LABORATORY MANUAL

18MEL77 DESIGN LAB

2019-2020

DEPARTMENT OF MECHANICAL ENGINEERING
ATRIA INSTITUTE OF TECHNOLOGY
Adjacent to Bangalore Baptist Hospital
Hebbal, Bengaluru-560024

Department of Mechanical Engineering

Vision

To be a center of excellence in Mechanical Engineering education and interdisciplinary research to confrontreal world societal problems with professional ethics.

Mission

1. To push the frontiers of pedagogy amongst the students and develop new paradigms in research.

2. To develop products, processes, and technologies for the benefit of society in collaboration withindustry and commerce.

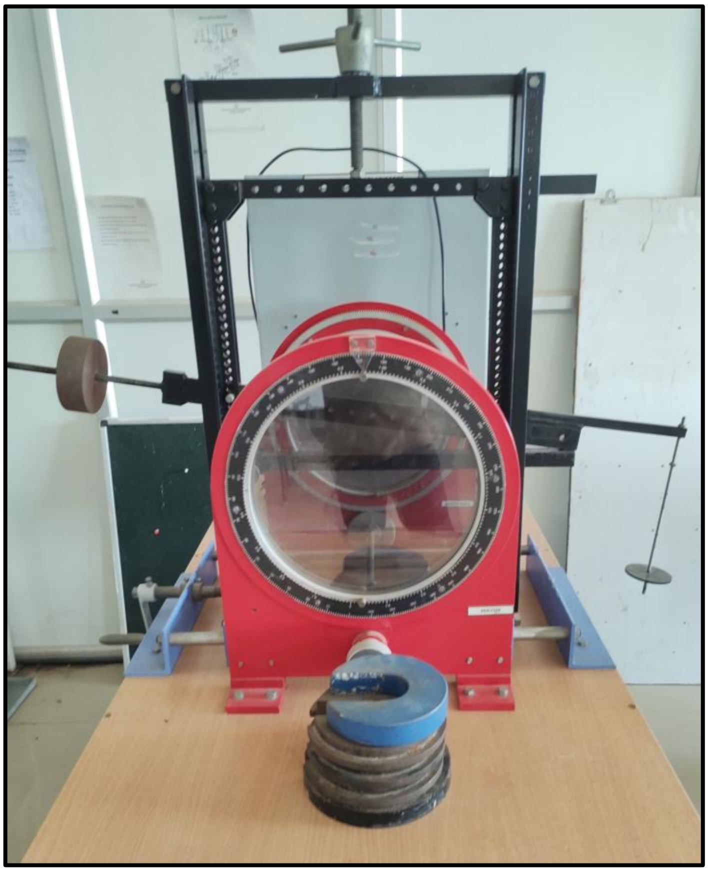
3. To mould the young minds and build a comprehensive personality by nurturing strong professionals with human ethics through interaction with faculty, alumni, and experts from academia/industry.

UNIVERSAL GOVERNOR

Centrifugal governors were used to regulate the distance and pressure between millstones in windmills since the 17th century. Early steam engines employed a purely reciprocating motion, and were used for pumping water – an application that could tolerate variations in the working speed. Between the years 1775 and 1800, Watt, in partnership with industrialist Matthew Boulton, produced some 500 rotative beam engines. At the heart of these engines was Watt's self-designed "conical pendulum" governor: a set of revolving steel balls attached to a vertical spindle by link arms, where the controlling force consists of the weight of the balls.

The theoretical basis for the operation of governors was described by James Clerk Maxwell in 1868 in his seminal paper 'On Governors'.

DETERMINATION OF FRINGE CONSTANT



Photoelasticity describes changes in the optical properties of a material under mechanical deformation. It is a property of all dielectric media and is often used to experimentally determine the stress distribution in a material, where it gives a picture of stress distributions around discontinuities in materials. Photo elastic experiments (also informally referred to as *photoelasticity*) are an important tool for determining critical stress points in a material, and are used for determining stress concentration in irregular geometries.

ATRIA INSTITUTE OF TECHNOLOGY
Department of Mechanical Engineering

LABORATORY CERTIFICATE

*This is to certify that Mr. /Ms.*____

*bearing USN*_________________*of*___________*semester and*_________*section has satisfactorily completed the course of experiments in* **DESIGN LAB**, *code* 15MEL76 *prescribed by the Visvesvaraya Technological University, Belagavi of this Institute for the academic year 20 20 .*

MARKS	
Maximum Marks	**Marks Obtained**

Signature of Faculty-In-Charge *Head of the Department*

Date:

PREFACE

Mechanical Design is the process by which resources is converted into useful mechanical forms, or the mechanisms so as to obtain useful output from the machines in the desired form as per the customer needs. The Design Laboratory contributes to educate the undergraduate students of 7th semester B.E, VTU Belagavi in the field of Mechanical Engineering.

The objectives of this laboratory are to impart practical knowledge on analysis of mechanisms for the specified type of motion in a machine. It also focuses on practical study of static and dynamic forces for balancing of rotating masses. With the study of vibrations and degrees of freedom for mechanical systems, undamped longitudinal and torsional vibrations can be well understood. Stress and strain analysis of several materials is understood using photoelasticity and strain rosettes. Pressure distribution of journal bearing is also understood.

Demonstration exercises are provided to understand machine kinematics and dynamics such as governors, gyroscopes, balancing machines and universal vibration facilities. Various experiments are made to calibrate photoelastic materials using photoelasticity.

I acknowledge Dr. Prashanth T, head of the department for his valuable guidance and suggestions as per Revised Blooms Taxonomy in preparing the lab manual.

Krishnappa R

SYLLABUS

Subject Code	: 15MEL76	IA Marks	: 20
No. of Practical Hrs. / Week	: 03	Exam Hours	: 03
Total No. of Practical Hrs.	: 42	Exam Marks	: 80

Students are expected-
1. To understand the natural frequency, logarithmic decrement, damping ratio and damping.
2. To understand the balancing of rotating masses.
3. To understand the concept of the critical speed of a rotating shaft.
4. To understand the concept of stress concentration using Photo elasticity.
5. To understand the equilibrium speed, sensitiveness, power and effort of Governor.

PART –A
1. Determination of natural frequency, logarithmic decrement, damping ratio and damping Coefficient in a single degree of freedom vibrating systems (longitudinal and torsional)
2. Determination of critical speed of rotating shaft.
3. Balancing of rotating masses.
4. Determination of fringe constant of Photo-elastic material using Circular disk subjected Diametric compression, Pure bending specimen (four point bending)
5. Determination of stress concentration using Photo elasticity for simple components like Plate with hole under tension or bending, circular disk with circular hole under compression, 2-d crane hook.

PART –B
1. Determination of equilibrium speed, sensitiveness, power and effort of Porter/ Proel / Hartnell Governor. (At least one)
2. Determination of pressure distribution in Journal bearing
3. Determination of principle stresses and strain in a member subjected to combined loading Using strain rosettes.
4. Determination of stresses in curved beam using strain gauge.
5. Experiments on Gyroscope (Demonstration only)

COURSE OUTCOMES
At the end of the course, the students will be able to:
1. To understand the working principles of machine elements such as Governors, Gyroscopes etc.,
2. To identify forces and couples in rotating mechanical system components.
3. To identify vibrations in machine elements and design appropriate damping methods and to determine the critical speed of a rotating shaft.
4. To measure strain in various machine elements using strain gauges.
5. To determine the minimum film thickness, load carrying capacity, frictional torque And pressure distribution of journal bearing.
6. To determine strain induced in a structural member using the principle of photo-elasticity.

Scheme of Examination:
One question from Part A: 40 Marks
One question from part B: 20 Marks
Viva- Voce: 20 Marks
Total: 80 Marks

CONTENTS

Experiment No.: 01 **Date:**

SPRING MASS SYSTEM

UNDAMPED FREE LONGITUDINAL VIBRATIONS

AIM: To study the longitudinal vibration of the spring mass system and to determine the natural frequency.

APPARATUS REQUIRED: Stop Watch, Measuring Scale, Weights.

THEORY:-

Components in a vibrating system have three properties of interest. They are: mass (weight), elasticity (springiness) and damping (dissipation). Most physical objects have all three properties, but in many cases one or two of those properties are relatively insignificant and can be ignored (for example, the *damping* of a block of steel, or in some cases, the *mass* of a spring). The property of mass (weight) causes an object to resist acceleration. It also enables an object to store energy, in the form of velocity (kinetic) or height (potential).

The property of elasticity enables an object to store energy in the form of deflection. A common example is a spring, but any piece of metal has the property of elasticity. The size of the deflection depends on the size of the applied force and the dimensions and properties of the piece of metal. The amount of deflection caused by a specific force determines the "spring rate" of the metal piece. The property of damping enables an object to DISSIPATE energy, usually by conversion of kinetic (motion) energy into heat energy.

The resonant frequency of an object (or system) is the frequency at which the amplitude of vibration is maximum. Increase in mass property or decrease the elasticity property of a system, the resonant frequency will decrease according to the relationship:

$$\omega_n = \sqrt{(k/m)}$$

According to energy method:

$$mx'' + kx = 0 \Rightarrow x'' + \frac{k}{m}.x = 0 \rightarrow (1)$$

&

Differential equation for SHM: $x'' + \omega_n x = 0 \rightarrow (2)$

Comparing (1) and (2) we get: $\omega_n = \sqrt{\frac{k}{m}}$

where "k" is the spring rate/spring stiffness and "m" is the mass value.

$$K = w/\delta = mg/\delta$$

$$\omega_n = \sqrt{\frac{mg}{m\delta}} = \sqrt{\frac{g}{\delta}}$$

Time period, $T = 2\pi/\omega_n$

Natural Frequency: $f_n = \frac{1}{T} = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$

Undamped vibration: If the vibratory system has no damper system i.e. if there is no reduction in amplitude over every cycle of operation then such a vibration is known as undamped vibration. Figure 1.1 shows simple spring mass system.

Free vibration: when no external force acts on the body after giving it an initial displacement, then the body is said to be under free vibration.

Longitudinal vibration: when the particles of the system vibrates parallel to the axis of the system then the vibration is known as longitudinal vibration.

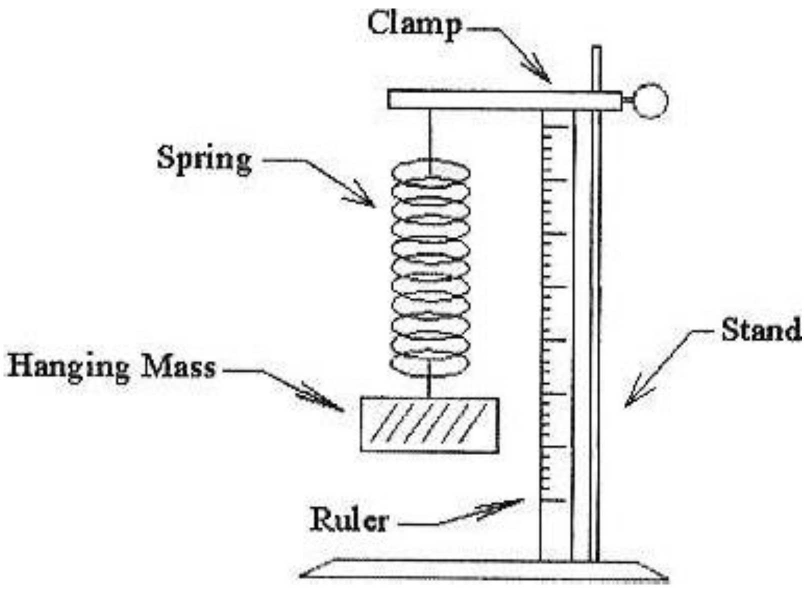


Fig.1.1: Spring mass system

PROCEDURE:

1. Hook the spring whose natural frequency has to be determined.
2. Determine the length of the spring at no load condition (no extra mass on hanger).
3. Add weights and measure the corresponding deflection of the spring.
4. For oscillations, stretch the spring for some distance and leave.
5. Note down the time taken for say 5, 10 or 15 oscillations.
6. Determine the time period and natural frequency.
7. Plot the graph of Load Vs Deflection.
8. Repeat the Procedure for different weights.
9. Compare the experimental values with theoretical valves and find the percentage error.

SPECIFICATIONS:

Motor speed N = 6rpm
Drum diameter d = 72 mm
Mass m = 5 kg

OBSERVATION:-

1 Mass of hanger, m_h = …………………….
2 Initial Length of Spring, L =……………………..

TABULAR COLUMN:-

Table: 1.1: Error estimation of spring mass system

Sl. No.	Mass attached (kg)	Final Length L_1(m)	Deflection $\int = L_1$-L(m)	Stiffness K (N/m)	Time taken for 10 oscillations (seconds)	T_{exp} (seconds)	F_{exp} (Hz)	F_{th} (Hz)	% error

FORMULAE USED

1. Stiffness of Spring, K = mg/∫(N/m)

Where ∫ = deflection

2. Experimental time period, T_{exp} = t/n(seconds)

Where t = time for n oscillations(seconds)
n = number of oscillations

3. Experimental frequency, $F_{exp} = 1/T_{exp}$(Hz)

4. Theoretical time period, $T_{th} = 2\pi\sqrt{M/K}$ (seconds)

Where M = mass of the body suspended from the spring (mh + Mass attached) (kg)

5. Theoretical Frequency, $F_{th} = 1/T_{th}$ (Hz)

CALCULATIONS:

GRAPH:

Draw load vs deflection graph

RESULTS:

The results obtained from the experiment are tabulated and the percentage error is………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION: The natural Frequency of system is calculated experimentally and compared with theoretically.

