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Department of Mechanical Engineering

Laboratory Manual

DESIGN LABORATORY

Design Laboratory

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Experiment - 1

LONGITUDINAL VIBRATION OF HELICAL SPRING

AIM:

To study of longitudinal vibration of helical spring and to determine the frequency and time period of oscillation theoretically and experimentally.

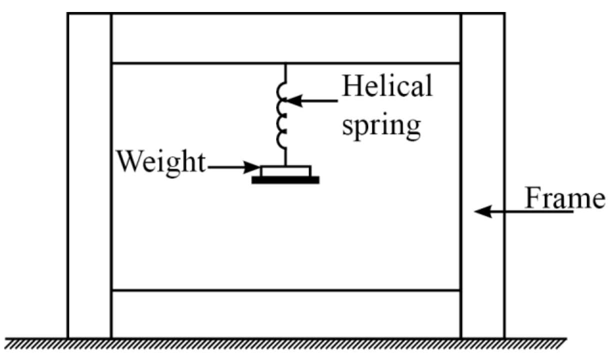
APPARATUS:

Helical spring, weights, weight pan, stopwatch, seals.

DESCRIPTION:

One end of open coil spring is fixed to the nut having a hoe which itself is mounted on a M.S strip fixed on one side of the main frame. The lower end of the spring is attached to the platform carrying the weight. The stiffness of the spring can be found out by varying the weight on the platform and by measuring the deflection of the spring. The time period of vibration can be calculated by measuring the number of oscillation & the time taken by them.

Experimental setup:



PROCEDURE:

1. For one end of helical spring to upper screw and this other end to the pan.
2. Determine free length.
3. Put some weight to pan and note down the deflection.
4. Stretch the spring through some distance and release
5. Count the time period in seconds for say 5, 10, 20 oscillations.
6. Determine the actual period
7. Repeat the procedure for different weights.

FORMULAE:

1. Stiffness of spring:

$$K_{exp} = \frac{W}{\delta} \quad (N/m) \qquad W = \text{weight add (N)}$$

δ = Deflection in spring (m)

2. Experimental period of vibration or Oscillation

$$T_{exp} = \frac{t}{n} \quad (sec) \qquad t = \text{time for 'n' oscillations (sec)}$$

3. Theoretical period of vibration

$$T_{theo} = 2\pi\sqrt{\frac{W}{k \times g}} \quad (sec)$$

W = weight added (N)

k = stiffness (N/m)

4. Experimental Frequency of Vibration

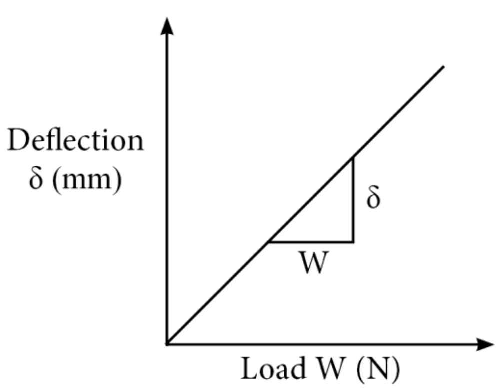
$$F_{exp} = \frac{1}{T_{exp}} \quad (Hz)$$

5. Theoretical Frequency of vibration

$$F_{theo} = \frac{1}{T_{theo}} \quad (Hz)$$

MODEL GRAPH:

Deflection Vs Load Added (To determine the stiffness of the Spring).



Slope :

$$\text{Stiffness } K = \frac{W}{\delta} (N/m)$$

TABULAR COLUMN:

Sl.No	Load added W		Length of spring before loading L_i (m)	Length of spring after loading L_f (m)	Deflection of spring $\delta = L_f - L_i$ (m)	Stiffness K_{exp} (N/m)
	Kg	N				
1						
2						
3						
4						

Sl.No	Number of oscillations 'n'	Time taken for 'n' oscillations t (sec)	Period of oscillation / vibration (sec)		Frequency of vibration / oscillation (Hz)	
			T_{exp}	T_{theo}	F_{exp}	F_{theo}
1						
2						
3						
4						

RESULT:

Time period & frequency of un damped free vibration (longitudinal vibration) of spring mass system were determined. The experimental and theoretical values were tabulated & compared.

Experiment - 2

TROSIONAL VIBRATION OF SINGLE ROTOR SYSTEM (UNDAMPED)

AIM:-

To determine the natural frequency of torsional vibration of single rotor system and compare it with the theoretical value.

APPARATUS USED:-

Shaft, Chuck key, Spanner, Measuring tape & Stopwatch.

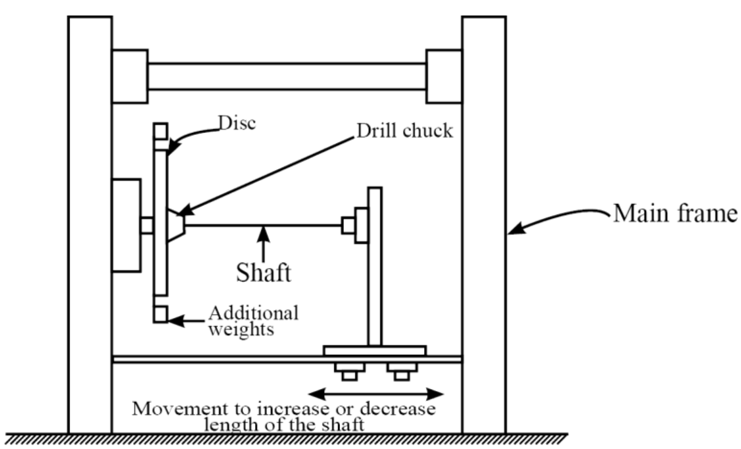
DESCRIPTION:-

In this experiment, one end of the shaft is gripped in the chuck. A heavy flywheel or disc free to rotate in ball bearing is fixed at the other end of shaft. The bracket with fixed end of the shaft can be moved to any convenient position along lower beam. The bearing housing is fixed to side member of main frame.

PROCEDURE:

1. Fix the bracket at a convenient position along the lower arm
2. Grip one end of the shaft at the bracket by chuck
3. Fix the rotor on the other end of shaft.
4. Twist the rotor through same angle and release.
5. Note down the time required for 5 or 10 or 20 oscillations
6. Repeat the procedure for different lengths of shaft.

EXPERIMENTAL SETUP:-



FORMULAE:

1. Periodic time or Experimental period of vibration : T_{exp} / T_{act}

$$T_{act} = \frac{t}{n} \qquad (sec)$$

where, t is time taken for ‘n’ oscillations

2. Theoretical period of vibration : T_{theo}

$$T_{theo} = 2p\sqrt{\frac{I}{K_t}}\,(sec)$$

where I = Moment of Inertia of disc (Nms^2)

K_t =Torsional stiffness (Nm)

3. Torsional stiffness: K_t

$$K_t = \frac{GJ}{L} \qquad N\text{-}m / rad$$

where G = modulus of rigidity (N/m^2)

L = length of shaft (m)

4. Polar moment of Inertia: J

$$J = \frac{pd^4}{32} \qquad \left(m^4\right)$$

where d = diameter of shaft (m)

5. Moment of Inertia of Disc: I

$$I = \frac{WD^2}{8g} + \frac{2WR_1^2}{8g} \quad N\text{-}\ m/\sec^2$$

where R = Radius of rotation of weights on the arm (m)

6. Experimental frequency of vibration

$$F_{exp} = \frac{1}{T_{exp}} \text{ or } \frac{1}{T_{act}} \quad (Hz)$$

7. Theoretical frequency of vibration

$$F_{theo} = \frac{1}{T_{theo}} \quad (Hz)$$

OBEERVITION:

1. Modulus of Rigidity : G = 80 GPa
2. Shaft Diameter : d = 3mm
3. Diameter of disc : D = 200mm = 0.2m
4. Man of disc : M = 2.5 Kg
5. Weight of disc : W = 24.525 N
6. Weight of additional Mass : $W_1 = 0.385 \times 9.81 = 3.77685$ N

TABULAR COLUMN:

Observation –

Sl.No	Length of shaft 'L' (m)	No. of oscillations (n)	Time for n oscillations 't' (sec)	Periodic time $T_{act} = t/n$
1				
2				
3				

Calculation –

Sl.No	Number of oscillations 'n'	Time taken for 'n' oscillations t (sec)	Period of oscillation / vibration (sec)		Frequency of vibration / oscillation (Hz)	
			T_{exp}	T_{theo}	F_{exp}	F_{theo}
1						
2						
3						

RESULT:

The frequency of torsional vibration of the single rotor system is determined and the experimental and theoretical values are tabulated and compared.

Experiment - 3

TORSIONAL VIBRATION OF TWO ROTOR SYSTEM (UNDAMPED)

AIM:-

To determine the period of frequency of torsional vibration of a two rotor system experimentally and compare it with theoretical values.

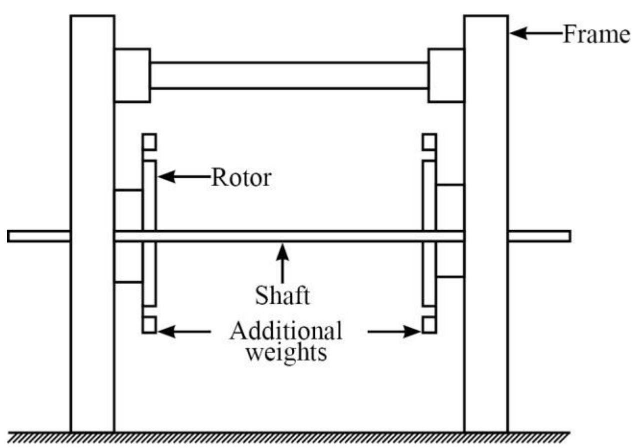
APPARATUS:-

Shaft, chuck key, spanner, measuring tape, stopwatch

DESCRIPTION:-

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

EXPERIMENTAL SETUP:-



PROCEDURE:

1. Fix disc A and disc B to the shaft and fit the shaft in the bearings
2. Deflect the discs A & B in opposite direction by hand release.
3. Note down the time required for ‘n’ oscillations 5, 10, 15, 20
4. Fit the cross arm to the disc A and attach equal masses to ends of cross arm.
5. Repeat the above procedure with different equal masses attached to ends of cross arm, and also by varying the length of this shaft

FORMULAE:

1. Periodic time or experimental period of vibration:

$$T_{act} = \frac{t}{n} \quad (sec)$$

where t = time for ‘n’ oscillations

n = no. of oscillations

2. Torsional stiffness:

$$K_t = \frac{GJ}{L} \quad N\text{-} \; m/rad$$

where G = Modules of Rigidity

J = Polar moment of inertia

$$J = \frac{pd^4}{32} \left(m^4\right)$$

3. Moment of inertia of Disc A:

$$I_A = \frac{W_A D_A^2}{8g} + \frac{2W_1 R^2}{8g} \quad N\text{-} \; m/sec^2$$

W_A = weight of disc A (N)

D_A = Diameter of disc A (m)

W_1 = Weight added (N)

4. Moment of inertia of Disc B:

$$I_B = \frac{W_B D_B^2}{8g} + \frac{2W_1 R^2}{8g} \quad N\text{-} \; m/sec^2$$

W_1 = Weight attached to cross arm (N)

R = Radius of rotation of weight.

5. Theoretical period of vibration:

$$T_{theo} = 2p\sqrt{\frac{I_A \,´\, I_B}{K_t\left(I_A + I_B\right)}} \quad (sec)$$

6. Theoretical Frequency of vibration

$$F_{theo} = \frac{1}{T_{theo}} \quad (Hz)$$

7. Experimental Frequency of vibration

$$F_{exp} = \frac{1}{T_{exp}} \quad (Hz)$$

NOTE: If no weights are added to the cross bar, $W_1 = 0$

$$I_B = \frac{W_B.D_B^2}{8g}$$

OBEERVITION:

1. Modular of Rigidity for shaft Material : $G = 80$ GPa
2. Shaft Diameter : $d = 3 \times 10^{-3}$ m
3. Diameter of disc A : $(D_A) = 0.25$m
4. Diameter of disc B : $(D_B) = 0.20$m
5. Weight of disc A : $W_A = 3.85 \times 9.81 = 37.768$ N
6. Weight of disc B : $W_B = 2.5 \times 9.81 = 24.525$ N
7. Weight of additional mass added : $W_1 = 0.385 \times 9.81 = 3.77685$ N

TABULAR COLUMN:

Sl.No	No of oscillations 'n'	Time for n oscillations 't' (sec)	Weight attached (N)	Radius of Rotation (m)	Moment of Inertial of Disc A (Nm/s^2)	Moment of Inertial of Disc B (Nm/s^2)	Periodic time t_{act} (sec)
1							
2							
3							

Sl.No	Moment of inertial of Disc A (Nm/s^2)	Moment of inertial of Disc B (Nm/s^2)	Period of oscillation (sec)		Frequency of vibration (Hz)	
			T_{exp}	T_{theo}	F_{exp}	F_{theo}
1						
2						
3						

RESULT:

The frequency of torsional vibration of the two rotor system is determined and the experimental and theoretical values are compared.

Experiment - 4

TORSIONAL VIBRATION (DAMPED)

AIM:-

To study the damped torsional vibrations of a single rotor system and to determine the damping factors and damping coefficient

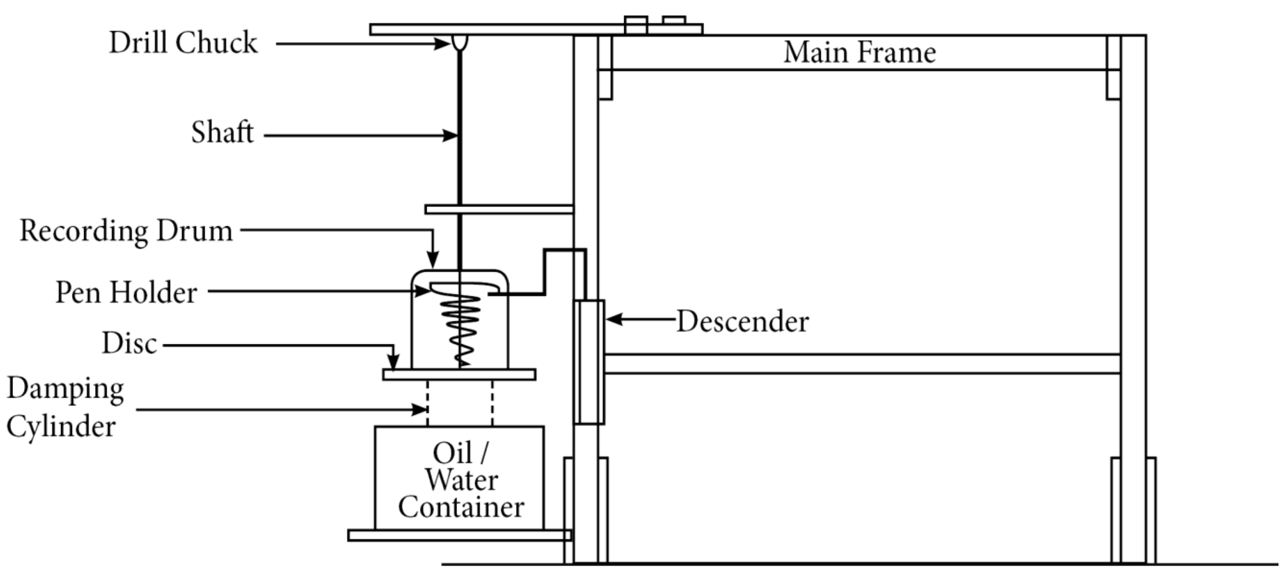
APPARATUS:-

Stand, shaft, rotor, measuring scale & stop watch

DESCRIPTION:-

The setup consists of a long elastic shaft gripped at the upper end by a chuck in the bracket. The bracket is clamped to the upper beam of the main frame. A heavy steel flywheel damped at the lower end of shaft is suspended from the bracket. A damping drum is fixed to the lower face of the flywheel. This drum is immersed in oil which provides damping. The rotor can be taken up and down for varying the depth of immersion of damping drum. Recording drum is mounted on the upper face of the flywheel. Paper is wrapped around the recording drum. Oscillations are sketched out on the paper, with the help of a specially designed piston of dashpot. The piston carries an attachment for fixing a pen in the holder.

