ACS COLLEGE OF ENGINEERING

Kambipura, Mysore Road, Bengaluru – 560074

DEPARTMENT OF AERONAUTICAL ENGINEERING

V SEMESTER B.E.

(Prescribed for V - Semester Aeronautical Engineering)

BAE502– AIRCRAFT STRUCTURES LABORATORY

LABORATORY MANUAL

2022 SCHEME

NAME OF THE STUDENT : ---

BRANCH : ---

UNIVERSITY SEAT NO. : ---

SEMESTER & SECTION : ---

BATCH : ---

LIST OF EXPERIMENTS

Experiments for Performance

EXPERIMENT – 1

DEFLECTION OF A SIMPLY SUPPORTED BEAM

Aim: To determine the deflection of a simply supported beam.

Equipment: Beam Test Set-Up with Load cells, steel scale, caliper, flat beam

Theory:

A beam shown in fig-1 shows the section which is simply supported at the ends and is subjected to bending about its major axis with a concentrated load anywhere in the beam. The beam is provided with strain gauge; the deflection of the beam can be determined whenever the load is applied on the beam. Strain gauge values may be noted for several further works.

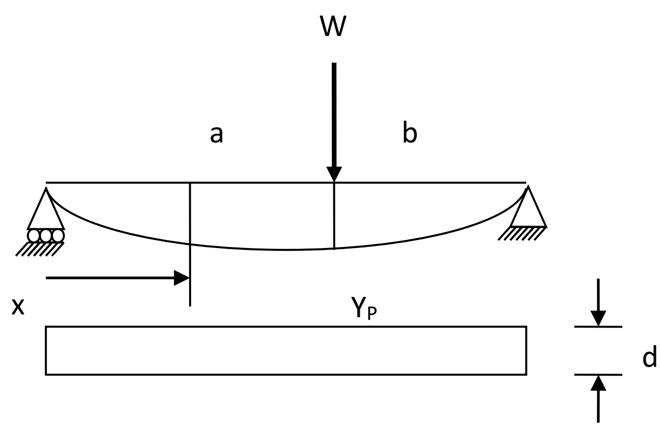


Fig-1: A simply supported beam

Deflections are given by following expressions. These expressions can be derived using Unit Load Method or Castigliano's Theorem.

$$Y_X = W\,b\,\frac{[(L^2 - b^2)\;x\; - x^3]}{(6\,E\,I\,L)}$$

For $0 < x < a$

$$Y_X = W\,b\,\frac{[\,x^3 - \frac{L}{b}(x - a)^3 - (L^2 - b^2)]}{(6\,E\,I\,L)}$$

For $a < x < L$

Where,

W is the load placed at a distance `a` from the left support in Newton
L = span of the beam in mm
Y_x = deflection at any point distance x from left end
I = moment of inertia of the beam in mm^4 (I_{xx})
E= Young's modulus in N/mm^2

Procedure:

1) Find the moment of inertia of beam from the following expression:

$$\frac{b_1 d_1^3}{12}$$

Where b_1 is width of beam and d_1 is depth.

2) Place the beam supporting from two wedge supports. The load position can be varied.
3) Set the load cell to read zero in the absence of load.
4) Set the deflection gauge to read zero in the absence of load.
5) Load the beam with 2.5 Kg. Note deflections before and after the load point through deflection gauge.
6) Increase the load to 5.0 Kg and repeat the experiment.
7) Find the deflections from the formula and verify.

Tabular column

Load (N)	I_{xx} (mm^4)	L (mm)	a (mm)	b (mm)	x (mm)	Theoretical value (mm)	Experimental Value (mm)

<u>CAUTION: Never exceed 10 kg load on the beam</u>

Conclusion:

Example:

Determine the deflection of a simply supported beam loaded with W = 50,000 N, Young's Modulus E = 2 x 105 N/mm2; Second moment of Inertia Ixx = 7332.9 x 104 mm4; & the load is placed at a distance a = 4800 mm; and the span of the beam L = 60000 mm. Find the deflection at x = 3000 mm from the left end.