Experiment No.:02 **Date:**

TORSIONAL VIBRATION OF A SINGLE ROTOR SYSTEM

UNDAMPED TORSIONAL VIBRATIONS

AIM: To study the undamped torsional vibration of single rotor shaft system.

APPARATUS REQUIRED: Rotor assembly, Weights, Stop clock, measuring tape

THEORY:

When the particles of the shaft or disc move in a circle about the axis of the shaft, then the vibrations are known as torsional vibrations. The shaft is twisted and untwisted alternatively and the torsional shear stresses are induced in the shaft. Since there is no damping in the system these are undamped vibrations. Also there is no external force is acting on the body after giving an initial angular displacement then the body is said to be under free or natural vibrations. Hence the given system is an undamped free torsional vibratory system as shown in figure 2.1.

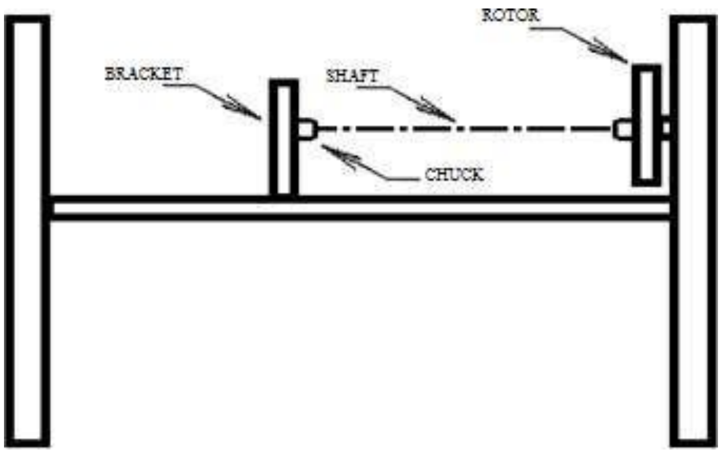


Fig. 2.1: Single rotor system

Undamped vibration: If the vibratory system has no damper system i.e. if there is no reduction in amplitude over every cycle of operation then such a vibration is known as undamped vibration.

Free vibration: when no external force acts on the body after giving it an initial displacement, then the body is said to be under free vibration.

Torsional vibration: when the particles of the shaft or disc moves in a circle about the axis of the shaft, i.e. if the shaft gets alternately twisted and untwisted on account of vibratory motion then the vibration is known as torsional vibration.

PROCEDURE:

1. Fix one end of the shaft at the rotor.
2. Fix the brackets at convenient position along the lower beam and grip the shaft at the bracket by chuck.
3. Twist the through some angle and release.
4. Note down the time required for 10, 20 oscillations.
5. Repeat the procedure for different length of the shaft.

FORMULAE USED:

1 Experimental time period of vibration, $T_{exp} = t/n$

Where t = time for n oscillations (seconds)

n = number of oscillations

2 Experimental Frequency, $F_{exp} = 1/T_{exp}$ (Hz)

3 Polar moment of inertia, $J = \frac{\pi d^4}{32}$ (m^4)

Where d = diameter of shaft

4 Moment of inertia of disc, $I = m \times K^2$ (kgm^2)

Where m = mass of rotating disc (kg)

K = Radius of gyration $= \frac{D}{2\sqrt{2}}$

5 Theoretical Frequency, $F_{th} = \frac{1}{2\pi}\sqrt{\frac{GJ}{IL}}$ (Hz)

OBSERVATIONS:

1. Mass of the rotor disc/rotor, m =
2. Diameter of the Disc/rotor, D =
3. Diameter of the shaft, d =
4. Length of the shaft, L_S =
5. Modulus of rigidity, G =
6. Moment of inertia of disc, I =
7. Polar moment of inertia, J =

MATERIAL DETAILS:

1. Modulus of Rigidity, G_{steel} = 0.86×10^{11} N/m^2
2. Modulus of Rigidity, G_{brass} = 0.37×10^{11} N/m^2
3. Mass of rotor disc (rotor) m_1 = 2.46 kg
4. Mass of rotor disc, m_2 = 3.85 kg
5. Weight of the disc, W_1 = 24.13 N
6. Weight of the disc, W_2 = 37.80 N
7. Diameter of the disc, D_1 = 0.20 m
8. Diameter of the disc, D_2 = 0.25 m

TABULAR COLUMN:

Table 2.1: Error estimation of single rotor system

Sl. No.	Length, L (m)	No. of Oscillation, n	Time for n Oscillation, t (s)	T_{exp}(s)	F_{exp} (Hz)	F_{th} (Hz)	% error

CALCULATIONS:

RESULTS :

The results obtained from the experiment are tabulated and the percentage error is ………….

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION: The natural Frequency of system is calculated experimentally and compared with theoretically.

Experiment No.:03 **Date:**

MOTORIZED GYROSCOPE

AIM: To determine the gyroscopic couple and compare it with the actual applied couple.

APPARATUS REQUIRED: Motorized gyroscope, tachometer, standard weights.

THEORY:

A body rotating about an axis offers a resistance to a change in direction of this axis, this is known as gyroscopic effect. Important applications of gyroscopic effect are the gyroscopic compass used in aircrafts, ships and in internal guidance system for missiles and space travel.

Force due to gyroscopic effect must be taken into account in the design of machines. These forces are encountered in the bearings of automobile engines as the automobile takes a turn, in marine turbine as ship pitches in a heavy sea and in a jet aircraft engine shaft when the aircraft changes direction. Figure 3.1 shows the schematic diagram of gyroscope.

The axis about which the body rotates is called axis of spin. The axis about which the shaft tends to tilt is called the axis of precession. The axis about which the torque is present is called torque axis. The three axes are mutually perpendicular to each other.

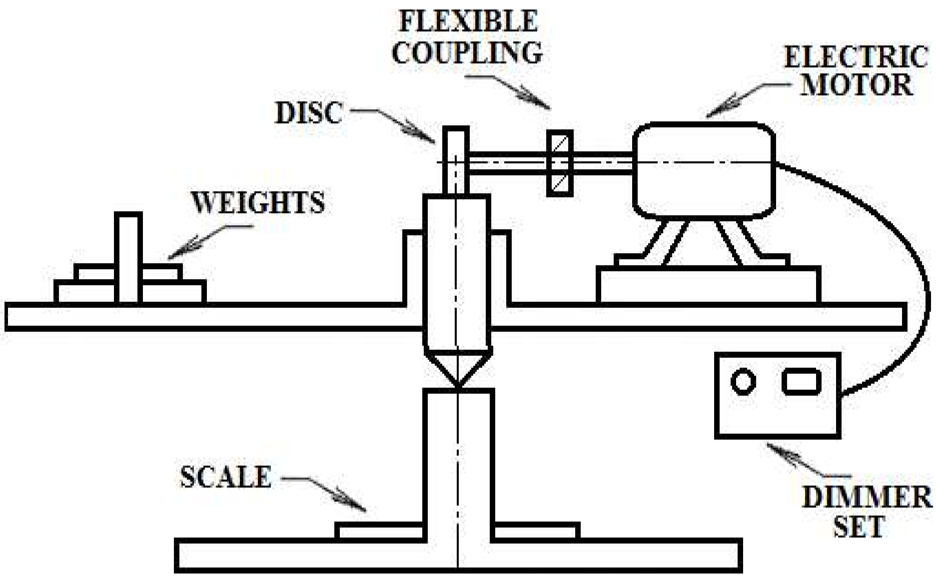


Fig. 3.1: Gyroscope

PROCEDURE:

1 Balance the rotor in the horizontal plane.

2 Start the motor and adjust the speed with the help of voltage regulation.

3 Allow the motor to stabilize & note down the rotor speed with the help of tachometer.

4 Put weights on the stud and at the same time start the stop watch.

5 Referring to the pointer provided note down the time taken for 45^0 precession.

6 Repeat the experiment for different weights and speed.

FORMULAE USED:

1. Moment of inertia, $I = mr^2/2$ (kg-m^2)

   Where m = mass of the rotor (kg)

   r = radius of the rotor (m)

2. Angular velocity of spin, $\omega = 2\pi N/60$(rad/sec)

   Where N = rotor speed (rpm)

3. Angular velocity of axis of spin, $\omega_p = \frac{\pi}{180} \times \frac{\theta}{t}$ (rad/sec)

   Where θ = angle of precession(degrees)

   t = time taken for 45^0precession(seconds)

4. Gyroscopic couple, $G_g = I.\omega.\omega_p$(N-m)

5. Applied couple, $G_a = W.X$(N-m)

   Where W = weight applied (N)

   X = Distance of stud from the disc (m)

OBSERVATIONS:

1. Mass of rotor, m =
2. Rotor diameter, d =
3. Distance of stud from the disc, X =

TABULAR COLUMN:

Table 3.1: Error estimation of couple in gyroscope

Sl. No.	Rotor speed, N (rpm)	Weight attached (N)	Time for 45^0 precession, t (s)	ω (rad/sec)	ω_p (rad/sec)	G_g (N-m)	G_a (N-m)	% error

CALCULATIONS:

RESULTS:

The error in the gyroscope is ……………………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The difference between the values of applied couple and gyroscopic couple is mainly due to the varying supply voltage and also due to the friction in the system.

Experiment No.:04 Date:

WHRLING OF SHAFTS

AIM: To determine the critical speed or whirling speed of a rotating shaft and to verify the value theoretically

APPARATUS REQUIRED: Whirling of shaft apparatus (rigid frame with motor, supporting ends), Shaft, Tachometer, Vernier caliper, Measuring scale.

THEORY:

When a shaft rotates, it may well go into transverse oscillations. If the shaft is out of balance, the resulting centrifugal force will induce the shaft to vibrate. When the shaft rotates at a speed equal to the natural frequency of transverse oscillations, this vibration becomes large and shows up as a whirling of the shaft. The angular velocity of the shaft at which this occurs is called a critical speed or whirling speed. It also occurs at multiples of the resonant speed. At a critical speed, the shaft deflection becomes excessive and may cause permanent deformation or structural damage. This can be very damaging to heavy rotary machines such as turbine generator sets and the system must be carefully balanced to reduce this effect and designed to have a natural frequency different to the speed of rotation. When starting or stopping such machinery, the critical speeds must be avoided to prevent damage to the bearings and turbine blades.

Machine components at a standstill may behave very differently when they are moving, even at relatively low speeds. A solid shaft able to support a hundred times its own weight plus the weight of the components mounted on it may, when rotating at certain speeds, bend and vibrate. The speeds are called 'critical speeds and the bending and the vibration is known as 'whirling'. If this 'critical speed of whirling' is maintained then the resulting amplitude becomes sufficient to cause buckling and failure. However if the speed is rapidly increased before such deleterious effects occur then the shaft is seen to restabilize and run true again until at another specific speed a double bow is produced.

Whirling is usually associated with fast-rotating shafts. When a shaft rotates it is subjected to radial or centrifugal forces, which cause the shaft to deflect from its rest position. These centrifugal forces are unavoidable, since material in homogeneities and assembly

Difficulties ensure that the center of gravity of the shaft or its attached masses cannot coincide with the axis of rotation. Drunkenly first investigated the centrifugal forces involved and determined that the only restabilizing or restoring force was that due to the elastic properties or stiffness of the shaft. Hence, he was able to deduce the speed at which the shaft would suffer an infinite deflection due to whirling. Figure 4.1 shows the schematic representation of whirling of shaft system.

When the speed of rotation is increased the centrifugal force also increases and so does the restoring force. Below the critical speeds, the restoring forces increase with increasing shaft deflection faster than the centrifugal forces, so the deflection is held in check. At the critical speeds, the restoring forces increase at the same rate as the unbalance forces, so they cancel each other out. Shaft deflection is unchecked and the shaft behaves as though it is very flexible. Above the critical speeds the unbalance forces hold sway, and the shaft rotates about the center of mass of the assembly (which is very close to the center of the shaft).

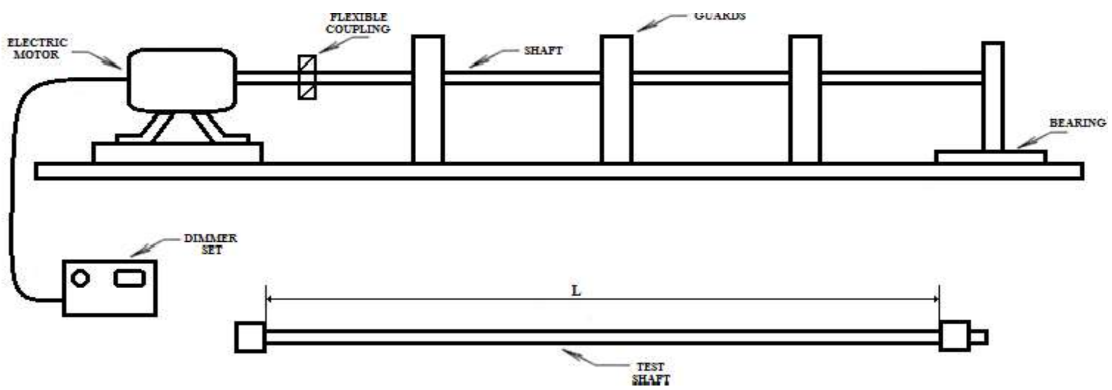


Fig. 4.1: Schematic diagram of whirling of shaft

Consider a shaft of negligible mass carrying a rotor as shown below figure 4.2 (a).Point G is the cg of the rotor, point S is on the shaft axis and point O is on the axis of rotation. Figure 4.2 (b) shown the position of G when the shaft is rotating.