EXPERIMENTAL SETUP



PROCEDURE:

1. Put the thin lubricating oil (20 - 30 grade) in the drum and note the depth of immersion.
2. Put the sketch pen in the bracket.
3. Allow the flywheel or disc to vibrate
4. Allow the pen to descend. Ensure that the pen always makes contact with the paper and record oscillations.
5. Determine the amplitude at any position and amplitude after (r) cycle.
6. After completion of experiment drain the oil

FORMULAE:

1. Torsional Stiffness [K_t]

$$K_t = \frac{GJ}{L} \; (N\text{-}m/rad)$$

where G = Modulus of Rigidity

J = Polar moment of inertia

2. Polar moment of Inertia of shaft

$$J = \frac{pd^4}{32} \; (m^4)$$

where d = diameter of shaft

3. Actual Time period

$$T_{act} = \frac{t}{n} \; (sec)$$

where t = time taken for 'n' oscillations

n = number of oscillations

4. Theoretical time period:

$$T_{theo} = 2p\sqrt{\frac{L}{K_t}}$$

where L = Length of shaft (m)

K_t = Torsional stiffness (N-m/rad)

5. Moment of Inertial of Disc

$$I = \frac{WD^2}{8g} \; (N\text{-}m/sec^2)$$

where D = Dia of disc (m)

W = Weight of disc (N)

6. Critical Damping factor

$$C_{tc} = 2\frac{W}{g}\sqrt{\frac{K_t}{I}}$$

7. Logarithmic Decrement

$$d = \frac{1}{r} \log_e \left| \frac{x_n}{x_{n+r}} \right|$$

8. Damping Ratio

$$\frac{C_t}{C_{tc}} = \frac{d}{\sqrt{4p^2 + d^2}}$$

OBEERVITION:

1. Length of shaft L = 0.87m
2. Diameter of shaft d = 0.0055m
3. Mass of disc m = 7.5kg
4. Weight of disc W = 73.575 N
5. Diameter of disc D = 0.35m
6. Modulus of Rigidity for shaft material G = 80Gpa

TABULAR COLUMN:

Sl.No	Length of suspension of shaft 'L' (m)	Amplitude of vibration in the graph 'X_n' (m)	Amplitude of vibration after 'r' cycles 'X_{n+r}' (m)	No. of cycles (r)	No. of oscillations (n)	Time for 'n' oscillation 't'
1						
2						

Sl.No	Torsional Rigidity K_t (N-m/rad)	Period of Oscillation		Critical damping factor 'c_{tc}'	Logarithmic Decrement 'δ'	Damping coefficient c_t / c_{tc}
		T_{act}	T_{theo}			
1						
2						

RESULT:

The damped torsional vibrations of a single rotor system were studied and the damping factor & the damping coefficient were determined.

Experiment - 5

CRITICAL OR WHIRLING OR WHIPPING SPEED OF SHAFT

AIM:-

To determine the whirling speed for various shaft sizes experimentally and compare the same with theoretical values.

APPARATUS:-

Shafts, digital tachometer, AC voltage regulator, chuck key.

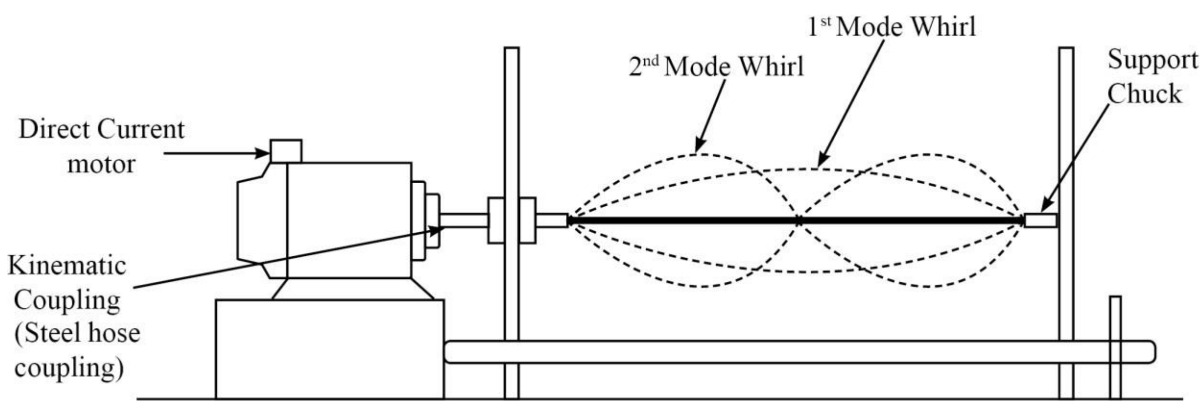
THEORY:-

The speed at which the rotating shaft vibrates violently in the transverse direction is called critical whirling speed. Critical speed of a shaft because due to self weight of the shaft or due to a disc mounted upon the shaft rotating about it. When a rotor is used the center of gravity of the disc must lie on the shaft axis.

Critical speed of shaft depends on the following factors:

1. Length of the shaft
2. Diameter of shaft
3. Bearing support i.e., fixed or free
4. Self weight of the shaft or the location of load carried by shaft when an extra disc is attached.

EXPERIMENTAL SETUP:



PROCEDURE:

1. Measure the dimensions of the shaft or specimen using calipers
2. Mount the shaft on to the machine by inserting it through the central hole of the retainer.

3. Switch on the speed control unit and the 1st natural frequency is read (1st mode of vibration). When the speed is increase further the shaft begins to vibrate violently as it nears the critical speed.
4. Once the critical speed is passed, the shaft restablishes and on further increase of the speed, the second natural frequency is reached.
5. Measure the speeds of rotation of the shaft at its first & second natural frequencies directly with the tachometer.
6. Measure speed three times and use the average value for calculation.

FORMULAE:

1. Weight of shaft : W = Density x Volume

$$\boxed{W = \rho \times \frac{\pi d^2}{4} \times L} \quad (N)$$

where d = Diameter of shaft (m)

L = Length of shaft (m)

2. Man moment of Inertia of shaft:

$$\boxed{I = \frac{\pi}{64} d^4} \quad (m^4)$$

3. Static deflection due to self weight of shaft:

$$\boxed{\delta_S = \frac{5WL^4}{385EI}}$$

where E = Young's Modulus (N/m^2)

4. Natural frequency of transverse vibration :

$$\boxed{fn = \left[\frac{EIg}{WL^4}\right]^{0.5} C} \quad (Hz)$$

where C is a constant value that depends on the end conditions of shaft.

Case	End Condition	Value of C 1st Mode	2nd Mode
1	Fixed – supported	1.572	6.3
2	Fixed – Fixed	3.75	8.82

5. Critical speed :

$Nc_{thy} = f_n$ rps

$\boxed{Nc_{thy} = f_n \times 60 \quad (rpm)}$

OBSERVATION:

1. Young's Modulus for steel E = 2.06×10^{11} N/m^2
2. Density of mild steel ρ = 7800 Kg/m^2
3. Length of shaft L = 1m

TABULAR COLUMN:

Diameter of shaft (m) d	Mass moment of Inertia of shaft I (m^4)	Weight of shaft per meter W (N/m)	Natural Frequencies	Whirling speed (rpm)	
				Nc_{exp}	Nc_{theory}
			1st Mode		
			2nd Mode		

RESULT:

The critical speed of the given shaft is obtained for various modes of vibration.

Experiment - 6

<u>CALIBRATION OF CIRCULAR DISC UNDER DIAMETRIAL COMPRESSION USING A CIRCULAR POLARISCOPE</u>

AIM:-

To determine the material fringe constant and model fringe constant using photo elastic material and under diametrical compression.

APPARATUS:-

Photo elastic apparatus with polarizer, analyzer and photo elastic specimen (circular disc).

THEORY:-

A circular disc under diametrical compression is frequently used as a calibration specimen. A circular disc can be easily machined & the loading is also simpler. If D is the diameter of disc and h is the thickness as show in figure. When subjected to a diametrical compression load ‘P’. Stresses along the horizontal diameter are given by:

$$\sigma_x = \sigma_1 = \frac{2P}{\pi D h}\left[\frac{D^2 - 4X^2}{D^2 + 4X^2}\right]^2$$

$$\sigma_y = \sigma_2 = \frac{2P}{\pi D h}\left[\frac{4d^4}{\left(D^2 + 4X^2\right)^2} - 1\right]$$

$$\tau_{xy} = 0$$

Then

$$\sigma_1 - \sigma_2 = \frac{8P}{\pi D h}\left[\frac{\left(1 - 4(X/D)^2\right)}{\left[1 + 4(X/D)^2\right]^2}\right]$$

$$\sigma_1 - \sigma_2 = \frac{8P}{\pi h W^2}\cdot\left(\frac{X}{D}\right)$$

At the centre:

$$X = 0$$

$$\sigma_1 = \frac{2P}{\pi D h}$$

And $\sigma_2 = \dfrac{-6P}{\pi D h}$

Hence;

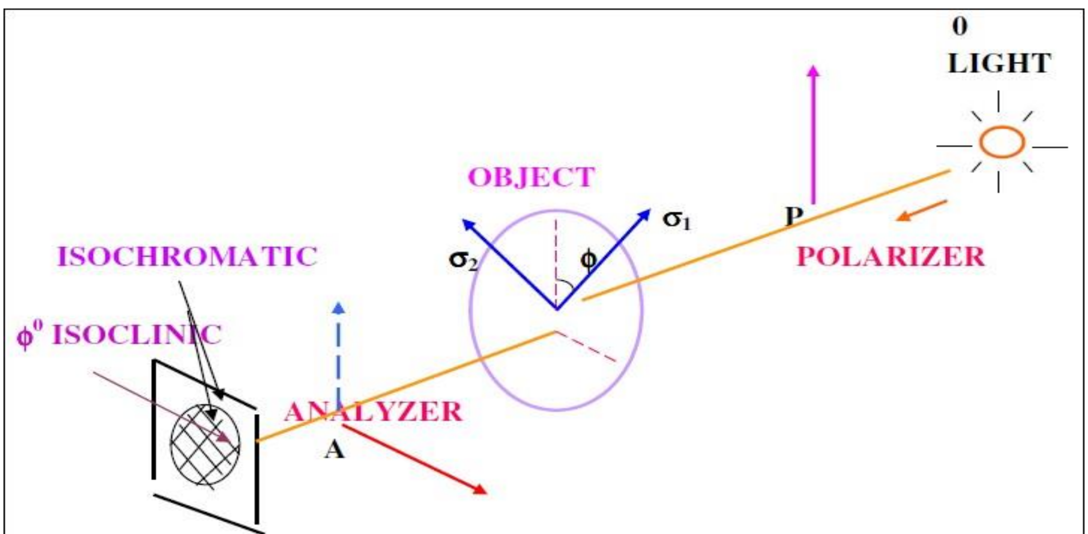
$$\sigma_1 = \sigma_2 = \frac{8P}{\pi D h} = \frac{N F_\sigma}{h}$$

$$\therefore \boxed{F_\sigma = \frac{8}{\pi D} \times \frac{P}{N}}$$

By knowing the value of ‘P’ for a particular fringe order N at the centre of disc, a graph can be plotted whose slope is substituted in the above equation to determine the value of F_σ.

PLANE POLARISCOPE:

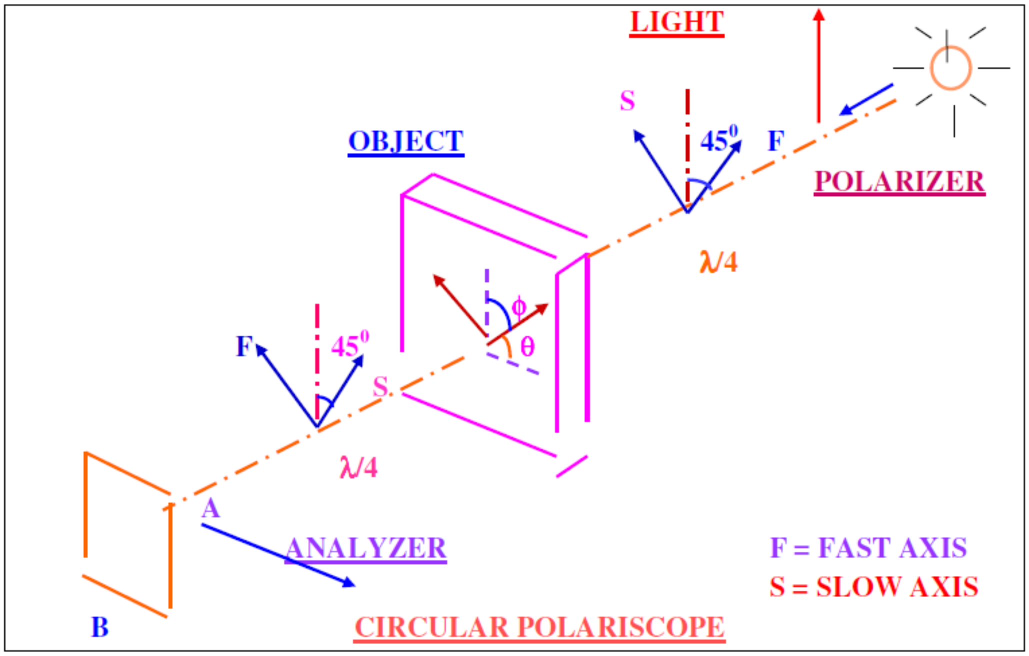
In plane polariscope, the light source may be mercury or a sodium vapour lamp, an incandescent filament lamp or a bank of bulbs. Mercury or sodium vapour lamps are used as monochromatic light sources and incandescent filament lamp is used as a white light source for the lens type polariscope. For the diffused light polariscope a bank of bulbs is used in an opaque box with a ground glass on one side to give diffused light. In plane polariscope, two types of set up are possible, first bright field, when polarizer and analyzer are parallel and then dark field when polarizer and analyzer are crossed.



