft Between short bearings





m = mass /unit length

Shaft Between long bearings

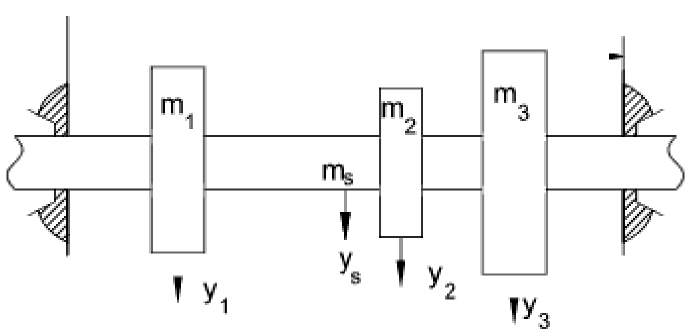


m = mass /unit length

Combined loading

This is known as Dunkerley"s method an is based on the theory of superposition….

$$\frac{1}{N_c^2} = \frac{1}{N_s^2} + \frac{1}{N_1^2} + \frac{1}{N_2^2} + \ldots\ldots$$



Procedure:

i. Measure the distance between the supports and fix the disc at the center of the shaft.
ii. Slowly rotate the variac so as to rotate the shaft.
iii. Keep on observing for the vibration on the shaft.
iv. When the shaft starts vibrating observe the amount of vibration in the shaft.
v. Note down the speed at which the shaft vibrates violently. This will be the critical speed of the shaft.
vi. Repeat the procedure at least three times and note down the critical speed. Tabulate the result and compare it with the theoretical critical speed.

Sl No.	Critical speed in rpm (actual)	Critical speed theoretical in rpm	Error
1.			
2.			
3.			

Inference:

Experiment # 5

SPRING CONTROLLED GOVERNOR

Aim: To determine the equilibrium speed at different radius of rotation and to draw the controlling force diagram.

Apparatus: Governor, Motor and Variance.

Theory:

The function of governor is, in all operating conditions, to adjust the fuel injection pump and ensure the stable operation of diesel engine at regulated speed when the load changes. The loads and speeds of cars, tractors and diesel engine are constantly changing. Climbing automobiles and sailing ships are often influenced by wind, water and other environmental factors, which leads to changes in engine load. If the ship is in a rough sea, the storm may make the ship swing left and right, after and forth, sometimes, even the propeller expose in the air and the load reduce suddenly, then if the supplication of the fuel can"t be reduced in time, the speed of the engine will escalate suddenly, and lead to "runaway". Conversely, a sudden increase in engine load, particularly in the low-speed operation, if the oil supplication cannot increase promptly, the engine may stops working. As the engine load changes large and quickly, it is difficult for manual control of fuel supplication, and even impossible. In order to adjust the fuel supplication to the load changes automatically, and ensure stabled engine operation, governor is equipped.

In addition, in the diesel generators, to ensure the stable voltage and constant speed operation, diesel is also equipped with governors. In short, the functions of governor can be summarized as follows:

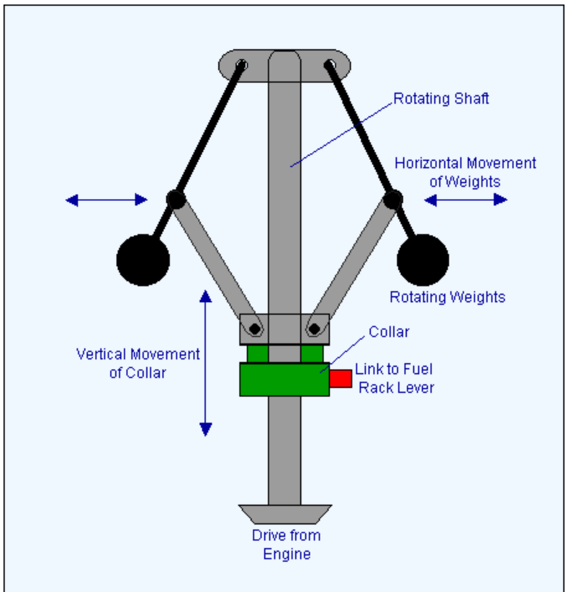
To prevent speeding diesel (runaway) operation-control the maximum speed;

Ensure stable operation at low speed-control minimum stable speed; control the engine run at regulated speed. With the load outside changes, automatically adjust fuel supplication; ensure the engine always operates at regular speed stabile.

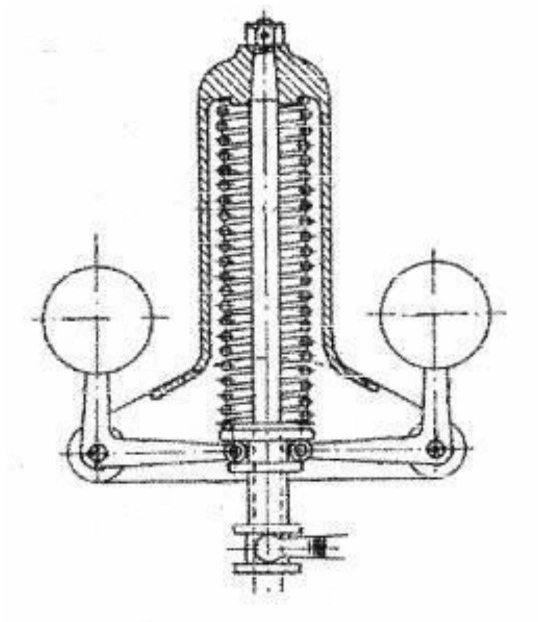
According to the different control mechanism, governor can be classified into:

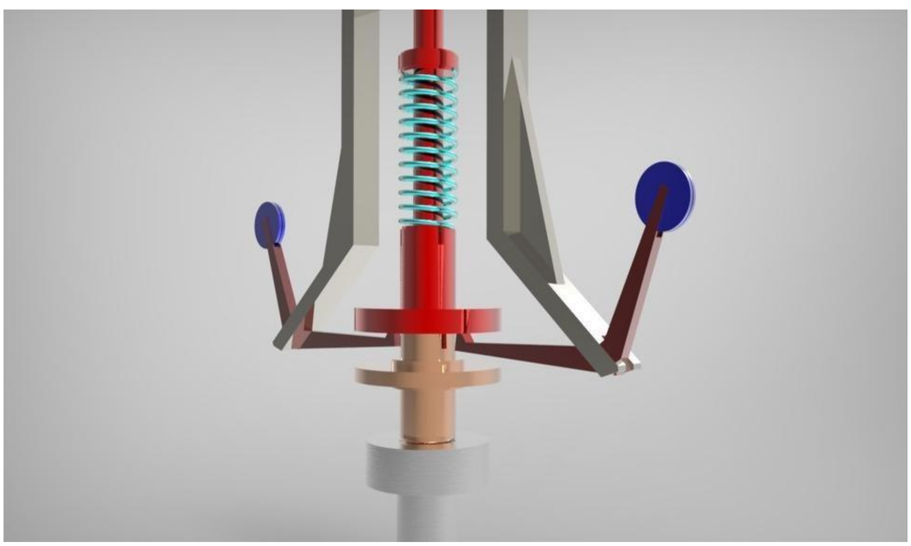
Dead weight governor and Spring Controlled governor. Dead weight governors are Porter and Proell Governor Hatnell Governor is a spring controlled governors.

Note: The student is expected to derive an express for spring controlled governor used in the set up. The student should also write about sensitiveness, equilibrium speed, isochronisms and stability and effort of a governor.