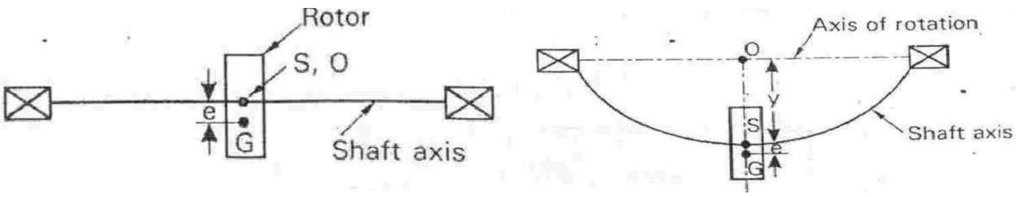


Fig. 4.2 (a) Shaft is stationary (b) Shaft is rotating

Let m=Mass of rotor=W\g

e=Eccentricity

y=Additional deflection due to centrifugal force.

δ = Static deflection under the load

ω= Angular speed of rotor

W=Weight of rotor=mg

K=stiffness of shaft

PROCEDURE:

1. Choose the required size of the shaft.
2. Mount the shaft ends on the frame to obtain the desired condition.
3. Start the motor; Increase the speed by varying voltage.
4. The amplitude of vibrations in lateral direction starts and mode shape is observed.
5. When the first mode appears the corresponding speed is measured using Tachometer.
6. To observe second mode shape the speed is increased further.
7. When the second mode appears the corresponding speed is again measured using Tachometer.
8. The theoretical and experimental frequencies are determined and tabulated.

FORMULAE USED:

1. Moment of Inertia, $I = \frac{\pi d^4}{64}$ (m^4)

   Where d = diameter of the shaft

2. Weight of the shaft, $W = A \times L \times \rho \times g$ (N)

   Where A = Area of the shaft

   L = length of the shaft

   ρ = Density of the shaft material

3. Angular speed, $\omega = 2\pi K_i \sqrt{\frac{gEI}{ML^4}}$ (rad/sec)

   Where i = modes (i = 1,2,3, ------)

   a. Fixed-Fixed: K_1 = 3.56 & K_2 = 8.82

b. Fixed-Free: K_1 = 2.46& K_2= 7.96

4. Theoretical critical speed, $N_{th} = \frac{m}{2n}60$ (rpm)

OBSERVATIONS:

1. End condition =
2. Shaft Material =
3. Diameter of the shaft, d =
4. Length of the Shaft, L =
5. Density of the Shaft material, ρ =
6. Young's Modulus of the shaft material, E =

TABULAR COLUMN:

Table 4.1: Error estimation of critical speed in whirling of shaft

Sl. No.	Mode	Experimental Critical speed, N_{exp} (rpm)	Angular speed, ω (rad/sec)	Theoretical critical speed, N_{th} (rpm)	% error

CALCULATIONS:

RESULTS:

The error between theoretical and experimental critical speed is ………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The speed of the shaft for different modes is determined and verified theoretically.

Experiment No.:05 **Date:**

TORQUE

AIM: To determine the torque applied to the given shaft experimentally & to compare the values theoretically.

APPARATUS REQUIRED: Digital torque indicator, dead weights etc.

THEORY:

Torque, moment or moment of force is the tendency of a force to rotate an object about an axis, fulcrum, or pivot. Just as a force is a push or a pull, a torque can be thought of as a twist to an object. Mathematically, torque is defined as the product of the lever-arm distance vector and the force vector, which tends to produce rotation. Figure 5.1 shows the schematic diagram of torque measurement.

Loosely speaking, torque is a measure of the turning force on an object such as a bolt or a flywheel. For example, pushing or pulling the handle of a wrench connected to a nut or bolt produces a torque (turning force) that loosens or tightens the nut or bolt.

The magnitude of torque depends on three quantities: the force applied, the length of the lever arm connecting the axis to the point of force application, and the angle between the force vector and the lever arm.

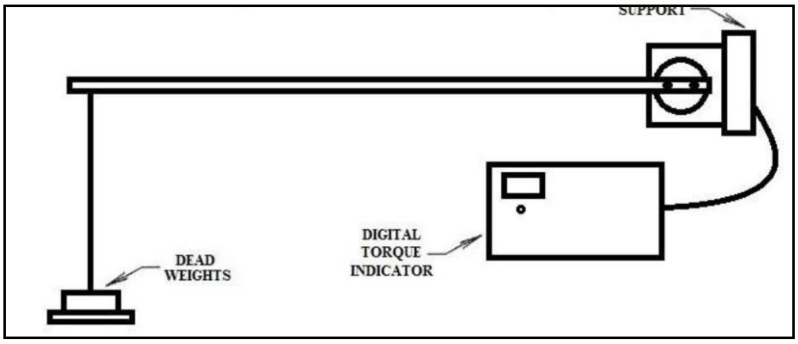


Fig. 5.1: Torque measurement

PROCEDURE:

1. First Check for the electrical connection.
2. Switch on the apparatus, allow it to stabilize.
3. Check the reading indicated by the digital indicator, set it to zero using the knob provided.

4. Apply dead weights on the hanger; allow the system to stabilize (say 5 to 10 seconds).
5. Note down the corresponding torque indicated.
6. Repeat the procedure for different weights and tabulate the readings.

FORMULAE USED:

Torque, T = m x L (kg-m)

Where m = mass added

L = length of arm (span length)

OBSERVATIONS:

Length of arm, L =

TABULAR COLUMN:

Table: 5.1: Error estimation of torque

Sl. No.	Mass attached (kg)	T_{exp}(kg-m)	T_{th}(kg-m)	% error

CALCULATIONS:

RESULTS:

The torque measured experimentally is compared with the theoretical value and the error is found to be…………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSIONS:

The torque is determined experimentally and verified theoretically.

Experiment No.:06 **Date:**

PERFORMANCE OF PORTER GOVERNOR

AIM: To determine the controlling force and sensitivity of the porter governor

APPARATUS REQUIRED: Porter governor, Tachometer, Dimmer set

THEORY:

The function of governor is to increase the supply of working fluid going to the prime-mover when the load on the prime-mover increases and to decrease the supply when the load decreases so as to keep the speed of the prime-mover almost at constant speed at different loads.

When there is change in load, variation in speed also takes place then governor operates a regulatory control and adjusts the fuel supply to maintain the mean speed nearly constant. Therefore, the governor automatically regulates through linkages, the energy supply to the engine as demanded by variation of load so that the engine speed is maintained nearly constant. The governors may, broadly, be classified as

1. Centrifugal governors, and
2. Inertia governors.

PORTER GOVERNOR: - is a modification of a Watt's governor, with central load attached to the sleeve as shown in Figure 6.1(a). The load moves up and down the central spindle. This additional downward force increases the speed of revolution required to enable the balls to rise to any predetermined level. Figure 6.1 (b) represents the force diagram of each ball of porter governor.

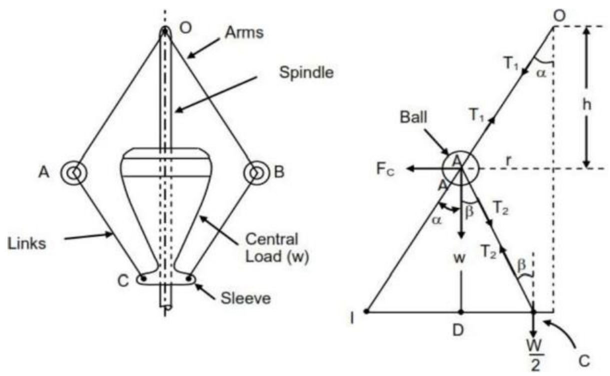


Fig. 6.1: (a) Porter governor (b) Force representation

PROCEDURE:

1. The porter governor assembly is mounted over the spindle and the motor is started.
2. The speed is gradually increased to get sleeve displacement (say 20 mm).
3. The speed of the governor at corresponding displacement is measured using tachometer.
4. The procedure is repeated for different displacements.
5. The readings are noted and tabulated.

FORMULAE USED:

1. Radius of rotation, $r_n = h\frac{s}{y} + r$
2. Controlling force, $F_c = m\,\omega^2 r_n$ (ω = 2nN/60)
3. Frictional force, $F_f = \frac{sN^2hw}{895y} - (W - 2w)$
4. Sensitivity, N = Range of speed/ Mean speed

$$= \frac{2(N_2 - N_1)}{(N_2 + N_1)}$$

OBSERVATIONS:

1. Radius of rotation at rest, r =
2. Length of vertical arm, x =
3. Length of horizontal arm, y =
4. Center load, M =
5. Mass of balls, m =

TABULAR COLUMN:

Table 6.1: Controlling and frictional force of porter governor

Sl. No	Speed, N (rpm)	Lift, h (mm)	Radius, r_n (mm)	Controlling force, F_c (N)	Frictional force, F_f (N)

CALCULATIONS:

RESULTS:

The frictional and controlling forces of porter governor are …………………… and ………………….. respectively.

Sensitivity of the porter governor is ……………………..

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The frictional force and sensitivity of the porter governor is determined.

Experiment No.07 **Date:**

PHOTOELASTICITY (CIRCULAR DISC)

AIM: To calibrate the given photo elastic material using circular disc under compression.

APPARATUS REQUIRED: Circular disc of photo elastic material (Epoxy resin) , Universal loading frame, 12” diffused light transmission polar scope.

THEORY:

Photo elasticity is an experimental technique for stress and strain analysis that is particularly useful for members having complicated geometry, complicated loading conditions, or both. For such cases, analytical methods (that is, strictly mathematical methods) may be cumbersome or impossible, and analysis by an experimental approach maybe more appropriate. While the virtues of experimental solution of static, elastic, two-dimensional problems are now largely overshadowed by analytical methods, problems involving three-dimensional geometry, multiple-component assemblies, dynamic loading and inelastic material behavior are usually more amenable to experimental analysis.

The name photo elasticity reflects the nature of this experimental method: photo implies the use of light rays and optical techniques, while *elasticity* depicts the study of stresses and deformations in elastic bodies. Through the photo elastic-coating technique, its domain has extended to inelastic bodies, too.

PHOTOELASTIC BEHAVIOR **-**The photo elastic method is based upon a unique property of some transparent materials, in particular, certain plastics. Consider a model of some structural part made from a photo elastic material. When the model is stressed and a ray of light enters along one of the directions of principal stress, a remarkable thing happens. The light is divided into two component waves, each with its plane of vibration (plane of polarization) parallel to one of the remaining two principal planes (planes on which shear stress is zero). Furthermore, the light travels along these two paths with different velocities, which depend upon the magnitudes of the remaining two principal stresses in the material.

The incident light is resolved into components having planes of vibration parallel to the directions of the principal stresses s1 and s2. Since these waves traverse the body with different velocities, the waves emerge with a new phase relationship, or relative retardation.2 specifically, the relative retardation is the difference between the number of wave cycles experienced by the two rays traveling inside the body.

Isoclinics are the locus of the points in the specimen along which the principal stresses are in the same direction. It is locus of the point at which the principal plane is inclined to the same extent with respect to reference direction.

Isochromatics are the locus of the points along which the difference in the first and second principal stress remains the same. Thus they are the lines which join the points with equal maximum shear stress magnitude.

Plane polariscope

The setup consists of two linear polarizer and a light source. The light source can either emit monochromatic light or white light depending upon the experiment. First the light is passed through the first polarizer which converts the light into plane polarized light. The apparatus is set up in such a way that this plane polarized light then passes through the stressed specimen. This light then follows, at each point of the specimen, the direction of principal stress at that point. The light is then made to pass through the analyzer and we finally get the fringe pattern.

The fringe pattern in a plane polariscope setup consists of both the isochromatics and the isoclinics. The isoclinics change with the orientation of the polariscope while there is no change in the isochromatics.

Position of quarter wave plates at D-D using the spring loaded pin

Circular polariscope

In a circular polariscope setup two quarter-wave plates are added to the experimental setup of the plane polariscope. The first quarter-wave plate is placed in between the polarizer and the specimen and the second quarter-wave plate is placed between the specimen and the analyzer. The effect of adding the quarter-wave plates is that we get circularly polarized light.

The basic advantage of a circular polariscope over a plane polariscope is that in a circular polariscope setup we only get the isochromatics and not the isoclinics. This eliminates the problem of differentiating between the isoclinics and the isochromatics.

Photoelastic method of stress analysis is based on optical fringe patterns formed when a transparent model of a member is loaded in the pressure of polarized light. An analysis of these fringe patterns yield the stress in the model which are then related to the prototype.

In circular polariscope there are four possible arrangements as given below table 7.1

Table 7.1: Quarter wave plate arrangement configuration

Arrangement	Quarter wave plate	Polarizer & analyzer	Field
A	Crossed	Crossed	Dark
B	Crossed	Parallel	Light
C	Parallel	Crossed	Light
D	Parallel	Parallel	Dark

Only A and B arrangements are recommended.

A little consideration of optical effects produced by different optical elements will show that:

a) Plane polariscope arrangement is suitable for obtaining isoclinics.

b) Circular polariscope arrangement is suitable for obtaining Isochromatics. With dark field arrangement (A),the order of Isochromatics will correspond to a retardation of intergral order of wavelength: i.e by using circular polariscope arrangement corresponding to λ,2λ,3λ etc.

c) With light field arrangement, the order of Isochromatics fringes will correspond to retardation $\lambda/2$,$3\lambda/2$,$5\lambda/2$.(or odd multiples of half).

Thus combining dark and light field arrangement we can get all points of the field nearest to ½ orders of fringes.