PLANE POLARISCOPE

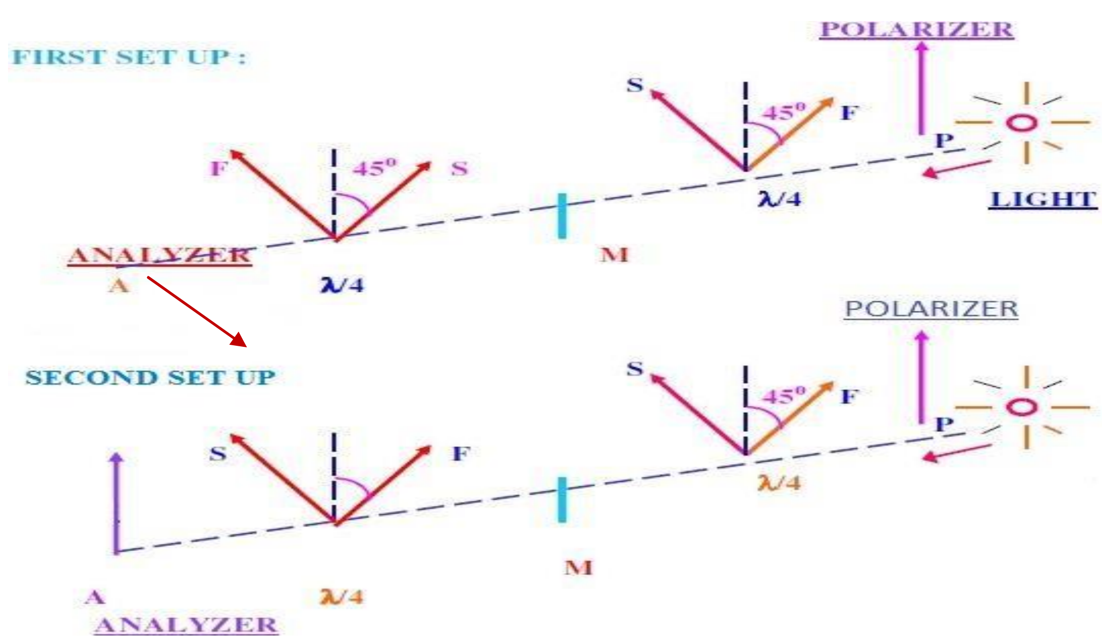
CIRCULAR POLARISCOPE:

In addition to all the elements of a plane polariscope, the circular polariscope has two more quarter wave plates, the first between the polarizer and model and second between the model and the analyzer. The fast and slow axes of the quarter wave plate are inclined at 45^o with the polarizer or the analyzer. The quarter wave plates are made of Polaroid film and produces a path difference of λ/4 or a phase difference of 90^0 in the two light vectors passing through them.



The crossed – crossed set up is called the standard set up of the circular polariscope. The first quarter wave plate converts plane-polarized light into circularly polarized light and the second quarter wave plate converts circularly polarized light into plane-polarized light.

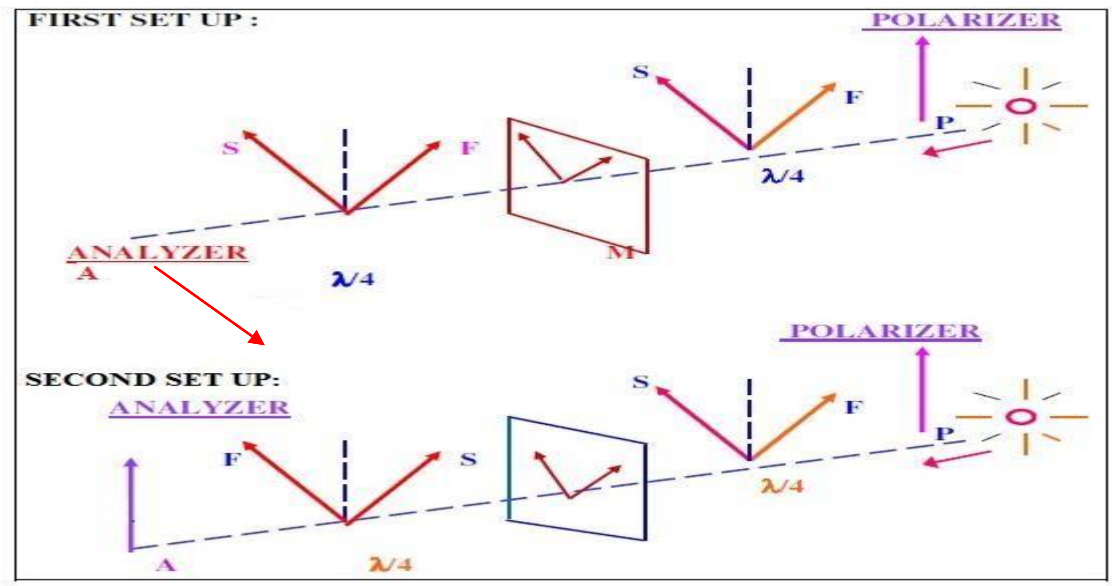
DARK FIELD ARRENGEMENT IN CIRCULAR POLARISCOPE:



LIGHT FIELD ARRENGEMENT IN CIRCULAR POLARISCOPE|

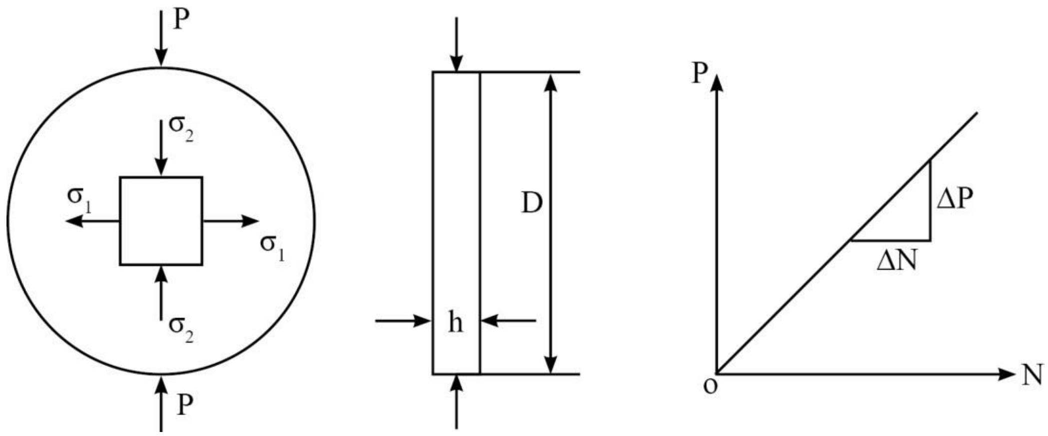
There are two possible arrangements in a circular polariscope to obtained light or bright field background -

- In first type, the two-quarter wave plates act as a half wave plate, and incident linearly polarized light get rotated by 90° and allowed through the analyzer.
- In second type, the two-quarter wave plates cancel each other and as the polarizer is parallel to the analyzer, the light is admitted through.



EXPERIMENTAL SET UP

MODEL USED:



PROCEDURE:

1. Ensure that the polarizer and analyzer at 0^o (i.e., 0 in scale is coinciding with marker line)
2. First quarter wave plate is at 45^o (ACW)
3. Second quarter wave plate is at 45^o CW or 315^o
4. This arrangement is called circular polar scope with dark field.
5. Now select the monochromatic light source for circular disc.
6. Now place the model between the supports.
7. Ensure that load indicator is at zero position. If not make it zero using balance potentiometer.
8. Apply the load by using hydraulic pump on the specimen by observing load value.
9. Now fringes will appear. To count the fringes in between two a odd numbers Tardy's Method is used.
10. Identify the symmetric fringe orders on both sides of the point of contact.
11. The load is adjusted to get full fringe order, the loading is continued till 5^{th} fringe order is obtained.

FORMULAE:

1. From the graph (Load V/s Fringe Number 'N')

$$\frac{DP}{DN} \text{ or } \frac{P}{N} = s \qquad \text{in N/fringe}$$

2. Material Fringe Constant or Material fringe value.

$$F_s = \frac{P}{N} \times \frac{8}{\pi D} \ (N/m/\text{fringe})$$

P = Load on Disc(N)

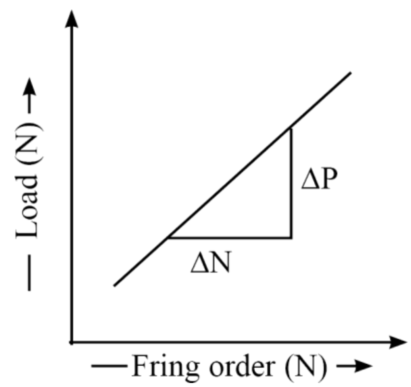
N = Fringe order

D = Diameter of disc

3. Model fringe constant

$$F_s = \frac{F_s}{h} \qquad N/m^2/\text{fringe}$$

MODEL GRAPH:-



Slope $\frac{\Delta P}{\Delta N}$ N/fringe

OBSERVATION:

Diameter of disc D = 7cm = 0.07m

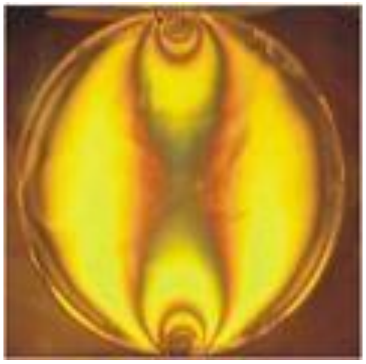
Thickness of circular disc h = 1 cm = 0.01m

Initial load =_______(Kg)

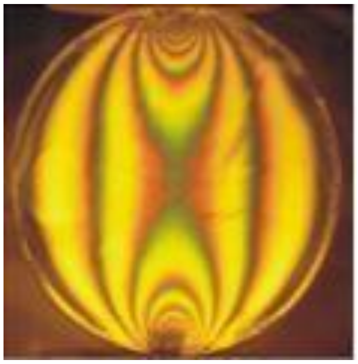
TABULAR COLUMN:

Sl. No	Load on specimen FL – IL (Kgf)	Load on specimen (N)	Fringe order N	Material Fringe constant Fσ(N/m/fringe)	Model fringe constant Fσ(N/m^2/fringe)
1			1		
2			2		
3			3		
4			4		
5			5		

FRINGE PATTERN FOR CIRCULAR DISC IN COMPRESSIVE LOAD



FRINGE ORDER – 1

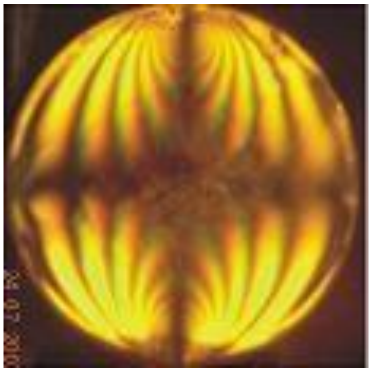


FRINGE ORDER - 2

FRINGE ORDER – 3

FRINGE ORDER - 4

FRINGE ORDER - 5



ISOCLINC & ISOCHROMATIC

RESULT:

The material fringe constant and model fringe constant for the given photo elastic material are obtained and tabulated in the tabular column.

Experiment - 7

CALIBRATION OF A FLAT RECTANGULAR BEAM UNDER PURE BENDING (4 POINT LOADING) USING A CIRCULAR POLARISCOPE

AIM:-

To determine the material fringe constant and model fringe constant using photo elastic material and also to calculate bending stress.

APPARATUS:-

Photo elasticity apparatus with polarizer, analyzer and photo elastic specimen (Beam).

DESCRIPTION:-

The rectangular beam of thickness ‘h’ and depth ‘w’ as show in figure is subjected to four point loading, i.e., pure bending. Pure bending in the beam may be produced by applying equal loads P at a distance ‘a’ from the ends of the beam of length ‘L’ as shown.

The uniform bending moment M in the beam is:

$$M = P\times a$$

The stress is the beam are

$$\sigma_1 = \frac{M}{I}Y = \frac{Pa}{\left(\frac{hw^3}{12}\right)} \times \frac{w}{2}$$

$$\sigma_1 = \frac{6Pa}{hw^2}$$

$$\sigma_2 = 0$$

$$\therefore \sigma_1 = \frac{6Pa}{hw^2} \quad \text{and} \quad \sigma_2 = 0$$

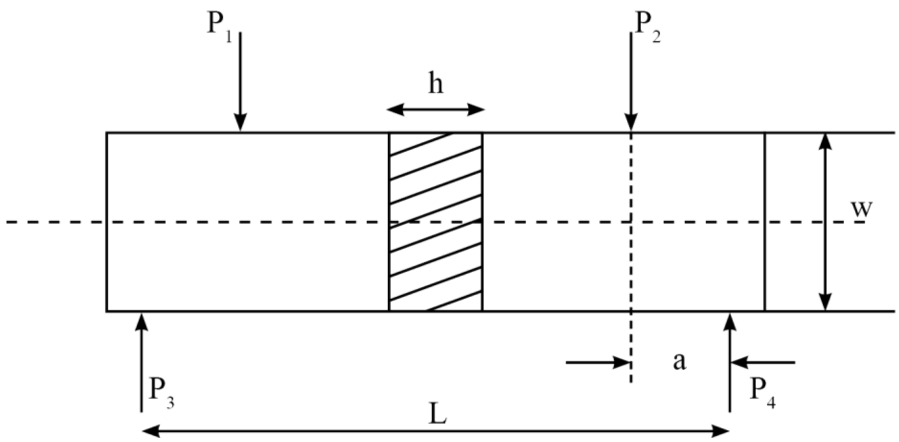
Applying stress optic law; we get

$$\sigma_1 - \sigma_2 = \frac{NF\sigma}{h}$$

$$\frac{6Pa}{hw^2} = \frac{NF\sigma}{h}$$

$$F\sigma = \frac{P}{N} \times \frac{6a}{w^2}$$

MODEL USED:-



PROCEDURE:

1. Ensure that the polarizer & analyzer are at 0^o (i.e., scale is coinciding with marked line)
2. Place the model between the shaded (4 point bending setup).
3. Apply a light load and watch for dark fringes (isochromatic).
4. Place the quarter wave plates.
5. First quarter wave plate is rotated at 45^o anticlockwise direction.
6. Second quarter wave plate is rotated at 45^0 clockwise direction.
7. After rotation of the quarter wave plate, the dark fringes are removed.
8. Ensure that the load indicator is at zero position. If not so, make it zero by using balance potentiometer.
9. The load is adjusted for full fringe order.
10. Record the load for different fringe orders / numbers.
11. Plot the graph, between load 'P' vs Fringe numbers N.