Simplified diagram of diesel engine governor





Total lift = $(x_1+x_2) = (b\,\theta_1 + b\,\theta_2)$

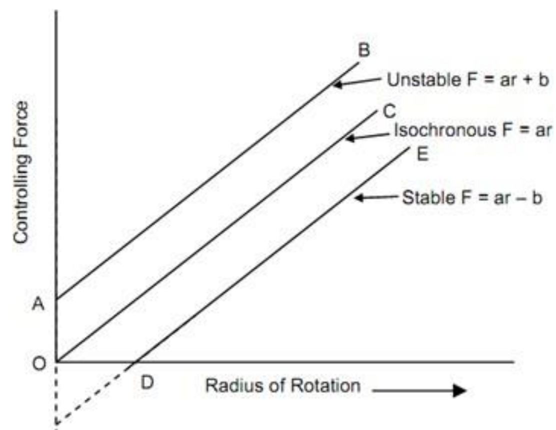
$= b\,(\theta_1+\theta_2)$

$= b\left(\frac{(r-r_1)}{a} + \frac{(r_2-r)}{a}\right) = \frac{b}{a}(r_2 - r_1)$

$s_2 - s_1$ = Total lift $\times$ s = $\frac{b}{a}(r_2 - r_1)\,s$

$\therefore \quad (F_c)_2 - (F_c)_1 = \left(\frac{b}{a}\right)^2 \frac{(r_2-r_1)}{2}\,s$

or stiffness of spring 's' = $2\left(\frac{a}{b}\right)^2 \frac{(F_c)_2-(F_c)_1}{(r_2-r_1)}$



Observations:

Mass of each ball = Kg

Height of the sleeve = mm

Radius of Rotation= mm

Procedure: Take the various dimension of the governor like radius of rotation, height of the sleeve and ball weight.

1. Using the variance increase the speed of the governor. The fly balls fly out and the sleeve is lifted. Not down the radius of rotation and the sleeve lift for that particular speed of the governor.

2. Repeat the above step for different speeds and tabulate the results

Sl No.	Speed in rpm	Radius of rotation in mm	Sleeve lift in mm	Centrifugal Force in N
1.				
2.				
3.				
4.				

Inference:

Experiment # 6

BALANCING OF ROTATING MASSES

Aim: To determine the balancing mass for the disturbing mass in the same plane and balancing of disturbing masses in different planes.

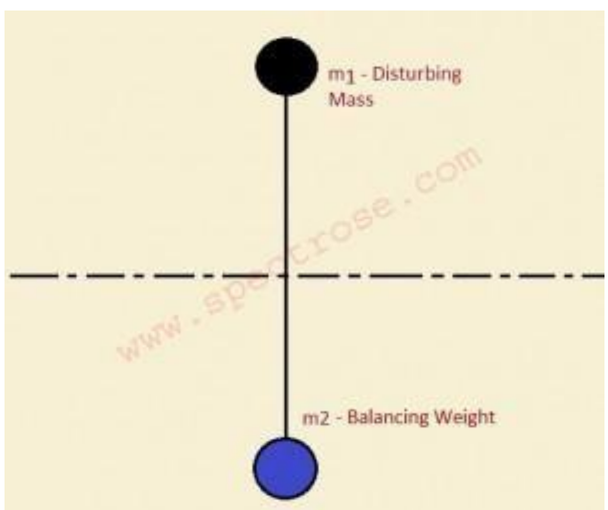
Apparatus: Balancing mass set up, weights and variance.

Theory: Consider a shaft rotating with an angular velocity „ω‟. If a mass, m is attached to that shaft, it exert a centrifugal force on the shaft which tends the shaft to bend or to vibrate it. This is an undesirable phenomenon in the case of a rotating shaft and in order to avoid these things, we are using the technique of Balancing of Rotating Masses. By this process, another mass is attached to the opposite side of the first mass so that the centrifugal force created by the second mass cancels out that produced by the first mass. By this, the vibrations in the shaft can be avoided.

The process of selecting and using the second mass in such a manner that its effect counter acts the effect of centrifugal force of the first mass is known as the Balancing of Rotating Masses.

The balancing of rotating masses at various situations is described below:

1. Balancing of Single Rotating Mass by a single mass rotating in the same plane:



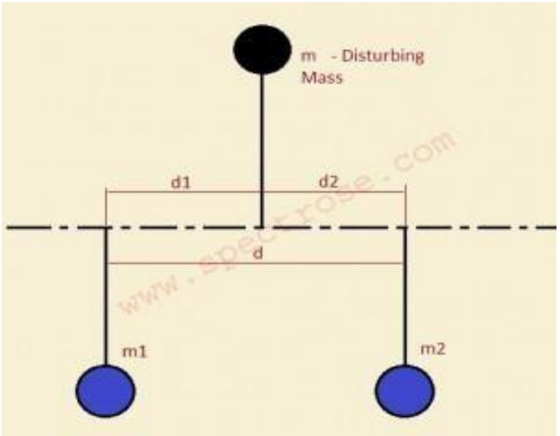
The figure shows the situation of balancing of a single rotating mass by a single mass rotating in the same plane. According to the situation,

$$FC_1 = FC_2$$

$$\rightarrow m_1 r_1(\omega)^2 = m_2 r_2(\omega)^2$$

$$\rightarrow m_1 r_1 = m_2 r_2$$

2. Balancing of Single Rotating Mass by Two rotating masses in different planes:



The figure shows the situation of balancing of a single rotating mass by Two rotating masses in different planes. According to the situation, two considerations can be taken into account.

a) $FC = FC_1 + FC_2$

$$\rightarrow mr(\omega)^2 = m_1 r_1(\omega)^2 + m_2 r_2(\omega)^2$$

$$\rightarrow mr = m_1 r_1 + m_2 r_2$$

Taking Moment about P:

$$FC_1 * d = FC * d_2$$

$$\rightarrow m_1 r_1 d = mrd_2$$

$$\rightarrow m_1 r_1 = mr(d_2/d)$$

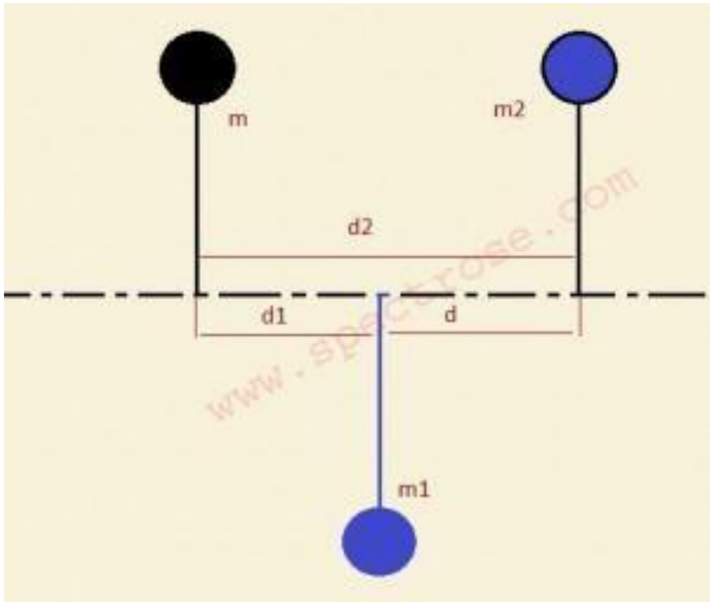
Taking Moment about Q:

$$FC_2 * d = FC * d_1$$

$$\rightarrow m_2 r_2 d = mrd_1$$

$$\rightarrow m_2 r_2 = mr(d_1/d)$$

b)



$FC+FC_2 = FC_1$

$\rightarrow mr+m_2r_2 = m_1r_1$

Taking Moments about P:

$\rightarrow m_1r_1 = mr(d_2/d)$

Taking Moments about Q:

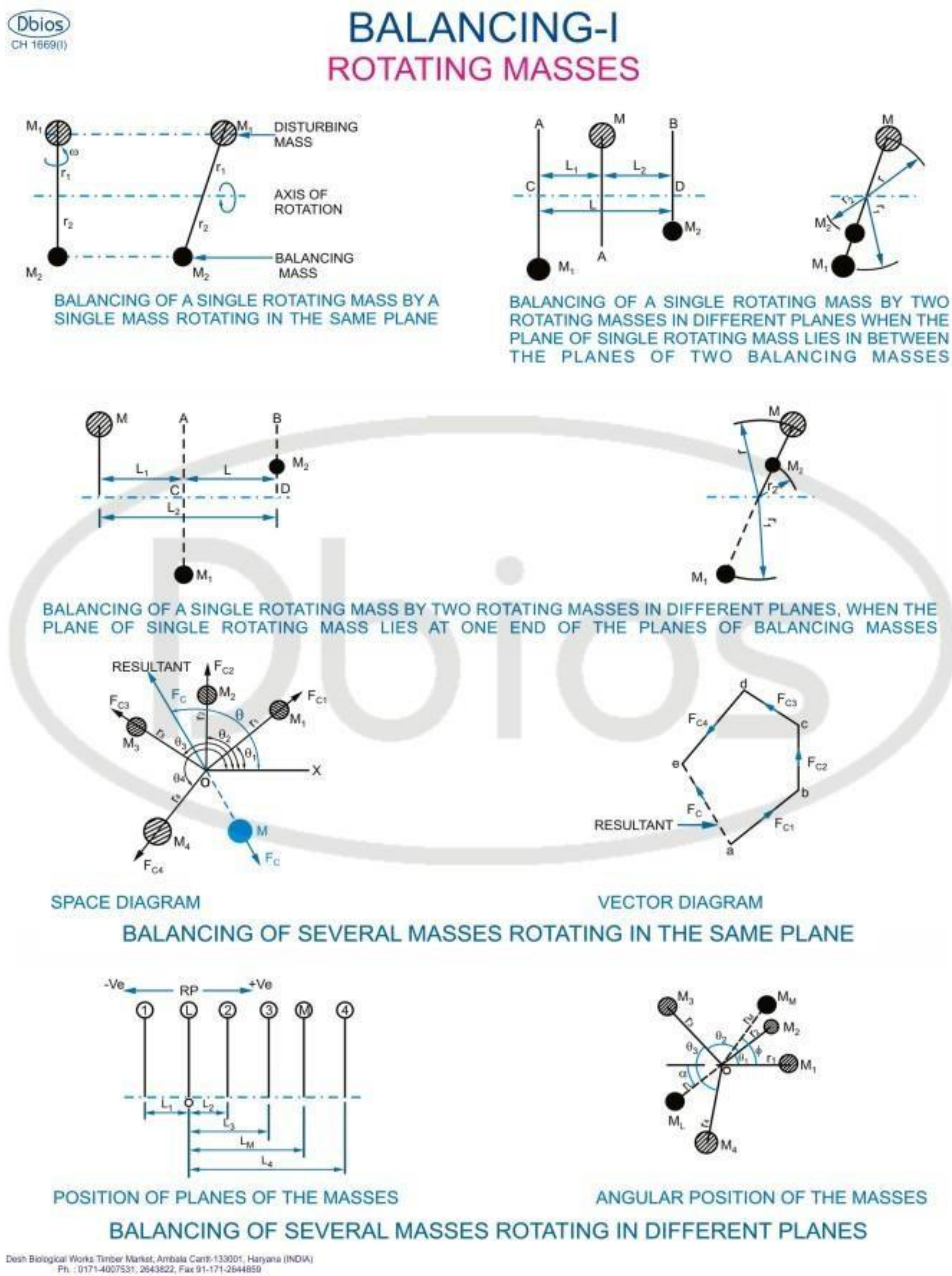
$\rightarrow m_2r_2 = mr(d_1/d)$

There are many other situations where balancing of rotating masses should be done. Examples are:

3. Several masses rotating in the same plane.

4. Several masses rotating in different planes.

These situations will be considered in the coming posts.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BALANCING-I
ROTATING MASSES
DISTURBING MASS
AXIS OF ROTATION
BALANCING MASS
BALANCING OF A SINGLE ROTATING MASS BY A SINGLE MASS ROTATING IN THE SAME PLANE
BALANCING OF A SINGLE ROTATING MASS BY TWO ROTATING MASSES IN DIFFERENT PLANES WHEN THE PLANE OF SINGLE ROTATING MASS LIES IN BETWEEN THE PLANES OF TWO BALANCING MASSES
BALANCING OF A SINGLE ROTATING MASS BY TWO ROTATING MASSES IN DIFFERENT PLANES, WHEN THE PLANE OF SINGLE ROTATING MASS LIES AT ONE END OF THE PLANES OF BALANCING MASSES
RESULTANT
RESULTANT
SPACE DIAGRAM
VECTOR DIAGRAM
BALANCING OF SEVERAL MASSES ROTATING IN THE SAME PLANE
-Ve
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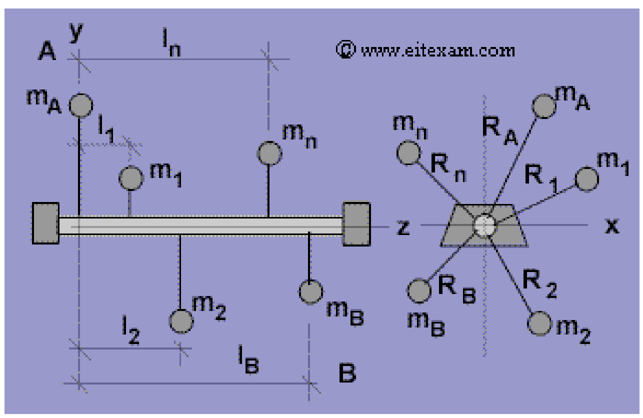
POSITION OF PLANES OF THE MASSES
ANGULAR POSITION OF THE MASSES
BALANCING OF SEVERAL MASSES ROTATING IN DIFFERENT PLANES
Desh Biological Works Timber Market, Ambala Cantt-133001, Haryana (INDIA)
Ph. : 0171-4007531, 2643822, Fax 91-171-2644859

Static (Single Plane) Balance -

$m_bR_{bx} = - S(m_iR_{ix})$ from i = 1 to n
$m_bR_{by} = - S(m_iR_{iy})$ from i = 1 to n
$f_b = \arctan[(m_bR_{by})/(m_bR_{bx})]$
$m_bR_{bx} = [(m_bR_{bx})^2 + (m_bR_{by})^2]^{1/2}$
m_b = balance mass
R_b = radial distance to CG of balance mass
m_i = i th point mass
R_i = radial distance to CG of the i th point mass
f_b = angle of rotation of balance mass CG with respect to the reference axis.
x, y = subscripts that designate orthogonal components



Two balance masses are added (or subtracted), one each on planes A and B.

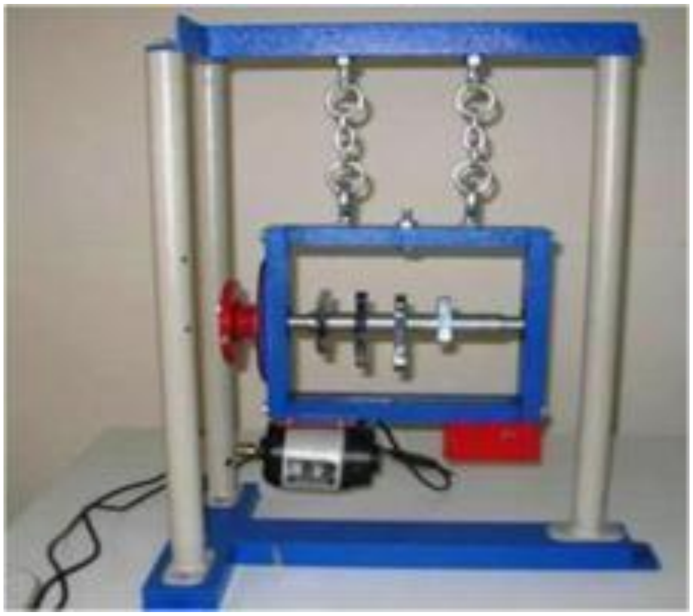
$m_BR_{Bx} = - S(m_iR_{ix}l_i)$ from i = 1 to n
$m_BR_{By} = - S(m_iR_{iy}l_i)$ from i = 1 to n

$m_AR_{Ax} = - S(m_iR_{ix}) - m_BR_{Bx}$ from i = 1 to n
$m_AR_{Ay} = - S(m_iR_{iy}) - m_BR_{By}$ from i = 1 to n
m_A = balance mass in the A plane
m_B = balance mass in the B plane
R_A = radial distance to CG of balance mass
R_B = radial distance to CG of balance mass
f_A, f_B, R_A, R_B are found using relationships in Static Balance above

Ignoring the weight of the tire and its reactions at 1 and 2,

$F_{1x} + F_{2x}\ m_A R_{Ax} w^2 + m_B R_{Bx} w^2 = 0$
$F_{1y} + F_{2y}\ m_A R_{Ay} w^2 + m_B R_{By} w^2 = 0$
$F_{1x}\ l_1 + F_{2x}\ l_2 + m_B R_{Bx} w^2\ l_B = 0$
$F_{1y}\ l_1 + F_{2y}\ l_2 + m_B R_{By} w^2\ l_B = 0$

$m_B R_{Bx} = (F_{1x}\ l_1 + F_{2x}\ l_2)/(\ l_B w^2)$
$m_B R_{By} = (F_{1y}\ l_1 + F_{2y}\ l_2)/(\ l_B w^2)$
$m_A R_{Ax} = (F_{1x} + F_{2x})/(w^2) - m_b R_{Bx}$
$m_A R_{Ay} = (F_{1y} + F_{2y})/(w^2) - m_b R_{By}$



Experiment # 7

JOURNAL BEARING TEST RIG

Aim of the experiment:-Determination of a pressure distribution in a journal bearing.