Effect of stressed model in circular polariscope (Dark field, arrangement A)

When a stressed photoelastic model is placed in the field of a circular polariscope with its normal coincident with z-axis of the polariscope, optical effects differ somewhat from those obtained in a plane polariscope. The use of a circular polariscope is more widely used than the plane polariscope. To illustrate this effect, consider the stressed model in the circular polariscope shown in the figure 7.1

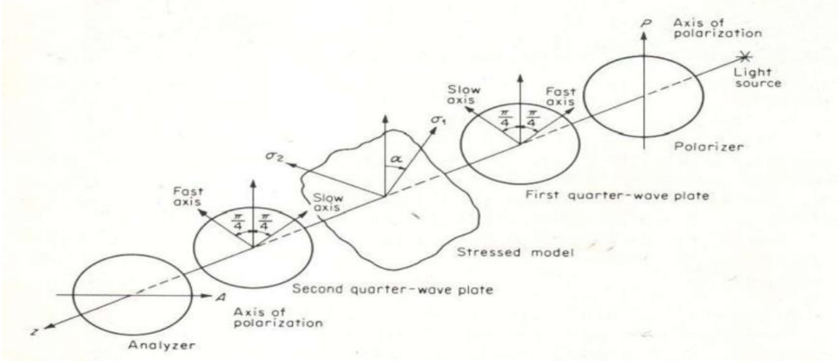


Fig. 7.1: Circular polariscope

Effects of a stressed model in a plane Polariscope

It has been established that the principal stress difference $\sigma 1 - \sigma 2$ can be determined in a two dimensional model if N is measured at each point in the model. Also, it was stated that the optical axes of the model coincide with the principal-stress directions. These two facts can be effectively utilized once a method to measure the optical properties of a stressed model has been established.

Consider first the case of the plane – stressed model inserted into the field of a plane polariscope with its normal coincident with the axis of the polariscope, as illustrated in the figure 7.2. Note that the principal- stress direction at the point under consideration in the model makes an angle α with the axis of polarization of the polarizer.

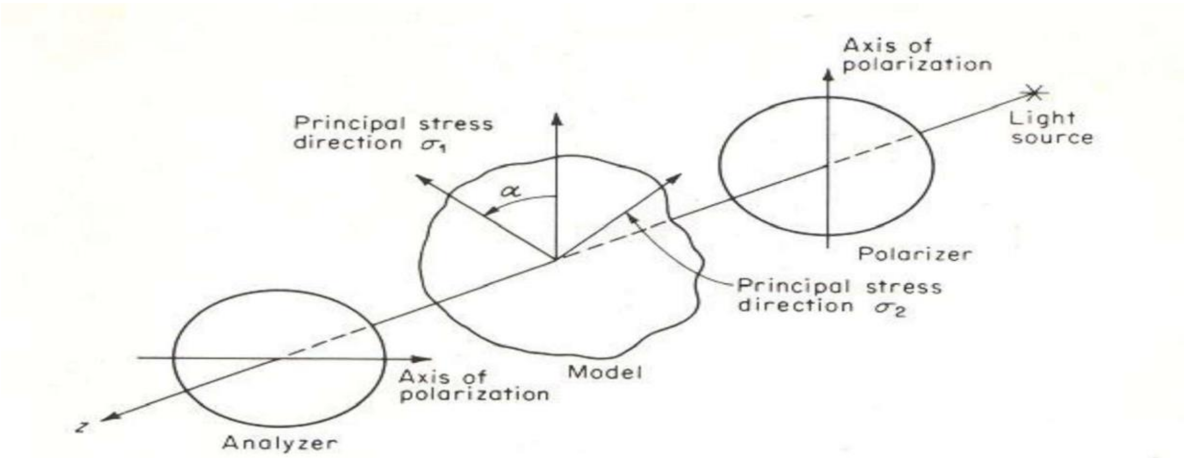


Fig. 7.2: A stressed photoelastic model in plane polariscope

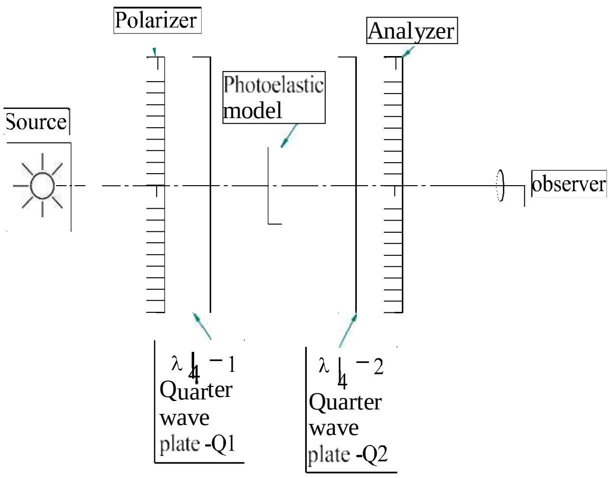
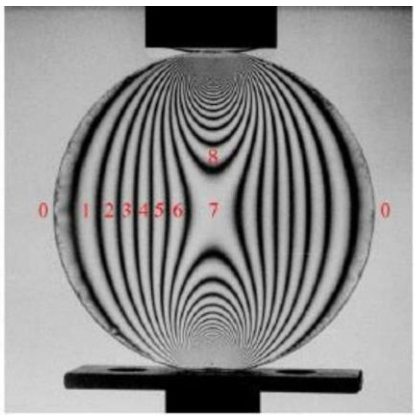


Fig:DIFFUSED LIGHT REASERCH POLARISCOPE



Circular disc under diametrical compression

(Calibration model)

Fig. 7.3: (a) Diffused light research polariscope (b) Circular disc under diametrical compression

TARDAY'S METHOD

This method is used for measuring fractional order by compensation at any desired point. There is every possibility that your point of interest may not be exactly on a integral fringe. In such case fractional fringe order may be found out by this method.

Determination of principle stress difference at a point:

This can be found by using the relation

$$\sigma 1 - \sigma 2 = \frac{N f_\sigma}{h}$$

Where, σ1=Major principal stress

σ2=Minor principal stress

N=Fringe order at a point

f_σ=Material fringe value

h= Thickness of photoelastic model

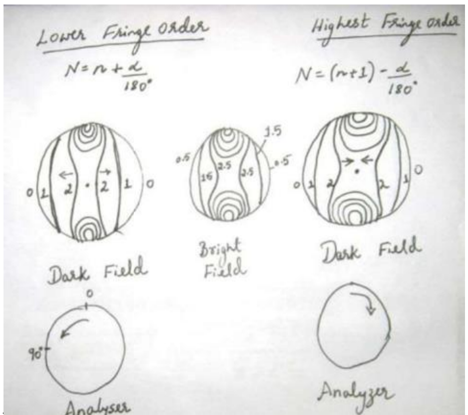


Fig.7.4: Lower and higher fringe order

PROCEUDRE:

1. Load the circular disc in universal loading frame, under diametrical compression as shown in figure 7.5.
2. The distance of X & Y must be measured initially.
3. Apply light load on plain polariscope (D-D) arrangement.
4. Observe the isoclinic fringe pattern and note the isoclinic reading at the center of the disc which is automatically zero.
5. Switch 'ON' the sodium lamp (monochromatic) 10 mins before conducting the test.
6. Load the specimen gradually and set to circular polariscope (M-M) position.
7. Use white light and Observe the isoclinic fringe pattern at the center of the disc.
8. Use Tardy's method if required ,to find fractional fringe order at the center of the disc. This can be done by rotating the analyzer either clock wise or anticlock wise to enable the coinciding at the center from lower or higher fringe order.
9. Determine the average fringe order.
10. Repeat the above procedure for different loads.

OBSERVATION:-

a. Distance X =…………..cm

b. Distance Y =…………..cm

c. Diameter of the disc d =……..……cm

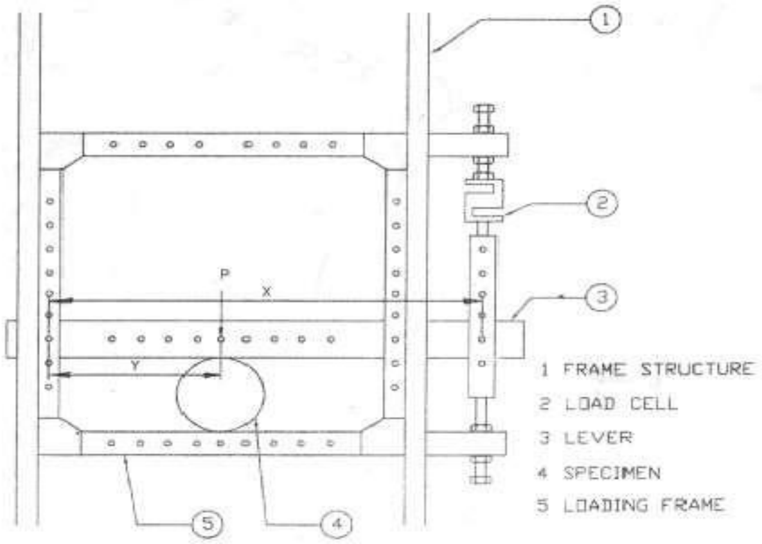


Fig.7.5: Experimental setup of disc under compression

TABULAR COLUMN:

Table 7.1: Determination of material fringe order

Load applied 'W'in Kg on load cell	**Load on model in Kg** $P = \frac{W.X}{Y}$	**Fractional Fringe 'N' at center**			**Material Fringe Order** $f_o = \frac{8P}{\pi DN}$ **Kg/cm**	**Average 'f_σ' Kg/cm**
		Lowest Fringe Order	**Higher Fringe Order**	**Average Fringe Order**		
4						
8						
10						
15						

FORMULA :

Material Fringe constant

$f_\sigma = \frac{8P}{\pi DN}$ **in Kg/cm**

CALCULATIONS :

RESULTS:

The material and model fringe value for the given photoelastic material is founded to be ……….

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSIONS:-

Circular disc model is easy to prepare and easy to load. Hence it is suitable for calculation of photoelastic model material. The avg. value of f_σ calculated is in good arrangements with values specified in the text for epoxy category.

OR

The stress distribution along the horizontal diameter in a circular disc under compression is given by

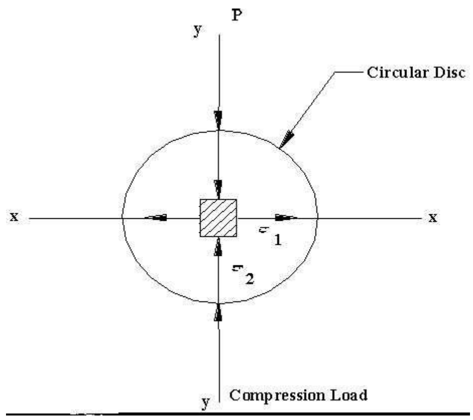


Fig. 7.6: Stress distribution of circular disc under compression

Circular disc subjected to compressive load

$\sigma_x = \sigma_1 = 2P/\pi dh[(d^2-4x^2)/(d^2+4x^2)]^2$

$\sigma_y = \sigma_2 = -2P/\pi dh[4d^4/(d^2+4x^2)^2-1]$

At the center i.e. x=0

$$\sigma_1 = 2P/\pi dh \quad \text{and} \quad \sigma_2 = -6P/\pi dh$$

$$\sigma_1 - \sigma_2 = 8P/\pi dh \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(1)}$$

From stress optic law for Two dimensional stress system

$$\sigma_1 - \sigma_2 = Nf/h \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(2)}$$

Equating eqs (1) and (2)

$$Nf/h = 8P/\pi dh$$

$$\Rightarrow \quad f = (8/\pi d)(P/N)$$

$$= (8/\pi d)(\Delta P/\Delta N) = \qquad \text{N/mm/fringe} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3)}$$

By knowing the loads required for producing different number of fringes, a graph of P Vs N is plotted and the slope of this linear graph gives ($\Delta P/\Delta N$) which is used to estimate fringe constant (f). This is referred as Calibration.

SPECIFICATIONS:

Diameter of the Specimen : d = mm

Thickness of the Specimen : h = mm

Distance from the fulcrum to the Applied load : Y = mm

Distance from fulcrum to the center of the specimen : X= mm

TABULAR COLUMN:

Table 7.2: Determination of material fringe constant using slope method

Sl. No	Fringe Order (N)	Load applied (W)		Effective load (P)	Slope of line ($\Delta P/\Delta N$)	Material fringe constant (f_σ)
		Kg	N	N	N/fringe	N/mm/fringe

GRAPH: P v/s N (linear)

SPECIMEN CALCULATIONS:

Effective load $P = W \times Y/X$ = N (By taking moments)

Slope from graph $= \Delta P/\Delta N$ = N/fringe

Material fringe constant $(f_\sigma) = (8/\pi d)\ (\Delta P/\Delta N)$ = /mm/fringe.

RESULTS AND DISCUSSIONS

The material fringe value of the given Photoelastic material ……………………….N/mm/fringe

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The material fringe value of a given photoelastic material is determined.

Experiment No. 08 **Date**:

DETERMINATION OF STRESS CONCENTRATION FACTOR

AIM: To determine the stress concentration for a circular disc with a circular hole at the Center under diametral compression.

APPARATUS REQUIRED: Circular Polariscope with Accessories (Photoelastic Bench), Photoelastic model in the form of a circular disc with central hole, Weights varies 50gms to 10kg.

THEORY: The stress distribution along the horizontal diameter in a circular disc under compression as shown in figure 8.1.