FORMULAE:

1. From the graph (Load 'P' vs Fringe Number 'N')

$$\frac{DP}{DN} = \frac{P}{N} = s$$

2. Material Fringe Constant

$$Fs = \frac{6\,Pa}{N\,w^2} \quad N/m/fringe$$

3. Model fringe constant

$$Fs = \frac{Fs}{h} \qquad N/m^2 / \text{fringe}$$

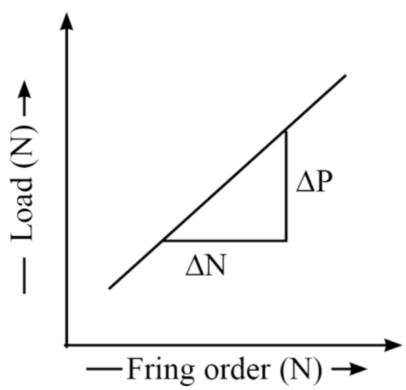
4. Actual Stress

$$s_{act} = \frac{NFs}{h} \qquad N/m^2$$

5. Theoretical stress

$$s_{theo} = \frac{6Pa}{bh^2} \qquad N/m^2$$

MODEL GRAPH



Slope $\frac{\Delta P}{\Delta N}$ N / fringe

OBSERVATION:

Thickness of beam : 1 cm = 0.01m

Width of beam : 4 cm = 0.04m

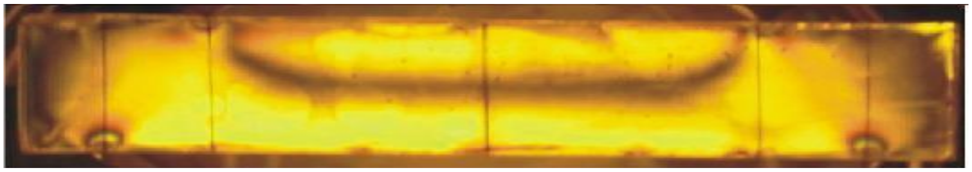
Distance between P_2 & P_4 : 2 cm = 0.02m

Initial Load :________(Kg)

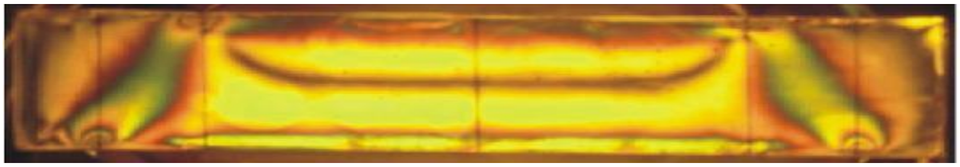
TABULAR COLUMN:

Sl. No	Load on specimen		Fringe order number N	Material Fringe constant Fσ(N/m/fringe)	Model fringe constant Fσ(N/m^2/fringe)	Actual stress σ_{act} (N/m^2)	Theoritical stress σ_{theo} (N/m^2)
	FL IL (Kgf)	(N)					
1							
2							
3							

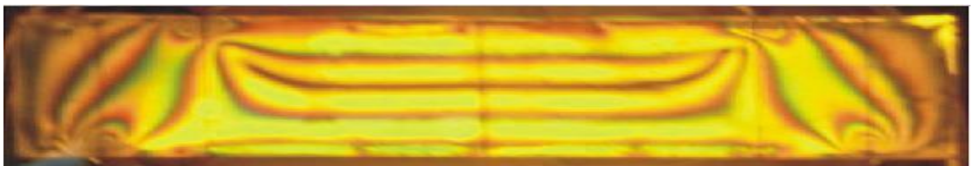
FRINGE PATTERN FOR FLAT WITH FOUR POINT LOADING



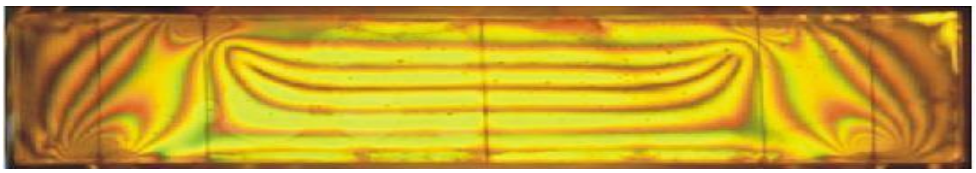
ZERO FRINGE ORDER



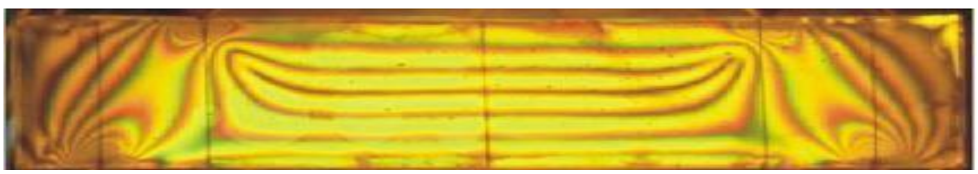
FIRST FRINGE ORDER



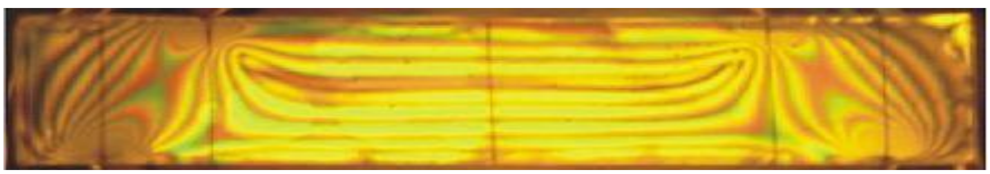
SECOND FRINGE ORDER



THIRD FRINGE ORDER



FOURTH FRING ORDER



FIFTH FRINGE ORDER

RESULT:

The material fringe constant and the model fringe constant were determined. The actual and theoretical bending stresses were determined & compared.

Experiment - 8

STRESS CONCENTRATION FACTOR OF A CIRCULAR RING UNDER COMPRESSIVE LOAD

AIM:-

To determine stress concentration factor for a circular ring subjected to compressive load.

APPARATUS:-

Photo elastic apparatus with polarizer, analyzer and photo elastic specimen,

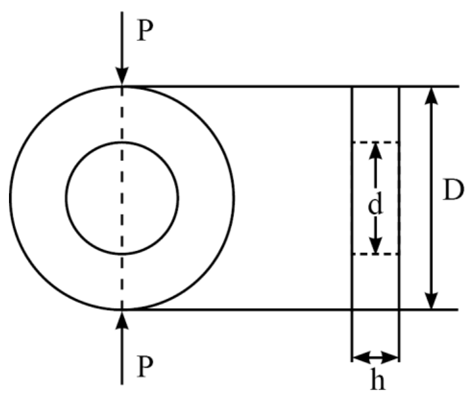
THEORY:-

Same theory to be followed as in circular disc

PROCEDURE:

Same procedure to be followed as in circular disc

MODEL USED:



D = Diameter of specimen
d = Diameter of the hole
h = Thickness of the specimen

FORMULAE:

1. Graphically obtain

$$\frac{DP}{DN} \text{ or } \frac{P}{N}$$

2. Material Fringe Constant

$$Fs = \frac{8}{\pi(D - d)} \times \frac{P}{N} \qquad (N/m/fringe)$$

3. Model fringe constant

$$Fs = \frac{Fs}{h} \qquad N/m^2\ /\ fringe$$

4. Nominal stress :

$$s_{nom} = \frac{P}{(D- \ d)h} \qquad N/m^2$$

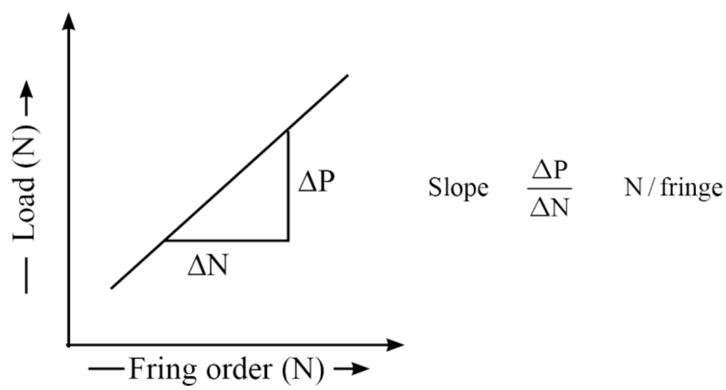
5. Maximum stress Induced:

$$s_{max} = \frac{NFs}{h} \qquad N/m^2$$

6. Stress concentration factor:

$$Ks = \frac{s_{max}}{s_{nom}}$$

MODEL GRAPH



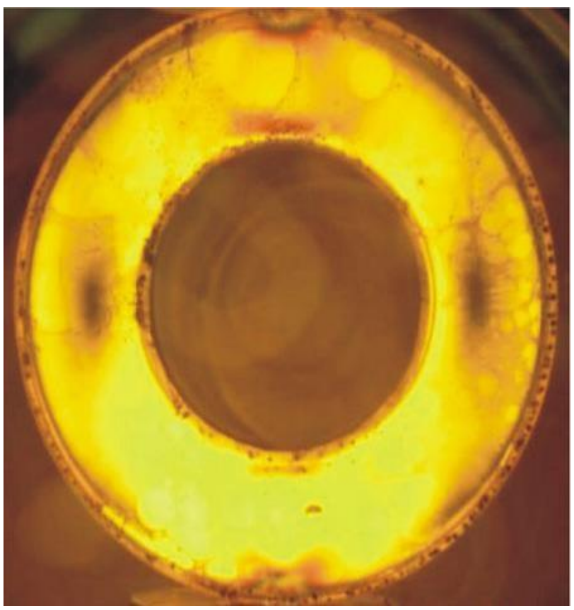
OBSERVATION:

1. Diameter of specimen = D = 100mm
2. Diameter of the hole = d = 50mm
3. Thickness of specimen = h = 8mm
4. Initial Load =_____(Kg)

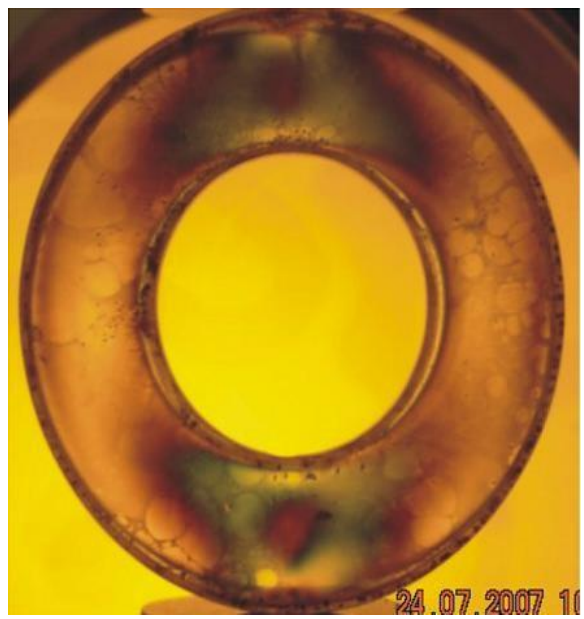
TABULAR COLUMN:

Sl. No	Load on specimen		Fringe order	Material Fringe constant $F\sigma$ N/m/fringe	Model Fringe constant $F\sigma$ N/m^2/fringe	Nominal stress σ_{nom} N/m^2	Maximum , induced stress σ_{max} N/m^2	Stress concentration factor $K\sigma$
	FL - IL Kgf	N	N					
1								
2								
3								
4								
5								

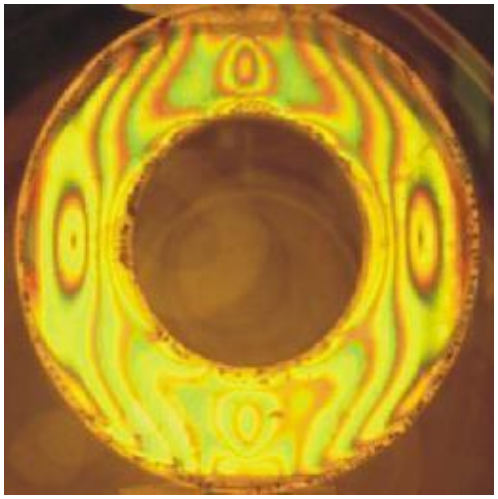
FRINGE PATTERN FOR CIRCULAR RING IN LIGHT AND DARK FIELD



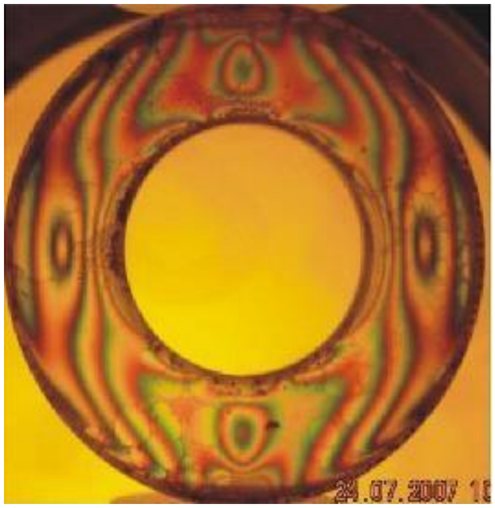
0 FRINGE FOR DARK FIELD (FULL FRINGE ORDER)



1/2 FRINGE ORDER LIGHT FIELD (HALF FRINGE ORDER)



FOR DARK FIELD (FULL FRINGE ORDER)



FOR LIGHT FIELD (HALF FRINGE ORDER)

RESULT:

The material fringe constant, model fringe constant and stress concentration factor were determined & tabulated.

Experiment - 9

STRESS CONCENTRATION FACTOR FOR A FLAT PLATE WITH A CIRCULAR HOLE AT CENTRE UNDER TENSION

AIM:-

To determine stress concentration factor for a flat plate with a hole at the centre.

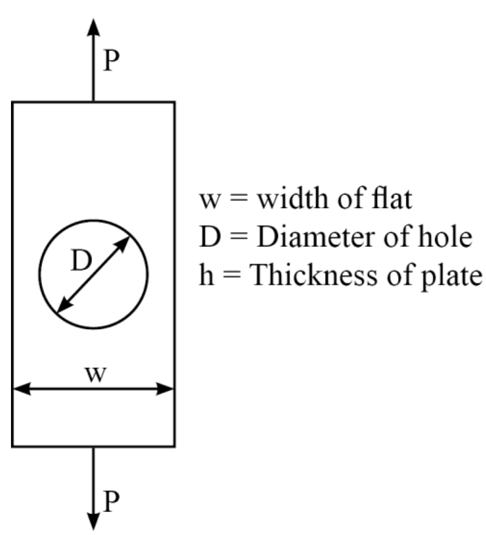
APPARATUS:-

Photo elastic apparatus with polarizer, analyzer and photo elastic specimen.