$$\boldsymbol{Y_X = W\, b\, \frac{[(L^2 - b^2)\ x\ - x^3]}{(6\, E\, I\, L)}}$$

$= 50000 \times 1200 \times [\ (6000^2 - 1200^2)\ 3000 - 3000^3] /(6 \times 2.0 \times 10^5 \times 7332.9 \times 10^4 \times 6000)$

$$Y_x = 8.714 \text{ mm}$$

EXPERIMENT – 2

VERIFICATION OF MAXWELL'S RECIPROCAL THEOREM

Aim: To verify the Maxwell's Theorem for the structures system

Equipment: Beam Test Set-Up with Load cells, steel scale, caliper, flat beam

Theory:

The displacement at point 'i', in a linear elastic structure, due to concentrated load at point 'j' is equal to the displacement at point 'j' due to a concentrated load of same magnitude at point 'i'.

The displacement at each point will be measured in the direction of the concentrated load at that point. The only other restrictions on this statement, in addition to the structure being linear elastic and stable, is that the displacement at either point must be consistent with the type of load at that point. If the load at a point is a concentrated force, then the displacement at that point will be a translation, while if the load is moment, then the displacement will be rotation. The displacement at any point will be in the same direction as the load at that point and its positive direction will be in the same direction as the load.

This theorem often referred to as **Maxwell's Reciprocal Displacement theorem.**

This can be proved through Unit Load Method i.e.; the deflection at A due to unit load at B is equal to deflection at B due to unit load at A.

$$\delta = \int M.m\, dx \;/\; EI$$

where,

M = Bending Moment at any point x due to external load

m= Bending Moment at any point x due to unit load applied at the point where deflection is required

let**,** $\mathbf{m_{xA}}$ = Bending Moment at any point x due to unit load at A

$\mathbf{m_{xB}}$ = Bending Moment at any point x due to unit load at B

when unit load (external load) is applied at A, $\mathbf{M=m_{xA}}$

To find deflection at B due to unit load at A, apply unit load at B

Then $\mathbf{m=m_{xB}}$

$$\text{Hence , } \delta_{BA} = \int M\, m\, dx \;/\; EI \;\; \delta = \int m_{xA}\, m_{xB}\, dx \;/\; EI \text{ ------(1)}$$

Similarly, when unit load (external load) is applied at B, $M=m_{xB}$

To find deflection at A, then $\mathbf{m=m_{xA}}$

Hence

$$\delta_{AB} = \int M\, m\, dx \;/\; EI \;\; = \int m_{xA}\, m_{xB}\, dx \;/\; EI \text{ ------------------(2)}$$

Comparing eqn. (1) and eqn. (2)

$$\delta_{AB} = \delta_{BA} \text{--- (3)}$$

The external load (W) can be taken as a multiple with unit load , therefore, this load W will appear as multiple with m_{xA} in eqn. (1) & as multiple with m_{xB} in eqn. (2). Thereby resulting in

$$W\,\delta_{AB} = W\,\delta_{BA} \text{-- (4)}$$

A beam shown in figure below which is simply supported at the ends and is subjected to bending about its major axis with a concentrated load anywhere in the beam

Deflections δ_x at any distance `x` from left support are given by following expressions; reference may be made to experiment no. 1.

$$\delta_x = W\, b\, [\, (L^2 - b^2)\, x - x^3] \,/(6\, E\, I\, L) \qquad \text{for} \quad 0<x<a$$

$$\delta_x = W\, b\, [\, X^3 - \; L/b\, (x-a)^3 \; - (L^2 - b^2)\, x \,] \,/(6\, E\, I\, L) \qquad \text{for} \quad a < x < L$$

Where,

W is the load placed at a distance \`a\` from the left support in Newton
b=distance of load from right side support
L = span of the beam in mm
Y_x = deflection at any point distance x from left end
I = moment of inertia of the beam in mm^4 (I_{xx})
E= Young\`s modulus in N/mm^2