Learning Objective:-Determination of Pressure distribution in Journal bearing.

Theory:-

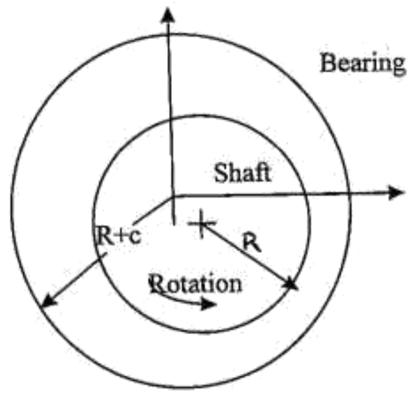


Figure: Plain journal bearing

To formulate the bearing action accurately in mathematical terms is a more complex job. However, one can visualize the pattern of bearing pressure distribution due to the hydrodynamic action with the help of experimental rig. This helps to understand the subject properly.

The experimental rig consists of a small journal bearing as shown in Figure. This apparatus helps to demonstrate and study the effect of important variables such as speed, viscosity and load, on the pressure distribution in a Journal bearing. This pressure distribution can be verified with Sommerfeld equations.

Procedure:-

1. Fill the oil tank with lubricating oil.
2. Drain out the oil bubbles from all manometer tubes.
3. Open the inlet valve and note down the initial manometer reading after getting uniform level.
4. Check and ensure that the dimmersrat is at zero position.
5. Rotate the dimmersrat knob gradually till the desired speed is reached.
6. Run the set –up at this speed for some time.
7. Note down the pressure of oil in all the manometer tubes and tabulate them.
8. Bring down the speed to zero and switch off the motor.
9. The difference in manometer pressure at each tapping is plotted.

THEORY OF JOURNAL BEARINGS

The mathematical analysis of the behaviour of a journal in a bearing fails into two distinct categories as given in the appendix to this Manual. They are:

1. Hydrodynamics of fluid flow between plates.
2. Journal bearing analysis where the motion of the journal in the oil film is considered.

According to equation the Sommerfield pressure function (when the velocity of the eccentricity and the whirl speed of the journal are both zero is given by :

$$P - P_o = \frac{-6\mu r^{2\omega}}{\delta^2 (2+n^2)} \text{ x } \frac{n \sin\theta in + n \cos\theta o}{(1+n \cos\theta o^2}$$

Where P is the pressure of the oil film at the point measured clockwise from the line of common centers (oo‟) and P = P_o at θ = 0 and θ = π (refer to Fig. No. 2)

Note : Some books on lubication gives the sommerfield function with a negative sign for n. This is true if it is measured from the point of minimum thickness of the oil films that is $h = \delta(1 - n\cos\theta)$

Maximum pressure occurs at $\cos\theta_m = \frac{3n}{2+n^2}$

Hence minimum pressure occurs at the point θ = -em. The total load (P) on the journal is given by equation (acting perpendicular the line of centers oo‟)

$$P = \frac{-12\mu r^3 L\omega\pi}{\delta^2} \text{ x } \frac{n}{2+n^2} x \frac{1}{\sqrt{1-n^2}}$$

Where L is width of the bearing and the total force along oo‟ is zero.

The total tractional couple „M‟ necessary to rotate the journal is given by

$$M = \frac{4\mu r^3 \omega\pi L}{\delta} \text{ x } \frac{1+2n^2}{(2+n^2)\sqrt{1-n^2}}$$

Note:

i) When comparing the above expressions for pressures, loads and so on, with experimental data obtained from the small journal bearing rig, θ must be measured from the point where the thickness of the oil films is maximum and in the anti clockwise direction.

ii) $P-P_o = 0$ at $\theta = 0$ and $\theta = \pi$

i.e. $P = P_o$ at 180° apart from zero.

That is on the pressure curve (head of oil/angular position) select two points of equal pressure 180 apart. Of these two points take as Origin the point where the thickness of the oil film is greater, and measure anti clockwise to plot the Summerfield pressure curve -after determining graphically the values of 'n' from:

$Cos\theta m = \frac{-3n}{2+n2}$ and the value of „k" in

Where „K" has some units of dimensions as „P", „n" is non-dimensional.

Determine the pressure distribution in the oil film of the bearing for various speeds and

a) Plot the Cartesian and polar pressure curves for various speeds.

$$P = \frac{12\sigma\mu r^3 Lw\pi}{\delta^2} \, x \, \frac{n}{2+n^2} \, x \, \frac{1}{\sqrt{1-n^2}}$$

And compare with load on the bearing. Determination of Tractional torque.

LOAD ON BEARING:

01. Total vertical load on bearing at N.R.P.M.

= Dry weight of bearing + weight added + weight of balancing load.

= 1.375 Kg. + 2 x 0.1150 Kg. + added weight Nil.

= 1.6775 Kg.

02. Referring to Fig. No.4, the mean positive pressure head of oil above supply head -

= (35.5 + 24 +12 + 8 +5 +2 +60 + 177 + 130)/10 = 45.5 cms.

Load carried by oil pressure on projected area of bearing –

= 45.5 x density of oil x (2R) L

= 45.5 x 0.8539 x 5.5 x 6.8

= 1.450 Kg.

Note: The underlined figures are recorded from graph and balanced are practical results.

03. Maximum theoretical load on Journal is „P".

$$P = \frac{12\mu Lr\omega\pi}{\delta^2} x \frac{n}{2+n^2} + \frac{1}{\sqrt{1-n^2}}$$

$$= \frac{K\,x\,2\pi r^3 L}{\sqrt{1-n^2}} x \; density\ of\ oil$$

$$= \frac{24.5\,x\,2\,x\,3.142\,x\,5/2\;x\,6.8}{\sqrt{1-0.8^2}}\,x\;0.8539$$

P = 3.724Kgs.

04. Tractional Torque = Balancing weight J x Length of arm of the weight L.

Table – 1

TYPICAL RESULTS w.r.t. MANOMETER TUBES

Ps = Supply head =

Weight of bearing =

TUBE No.	
1	
2	
3	
4	
5	
6	
7	
8	
9	
10	
12	
A	
B	
C	
D	

Note : PO = Supply head of Oil.

TABLE –2

PRESSURE HEAD OF OIL FILM ABOVE HEAD

$(P-P_s)$ cm. : __________

Shaft Speed:__________Rev/.min.

TUBE No.	
1	
2	
3	
4	
5	
6	
7	
8	
9	
10	
12	
A	
B	
C	
D	

OBSERVATIONS:

The Summerfield pressure function agrees with the experimental pressure curve within reasonable limits as indicated in Fig. Any deviations between the experimental and theoretical curves can be due to:

01. Human error in taking readings, for example in deciding whether or not the oil levels in the manometer are absolutely steady before taking a reading.
02. The theoretical analysis is based on the assumption that the thickness of the oil film h = + e cos which is true only if the radial clearance is very small. In practical journal bearings this assumption is true but this test rig = 2.5 mm which is very large. This has been purposely done so that the oil film profile is clearly visible.
03. The total weight of the bearing is = 1.375 Kg. It can be seen that the oil in the bearing does not carry this full weight, a part of the weight appears to be taken by the seal, and the flexible plastic tubes attached to the bearing.