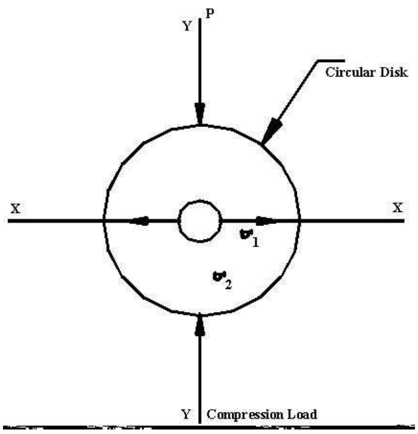


Fig. 8.1: Stress distribution of circular disc under compression

Circular disc with central hole subjected to pure compression

$$\sigma_x = \sigma_1 = 2P/\pi dh[(d^2-4x^2)/(d^2+4x^2)]^2$$

$$\sigma_y = \sigma_2 = -2P/\pi dh[4d^4/(d^2+4x^2)^2-1]$$

At the center i.e.at x=0

$$\sigma_1 = 2P/\pi dh \quad \text{and} \quad \sigma_2 = -6P/\pi dh$$

$$\sigma_1 - \sigma_2 = 8P/\pi dh \ldots\ldots (1)$$

For circular disc with central hole

$$\sigma_1 - \sigma_2 = 8P/\pi(D-d)h \ldots\ldots (2)$$

From the stress optic law for Two dimensional stress system

$$\sigma_1 - \sigma_2 = Nf/h \ldots\ldots (2)$$

Equating eqs (1) and (2)

$$Nf/h=8P/\pi(D-d)h$$

$$\Rightarrow \quad f = [8/\pi(D-d)]\ [P/N]$$

$$= [8/\pi(D-d)]\ [\Delta P/\Delta N] \text{ N/mm/fringe} \ldots\ldots\ldots\ldots(3)$$

Estimating the fringe value is called Calibration.

EXPERIMENT SETUP: The arrangement of loading as shown in figure 8.2.

Circular disc with central hole subjected to diametral compression

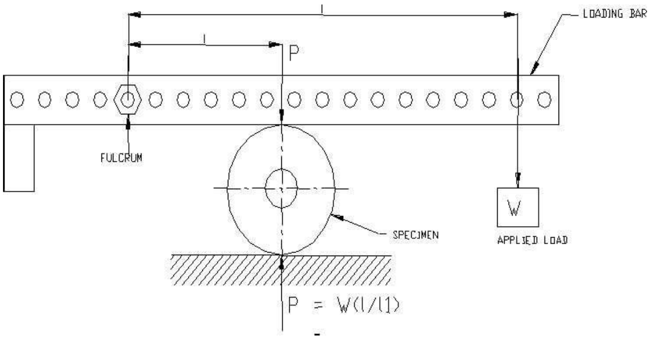


Fig. 8.2: Experimental setup of circular disc with circular hole

PROCEDURE:

1. Hang a pan to the loading bar for placing weights for loading so as to make the lever horizontal.
2. Place the model between the loading arm and the bottom surface of the frame.
3. Measure the distances from the fulcrum to the specimen (l_1) and fulcrum to the load (l).
4. Observe for each fractional load placed on the pan, the specimen through the analyzer.
5. Determine the effective loads required for getting integral fringe orders (0,1,2,3.) at the centre of the circular disk and tabulate.
6. Draw the graph between effective load Vs fringe pattern (linear graph)
7. Calculate the slope of the line.
8. Calculate material fringe constant by using the equation (3)

OBSERVATION:

Diameter of the specimen: D = mm

Diameter of the hole: d = mm

Thickness of the specimen: h = mm

Distance from the fulcrum to the Applied load: Y = mm

Distance from fulcrum to the center of the specimen: X= mm

TABULAR COLUMN:

Table 8.1: Determination of material fringe constant

Sl. No	**Fringe order (N)**	**Load applied (W)**		**Effective load (P)**	**Slope of line ($\Delta P/\Delta N$)**	**Material fringe constant (f_σ)**
		Kg	**N**	**N**	**N/fringe**	**N/mm/fringe**

GRAPH: P v/s N

SPECIMEN CALCULATIONS:

Effective load P = W x Y/X = N (By taking moments)

Slope from graph $\Delta P/\Delta N$ = N/fringe

Material fringe constant f = $[8/\pi(D-d)]\,[\Delta P/\Delta N]$ N/mm/fringe.

Nominal stress σ_{nom} = P/(D-d) h = N/mm^2

Maximum induced stress σ_{max} =Nf/h = N/mm^2

Stress concentration factor K_σ = $\sigma_{max}/\sigma_{nom}$ =

TABULAR COLUMN:

Table 8.2: Determination of stress concentration

Sl No	Fringe No	Load	Nominal stress (σ_{nom})	Max induced stress (σ_{max})	Stress concentration (K_σ)
1					
2					
3					
4					

RESULTS:

The material fringe value of the given Photoelastic material is …………. N/mm/fringe.

The Stress concentration factor at the hole for the given Model is ……………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The material fringe value and stress concentration factor for a given photoelastic material is determined.

Note: Colour Code to determine the fringes

Table 8.3: Colour code

Colour	Fringe No. (Approx.)
Black/Brown	0
Blue	1
Rose red	2
Green(light)	Fringe No. 3
Orange red	3

Experiment No. 09 **Date:**

PHOTOELASTICITY (PURE BENDING)

AIM: To calibrate photo-elastic material model by using a beam subjected to pure bending (four point bending).

APPARATUS REQUIRED: Universal loading frame, 1 2" diffused light transmission polariscope beam model prepared out of photoelastic material.

THEORY:

Photoelasticity is an experimental technique for stress and strain analysis that is particularly useful for members having complicated geometry, complicated loading conditions, or both. For such cases, analytical methods (that is, strictly mathematical methods) may be cumbersome or impossible, and analysis by an experimental approach maybe more appropriate.

While the virtues of experimental solution of static, elastic, two-dimensional problems are now largely overshadowed by analytical methods, problems involving three-dimensional geometry, multiple-component assemblies, dynamic loading and inelastic material behavior are usually more amenable to experimental analysis.

Isoclinics are the locus of the points in the specimen along which the principal stresses are in the same direction. It is locus of the point at which the principal plane is inclined to the same extent with respect to reference direction.

Isochromatics are the locus of the points along which the difference in the first and second principal stress remains the same. Thus they are the lines which join the points with equal maximum shear stress magnitude.

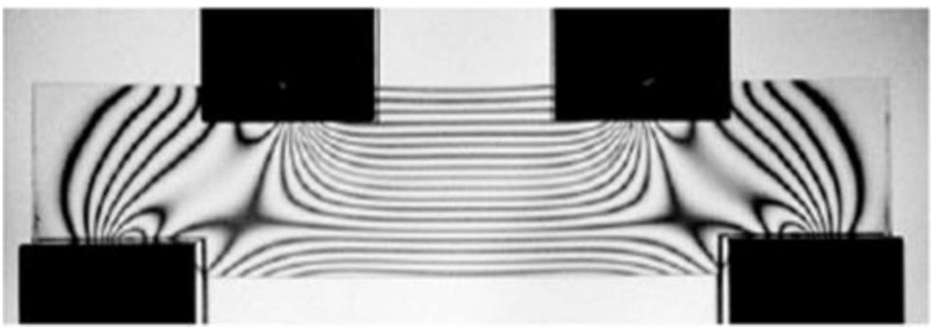


Fig. 9.1: Beam in 4-point bending

EXPERIMENT SETUP: The arrangement of loading as shown in figure 9.2.

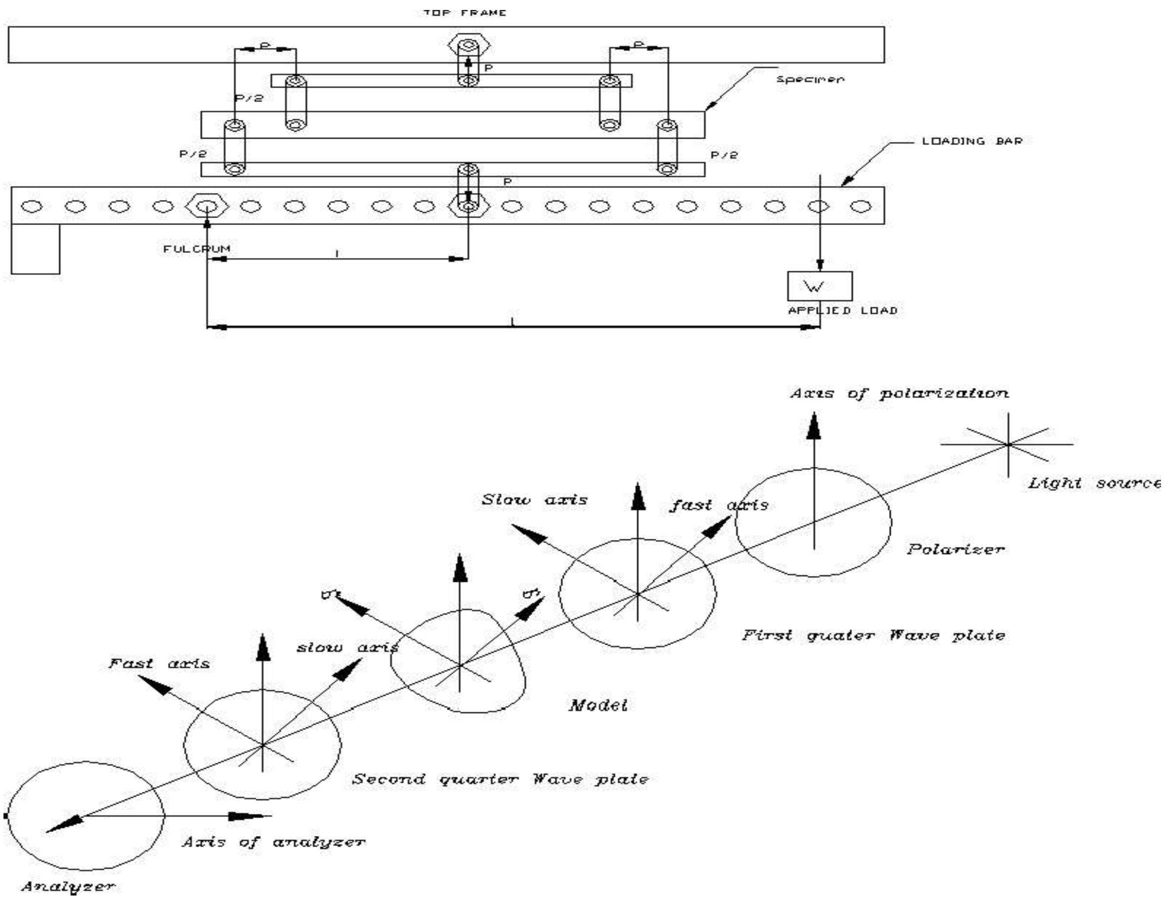


Fig. 9.2: Stressed model of a circular polariscope

PROCEDURE:

1. Hang a pan to the loading bar for placing weights for loading so as to make the lever horizontal.
2. Place the model between the loading arm and the bottom surface of the frame.
3. Measure the distances from the fulcrum to the specimen (l_1) and fulcrum to the load 'W' (l).
4. Observe for each fractional loads placed on the pan, the specimen through the analyzer.
5. Determine the effective loads required for getting integral fringe orders (0,1,2,3..) at the centre of the circular disk and tabulate.
6. Draw the graph between effective load Vs fringe pattern (linear graph)
7. Calculate the slope of the line.
8. Calculate material fringe constant by using the equation (4)

OBSERVATION:

Width of the specimen	: d	=	mm
Thickness of the specimen	: h	=	mm
Distance from the fulcrum to the Applied load	: Y	=	mm
Distance from fulcrum to the centre of the specimen	:X	=	mm
Eccentricity (Distance between pt. of loading and supports)	:e or L	=	mm

FORMULA USED:

The Bending stress in the rectangular specimen can be calculated by the formula using figure 9.3.

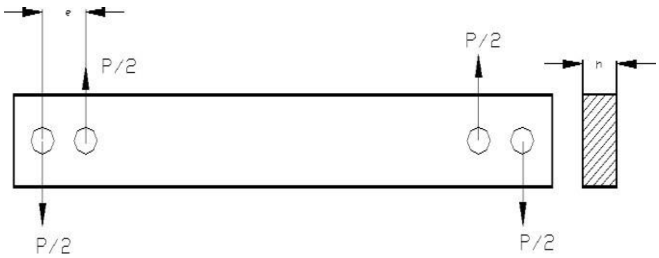


Fig. 9.3: Bending stress rectangular specimen

$$\sigma_b = \frac{M}{I} y$$

Where M_b= Constant Bending moment =

y = Distance the outer most fiber from the neutral axis =

I = Moment of inertia about neutral axis =

$$\sigma_b = \frac{3Pe}{hd^2} \rightarrow (1)$$

From stress optic law: (σ_1 -σ_2) = Nf/h …………… (2)

But σ_2 = 0 on the free boundary surface and σ_1 = σ_b:σ_b = Nf/h ………… (3)

Equating equations (1) and (3): $\frac{3Pe}{hd^2} = \frac{Nf}{h}$

$$\therefore f = \frac{3e}{d^2}\left[\frac{P}{N}\right] = \frac{3e}{d^2}\left(\frac{\Delta P}{\Delta N}\right) \text{………… (4)}$$

(ΔP/ΔN) From graph. $f = \frac{3e}{d^2}\left(\frac{\Delta P}{\Delta N}\right) =$

TABULAR COLUMN:

Table 9.1: Determination of material fringe constant

Sl. No	Fringe order (N)	Load applied (W)		Effective load (P)	Slope of line ($\Delta P/\Delta N$)	Material fringe constant (f_σ)
		Kg	N	N	N/fringe	N/mm/fringe
1						
2						
3						
4						
5						

GRAPH: P v/s N (linear)

SPECIMEN CALCULATIONS:

Effective load	P = W x Y/X =	N (By taking moments)
Slope from graph	$\Delta P/\Delta N$ =	N/fringe
Material fringe constant	$f_c = \frac{3e}{d^2}\left(\frac{\Delta P}{\Delta N}\right)$ =	N/mm/fringe

RESULTS:

The material fringe value of the given Photoelastic material is ………………….N/mm/fringe.