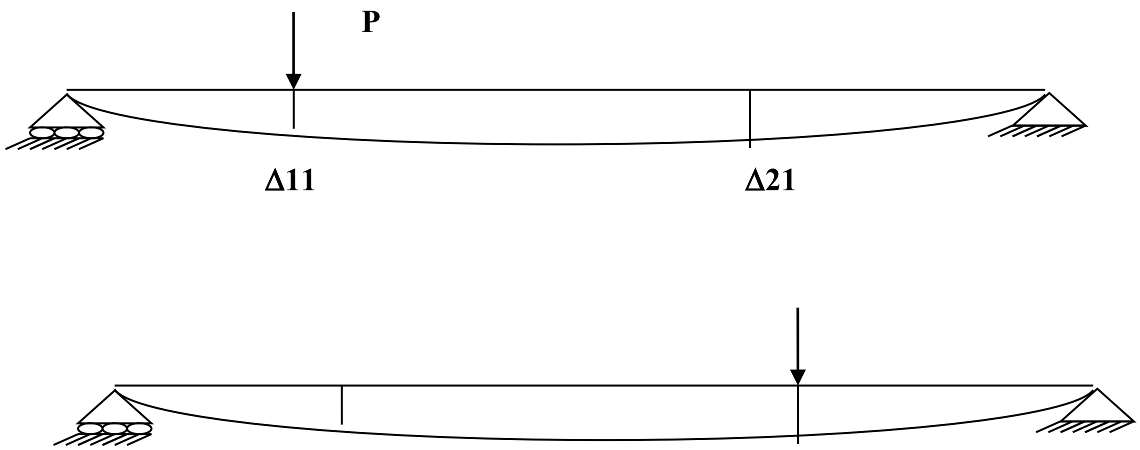


Fig-3: A simply supported Beam loaded at position 1 & 2

The Maxwell's Reciprocal Displacement theorem is very useful in the analysis of statistically indeterminate structures for evaluating the flexibility coefficients.

The displacement relationship can be expressed at point i and j

$$\delta_{i,B} = f_{i,j}\ W \text{--} \quad (5)$$

$$\delta_{j,A} = f_{j,i}\ W \text{--} \quad (6)$$

Where, $f_{i,j}$ is the displacement at point i due to a unit load at point j and $f_{j,i}$ is the displacement at point j due to a unit load at point 'i'. If we now substitute these expressions in Betti`s law and cancel out the term W on each side, we obtain

$$f_{i,j} = f_{j,i}$$

The theorem can be restated as the displacement at point i, in an elastic structure, due to a unit load at point j is equal to the displacement at point j due to unit load at point i.

Procedure:

1. Find the moment of inertia of beam from the following expression: $1/12\ b_1 d_1^{\ 3}$, where b_1 is width of beam and d_1 is depth.
2. Place the beam supporting from two wedge supports. The load position can be varied.
3. Set the load cell to read zero in the absence of load. Set the deflection gauge to read zero in the absence of load.
4. Load the beam with 2.5 Kg . Note deflections at any point through deflection gauge.
5. Interchange the load location with the point of deflection measurement and repeat the readings.
6. Increase the load to 5.0 Kg and repeat the experiment.
7. Find the deflections from the formula and verify.

CAUTION: NEVER EXCEED 10 Kg LOAD ON THE BEAM

Tabular column

Load (N)	I_{xx} mm^4	L mm	a mm	b mm	x mm	Theoretical value mm δ_{BA} or δ_{AB}	Experimental value mm δ_{BA} or δ_{AB}

Conclusions:

EXPERIMENT – 3

DETERMINATION OF YOUNG'S MODULUS USING STRAIN GAGES

Aim: To determine the young's modulus of a simply supported beam or a cantilever beam.

Theory:

A beam shown in fig-1 shows the section which is simply supported at the ends and is subjected to bending about its major axis with a concentrated load anywhere in the beam. The beam is provided with strain gauge, the deflection of the beam can be determined wherever the load is applied on to the beam. An available cantilever beam can also be utilized for this experiment.