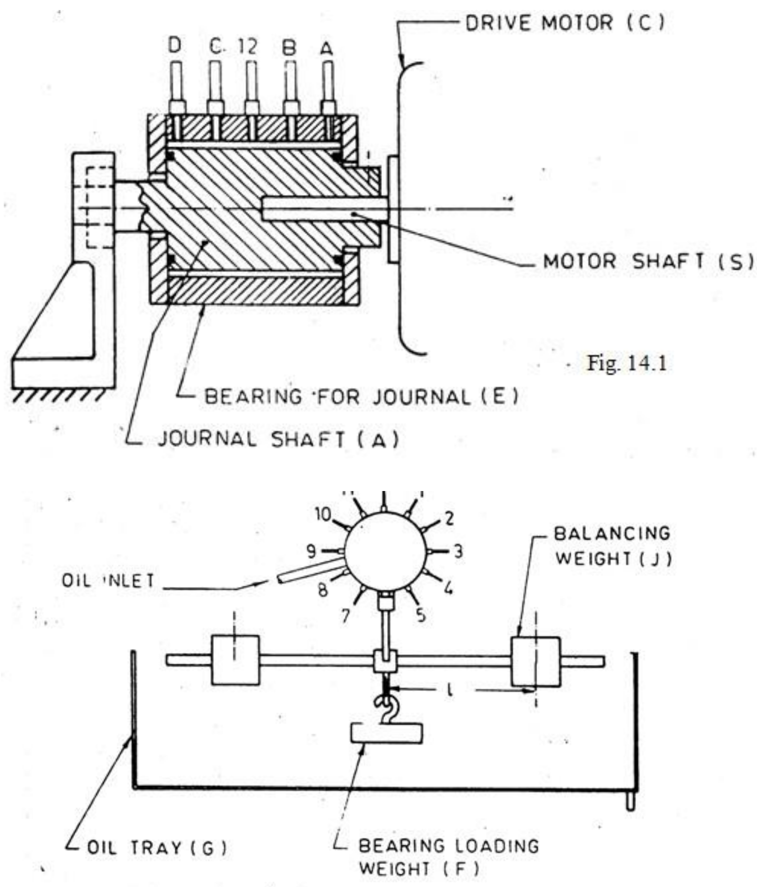
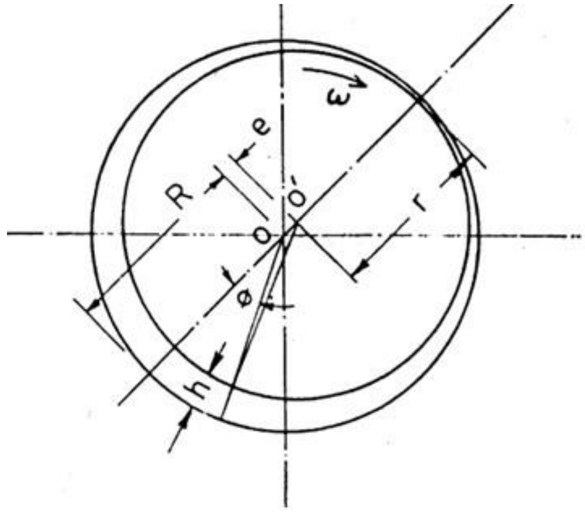


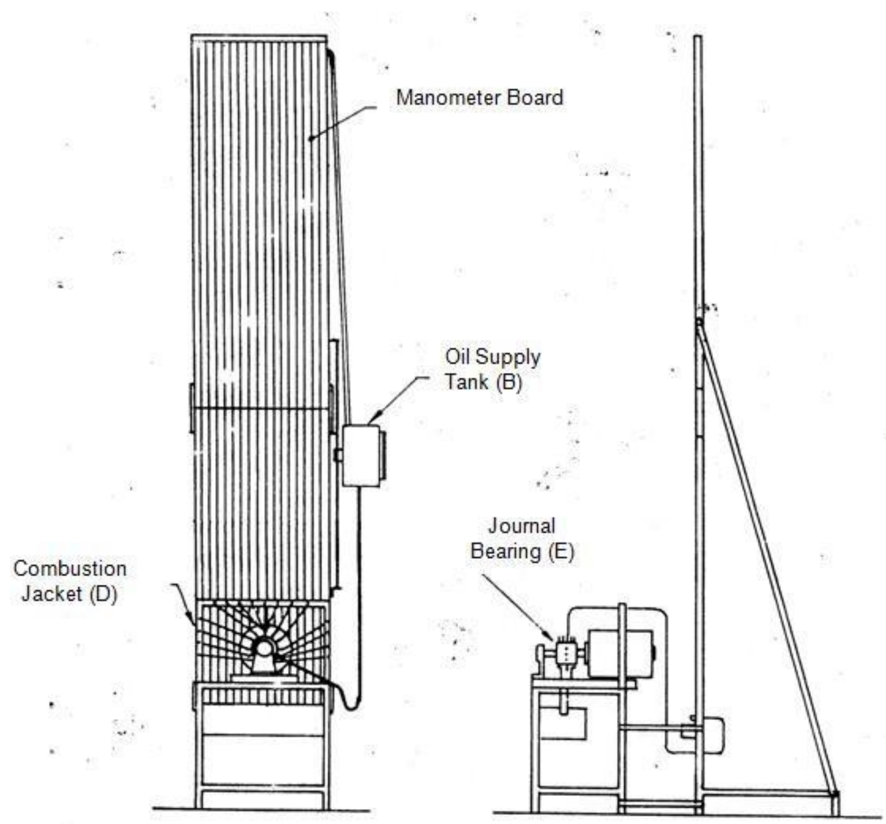
Fig. 14.1

DETAILS OF JOURNAL, BEARING & LOADING ARRANGEMENT



ω - Speed of rotation of Journal
r - Radius of Journal
δ - Radial clearance (R - r)
e - Eccentricity 00'
n - e/δ
μ - viscocity of oil

GEOMETRY OF JOURNAL BEARING



Schematic Layout of Journal Bearing Apparatus

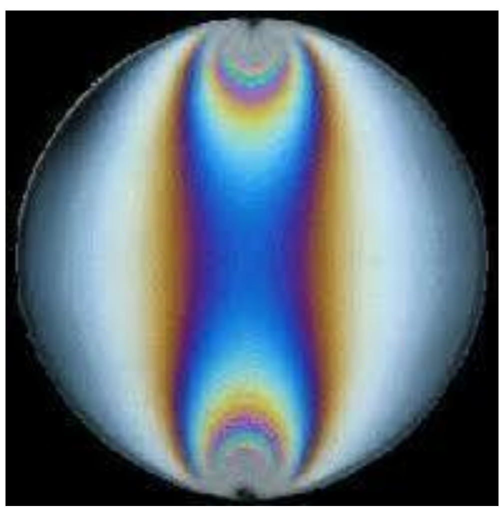
Experiment # 8

PHOTO ELASTIC TEST BENCH

Aim: To determine the fringe constant and stress concentration Factor for the given Specimen using photo elastic test bench.

Apparatus:

Theory: Photoelasticity is an experimental method to determine the stress distribution in a material. The method is mostly used in cases where mathematical methods become quite cumbersome. Unlike the analytical methods of stress determination, photoelasticity gives a fairly accurate picture of stress distribution, even around abrupt discontinuities in a material. The method is an important tool for determining critical stress points in a material, and is used for determining stress concentration in irregular geometries.



The method is based on the property of birefringence exhibited by certain transparent materials. Birefringence is a property where a ray of light passing through a birefringent material experiences two refractive indices. The property of birefringence (or double refraction) is observed in many optical crystals. Upon the application of stresses, photoelastic materials exhibit the property of birefringence, and the magnitude of the refractive indices at each point in the material is directly related to the state of stresses at that point. Information such as maximum shear stress and its orientation are available by analyzing the birefringence with an instrument called polariscope.