PROCEDURE:-

1. Ensure that the polarizer and analyzer are at 0^o.
2. Place the model between the shackles.
3. Apply a light load value, and watch for dark fringes.
4. Place the quarter wave plate and rotate it 45^o in anticlockwise direction.
5. The second quarter wave plate is rotated in 45^o in clockwise direction.
6. After the rotation of the quarter wave plates the dark fringes are removed.
7. Increase the load for different fringe order.
8. The load is adjusted to get a full fringe order.
9. Plot the graph between load ‘P’ Vs fringe order N.

MODEL USED:



FORMULAE:

1. Graphically obtain

$$\frac{DP}{DN} = \frac{P}{N}$$

2. Material Fringe Constant

$$Fs = \frac{P}{N} ´ \frac{1}{(W- D)} \qquad N/m/fringe$$

3. Model fringe constant

$$Fs = \frac{Fs}{h} \qquad N/m^2/fringe$$

4. Nominal stress

$$s_{nom} = \frac{P}{(W- D)h} \qquad N/m^2/fringe$$

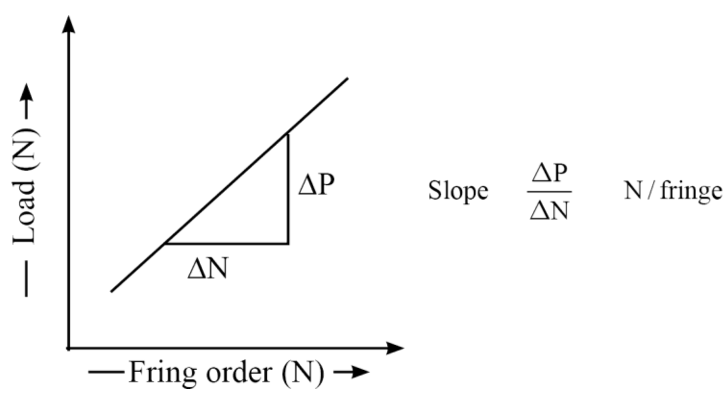
5. Maximum stress Induced / Actual stress

$$s_{act} = s_{max} = \frac{Fs ´ N}{h} \qquad N/m^2$$

6. Stress concentration factor

$$Ks = \frac{s_{max}}{s_{min}}$$

MODEL GRAPH



OBSERVATION:

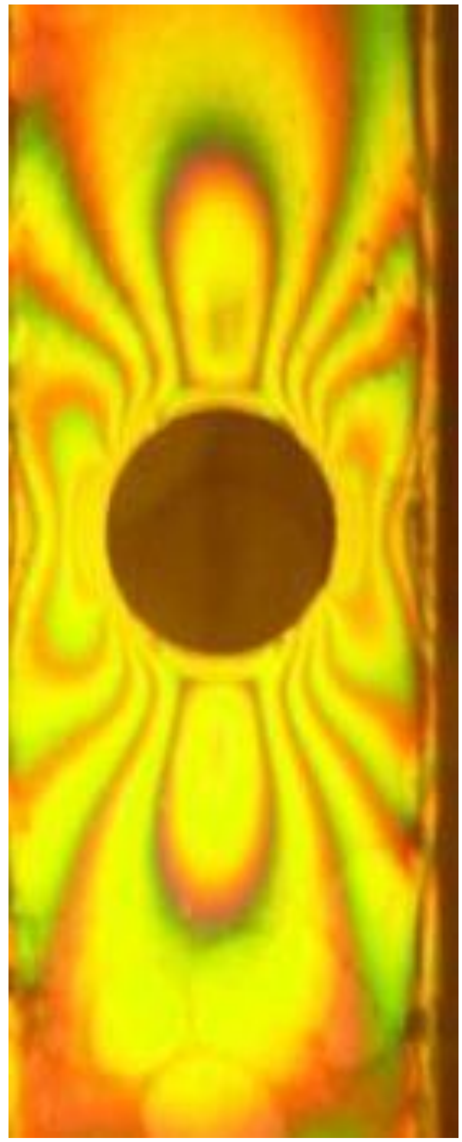
1. Width of plate = 0.04m
2. Diameter of the hole = 19.5 x 10^3 = 0.195m
3. Thickness of flat = 0.01m
4. Initial Load =_______(Kg)

TABULAR COLUMN:

Sl. No	Load on specimen		Fringe order N	Material Fringe constant N/m^2/fringe	Model Fringe constant $F\sigma$ N/m^2/fringe	Normal stress (σ_{nom}) N/m^2	Maximum induced stress (σ_{max}) N/m^2	Stress concentration factor $K\sigma$
	FL- IL Kgf	N						
1								
2								
3								
4								

FLAT PLATE WITH CIRCULAR HOLE UNDERTENSION

FIRST FRINGE ORDER



SECOND FRINGE ORDER

RESULT:

The material fringe constant, model fringe constant and stress concentration factor for the given photo elastic material are determined & tabulated.

Experiment - 10

CALIBRATION OF I – SECTION SPECIMEN IN TENSION USING CIRCULAR POLARISCOPE

AIM:-

To determine the material fringe constant and model fringe constant using I section photo elastic model under tensile load.

APPARATUS:-

Photo elastic apparatus with polarizer, analyzer and photo elastic specimen, I section (tensile specimen)

DESCRIPTION:-

If we prepare a simple tension specimen as shown in figure whose width is ‘w’ and thickness is ‘h’ under load ‘P’ : the uniform stress in the specimen is: $\sigma_1 = \frac{P}{w \times h}$ and $\sigma_2 = 0$

$$\therefore \sigma_1 - \sigma_2 = \frac{P}{w \times h}$$

Applying stress optic law, we get

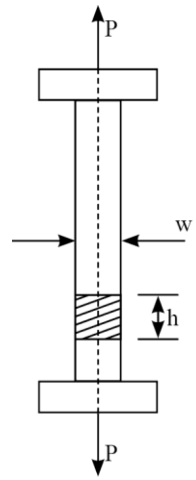
$$\sigma_1 - \sigma_2 = \frac{N\,F\sigma}{h}$$

$$\frac{P}{wh} = \frac{N\,F\sigma}{h}$$

$$F\sigma = \frac{P}{N} \times \frac{1}{w}$$

In the tensile specimen we get escaping type of fringe i.e., as load is increased from zero, successive fringes appear in the field of view and disappear as the load is increased. Generally a graph is plotted between the load applied and fringe order N. Its slope is used to determine Fσ. In this method the load ‘P’ has to be adjusted to have full fringe order in field of view.

MODEL USED:



FORMULAE:

1. From the graph (Load ‘P’ V/s Fringe Number ‘N’)

$$\frac{DP}{DN} \text{ or } \frac{P}{N} = s \qquad \text{in N / fringe}$$

2. Material Fringe Constant

$$Fs = \frac{P}{N} \times \frac{1}{W} \qquad \text{N / m / fringe}$$

N= Fringeorder

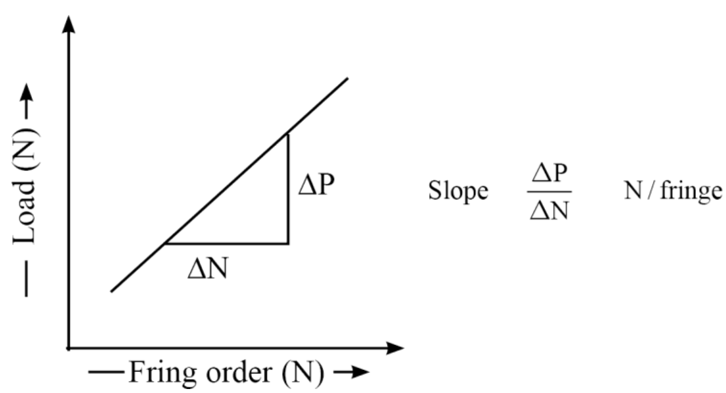
P= Load on specimen(N)

3. Model fringe constant

$$Fs = \frac{Fs}{h} \qquad \text{N / m}^2\text{ / fringe}$$

h = thicknessof specimen

MODEL GRAPH:-



OBSERVATION:

Width of section W = 1.4cm = 0.014m

Thickness of section h = 1cm = 0.01m

Initial load IL=______(Kg)

TABULAR COLUMN:

<table>
<tr><th rowspan="3">Sl.No</th><th colspan="2">Load specimen</th><th rowspan="3">Fringe order
N</th><th rowspan="3">Material Fringe Constant Fσ
(N/m/fringe)</th><th rowspan="3">Model Fringe Constant Fσ
(N/m^2/fringe)</th></tr>
<tr><th>FL - IL</th><th rowspan="2">N</th></tr>
<tr><th>Kgf</th></tr>
<tr><td>1</td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>2</td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>3</td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>4</td><td></td><td></td><td></td><td></td><td></td></tr>
</table>

RESULT:

The material fringe constant and the model fringe constant for the given photo elastic material is obtained and tabulated.

Experiment - 11

BALANCING OF ROTATING MASSES

AIM:

To find static the position and magnitude of balancing masses in a given system of rotating masses.

APPARATUS:

It consists of a frame, which is hung by chains from the main frame. A shaft rotates within bearings in the frame. Four eccentric weights are supplied which can be easily fitted over the shaft.

THEORY:

Balancing of masses is an important aspect of Machine Design. When a mass is stationary it can be easily balanced by putting suitable counter weight on the opposite side of mass. When a mass is revolving and if it is left unbalanced, then a centrifugal force is developed which changes its direction during rotation. This causes pre- mature failure of bearings and shafts and hence balancing is essential in Machine Design.

EXPERIMENTAL SETUP

PROCEDURE:

I. **Static balancing:-**

1. Remove the leather rope over the pulley.
2. Fix the pointer to 0° position.
3. Attach the balancing pans.
4. Remove locking screw and go on adding steel balls to the pan till, the pointer rotates through 90^{o} count number of balls.The weight of balls is the balancing weight for the eccentric weight similarly, find out relative weight for all eccentric weights and note down.

II. **Dynamic balancing:-**

From the relative weight (number of balls) assume position of two of the weights over shaft, draw the force polygon and find out position of other weights.

5) Mount the weights at proper position over the shaft.
6) Put the leather belt over the pulley and start the motor.
7) If the system is balanced, the shaft will rotate free from vibrations.

Calculations:

1) Static balancing: -

Let the given weights be m_1r_1 and m_2r_2 with an angle between them be 0, m_3r_3 and m_4r_4 are the balancing weights whose angular position are to be determined. Draw positions of m_1r_1 and m_2r_2 in position diagram. To draw force polygon (Fig.b.) draw 'ab' parallel to min to some scale From 'a', draw an arc whose radius is proportional to m_4r_4 and from 'c' draw an arc with radius proportional to m_3r_3 The intersection of the arc gives point 'd.' Join 'ad' and 'cd' draw parallel lines to 'cd' and ab in position diagram, this will give angular position of m_3r_3 and m_4r_4 respectively.

2) Dynamic balancing: -

Follow the procedure for static balancing of the system and find out angular position of balance weights. To find the linear position couple polygon (Fig.d.) is required, assume linear position of m_1r_1 taking moments about rotating plane of m_3r_3 couples are 1) m_1r_1x, 2) $m_2r_2a_2$, 3) $m_4r_4a_1$

Draw 'ab' parallel to m_1r_1 to the scale of m_1r_1 couple. From 'b' draw parallel to m_2r_2 from 'a' draw parallel line to m_4r_4 The intersection gives point 'c' "bo" is proportional to $m_2r_2a_2$ and "ac" is proportional to $m_4r_4a_1$. As m_2r_2 and m_4r_4 are known values then a. & a_2 can be determined.

Calculation:

$m_1 =$ gr

$m_2 =$ gr

$m_3 =$ gr

$m_4 =$ gr

$r_1 =$ cm

$r_2 =$ cm

$r_3 =$ cm

$r_4 =$ cm

To find θ_3 and θ_4:-

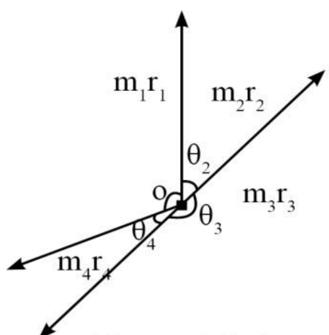


Fig. a. End view

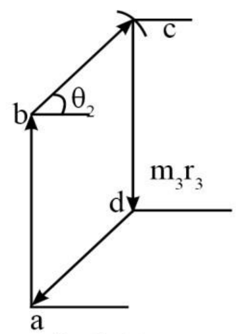
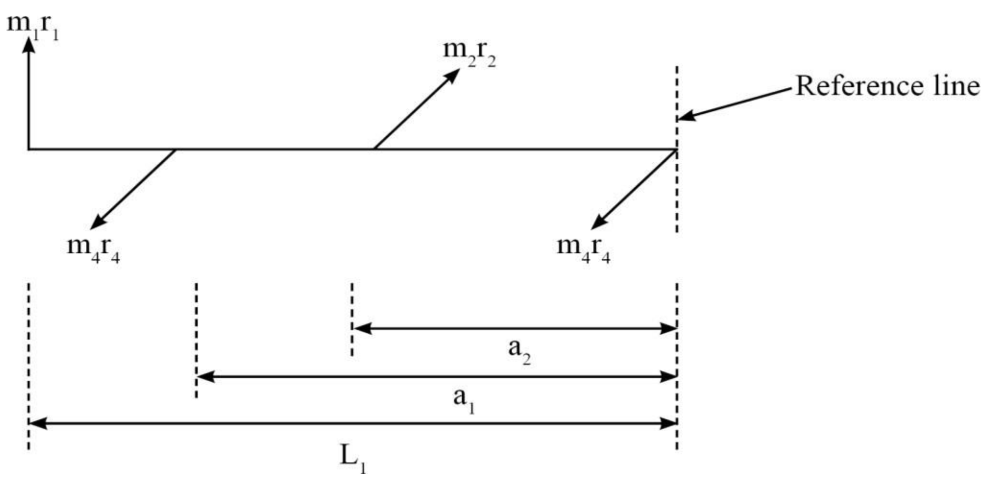


Fig. b. Force polygon



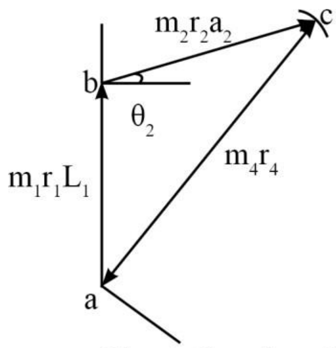


Fig. d. Couple polygon

From Fig.d.