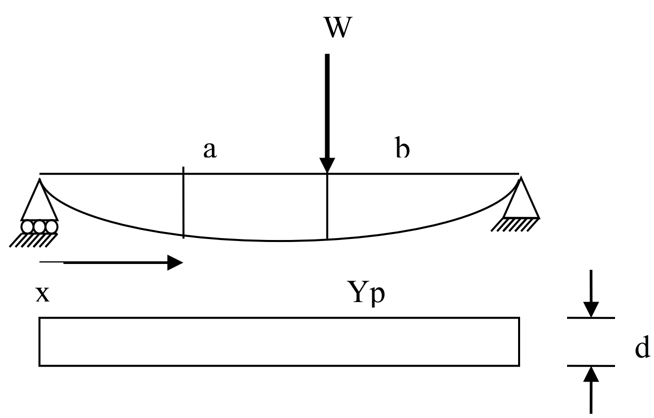


Fig-1: A simply supported beam

The strain gauge is at a fixed position in the beam and load position can be varied. A strain gauge is mounted on a free surface, which in general, is in a state of plane stress where the state of stress is with regards to a specific xy rectangular rosette. Consider the three element rectangular rosette shown in fig-2, which provides normal strain components in three directions spaced at angles of 45^0. If an xy coordinate system is assumed to coincide with the gauge A and C then $\varepsilon x = \varepsilon_A$ and $\varepsilon y = \varepsilon c$. Gauge B provides information necessary to determine γ_{xy}. Once εx, εy and γ_{xy} are known, then Hooke's law can be used to determine σx, $\sigma_{y,}$ and $\tau_{xy.}$ However in this case the requirement is to determine Young`s Modulus (E), which can be determined from equation (1) below.

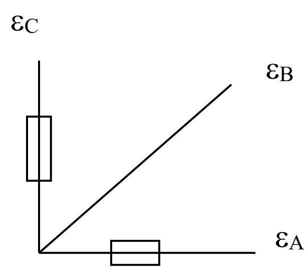


Fig-2

$$\sigma_x = \frac{E}{(1-\nu^2)}(\varepsilon_x + \nu\varepsilon_y);\sigma_y = \frac{E}{(1-\nu^2)}(\varepsilon_y + \nu\varepsilon_x);\tau_{xy} = \frac{E}{2(1+\nu)}\gamma_{xy}$$

$$\varepsilon_x = \varepsilon_A; \qquad \varepsilon_y = \varepsilon_C; \qquad \gamma_{xy} = 2\,\varepsilon_B - \varepsilon_A - \varepsilon_C\,;$$

Sl.No	Load (N)	ε_A	ε_B	ε_C	εx	εy	γ_{xy}	$\nu=\varepsilon y/\varepsilon x$

M/I = σ/y ---1

also

Shear Modulus $G = \dfrac{\tau_{xy}}{\gamma_{xy}}$ **---2**

Young's Modulus E = 2G (1+ν) ---3

Procedure: Mount the cantilever beam at the left support of beam test set-up. Connect the strain gauges wires with the strain measuring equipment. Use the following color codes

ε_A Blue wires
ε_B Green wires
ε_C Red wires

Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with 2.5 Kg at some point and record the strains in three directions. Record the load value at the load cell.

Repeat the experiment with load value of 5 Kg. Compute the values of Poisson's ratio from:

$$\nu=\varepsilon y/\varepsilon x$$

CAUTION: NEVER EXCEED 10 Kg LOAD ON THE BEAM .

Tabular Column:

Sl.No	Load (N)	ε_A	ε_B	ε_C	εx	εy	γ_{xy}	$\nu=\varepsilon y/\varepsilon x$

Find Young `s modulus through formulas above.

Example:

Sl No.	Load (N)	ε_A	ε_B	ε_C	εx	εy	γ_{xy}	$\nu=\varepsilon y/\varepsilon x$
1	10N	92μ	46μ	-18μ	92μ	-18μ	18μ	0.1957
2	20N	64μ	304μ	-18μ	64μ	-18μ	14μ	0.2188
3	30N	37μ	19μ	-10μ	37μ	-10μ	11μ	0.2973

EXPERIMENT – 4

POISSON`S RATIO (ν) DETERMINATION

Aim: To determine the Poisson's ratio of cantilever beam

Equipment: Beam Test Set-Up with load cells, Cantilever beam with calibrated rosette strain gauge, strain measuring equipment.

Theory: A cantilever beam is subjected to bending about its major axis with a concentrated load anywhere in the beam. The beam is provided with rosette strain gauge.