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

a) Model of beam subjected to pure bending can be effectively used to calibrate the photoelastic material
b) There is no need to change load from an value of load given a set of readings
c) Average f_σ is calculated which is in close agreements with volume specified for epoxy resins

Note:

We know that from theory of bending

$$\frac{M}{I} = \frac{f}{Y} \therefore f = \frac{MY}{I}$$

But $\sigma 1 - \sigma 2 = \frac{MY}{I}$

Also w.k.t $\sigma 1 - \sigma 2 = \frac{Nf_\sigma}{b}$

$$\frac{Nf_\sigma}{b} = \frac{MY}{I}$$

$$\therefore f_\sigma = \frac{MYb}{IN}$$

Experiment No. 10 **Date:**

CALIBRATION OF PHOTOELASTIC MATERIAL UNDER TENSILE LOAD

AIM: To calibrate the Photoelastic material under tensile load.

APPARATUS REQUIRED: Circular Polariscope with Accessories (Photoelastic Bench), Photoelastic model (tensile specimen), Weights.

THEORY: When a specimen of uniform cross section of width 'w' and the thickness 'h' is subjected to an external load 'P',

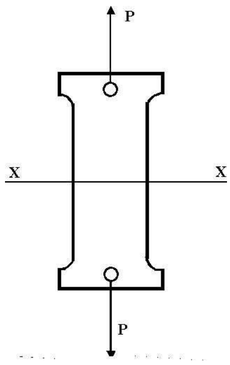


Fig.10.1: Tensile specimen of photoelastic model

The axial stress in the specimen(at the center) is given by

$$\sigma = P/bh$$

The specimen is in a plane state of stress with

$$\sigma_1 = P/bh \quad \text{and} \quad \sigma_2 = 0$$

$$\sigma_1 - \sigma_2 = P/bh \ldots\ldots\ldots\ldots (1)$$

From stress optic law for Two dimensional stress system

$$\sigma_1 - \sigma_2 = Nf/h \ldots\ldots\ldots\ldots (2)$$

Equating eqs (1) and (2)

$$Nf/h = P/bh$$

$$\Rightarrow f = (1/b) * (P/N)$$

$$= (1/b)\,(\Delta P/\Delta N) \quad = \qquad \text{N/mm/fringe} \ldots\ldots\ldots (3)$$

Fringe value for a plate with a hole

$$f = \frac{1}{b-d}\left(\frac{\Delta P}{\Delta N}\right)$$

Estimating the fringe constant (f) is referred as Calibration.

PROCEDURE:

1. Hang a pan to the loading bar for placing weights for loading so as to make the lever horizontal.
2. Place the model between the loading arm and the bottom surface of the frame.
3. Measure the distances from the fulcrum to the specimen (l_1) and fulcrum to the load 'W' (l).
4. Observe for each fractional loads placed on the pan, the specimen through the analyzer.
5. Determine the effective loads required for getting integral fringe orders(0,1,2,3..) at the center of the circular disk and tabulate.
6. Draw the graph between effective load vs fringe pattern(linear graph)
7. Calculate the slope of the line.
8. Calculate material fringe constant by using the equation(3)

EXPERIMENT SETUP: The arrangement of loading as shown in figure 10.2.

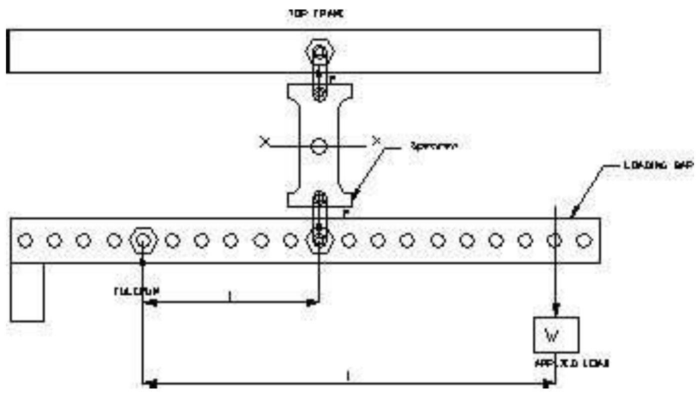


Fig.10.2: Experimental setup of photoelastic tensile specimen

OBSERVATION:

Width of the specimen	: b =	mm
Diameter of the hole	:d=	mm
Thickness of the specimen	: h =	mm
Distance from the fulcrum to the Applied load	: Y =	mm
Distance from fulcrum to the center of the specimen	: X=	mm

TABULAR COLUMN:

Table 10.1: Determination of material fringe constant

Sl. No	Fringe order (N)	Load applied (W)		Effective load (P)	Slope ($\Delta P/\Delta N$)	Material fringe constant (f_σ)
		Kgf	N	N	N/fringe	N/mm/fringe

GRAPH: P v/s N

SPECIMEN CALCULATIONS:

Effective load P = W x l/l_1 = N (By taking moments)

Slope from graph = $\Delta P/\Delta N$ = N/fringe

Material fringe constant $f = \frac{1}{b-d}\left(\frac{\Delta P}{\Delta N}\right)$ N/mm/fringe.

TABULAR COLUMN:

Table 10.2: Determination of material fringe constant

Sl No	Fringe No	Load	Nominal stress (σ_{nom})	Max induced stress (σ_{max})	Stress concentration (K_σ)
1					
2					
3					

RESULTS AND DISCUSSIONS:

The material fringe value of the given Photoelastic material is……………………N/mm/fringe.

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

The material fringe value of the given photoelastic material is determined and graph of load versus material fringe number is plotted and said to be linear.

Experiment No. 11 **Date:**

JOURNAL BEARING

AIM: To study the pressure distribution in a Journal bearing.

APPARATUS REQUIRED: Journal bearing setup, Tachometer, weights and lubricating oil.

THEORY: A Journal bearing supports a shaft and permits rotary motion. This cause wear of surfaces due to friction between the contact surfaces and heat is generated, resulting in loss of power. To minimize this, lubricating oil is introduced in the clearance between the journal and bearing. Pressure developed in the oil film due to viscous force while the journal is rotating and this separates the contact surfaces (lift the journal). The study of pressure distribution and variables associated with the bearing and can be used for design purposes. The operating characteristics such as load carrying capacity and coefficient of friction of a full journal bearing will be discussed.

EXPERIMENTAL SETUP: It consists of a Journal and bearing assembly connected to a D.C. motor the motor is fixed on a rigid support. The bearing carries a hook on which weights can be placed. Lubricating oil (SAE 40) is supplied to the bearing through the tubes from the oil tank, which is placed above the bearing. The bearing has 16 tapping, 12 for radial and 5 (including one radial tap, which is also in axial direction) for axial pressure distribution. These tapping are connected to flexible tubes, which are supported vertically. These tubes form manometers for reading the pressure. Each of the tubes is provided with an adjacent scale for measuring the head of the oil.

PROCEDURE:

1. Fill the oil tank with lubricating oil (SAE 40).
2. Drain out the air bubbles from all the manometer tubes on the manometer board as well as from the inlet tubes.
3. Open the inlet valve, after some time note down the initial manometer reading.
4. Check and ensure that the dimmer stat is at zero position.
5. Rotate the dimmer stat knob gradually till the desired speed is reached.
6. Run the setup at this speed, till the oil levels in all the manometer tubes are in steady state.
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 and the main supply.
9. The difference in manometer pressure at each tapping is plotted.

SPECIFICATIONS:

Diameter of the journal d = mm = m

Inside diameter of bearing D = mm = m

Bearing width l = mm = m

Speed of the journal N = rpm

Speed of the journal n =N/60 = rps

Lubricating Oil used = SAE 40

Viscosity of the oil around 40^0 = η = 150×10^{-3}

Self weight of the bearing W_1 = 3.0 kg

Load on the shaft after applying a load $w_a W_2 = W_1+W_a$

TABULAR COLUMN (RADIAL DIRECTION):

Table 11.1: Determination of pressure in radial direction

Sl No	Tube No	Position of the tap	Supply head P_s(Cm)	Absolute pressure head while running the motor. p_a (Cm)		Pressure developed $p = p_a - p_s$(Cm)	
				With Self Weight	With Additional Load	With Self Weight	With Load W_2
01							
02							
03							
04							
05							
06							
07							
08							
09							
10							
11							
12							

TABULAR COLUMN (AXIAL DIRECTION)

Table 11.2: Determination of pressure in axial direction

Sl No	Tube No	Supply head p_s(Cm)	Absolute pressure head while running the motor. p_a (Cm)		Pressure developed $p = p_a - p_s$(Cm)	
			With Self Weight	With Additional Load	With Self Weight	With Load W_2
01						
02						
03						
04						
05						

Graph: Graph to be plotted for pressure distribution (Cm) in radial direction at intervals of 30^0.Graph to be drawn for pressure in axial direction (tube No v/s Pressure (p))

STEPS FOR PLOTTING THE GRAPH:

1. Select a suitable scale to plot the pressure distribution curve
2. With the initial pressure head as the radius draw a circle.
3. Divide the circle in to 8 equal divisions to represent the location of the pressure tapping on the bearing along the circumference.
4. Draw radial lines from the center of the circle along these 12 points, starting from the tube 1.
5. Mark the pressure heads along these radial lines corresponding to the tapping.
6. Join these points with a smooth curve.
7. Mark the direction of rotation of the journal as shown in the figure 11.1.

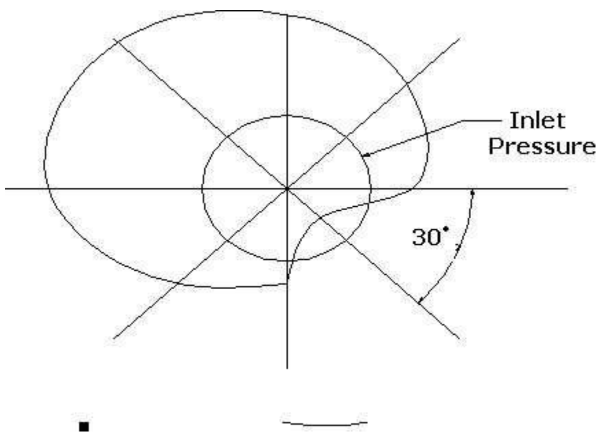


Fig 11.1: Pressure distribution curve in radial direction

SPECIMEN CALCULATIONS:

Unit Load or Pressure $p = \frac{\text{Load}}{\text{Projected Area}} = \frac{W}{ld} =$ N/m^2

Diametral Clearance = $\psi = \frac{c}{d} = \frac{D - d}{d} =$

Petroffs equation for Coefficient of friction: $\mu = 2\pi^2 \left(\frac{\eta n}{p}\right)\left(\frac{1}{\psi}\right) =$

Table 11.3: Pressure and co-efficient of friction wrt different loads

Load (N)	Pressure (p) N/m^2	Coefficient of friction μ
W_1		
W_2		

RESULTS AND DISCUSSIONS:

The co-efficient of friction is …………………………………..

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION: Pressure developed in bearing is measured in axial and radial direction and drawn the pressure distribution diagram.

Experiment No. 12 **Date:**

STRAIN ROSETTES

RECTANGULAR ROSETTE

AIM: To determine Principal stresses and strains induced at a point on the surface of the specimen when it is subjected to combined Bending moment & torque.

APPARATUS USED: Experimental setup, which includes strain gauges, mounted on the specimen, weights and strain indicator.

THEORY: Electrical resistance strain gauges are widely used because of its negligible mass, their Small size and faithful response to rapidly fluctuating strains. As the output is electrical, remote observation is possible. The output can be displayed, recorded or processed as required.

Electrical resistance strain gauges are widely used in

1. Experimental study of stresses in transports vehicles, Aircraft, Ships, Automobiles and Trucks.
2. Experimental analysis of stresses in structures and machines, Apartment buildings, Pressure vessels, Bridges, Dams, Transmission towers, Steam and Gas turbines.
3. Experimental verification of theoretical analysis
4. Assist failure analysis.
5. As a sensing element in Transducers for measurement of force, load, pressure, displacement and Torque.

Strain gauges are very sensitive to temperature. The error in strain measurement due to temperature variation can be reduced to a minimum either through the use of suitable compensate gauge or By using self compensated gauges.

Strain gauges can be used for the measurement of strains on the free surface of any member. In electrical strain resistance gauge a change in length or strain produces a change in resistance. It is necessary to measure 3 strains at a point ($\varepsilon_x, \varepsilon_y, \gamma_{XY}$) to completely define either the strain or stress field. To determine Principal strains (ε_1 and ε_2) and the direction ϕ of ε relative to the X-axis. It is necessary to employ multiple element strain gauges and they can be arranged in combination to get three-element rectangular rosette or three-element delta rosette four-element rectangular rosette etc.

For Three element rectangular rosette with gauges A, B and C with angles of θ_A, θ_B and θ_C respectively, the strains induced are

$\varepsilon_A = \varepsilon_x \cos^2\theta_A + \varepsilon_y \sin^2\theta_A + \gamma_{XY}\cos\theta_A .\sin\theta_A \quad \rightarrow \quad (1)$

$\varepsilon_B = \varepsilon_x \cos^2\theta_B + \varepsilon_y \sin^2\theta_B + \gamma_{XY}\cos\theta_B .\sin\theta_B \quad \rightarrow \quad (2)$

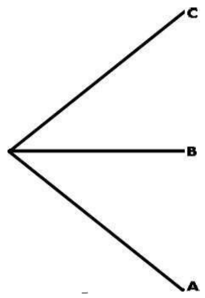
$\varepsilon_C = \varepsilon_x \cos^2\theta_C + \varepsilon_y \sin^2\theta_C + \gamma_{XY}\cos\theta_C .\sin\theta_C \quad \rightarrow \quad (3)$

At a point on the member, the strain gauge B is mounted along the axis of the shaft and the strain gauges A and C are mounted 45^0 the strain gauge B clock wise and anticlock wise respectively.