$m_2r_2a_2 =$ **'x'** mm

$\therefore a_2 = x / m_2r_2$

$m_4r_4a_1 =$ **'y'** mm

$\therefore a_1 = 'y' / m_4r_4$

Experiment - 12

WATT GOVERNER

AIM:-

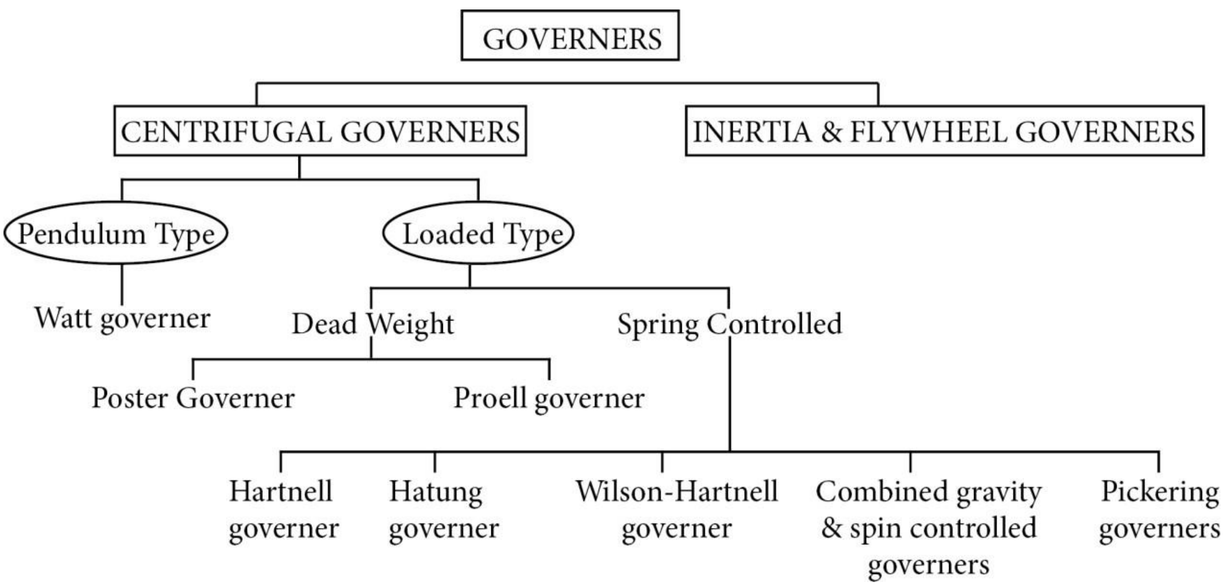
To study the characteristics of a watt governor and to determine the controlling force and frictional force.

APPARATUS:-

Universal governer experimental setup.

THEORY :-

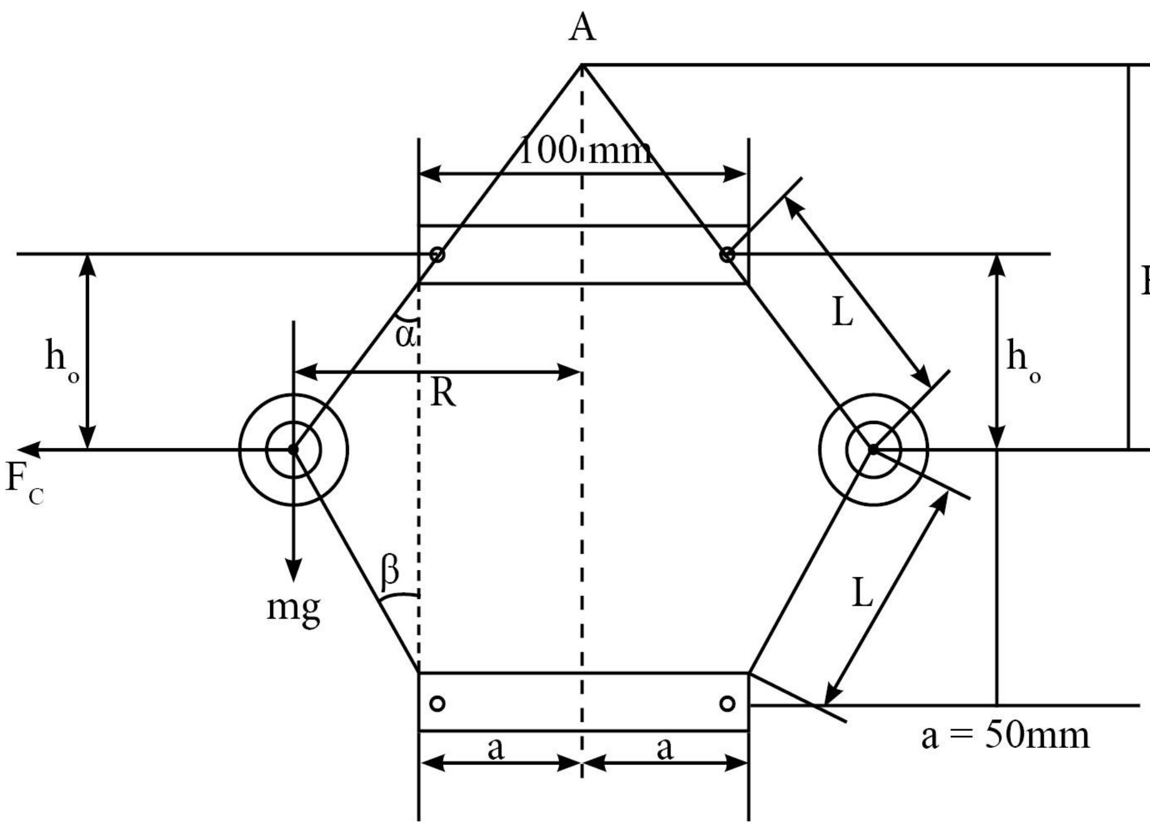
Governers is a device used to regulate the mean speed of an engine. Governers can be classified as follows.



Applications:

1. Governers are employed in a wide range in automobile of regulation of speed (eg, in cars. mopeds, trucks etc).
2. Governers are used in aircraft propellers. The governer senses shaft rpm, and adjusts or controls the angle of blades to vary the torque load on the engine. As the aircraft speeds up (disc) or slows (limb) the rpm is held constant.
3. Small engines, used to power lawn movers, portable generators are equipped with a governer to limit engine to a safe speed, and to maintain constant speed despite load changes.

EXPERIMENTAL SETUP:



PROCEDURE:

1. Place the watt governer assembly over the spindle of universal governer apparatus.
2. Tighten the bolt and nuts.
3. Start the motor, adjust the speed to the required value.
4. Note down the sleeve lift and the speed
5. Repeat the procedure for different speeds

FORMULAE:

1. Centrifugal or Controlling Force Due to Each Ball: (Actual)

$$\boxed{Fc_{act} = m_b^1 w^2 r} \qquad (N)$$

where $m_b^1 = m_b \times 2 =$ man of ball(Kg)

w = Angluar velocity $= 2\pi N/60$ (rad / sec)

R = radius of rotation(m)

$= [a + L\sin\alpha]$

$\alpha = \cos^{-1}\left(\frac{\text{height}}{\text{length of arm}}\right) = \cos^{-1}\left(\frac{h}{L}\right)$

a = Distance of pivot to centre of spindle

2. Theoretical Centrifugal Force:

$$\boxed{Fc_{Thy} = \frac{m_b^1 Rg}{H}} \quad (N)$$

where H = Height of governer(mm): $H = \left[\frac{a}{\tan x} + h\right]$(mm)

h = final height(mm) $= h = \left[h_o - \frac{X}{2}\right]$(mm)

X = final height of sleeve(mm)

g = Acceleration due to gravity: $g = 9.81m^2/s$

OBEERVITION:

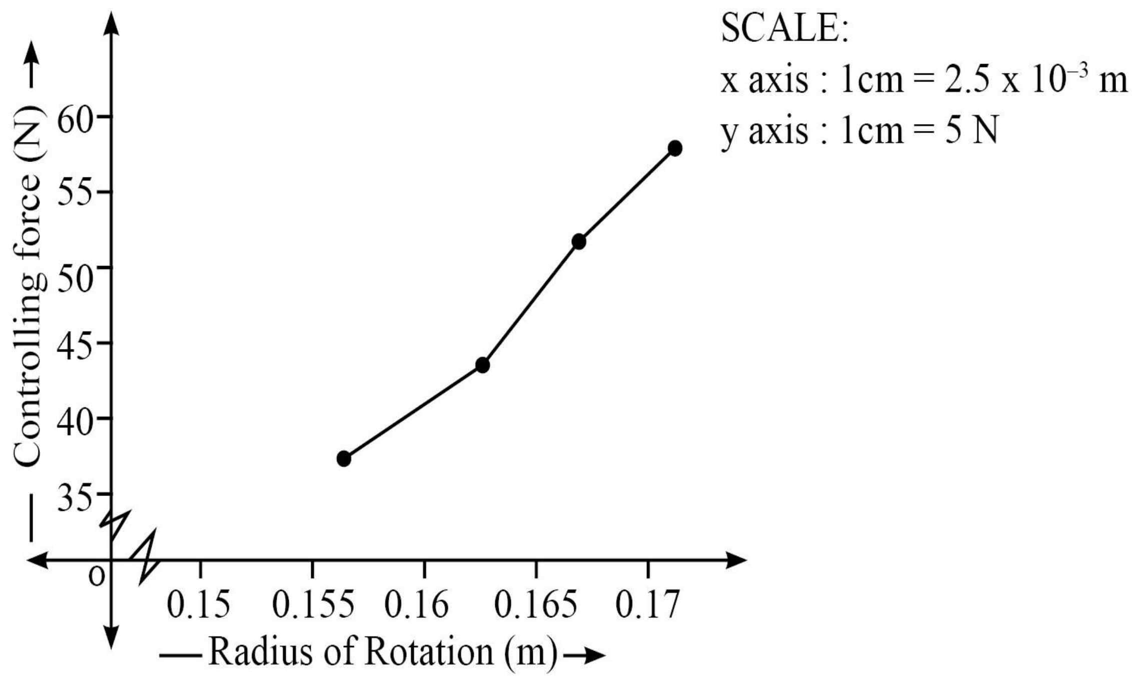
Length of each link or arm	l = 140mm = 0.14m
Mass of sleeve assembly	M = 4.8 Kg
Mass of governer ball	M_b = 0.120kg x 2 = 0.240 Kg
Initial radius of rotation	r_o = 0.145m
Initial height	h_o = 100mm = 0.1m
Distance of pivot to center of spindle	a = 50mm

TABULAR COLUMN:

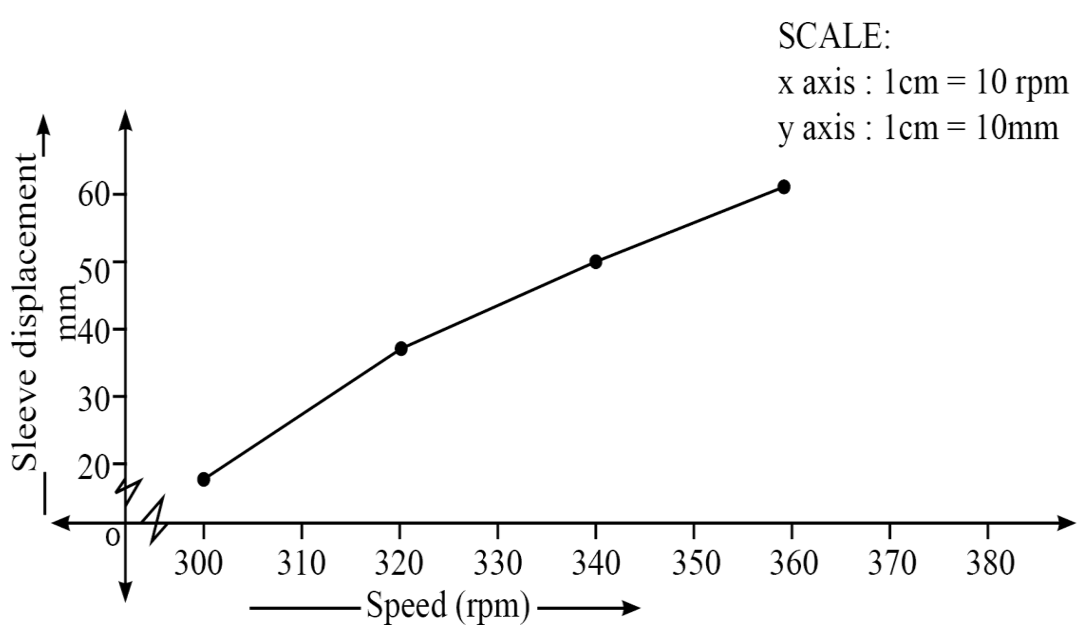
Sl. No	Governer speed N (rpm)	Angular velocity ω (rad/sec)	Sleeve Displacement X (mm)	Height of Governer H (m)	Radius of rotation r (m)	Controlling force F_{act} (N)	Controlling force Fc_{thy} (N)
1							
2							
3							
4							
5							

MODEL GRAPHS:-

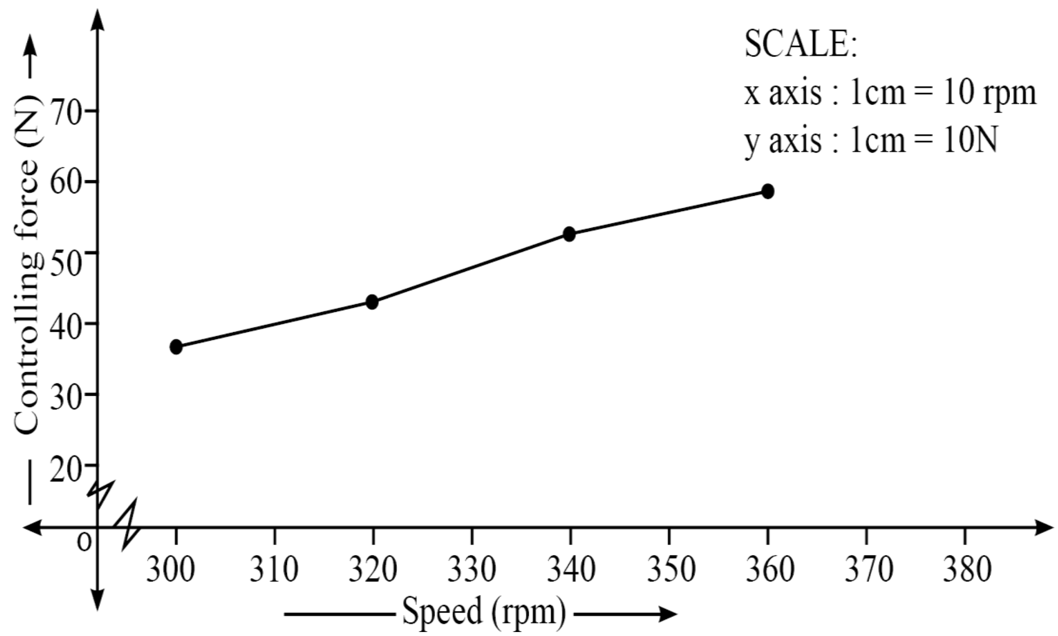
I. Controlling Force Vs Radius of Rotation



II. Speed Vs Sleeve Displacement



III. Controlling Force Vs Speed



RESULT:

The characteristics of watt governer were studied, controlling force and frictional forces were determined at different speeds.