When a ray of light passes through a photoelastic material, its electromagnetic wave components gets resolved along the two principal stress directions and each of these components experiences different refractive indices due to the birefringence. The difference in the refractive indices leads to a relative phase retardation between the two components. Assuming a thin specimen made of isotropic materials, where two-dimensional photoelasticity is applicable. The magnitude of the relative retardation is given by the *stress-optic law*:[3]

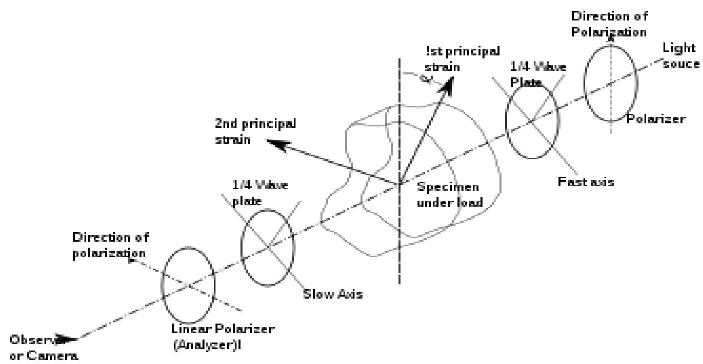
$$\Delta = \frac{2\pi t}{\lambda} \; C(\sigma_1 - \sigma_2)$$

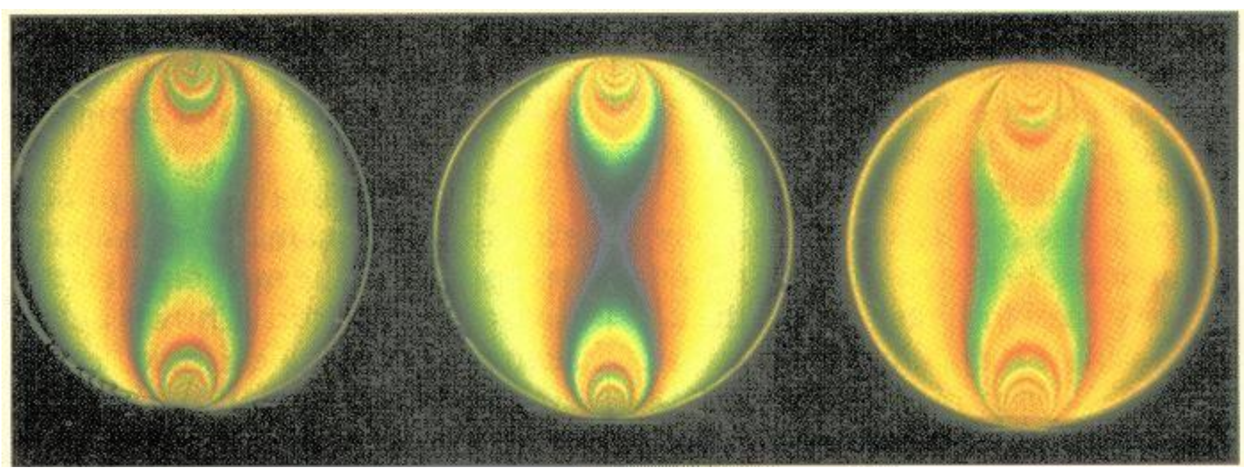
where Δ is the induced retardation, C is the stress-optic coefficient, t is the specimen thickness, σ_1 and σ_2 are the first and second principal stresses, respectively. The retardation changes the polarization of transmitted light. The polariscope combines the different polarization states of light waves before and after passing the specimen. Due to optical interference of the two waves, a fringe pattern is revealed. The number of fringe order N is denoted as

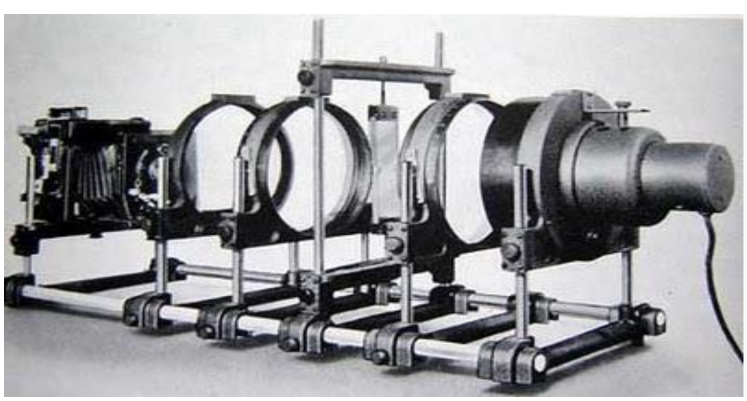
$$N = \Delta/2\pi$$

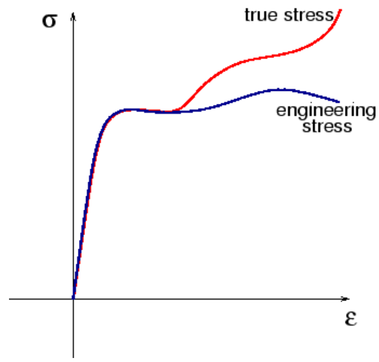
which depends on relative retardation. By studying the fringe pattern one can determine the state of stress at various points in the material.

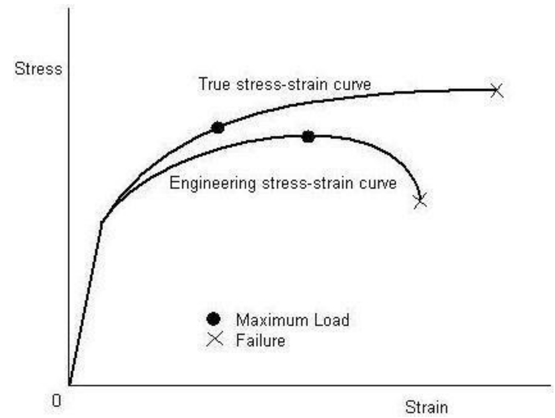
For materials that do not show photoelastic behavior, it is still possible to study the stress distribution in such materials. The first step is to build a model using photoelastic materials, which has similar geometry as the real structure to be investigated. The loading is then applied in the same way to ensure that the stress distribution in the model is similar to the stress in the real structure.