Hence $\theta_A = -45^0$, $\theta_B = 0^0$ and $\theta_C = 45^{04}$

Substituting these values in the equations (1), (2) and (3), We can get the values of ε_x, ε_y and γ_{XY}

$\varepsilon_x = \varepsilon_B$



$\varepsilon_y = (\varepsilon_A + \varepsilon_C - \varepsilon_B)$ and $\gamma_{XY} = \varepsilon_C - \varepsilon_A$

Principal Strains:

$$\varepsilon_1 = \frac{\varepsilon_A + \varepsilon_C}{2} + \frac{1}{2}\sqrt{(2\varepsilon_B - \varepsilon_A - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

$$\varepsilon_2 = \frac{\varepsilon_A + \varepsilon_C}{2} - \frac{1}{2}\sqrt{(2\varepsilon_B - \varepsilon_A - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

Principal Stresses:

$$\sigma_1 = \frac{E}{1-\gamma^2}\left(\varepsilon_1 + \gamma\varepsilon_2\right)$$

$$\sigma_2 = \frac{E}{1-\gamma^2}\left(\varepsilon_2 + \gamma\varepsilon_1\right)$$

Principal Directions:

$$\phi = \frac{1}{2}\tan^{-1}\left[\frac{\varepsilon_C - \varepsilon_A}{2\varepsilon_B - \varepsilon_A - \varepsilon_C}\right]$$

$$\tan 2\phi = \left[\frac{\gamma_{xy}}{\varepsilon_x - \varepsilon_y}\right]$$

PROCEDURE:

1. Connect the strain gauges A, B and C to the channels 1, 2 & 3 of strain indicator.
2. Set the gauge factor to the desired value on the indicator.
3. Set initial reading on the indicator to zero.
4. Apply the load (increments of 500gms) at a distance of l.
5. Note down the readings for three strain gauges A,B and C and find the direction of Principal stresses and strains.

TABULAR COLUMN:

Table 12.1: Determination of shear stress and shear strain of rectangular rossette

Sl No	Load (Kgf)	Load (F) (N)	Strain Indicator Reading (με)			Principal Strains (με)		Principal Stresses (MPa) (Exp)		Principal Angles		Max Shear Stress (MPa) (Exp)	Max Shear Stress (MPa) (The)	Max Shear Strain (με)
			ε_A	ε_B	ε_C	ε_1	ε_2	σ_1	σ_2	ϕ_1	ϕ_2	τ_{max}	τ_{max}	max
01														
02														
03														
04														
05														

SPECIFICATIONS:

Material of the shaft :

Youngs modulus : $E =$

Modulus of rigidity : $G =$

Poison's ratio : $\gamma =$

Outside diameter of the shaft : $d_o =$

Inside diameter of the shaft : $d_i =$

Arm length for bending : $l_b =$

Arm length for twisting : $l =$

Resistance of each strain gauge : $R_G = 350\Omega$

Gauge factor : $S_a =$

SPECIMEN CALCULATIONS (EXPERIMENTAL):

Principal Strains:

$$\varepsilon_1 = \frac{\varepsilon_A + \varepsilon_C}{2} + \frac{1}{2}\sqrt{(2\varepsilon_B - \varepsilon_A - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

$$\varepsilon_2 = \frac{\varepsilon_A + \varepsilon_C}{2} - \frac{1}{2}\sqrt{(2\varepsilon_B - \varepsilon_A - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

Principal Stresses:

$$\sigma_1 = \frac{E}{1-\gamma^2}(\varepsilon_1 + \gamma\varepsilon_2)10^{-6} =$$

$$\sigma_2 = \frac{E}{1-\gamma^2}(\varepsilon_2 + \gamma\varepsilon_1)10^{-6} =$$

Principal Directions:

$$\phi = \frac{1}{2}\tan^{-1}\left[\frac{\varepsilon_C - \varepsilon_A}{2\varepsilon_B - \varepsilon_A - \varepsilon_C}\right]$$

$\phi = \phi_1 =$

$\phi_2 = \phi_1 + 90^0 =$

Max Shear Strain:

$$\gamma_{max} = \frac{\tau_{max}(exp)}{G} =$$

Max Shear Stress (Exp):

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} =$$

THEORETICAL CALCULATIONS:

Bending Stress $\sigma_b = \frac{M_b}{Z} =$

Torsional Shear Stress $\tau = \frac{M_t}{J} r_o =$

Principal Stresses:

$$\sigma_1 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

$$\sigma_2 = \frac{\sigma_b}{2} - \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

Max Shear Stress (The): $\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} =$

RESULTS AND DISCUSSION:

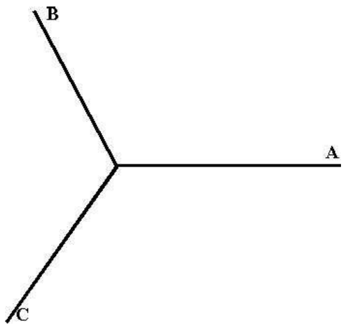
The principal stresses and strains are given by ……………………………...

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION: Magnitude and direction of Principal stresses and strains of the given member at a point are calculated by using the rectangular strain rosette at that point.

DELTA ROSETTE



TABULAR COLUMN:

Table 12.2: Determination of shear stress and shear strain of delta rosette

Sl. No	Load (kgf)	Load (F) (N)	Strain indicator reading (με)			Principal strains (με)		Principal stresses (MPa) (Exp)		Principal angles		Max shear stress (MPa) (Exp)	Max shear stress (MPa) (The)	Max shear strain (με)
			ε_A	ε_B	ε_C	ε_1	ε_2	σ_1	σ_2	ϕ_1	ϕ_2	τ_{max}	τ_{max}	γ_{max}
01														
02														
03														
04														
05														

SPECIFICATIONS:

Material of the shaft	:
Youngs modulus	: E =
Modulus of rigidity	: G =
Poison's ratio	: γ =
Outside diameter of the shaft	: d_o =
Inside diameter of the shaft	: d_i =
Arm length	: l =
Resistance of each strain gauge	: R_G =

Gauge factor : S_a=

SPECIMEN CALCULATIONS (EXPERIMENTAL):

Principal Strains:

$$\varepsilon_1 = \frac{\varepsilon_A + \varepsilon_B + \varepsilon_C}{3} + \frac{\sqrt{2}}{3}\sqrt{(\varepsilon_A - \varepsilon_B)^2 + (\varepsilon_B - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

$$\varepsilon_2 = \frac{\varepsilon_A + \varepsilon_B + \varepsilon_C}{3} - \frac{\sqrt{2}}{3}\sqrt{(\varepsilon_A - \varepsilon_B)^2 + (\varepsilon_B - \varepsilon_C)^2 + (\varepsilon_C - \varepsilon_A)^2}$$

Principal Stresses:

$$\sigma_1 = \frac{E}{1-\gamma^2}(\varepsilon_1 + \gamma\varepsilon_2)10^{-6} =$$

$$\sigma_2 = \frac{E}{1-\gamma^2}(\varepsilon_2 + \gamma\varepsilon_1)10^{-6} =$$

$$\phi = \frac{1}{2}\tan^{-1}\left[\sqrt{3}\,\frac{\varepsilon_C - \varepsilon_B}{2\varepsilon_A - (\varepsilon_B + \varepsilon_C)}\right]$$

Principal Directions: $\phi = \phi_1 =$

$\phi_2 = \phi_1 + 90^0 =$

Max Shear Strain $\gamma_{max} = \dfrac{\tau_{max}(exp)}{G} =$

Max Shear Stress (Exp): $\tau_{max} = \dfrac{\sigma_1 - \sigma_2}{2} =$

Theoretical Calculations:

Bending Stress $\sigma_b = \dfrac{M_b}{Z} =$

Torsional Shear Stress $\tau = \dfrac{M_t}{J} r_o =$

Principal Stresses:

$$\sigma_1 = \frac{\sigma_b}{2} + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2} =$$

$$\sigma_2 = \frac{\sigma_b}{2} - \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

Max Shear Stress (The): $\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} =$

RESULTS AND DISCUSSION:

The principal stresses and strains are given by ……………………………...

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION: Magnitude and direction of Principal stresses and strains of the given member at a point are calculated by using Delta strain rosette at that point.

Experiment No. 13 **Date:**

STATIC AND DYNAMIC BALANCING OF A SHAFT

AIM: 1. To calculate angular and longitudinal positions of counter balancing weights for static and dynamic balancing of an unbalanced rotating mass system.

2. To check experimentally that the positions of counter balancing weights calculated as above are correct.

THEORY: A shaft is said to be statically balanced if the shaft can rest, without turning, at any angular position in its bearings. This condition is attained when the sum of the centrifugal forces on the shaft due to unbalanced masses is zero in any radial direction. The centrifugal force due to unbalanced mass of weight W_i with its centre of gravity at a radial distance r_i is proportional to $W_i r_i$. For a shaft to be statically balanced, the summation of components of all such forces should be zero in any radial direction. That is,

$$\sum_i W_i r_i = 0$$

A shaft is said to be dynamically balanced when it does not vibrate in its running state. To make a shaft dynamically balanced, it must first be statically balanced. In addition, the sum of the moments of centrifugal forces due to the attached masses about any axis erpendicular to the axis of the shaft must be zero. This condition is fulfilled when

$$\sum_i W_i r_i l_i = 0$$

where l_i is the distance of the attached mass from one end of the shaft.

APPARATUS REQUIRED:

Static and dynamic balancing machine.

The machine consists of two frames - a small rectangular main frame and a large rectangular support frame which stands vertically up on a platform. The shaft to be balanced is mounted in the main frame and may be run by an electric motor attached to the lower member of the frame. The axial distance of the masses can be measured by a scale attached to the lower member. The position of masses is determined with the help of a protractor fitted to one end of the shaft. Four different masses are provided which may be clamped on to the shaft at any axial and angular positions.

PROCEDURE:

A. STATIC BALANCING

1. Clamp blocks 1 and 2 on to the shaft at given (known) angular positions and at any convenient axial positions. The shaft becomes statically unbalanced as shown in figure 13.1 (a).

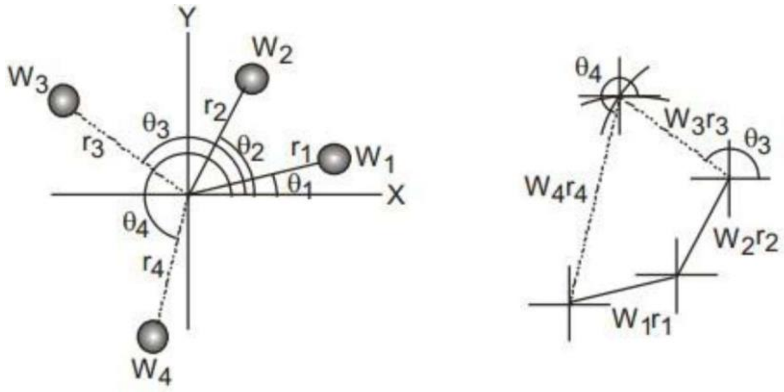


Fig. 13.1: (a) Balancing of rotating masses for static balancing (b) Couple polygon

2. To balance the shaft, blocks 3 and 4 are to be clamped at some angular positions which will satisfy the following equations for static balancing from figure 13.1 (b).

$$\sum_{c} (W_i r_i)_x = \sum_{t} (W_i r_i)\cos\theta_i = 0$$

$$\sum_{t} (W_i r_i)_y = \sum_{t} (W_i r_i)\sin\theta_i = 0$$

The angular positions of blocks 3 and 4 can be found from the above equations. Knowing the W_r-values of the four blocks, one should be able to find the unknown angles with the help of the force polygon.

3. Clamp blocks 3 and 4 on the shaft at the determined angles.

4. They should be statically balanced. Verify that the shaft rests in its bearings at any angular positions.

B. DYNAMIC BALANCING:

1. Take the main frame off from its rigid support and suspend it parallel to the support frame with the help of three springs. Put on the motor belt as shown in figure 13. 2 (a).

2. Place blocks 1 and 2 at given axial and radial positions. Radial positions being calculated earlier, axial positions of blocks 3 and 4 have to be determined for dynamic balancing analytically be using the following equations or graphically by using the couple polygon as shown in figure 13.2 (b).

$$\sum_{c} (W_i r_i l_i)_x = \sum_{t} (W_i r_i \sin\theta_i) L_i = 0$$

$$\sum_{c} (W_i r_i l_i)_y = \sum_{t} (W_i r_i \cos\theta_i) L_i = 0$$

Let their axial positions be indicated by L3 and L4 as required for dynamic balancing.