Experiment - 13

PORTER GOVERNER

AIM:-

To study the characteristics of a porter governer and to determine the controlling force and frictional force

APPARATUS:-

Universal governer experimental setup

THEORY:-

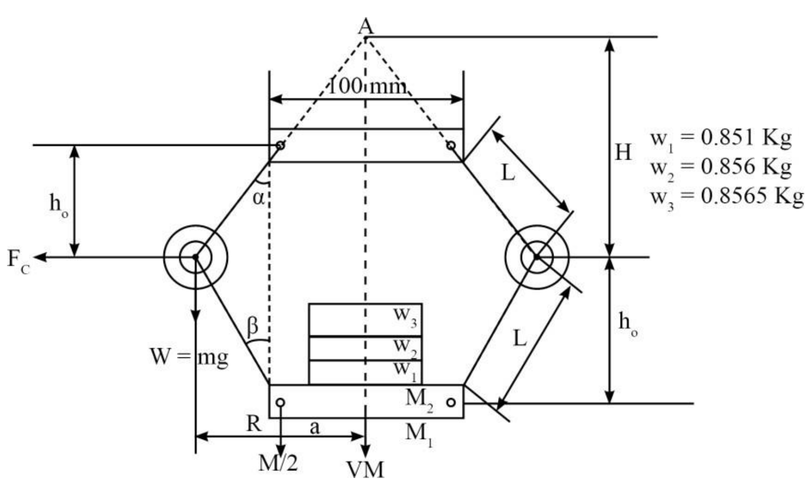
Porter governer is the modification of Watt governer. If the sleeve of watt governer is loaded with a heavy mass, it become a porter governer. Though there are several methods to determine the relation between the speed of the balls and height of governer. The following methods are important.

a) Method of resolution of forces
b) Instantaneous center method

PROCEDURE:

1. Place the porter governer assembly over the spindle of universal governer apparatus.
2. Place the required load on the spindle.
3. Tighten the bolts and nuts.
4. Start the motor, adjust the speed to required value
5. Note down the sleeve lift and the speed.
6. Repeat the procedure for different loads and speeds.

EXPERIMENTAL SETUP:



FORMULAE:

1. **Final Height** : $h = h_o - \frac{X}{2}$

where X = final height of sleeve (mm)

h_o = Initial length

where h = final height (mm)

L = Length of link (mm)

2. $\alpha = \cos^{-1}\left(\frac{h}{L}\right)$

3. **Radius of Rotation:** $R = [a + L\sin\alpha]$ (mm)

4. **Height of Governer :** $H = \left\{\left(\frac{a}{\tan\alpha}\right) + h\right\}$ (mm)

5. **Angular Velocity ;**

$\omega = \frac{2\pi N}{60}$ (rad / sec) where N = actual speed of governer (rpm)

6. **Actual Centrifugal Force / Controlling Force:**

$Fc_{act} = m_b \omega^2 R$ (N) where m_b = mass of each ball (Kg)

7. **Theoretical Centrifugal Force / Controlling Force:**

$Fc_{theo} = \left[m_b g + \frac{M}{2} g(1 + k)\right]\tan\alpha$ where M = total dead weight or total sleeve mass

Constant K = 1

Mass(Sleevassembly and added weights)

$M = M_1 + M_2 + M_3 + M_4 + M_5$

8. **Frictional Force / Equivalent Load to Friction - f is obtained from**

$$N^2 = \left(\frac{m_b + M \pm f}{m_b}\right)\frac{g}{H} \times \left(\frac{60}{2\pi}\right)^2$$

TABULAR COLUMN:

Sl. No	Actual speed of governer N_{act} (rpm)	Final height of sleeve X (mm)	Height of Governer H (mm)	Radius of Rotation R (mm)	Actual controlling Force (F_{act}) (N)	Theoretical controlling Force (Fc_{theo}) (N)	Frictional Force (f) (N)
1							
2							
3							
4							

Model Calculations –

1. Final height : $h = h_o - \frac{X}{2} = 100 - \frac{5}{2} = 97.5\text{mm}$

2. $\alpha = \cos^{-1}\left(\frac{97.5}{140}\right) = 45.859^o$

3. Radius of Rotation: $R = a + L\sin\alpha = 50 + 140\sin(45.859)$

$R = 150.467\text{mm}$

4. Height of governer: $H = \frac{a}{\tan\alpha} + h = \frac{50}{\tan(45.859)} + 97.5$

$H = 146.023\text{mm}$

5. Angular velocity: $\omega = \frac{2\pi N_{act}}{60} = 36.652\ \text{rad/sec}$

6. Actual centrifugal force = $Fc_{act} = m_b\omega^2 R$

$= 48.511\text{N}$

7. Theoretical controlling force:

$Fc_{theo} = [m_b + M]g\tan\alpha$

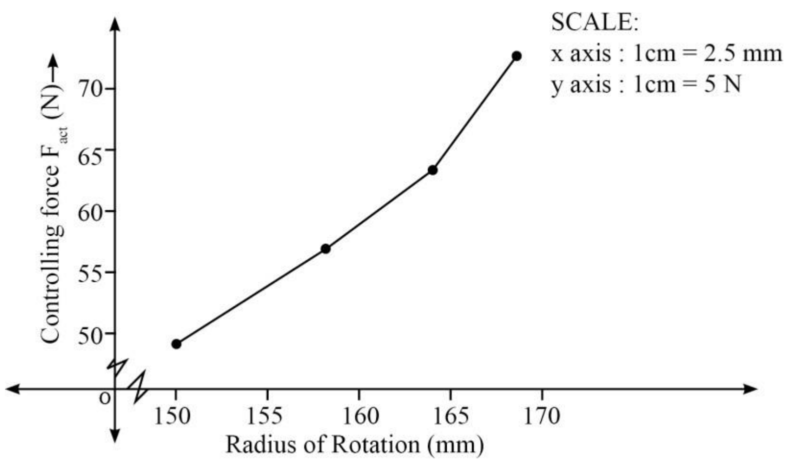
$M = M_1 + M_2 + M_3 + M_4 + M_5 = 7.727\text{Kg}$

$Fc_{theo} = 80.535\text{N}$

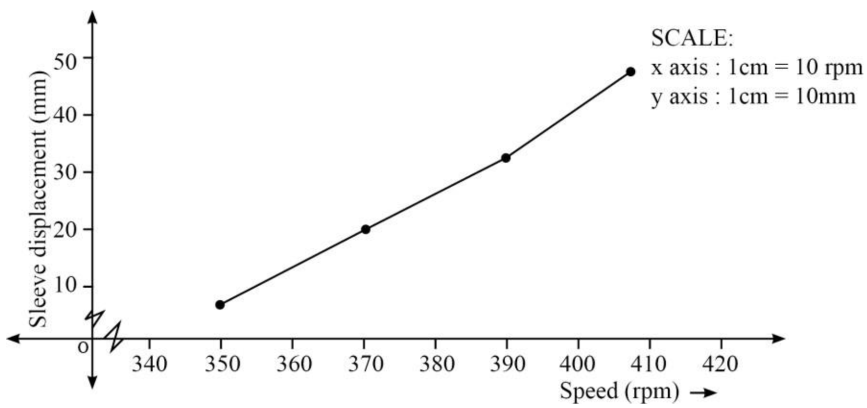
8. Frictional force : $N^2 = \left(\frac{m_b + M + f}{m_b}\right)\frac{g}{H} \times \left(\frac{60}{2\pi}\right)^2 \Rightarrow f = -3.1675\text{N}$

Model Graph:

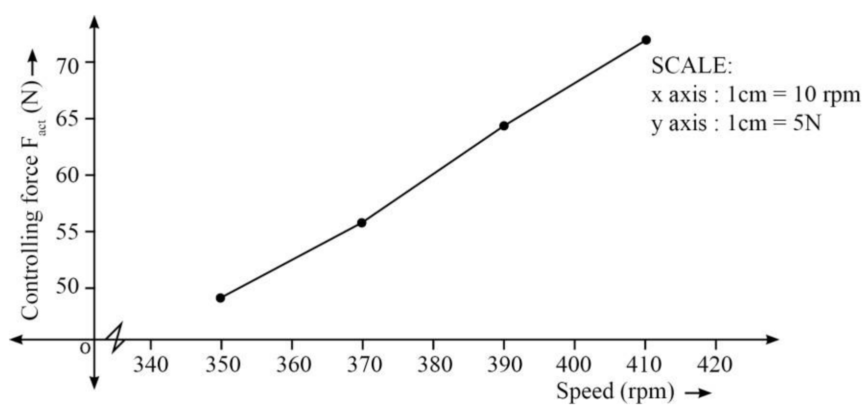
I. Controlling Force Vs Radius Of Rotation



II. Sleeve Displacement Vs Speed



III. Controlling Force Vs Speed



RESULT:

The characteristics of a porter governer were studied and the controlling force and frictional force were determined for various speeds and weights on sleeve.

Experiment - 14

STRAIN GAUGE ROSETTES

AIM:-

Determine of principle stresses and strains using strain gauge rosettes.

APPARATUS:-

Strain Rosette, weights and strain indicator.

THEORY:-

Electrical resistance strain gauges are widely used because of its negligible mass, small size and response to rapidly fluctuating strains. As the output is electrical remote observation is possible.

Electrical resistance strain gauges are widely used in

1. Experimental study of stress in transport vehicles, aircrafts, ships, automobiles & trucks.
2. Experimental Analysis of stress is structures, machines & apartment buildings, pressure vehicles, bridges, dams, transmission towers, steam & gas turbine.
3. Experimental verification of theoretical analysis.
4. Assist failure analysis
5. As a sensing element in transducers for measurement of force, load, pressure displacement & torque.

For a three element rectangular rosette with gauges A, B & C with angle θ_A, θ_B & θ_C respectively the strains induced are:

$$\varepsilon_A = \varepsilon_x \cos^2\theta_A + \varepsilon_y \sin^2\theta_A + \gamma_{xy} \cos\theta_A . \sin\theta_A$$

$$\varepsilon_B = \varepsilon_x \cos^2\theta_B + \varepsilon_y \sin^2\theta_B + \gamma_{xy} \cos\theta_B . \sin\theta_B$$

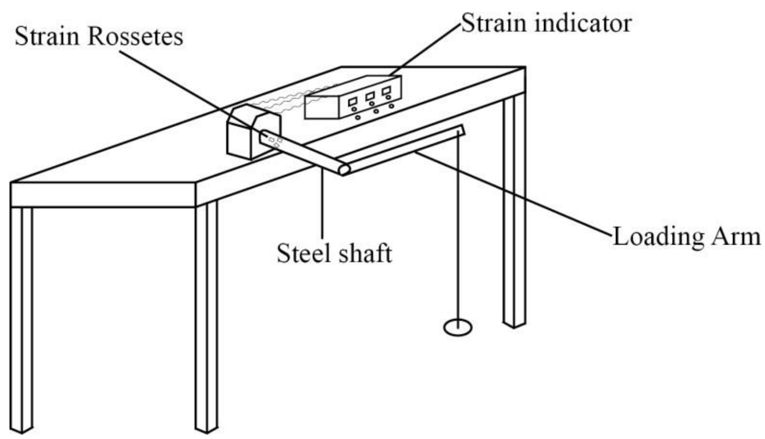
$$\varepsilon_C = \varepsilon_x \cos^2\theta_C + \varepsilon_y \sin^2\theta_C + \gamma_{xy} \cos\theta_C . \sin\theta_C$$

where ε_x is strain along x axis

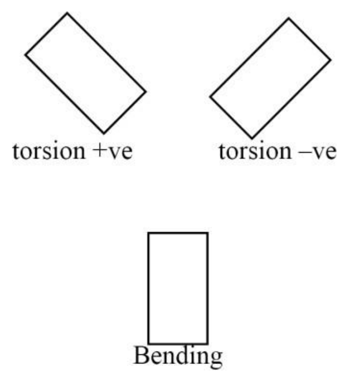
ε_y is strain along y axis

γ_{xy} is shear strain in xy plane

EXPERIMENTAL SETUP:-



Strain Gauge's Placement



PROCEDURE:

1. Make the necessary connections to the digital strain indicator from sensor. Adjust the indicator knob to zero.
2. Load the specimen with the aid of loading arm in steps.
3. Record the strain in microstrains by connecting corresponding strain gauges to indicator with help of probes.
4. Three readings are recorded which indicate bending and torsional strain.
5. Repeat the above steps for different loads.
6. Compute the required parameters by using appropriate formulae.