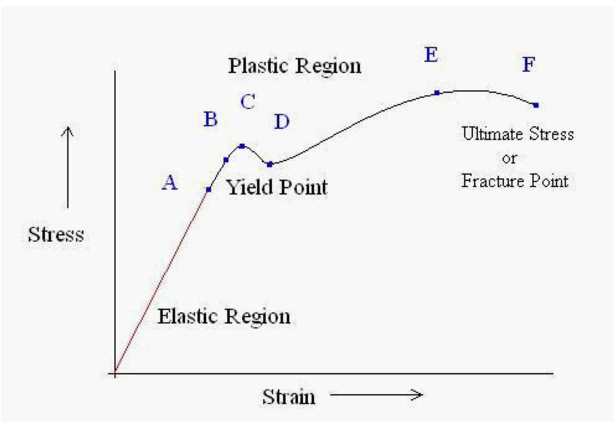


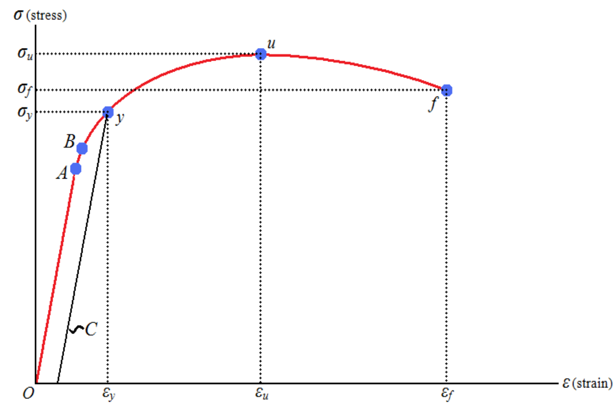


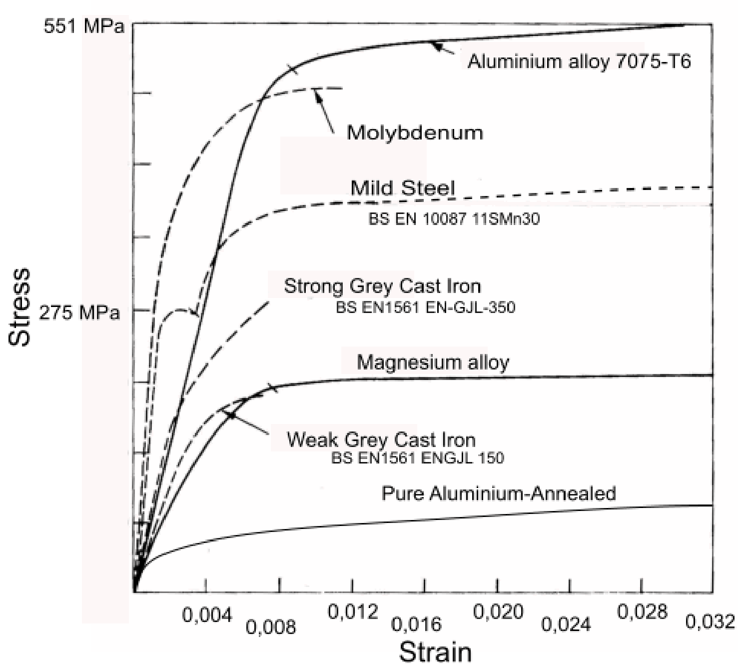


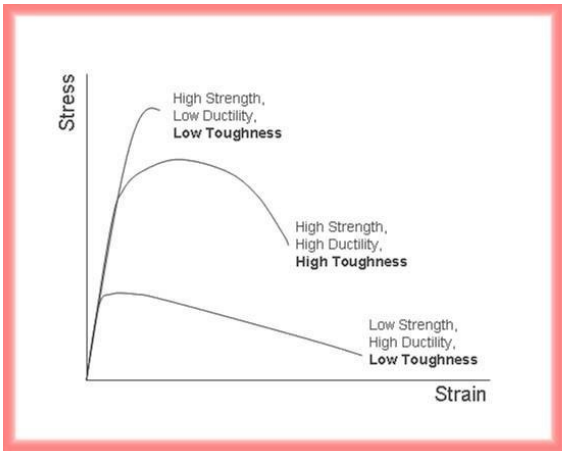

σ
true stress
engineering
stress
ε

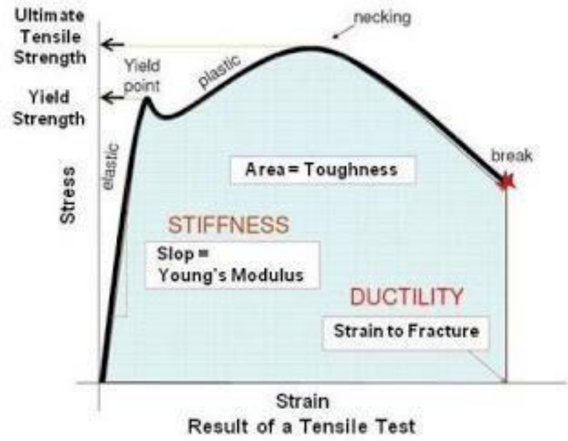

Stress
True stress-strain curve
Engineering stress-strain curve
Maximum Load
Failure
0
Strain

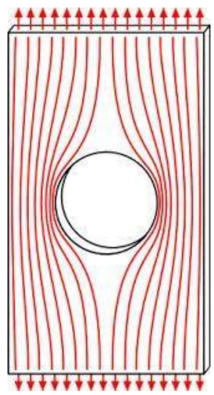

Plastic Region
E
F
B
C
D
A
Yield Point
Ultimate Stress
or
Fracture Point
Stress
Elastic Region
Strain

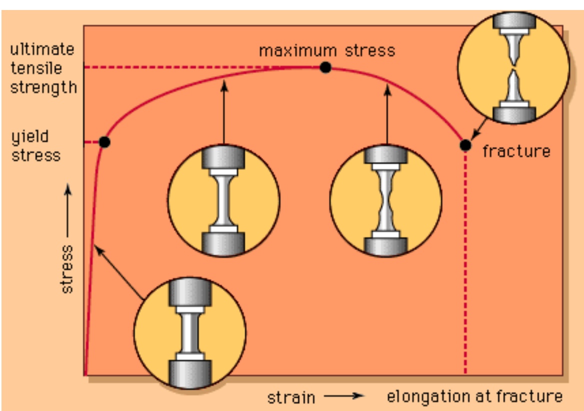

σ (stress)
u
B
A
C
y
f
O
ε (strain)

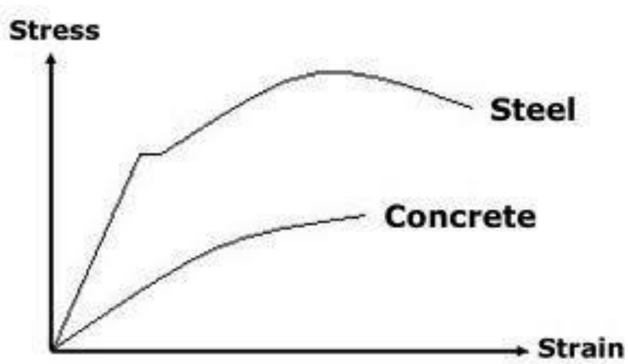

551 MPa
Aluminium alloy 7075-T6
Molybdenum
Mild Steel
BS EN 10087 11SMn30
Strong Grey Cast Iron
BS EN1561 EN-GJL-350
Stress
275 MPa
Magnesium alloy
Weak Grey Cast Iron
BS EN1561 ENGJL 150
Pure Aluminium-Annealed
0,004
0,008
0,012
0,016
0,020
0,024
0,028
0,032
Strain


Stress
High Strength,
Low Ductility,
Low Toughness
High Strength,
High Ductility,
High Toughness
Low Strength,
High Ductility,
Low Toughness
Strain


Ultimate
Tensile
Strength
necking
Yield
point
plastic
Yield
Strength
elastic
Stress
Area = Toughness
break
STIFFNESS
Slop =
Young's Modulus
DUCTILITY
Strain to Fracture
Strain
Result of a Tensile Test




ultimate
tensile
strength
maximum stress
yield
stress
fracture
stress
strain
elongation at fracture


Stress
Steel
Concrete
Strain

Experiment # 9

GYROSCOPE

AIM: To study the gyroscopic behavior of rotating masses and verify the gyroscopic relationship.

Apparatus: Gyroscope, weights, stop watch, tachometer, spirit level

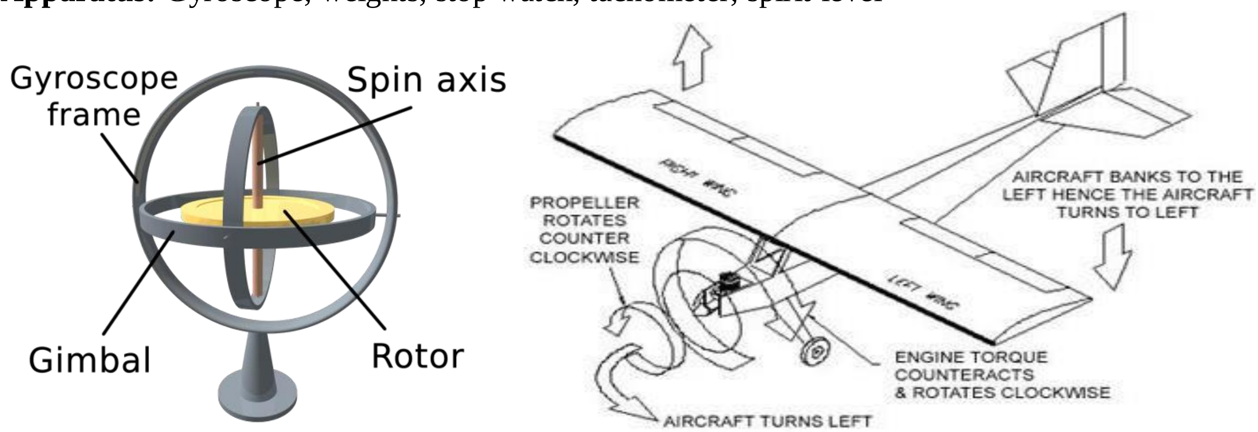


Fig. 10: Schematic representation of Gyroscope axis and aircraft

Theory: *For theory answer the following questions*

1. Write a short note on gyroscope.
2. What do you understand by gyroscopic couple? Derive a formula for its magnitude.
3. Explain the application of gyroscopic principles to aircrafts.
4. Discuss the effect of the gyroscopic couple on a two wheeled vehicle when taking a turn.