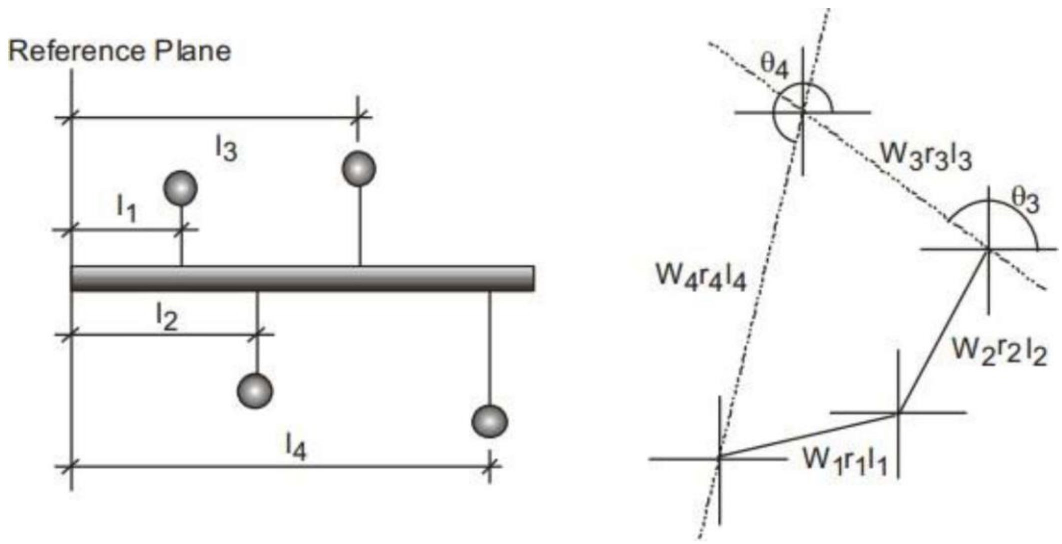


Fig. 13.2: (a) Balancing of rotating masses for dynamic balancing (b) Couple polygon

3. Clamp locks 3 and 4 at the calculated angular and axial positions.

4. Switch on the motor to run the shaft and verify that the shaft does not vibrate.

SPECIFICATIONS:

Dia of rotor D= mm

Dia of shaft d= mm

TABULAR COLUMN: -

Table 13.1: Determination of couple

Plane	Mass 'm' (kg)	Radius 'R' (cm)	Centrifugal Force 'mr' (kg-cm)	Distance Of Planes from Ref. Plane A 'l' (cm).	Couple 'mrl' (kg-cm)

CALCULATIONS:

From couple polygon

From force polygon

RESULTS AND DISCUSSION:

Couple is given by ……………………………………..kg-cm

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

Angular and longitudinal positions of counter balancing weights for static and dynamic balancing of an unbalanced rotating mass system is balanced.

STATIC BALANCING OF A SHAFT

AIM :- To perform the experiment for static balancing on static balancing machine.

APPARATUS USED:- Static Balancing m/c.

THEORY :- A system of rotating masses is said to be in static balance if the combined mass centre of the system lies on the axis of rotation. Whenever a certain mass is attached to a rotating shaft, it exerts some centrifugal force, whose effect is to bend the shaft and to produce vibrations in it. In order to prevent the effect of centrifugal force, another mass is attached to the opposite side of the shaft. The process of providing the second mass in order to counteract the effect of the centrifugal force of the first mass, is called balancing of rotating masses.
The following cases are important from the subject point of view :

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 different masses rotating in the same plane.
4. Balancing of different masses rotating in different planes.

PROCESURE :- Remove the belt, the value of weight for each block is determined by clamping each block in turn on the shaft and with the cord and container system suspended over the protractor disc, the number of steel balls, which are of equal weight are placed into one of the containers to exactly balance the blocks on the shaft. When the block becomes horizontal, the number of balls N will give the value of wt. for the block.
For finding out W_r during static balancing proceed as follow:

1. Remove the belt.
2. Screw the combined hook to the pulley with groove. This pulley is diff. than the belt pulley.
3. Attached the cord end of the pans to above combined hook.
4. Attached the block no.-1 to the shaft at any convenient position and in vertical downward direction.
5. Put steel balls in one of the pans till the blocks starts moving up. (upto horizontal position).
6. Number of balls give the W_r value of block-1. repeat this for 2-3 times and find the average no. of balls.
7. Repeat the procedure for other blocks.

OBSERVATION :-

S.no.	Plane	Mass (m) kg.	Radius ® m	Cent. Force ÷ ω^2 (m.r) kg-m	Distance from plane x(l) m	Couple ÷ ω^2 (m.r.l) kg- m^2

CALCULATION :- The balancing masses and angular positions may be determined graphically as given below :-

1. First of all, draw the couple polygon from the data which are calculated in table to some suitable scale. The vector distance represents the balanced couple. The angular position of the balancing mass is obtained by drawing, parallel to vector distance. By measurement will be find the angle.
2. Then draw the force polygon from the data, which are calculated in table to some suitable scale. The vector distance represents the balanced force. The angular position of the mass is obtained by drawing, parallel to vector distance. By measurement will be find the angle in the clockwise direction from mass.

PRECAUTIONS :-

1. Couple should be represented by a vector drawn perpendicular to the plane of the couple.
2. Angular position measure carefully in clockwise direction.
3. Vector diagram should be represent with suitable scale.

CALCULATIONS:

RESULTS AND DSICUSSION:

Couple is given by ………………………………..

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

Static balancing of shaft is successfully made using static balancing machine.

DYNAMIC BALANCING OF A SHAFT

AIM :- To perform the experiment for dynamic balancing on dynamic balancing machine.

APPARATUS USED:- Dynamic balancing m/c.

THEORY :- When several messes rotates in different planes, the centrifugal force, in addition to being out of balance, also forms couples. A system of rotating masses is in dynamic balance when there does not exit any resultant centrifugal force as well as resultant couple.

Pivoted-cradle Balancing M/C :-

In this type of m/c., the rotor to be balanced is mounted on half-bearing in a rigid carriage and is rotated by a drive motor through a universal joint. Two balancing planes A and B are chosen on the rotor. The cradle is provided with pivots on left and right sides of the rotor which are purposely adjusted to coincide with the two correction planes. Also the pivots can be put in the locked or unlocked position. Thus, if the left pivot is released, the cradle and the specimen are free to oscillate about the locked (right) pivot. At each end of the cradle, adjustable springs and dashpots are provided to have a single degree of freedom system. Usually, their natural frequency is tuned to the motor speed.

PROCEDURE :-

1. First either of the two pivots say left is locked so that the readings of the amount and the angle of location of the correction in the right hand plane can be taken. These readings will be independent of any unbalance in the locked plane as it will have no moment about the fixed pivot.
2. A trial mass at a known radius is then attached to the right hand plane and the amplitude of oscillation of the cradle is noted.
3. The procedure is repeated at various angular positions with the same trial mass.
4. A graph is then plotted of amplitude Vs angular positions of the trial mass to know the optimum angular position for which amplitude is minimum. Then at this position, the magnitude of the trial mass is varied and the exact amount is found by trial and error which reduces the unbalance to almost zero.
5. After obtaining the unbalance in one plane, the cradle is locked in the right hand pivot and released in the left hand pivot. The above procedure is repeated to obtain the exact balancing mass required in that plane.
6. Usually, a large number of test runs are required to determine the exact balance masses in this type of machine. However, by adopting the following procedure, the balance masses can be obtained by making only four test runs :

First make a test run without attaching any trial mass and note down the amplitude of the cradle vibrations. Then attach a trial mass m at some angular position and note down the amplitude of the cradle vibrations by moving the rotor at the same speed. Next detach the trial mass and again attach it at 90° angular position relative to the first position at the same radial distance. Note down the amplitude by rotating the rotor at the same speed. Take the last reading in the same manner by fixing the trial mass 180°. Let the four reading be

OBSERVATION :-

S.No.	Trial Mass	Amplitude
1.	0	X_1
2.	m at 0°	X_2
3.	m at 90°	X_3
4.	m at 180°	X_4

CALCULATION & CONSTRUCTION :-

Draw a triangle OBE by taking OE = 2 X_1, OB = X_2 and BE = X_4. Mark the mid-point A on OE. Join AB.

Now, **OB = OA + AB**

Where, OB = Effect of unbalance mass + Effect of the trial mass at 0°

OA = Effect of unbalanced mass

Thus, AB represents the effect of the attached mass at 0°. The proof is as follows:

Extend BA to D such that AD = AB. Join OD and DE. Now when the mass m is attached at 180° at the same radial distance and speed, the effect must be equal and opposite to the effect at 0° i.e. if AB represents the effect of the attached mass at 0°, AD represents the effect of the attached mass at180°.

Since, **OD = OA +AD**

OD must represent the combined effect of unbalance mass and the effect of the trial mass at 180° (X_4).

Now, as the diagonals of the quadrilateral OBED bisect each other at A, it is a parallelogram which means BE is parallel and equal to OD. Thus, BE also represents the combined effect of
unbalance mass and the effect of the trial mass at 180° or X_4 which is true as it is made in the construction.

Now as OA represents the unbalance, the correction has to be equal and opposite of it or AO. Thus, the correction mass is given by

$$\mathbf{m_c = m.OA/AB} \text{ at an angle } \boldsymbol{\theta} \text{ from the second reading at } \mathbf{0°}.$$

For the correction of the unbalance, the mass m_c has to be put in the proper direction relative to AB which may be found by considering the reading X_3.

Draw a circle with A as centre and AB as the radius. As the trial mass as well as the speed of the test run at 90° is the same, the magnitude must equal to AB or AD and AC or AC′ must represent the effect of the trial mass. If OC represents X_3, then angle is opposite to the direction of angle measurement. If OC′ represents X_3, then angle measurement is in taken in the same direction.

PRECAUTIONS :-

1. Measure the amplitude carefully.
2. Draw the triangle and parallelogram in correct scale.
3. Vector diagram should be represent with suitable scale.

CALCULATIONS:

RESULTS AND DISCUSSIONS:

Couple is given by……………………………………

ANALYSIS OF RESULTS:

DISCUSSIONS:

CONCLUSION:

Dynamic balancing of shaft is successfully made using dynamic balancing machine.

Viva QUESTIONS

Blooms Level 1-Remembering

1. What do you mean by the term vibration?
2. Can you define damped vibrations?
3. What is an undamped vibration?
4. How is damped vibration different from undamped vibration?
5. Can you recall few examples of damped vibrations?
6. Why vibration occurs?
7. Can you define degrees of freedom?
8. What are the different types of vibrations?
9. Define stiffness.
10. What is logarithmic decrement?
11. What do you mean by damping ratio?
12. What do you mean by damping co-efficient?
13. How the term governor is defined?
14. What do you mean by power of the governor?
15. What do you mean by whirling of shaft?
16. Define critical speed of the shaft.
17. What do you mean by 1N-m?
18. What do you mean by couple?
19. Define bearing.
20. What do you mean by viscosity?
21. What is lubrication?
22. What is journal?
23. What do you mean by polarization?
24. What is material fringe constant?
25. Define principal stresses and principal strains.
26. What do you mean by tensor?
27. Define stress concentration factor.
28. How many types of methods are there to solve any engineering problem? Name them.
29. What do you mean by gauge factor?

Blooms Level 2-Understanding

1. How would you classify vibrations?
2. How would you compare longitudinal vibrations with torsional vibrations?
3. State in your own words about degrees of freedom with respect to vibrations.
4. How is damped vibration different from undamped vibration?
5. What is the unit of stiffness?
6. How would you compare critical damping with normal damping?
7. How do you classify governors?
8. How power of the governor is different from the effort of the governor?
9. How the whirling of shaft takes place?
10. What is the importance of eccentricity in whirling of shaft experiment?
11. How do you classify balancing of rotating shaft?
12. What is purpose of balancing the machines?
13. Distinguish between static and dynamic force?
14. How would you classify lubrication?
15. What are the methods of lubrication?
16. What are the properties of good lubricant?
17. What is SAE 1040?
18. What do you mean by calibration?
19. Define Bearing characteristic number and bearing modulus for Journal bearing.
20. What is meant by Hydrodynamic lubrication?
21. How would you differentiate nominal and true stress?
22. Distinguish between plane and circular polariscope?
23. How polariscope is classified?
24. What is a strain gauge? Mention different types of strain gauge.

Blooms Level 3-Applying

1. What examples can you give with respect to single and multi degrees of freedom?
2. How would you arrive at mathematical expression of spring mass damper system?

3. What approach would you like to use to arrive at mathematical expression of two spring mass system?
4. How would you apply what you learned to develop a comfort seating system in an automobile vehicle (car)?
5. Where you can apply torsional vibrations practically?
6. Where you can apply the concept of whirling of shaft in practical usage?
7. Mention few practicality of applying torque in daily usage.
8. How to apply force polygon in static balancing?
9. How you would like to classify bearings? Give examples for each.
10. List the applications of journal bearing.
11. What are the applications of photoelasticity?
12. What is the use of quarter wave plate?
13. Is a stress is a first order tensor? Justify the statement.
14. List the reasons to cause the stress concentration factor.
15. What is the use of strain rosette?
16. Mention the methods to minimize the stress concentration.

Blooms Level 4-Analysing

1. How would you categorize dampers?
2. Can you distinguish natural frequency and damped frequency?
3. What is the function of Porter governor?
4. Can you distinguish between centrifugal force and controlling force?
5. What is the advantage of aligning the shaft?
6. What is the importance of modes which are generated by whirling of shaft?
7. What is the significance of couple polygon?
8. Is pressure and stress are same? Justify with examples.
9. What is relationship between Young's Modulus, Modulus of Rigidity and Poisson's ratio?
10. What is the relationship between Young's Modulus, Modulus of Rigidity and Bulk Modulus?
11. How to analyse stressed model using photoelasticity?
12. With suitable examples analyse zero order, first order and second order tensor.

Blooms Level 5-Evaluating

1. Based on what you know, how would you explain the derivation of mathematical expression of single degree of freedom?
2. Would it be better if damping is provided in all automobile vehicles? Justify.
3. Does a mode (mode shape) play a significance role in determination of natural frequency? Justify the statement.
4. Based on what you know, how would you derive co-efficient of friction for journal loading?
5. Based on what you know, explain strain energy stored in spring with mathematical expression?
6. Would it better to evaluate photoelastic models using dark or bright fringes using circular polariscope? Explain.

Blooms Level 6-Creating

1. What way would you design the best spring mass system to minimize vibrations in an automobile?
2. Can you propose an alternative method for controlling the speed of a machine?
3. Is it possible to propose an alternative method for producing the modes? Justify.
4. Stress analysis of bolts can be proposed using photoelasticity. Justify the statement.