FORMULAE:

I. EXPERIMENTAL CALCULATIONS:-

1) Principal Strains:

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

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

2) Principal stress:

$$\sigma_1 = \frac{E}{1-\mu^2}(\varepsilon_1 - \mu\varepsilon_2) \quad ; \quad \sigma_2 = \frac{E}{1-\mu^2}(\varepsilon_2 - \mu\varepsilon_2) \qquad \text{where } \mu = 0.3$$

3) Principal Directions:

$$\phi_1 = \frac{1}{2}\tan^{-1}\left[\frac{2\varepsilon_B - \varepsilon_A - \varepsilon_C}{\varepsilon_A - \varepsilon_C}\right] \quad ; \quad \phi_2 = 90^o + \phi_1$$

4) Maximum shear strain:

$$\gamma_{max} = \sqrt{(\varepsilon_A - \varepsilon_C)^2 + (2\varepsilon_B - \varepsilon_A - \varepsilon_C)^2}$$

5) Maximum shear stress:

$$\tau_{max} = \frac{E}{2(1+\mu)}\gamma_{max}$$

II. THEORITICAL CALCULATIONS:-

1) Principal stress:

$$\sigma_1 = \left(\frac{\sigma_b}{2}\right) + \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2} \quad ; \quad \sigma_2 = \left(\frac{\sigma_b}{2}\right) - \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

where $\sigma_b = \dfrac{32M_b}{\pi d^3}$; bendingstress

Torisionalshearingstress $\tau = \dfrac{16T}{\pi d^3}$

M_b = Load applied $\times$ length of shaft

T = Load $\times$ Length of arm

2) Principal Strains:

$$\varepsilon_1 = \frac{1}{E}(\sigma_1 - \mu\sigma_2) \quad ; \quad \varepsilon_2 = \frac{1}{E}(\sigma_2 - \mu\sigma_1)$$

3) Maximum shear stress:

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} \qquad \text{or} \qquad \tau_{max} = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \tau^2}$$

4) Maximum shear strain:

$$\nu_{max} = \frac{\epsilon_1 - \epsilon_2}{2}$$

OBSERVATIONS:

1. Material of specimen = Mild steal
2. Diameter of specimen = d = 24.7mm
3. Length of torque arm = L = 1000mm
4. Modulus of Elasticity = E = 210 Gpa
5. Modulus of Rigidity = G = 73 Gpa
6. Poisson's Ratio = μ = 0.3
7. Length of shaft = l = 265mm

TABULAR COLUMN:

Sl. No.	Load applied W		Strain Indicator Reading $\mu\epsilon$			Principle strains x 10^{-6}				Principle angles	
						Experimental		Theoritical			
	Kg	N	ϵ_A	ϵ_B	ϵ_C	ϵ_1	ϵ_2	ϵ_1	ϵ_2	ϕ_1	ϕ_2
1											
2											
3											

Sl.No	Principal stress (MPa)				Maximum shear stress τ_{max} (MPa)		Maximum shear strain ν_{max}	
	Experimental		Theoretical					
	σ_1	σ_2	σ_1	σ_2	Experimental	Theoretical	Experimental	Theoretical
1								
2								
3								

RESULT:

The Principle stress and strains were determined using strain gauge rosettes. The theoretical and experimental values were tabulated and compared.

Experiment - 15

GYROSCOPE

AIM:-

To determine the gyroscopic couple and compare it with the applied couple.

APPARATUS:-

Motorized gyroscope, dimmer stat set, standard weights digital tachometer.

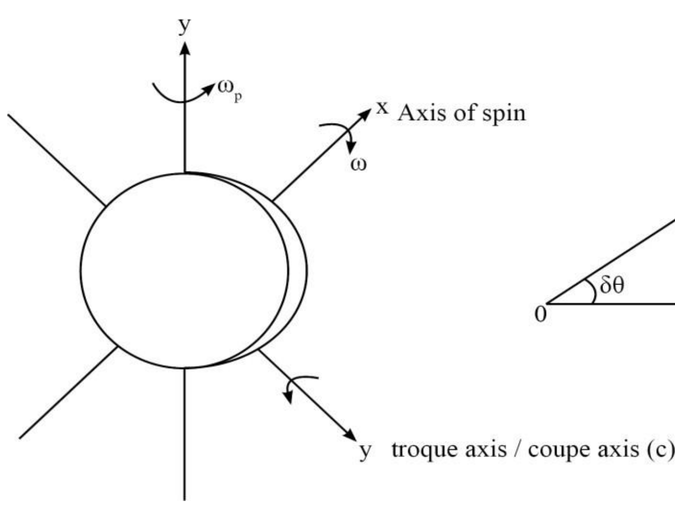
THEORY:

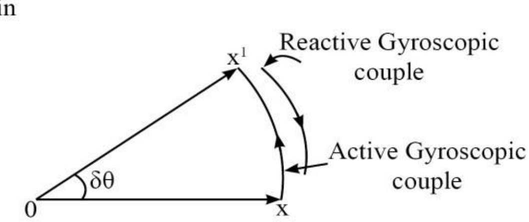
A gyroscope is a body which while spinning about the axis is free to move in other directions under the action of external force. It is a mechanical instrument that maintains an angular reference direction by virtue of a rapidly spinning, heavy mass. Application of gyroscope depend on a form of Newton's second law, which states that a massive, rapidly spinning body resists being disturbed and tends to react to a disturbing torque of PRECESSING (rotating slowly) in a direction perpendicular to the direction of torque. (GYRO)

The properties of gyroscope are observed in celestial bodies, artillery shells, turbine rotor on ships, while their applications are automatic guidance of airplanes, ships, rockets, torpedoes, determination of geographic meridian etc.

The properties of gyroscope are manifested only when:

1. The axis of rotation is capable of changing orientation.
2. The angular velocity of gyroscope's rotation about its axis must be very much greater than angular velocity of axis itself.





Properties of Gyroscope

1. RIGIDIITY: The axis of rotation of the wheel tends to remain in a fixed direction in space if no force is applied on it.

   In a balanced gyroscope with three degrees of freedom, its axis persistently tends to preserve its initial orientation in space.
2. PERCESSION: Axis of rotation tends to turn at right angle to applied force. It can be defined as change in direction of rotation axis in which the second Euler angle (nutation) is constant. There are two types: torque free and torque induced (gyroscopic) precession.
3. NUTATION: It is a rocking, swaying or nodding motion in the axis of rotation of a largely symmetric object such that the first Euler angle (precession) is constant.

PROCEDURE:

1. Switch on the apparatus.
2. The dimmer stat is adjusted to get a desired speed.
3. The gyroscope is given some time to obtain a constant speed.
4. Standard weights are now added to the gyroscope and the time taken by it to move by an angle of 45^o or 90^o is noted.
5. Step 1 to 4 or repeated for different speeds.

FORMULAE:

1. GYROSCOPIC COUPLE:

$$\boxed{C_g = I w w_p} \qquad (Nm)$$

where I = Mass moment of Inertia of disc

$$I = \frac{m_d r^2}{2} Kgm^2 \qquad \text{where, } r = \text{radius of disc in 'm'}$$

ω = Angular velocity of disc or rotor or velocity of spin (rad / sec)

ω_p = Angular velocity of precession or axis of spin (rad /esec)

$$= \frac{\text{angle turned}}{\text{time taken}} \quad (\text{rad/sec})$$

m_d = mass of disc (Kg)

x = distance between C.G of weight stud to the C.G of disc (m)

2. APPLIED COUPLE:

$$\boxed{C_a = W.x} \qquad (Nm)$$

W = Weight applied (N)

OBEERVITION:

W_d = Weight of disc = 4 x 9.81 = 39.24 N

d = Diameter of disc = 280 mm = 0.28m

t – Thickness of disc = 10 mm = 0.01m

x = Distance between the C G of weight stud and the C G of the disc

= 260 mm = 0.26m

TABULAR COLUMN:

Sl. No	Rotor speed N (rpm)	Weight sadded		Time for 90º /45º Precession(sec)	Angular velocity of rotor ω (rad/sec)	Angular velocity of precession ω_p (rad/sec)	Gyroscopiccouple Cg N - m	Appliedcouple C_a N - m
		M (Kg)	W (N)					
1								
2								

RESULT:

The gyroscopic couple was determined & compared with applied couple, are tabulated in the tabular column.

Experiment – 16

Journal Bearing

AIM:

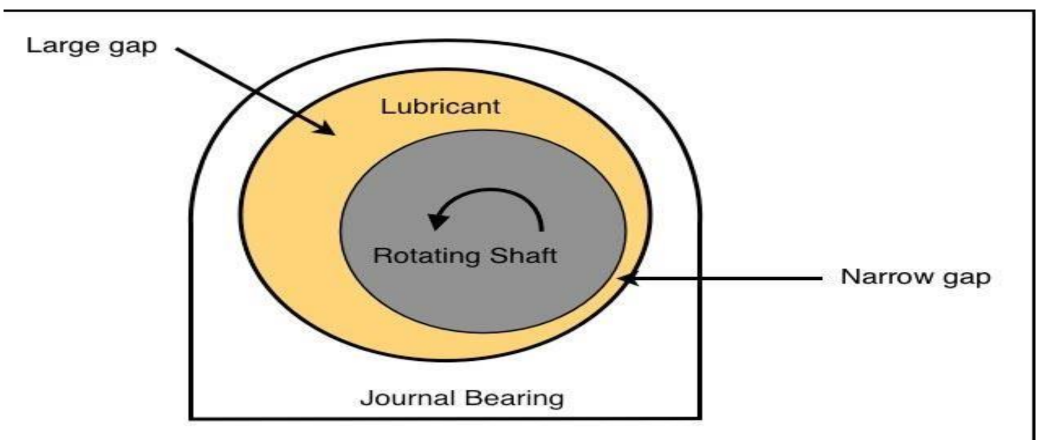
To determine the pressure distribution in the oil film of a journal bearing

APPARATUS:

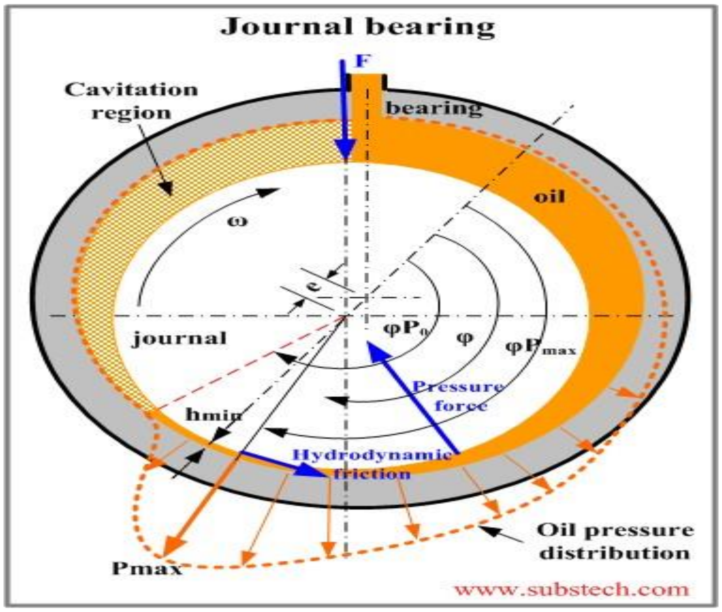
Computerized Journal bearing apparatus

THEORY:

The purpose of a bearing is to support a load. Deflection of the journal within the bearing can adversely affect the load carrying capacity of the bearing. This deflection can be greatly reduced by increasing the diameter of the journal and decreasing its length. This results in a short bearing with a consequential greater flow of oil out of the ends of the bearing. This outflow of oil transfers heat from the bearing and helps to reduce the bearing temperature.



The factor which controls the length of the bearing to the journal diameter is known as the L/D ratio. For ratios greater than unity a long bearing would result and for ratios less than unity a short bearing results. If the roughness of the sliding surfaces is reduced, the load carrying capability of the bearing increases Applying suitable surface treatments is a useful addition to bearing design.



Clearances should be small enough to obtain the maximum load carrying capability of the bearing. If the clearance is too small the bearing temperature will be too high and the minimum film thickness will be too low. As the bearing wears, the effect on the bearing performance must be considered, as this leads to a decrease in the bearing temperature and an increase in the flow of oil through the bearing with a knock on effect on hydrodynamic and boundary lubrication.

PROCEDURE:

1) Fill the oil tank using required SAE grade oil under test & position the tank at the desired level
2) Drain out the air from all the tubes.
3) Check for any leakage of the oil.
4) Some leakage of the oil is necessary for cooling purpose
5) Check the direction of rotation & increase the speed of motor slowly.
6) Set the speed & let the journal run for about 30 minutes. Until the oil in the bearing is warmed up
7) Check the steady oil level at various tapping & record the reading from the digital Indicator.
8) Add required load & keep the balancing load in horizontal position & observer the steady level.

9) Take the pressure readings of 16 tubes
10) Repeat the experimental for various speeds & constant loads or constant speed &.varying load.

Specification & observation

Diameter of journal d = ...mm

Length o f journal L = ... mm

Diameter of bearing D =.......... mm

Speed of the motor = ... rpm

Hanger Weight = ... N

Total weight = ... N

Initial Pressure P_0 = ... N /m^2

Tabular column:

Manometer	Pressure P N /m^2	Pressure Diff (P-Po) N/m^2
1		
2		
3		
4		
5		
6		
7		
8		
9		
10		
12		

FORMULAE:

Pressure distribution in Journal

1) $(P\ P_0)_{max} = K \sin\theta_{max} (2+\varepsilon \cos \theta_{max}) / (1+ \varepsilon \cos \theta_{max})^2$

Find K = ………….

Where θ_{max} =............ from the Graph

$$\cos\theta_{max} = \frac{-3\varepsilon}{\left(2+\varepsilon^2\right)}$$

Find Attitude ε = c/ ε .....From the above Equation

2) Load Carried by oil in the projected area

W = hρdL

Where Avg Film Thickness h = sum of (P- P_0) / number of positive Value

3) Theoretical Load $W = \left\{ \left(\frac{12\eta r^3\omega\pi}{d}\right)\left(\frac{\varepsilon}{2+\varepsilon^2}\right)\left(\frac{1}{\sqrt{1-\varepsilon^2}}\right) \right\}$

Where δ = Radial clearance = (D-d/2)

4) Frictional Couple $M = \left\{ \left(\frac{4\eta r^3\omega\pi}{d}\right)\left(\frac{1+2\varepsilon^2}{2+\varepsilon^2}\right)\left(\frac{1}{\sqrt{1-\varepsilon^2}}\right) \right\}$

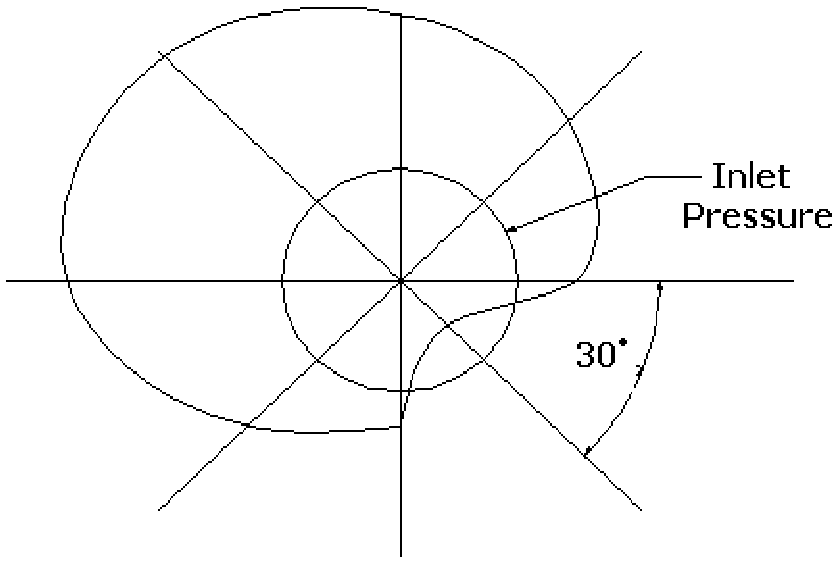
5) Frictional Force F = $(2\,\eta N L d \pi^2)/\,\psi$

6) Coefficient of friction μ = F/w

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 on the fig.



Pressure Distribution Curve (Radial)

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