The earliest observation and studies on gyroscopic phenomenon carried out during Newton‟s time. These were made in the context of the motion of our planet which in effect in a massive gyroscopic. The credit of the mathematical foundation of the principles of gyroscopic motion goes to Euler who derived at set of dynamic equation relating applied mechanics and moment inertia, angular acceleration and angular velocity in many machines, the rotary components are forced to turn about their axis other than their own axis of rotation and gyroscopic effects are thus setup. The gyroscopes are used in ships to minimize the rolling & pitching effects of water.

A Gyroscope is a spinning body mounted universally to turn with an angular velocity of precession in a direction at right angles to the direction of the moment causing it but its center of gravity will be in a fixed position.

The gyroscope has 2 degrees of freedom. The first axis is OX called spin axis on which the body is spinning. The second axis is OY called Torque axis. Third axis OA is called precession axis on which the body moves opposing the original motion. All the 3 axis are mutually perpendicular. Such a combined effect is known as Gyroscopic effect.

The analyses of gyroscopic principles are based on Newton‟s Laws of Motion and inertia. When the rotor is spinned, the gyroscope exhibits the following two important characteristics:

1. Gyroscopic Inertia 2. Precession

Applications: The gyroscopic principle is used in an instrument or toy known as gyroscope. The gyroscopes are installed in ships in order to minimize the rolling and pitching effects of waves. They are also used in aeroplanes, monorail cars, gyrocompasses etc.

Gyroscopic Inertia: $I = \frac{1}{2} m r^2 = \frac{1}{2} \frac{\omega_d}{g} \left(\frac{D}{2}\right)^2$ Kg-m^2

Where, m= mass of disc

W_d = wt of disc

D= dia of disc

Precession: When a force is applied to the gyroscope about the horizontal axis it may be found that the applied force meets with resistance and that the gyro, instead of turning about the horizontal axis, turns about its vertical axis and vice versa. It follows right hand thumb rule. Thus the change in direction of plane of rotation of rotor is known as precession.

Velocity of Spin:

$\omega = \frac{2\pi n}{60}\ rad/s$ Where n = speed of motor in rpm

Velocity of Precession:

$\omega_p = \frac{d\theta}{dt} = \frac{\pi\theta}{180}$ rad/s Where θ = angle of precession, t= time for precession

Gyroscopic Couple: The couple generated due to change in direction of angular velocity of rotor is called gyroscopic couple.

$C_g = I\omega\omega_p$, N-m

Applied Torque: The torque applied to change the direction of angular velocity rotor is called applied torque.

C_a = W.a. Where W = wt placed in the wt. stud, a= its distance from center of disc.

Angular momentum: C = Iω

Effect of the Gyroscopic Couple on an Aeroplane:

1. When the aeroplane takes a right turn under similar conditions. The effect of the reactive gyroscopic couple will be to dip the nose and raise the tail of the aeroplane.
2. When the engine or propeller rotates in anticlockwise direction when viewed from the rear or tail end and the aeroplane takes a left turn, then the effect of reactive gyroscopic couple will be to dip the nose and raise the tail of the aeroplane.
3. When the aeroplane takes a right turn under similar conditions as mentioned in note 2 above, the effect of reactive gyroscopic couple will be to raise the nose and dip the tail of the aeroplane.

4. When the engine or propeller rotates in clockwise direction when viewed from the front and the aeroplane takes a left turn, then the effect of reactive gyroscopic couple will be to raise the tail and dip the nose of the aeroplane.
5. When the aeroplane takes a right turn under similar conditions as mentioned in note 4 above, the effect of reactive gyroscopic couple will be to raise the nose and dip the tail of the aeroplane.

The effect of gyroscopic couple on the naval ship in the following three cases :

1- Steering 2- Pitching and 3- Rolling

APPLICATIONS: The gyroscopic principle is used in an instrument or toy known as gyroscope. The gyroscopes are installed in ships in order to minimize the rolling and pitching effects of waves. They are also used in aeroplanes, monorail cars, gyrocompasses etc.

Effect of the Gyroscopic Couple on a Naval ship during Steering:-

1. When the ship steers to the right under similar conditions as discussed above, the effect of the reactive gyroscopic couple will be to raise the stern and lower the bow.
2. When the rotor rotates in the anticlockwise direction, when viewed from the stern and the ship is steering to the left, then the effect of reactive gyroscopic couple will be to lower the bow and raise the stern.
3. When the ship is steering to the right under similar conditions as discussed in note 2 above, then the effect of reactive gyroscopic couple will be to raise the bow and lower the stern.
4. When the rotor rotates in the clockwise direction when viewed from the bow or fore end and the ship is steering to the left, then the effect of reactive gyroscopic couple will be to raise the stern and lower the bow.
5. When the ship is steering to the right under similar conditions as discussed in note 4 above, then the effect of reactive gyroscopic couple will be to raise the bow and lower the stern.
6. The effect of the reactive gyroscopic couple on a boat propelled by a turbine taking left or right turn is similar as discussed above.

Effect of the Gyroscopic Couple on a Naval ship during Pitching:-

1. The effect of the gyroscopic couple is always given on specific position of the axis of spin, i.e. whether it is pitching downwards or upwards.
2. The pitching of a ship produces forces on the bearings which act horizontally and perpendicular to the motion of the ship.
3. The maximum gyroscopic couple tends to shear the holding-down bolts.
4. The angular acceleration during pitching.

$\boldsymbol{\alpha = d^2\theta/dt^2 = -\Phi(\omega_1)^2 \sin \omega_1 t}$ (Differentiating $d\theta/dt$ with respect to t)

The angular acceleration is maximum, if sin $\omega_1 t = 1$

Therefore Maximum angular acceleration during pitching,

$$\alpha_{max} = \Phi(\omega_1)^2$$

Effect of the Gyroscopic Couple on a Naval ship during Rolling:- We know that, for the effect of gyroscopic couple to occur, the axis of precession should always be perpendicular to the axis of spin. If, however, the axis of precession becomes parallel to the axis of spin, there will be no effect of the gyroscopic couple acting on the body of the ship.

In case of rolling of a ship, the axis of precession (i.e. longitudinal axis) is always parallel to the axis of spin for all positions. Hence, there is no effect of the gyroscopic couple acting on the body of a ship.

Procedure of conduction of experiment:

1. Connect the motor of the gyroscope to an A.C. supply through dimmer stat.
2. Using spirit level, check the rotor for vertical position. Adjust the balance weight slightly if required using the bottom clamp screws.
3. Set the dimmer at zero position and put ON the supply.
4. Start the motor by applying the voltage of around 170(for instant build up of voltage) and then reduce gradually.
5. Adjust the rotor speed if required
6. Note down the rotor speed with the help of a tachometer when it becomes steady.
7. Place the required wt. on the wt. stud and at the same instant start the stop watch. Note down the time required for θ degree of precession.
8. Repeat the procedure for different weights and precessions.
9. Measure and record the distance between the center of the disc and center of weight stud.
10. Tabulate the results.
11. Determine and compare the gyroscopic couple with that of applied torque and plot the following curves:

    i. Calibration curve ii. Gyroscopic couple Vs Precession.

Observations:

Rotor Diameter	=	245 mm
Rotor thickness	=	10 mm
M.I. of all Rotating Parts	=	0.02986 Nm- sec ^{2}

Distance from centre of Disc

to Centre of Dead Weights = 0.155 m

Motor speed Max = 6000 rpm

Speed of motor N rpm	Applied load W		Angle of precession θ Degrees	Time taken for precession „t" Sec	Angular velocity ω rad / s	Torque applied T Nm	Experiment velocity of precession ϖ_{Pe} rad / s	Theoretical velocity of precession ϖ_{PT} rad / s
	Kg	N						
			60					
			60					
			60					
			60					

Specimen calculations:

Angular Velocity of rotor $\varpi = \frac{2\pi N}{60}$

Gyroscopic couple (or) Torque Applied $T = W \times r$ N-m

Experimental Velocity of precision, $\varpi_{Pe} = \frac{\theta \times \frac{\pi}{180}}{t}$ rad/sec

Theoretical Velocity of Precision, $\varpi_{Pt} = \frac{T}{I\varpi}$ rad/sec

Result:

Percentage error between applied torque and gyroscopic couple is: _____________