A calibrated strain gauge rosette is fixed at a location with-in the span of the beam, and load position can be varied. Calibration has been done to read strains in microns (μ). Consider the three element rectangular rosette shown in fig-2, which provides normal strain components in three directions spaced at angles of 45^0. If an xy coordinate system is assumed to coincide with the gauge A and C then $\varepsilon x = \varepsilon_A$ and $\varepsilon y = \varepsilon c$.
Gauge B provides information necessary to determine shear strain (γ_{xy}).
Once εx, εy and γ_{xy} are known, then Hooke's law can be used to determine σx, σ_y, and τ_{xy}. Subsequently principal stresses can be determined.
Poisson's ratio (ν) can be determined from:

$$\nu = \varepsilon y / \varepsilon x$$

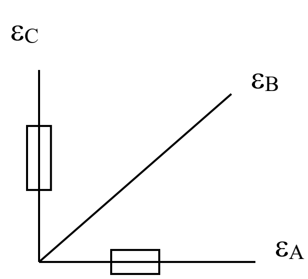


Fig-2

$$\sigma_x = \frac{E}{(1-\nu^2)}(\varepsilon_x + \nu\varepsilon_y);\ \sigma_y = \frac{E}{(1-\nu^2)}(\varepsilon_y + \nu\varepsilon_x);\quad \tau_{xy} = \frac{E}{2(1+\nu\ )}\gamma_{xy}$$

$$\varepsilon_x = \varepsilon_A\ ;\quad \varepsilon_y = \varepsilon_c\quad ;\quad \gamma_{xy} = 2\,\varepsilon_B - \varepsilon_A - \varepsilon_c$$

<u>Principal stress axes:</u>

Principal stress axes is located with the angle θ according to :
$\tan 2\theta = (2\,\varepsilon_B - \varepsilon_A - \varepsilon_c)\ /\ (\varepsilon_A - \varepsilon_c)$

<u>Principal Stresses are given by following expressions:</u>

$$\sigma_1 = A + \sqrt{B^2 + C^2}\quad ,\qquad \sigma_2 = A - \sqrt{B^2 + C^2}$$

$$\tau_{max} = (\sigma_1 - \sigma_2)/2 = \sqrt{B^2 + C^2}$$

where, $A = (\sigma_x + \sigma_y)/2$, $B = (\sigma_x - \sigma_y)/2$ and $C = \tau_{xy}$

E= Young's modulus in N/mm^2. Strains are in microns

Procedure: Mount the cantilever beam at the left support of beam test set-up. Connect the strain gauges wires with the strain measuring equipment. Use the following color codes

ε_A	Blue wires
ε_B	Green wires
ε_C	Red wires

Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with 2.5 Kg at some point and record the strains in three directions. Record the load value at the load cell.

Repeat the experiment with load value of 5 Kg. Compute the values of Poisson's ratio from: $\nu = \varepsilon_y/\varepsilon_x$

<u>CAUTION: NEVER EXCEED 10 Kg LOAD ON THE BEAM .</u>

Students may determine stresses using formulas above. MATLAB program is provided at the end of this manual to compute stresses from given strain data.

Tabular Column:

Sl.No	Load (N)	ε_A	ε_B	ε_C	εx	εy	γ_{xy}	$\nu=\varepsilon_y/\varepsilon_x$

Example: A typical data is shown for the purpose of computing shear strain and Poisson's ratio is given below.

Sl.No	load	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}	$\nu=\varepsilon_y/\varepsilon_x$
1	10N	92μ	46μ	-18μ	92μ	-18μ	18μ	0.1957
2	20N	64μ	304μ	-18μ	64μ	-18μ	14μ	0.2188
3	30N	37μ	19μ	-10μ	37μ	-10μ	11μ	0.2973

Example: A typical data is shown for the purpose of computing stresses, principal stresses, and planes of principal stresses.

Sl No	ε_A	ε_B	ε_C	σ_x Mpa	σ_y Mpa	τ_{xy} Mpa	Principle stress σ_1 Mpa	Principle Stress σ_2 Mpa
1	200μ	900μ	1000μ	105.58	230.09	46.69	342.04	-6.371

	Normal, shear stress,& principal stress					Angle of principal stress	
Sl No	σ_x Mpa	σ_y Mpa	τ_{xy} Mpa	σ_1 Mpa	σ_2 Mpa	θ_1	θ_2
1	5.6787	0.8159	0.3879	6.8156	-0.242	64.65	-86.00
2	12.064	1.1000	0.9842	13.299	-0.651	51.45	-85.62
3	17.944	1.2762	1.5695	19.347	-1.273	41.80	-85.04

EXPERIMENT - 5

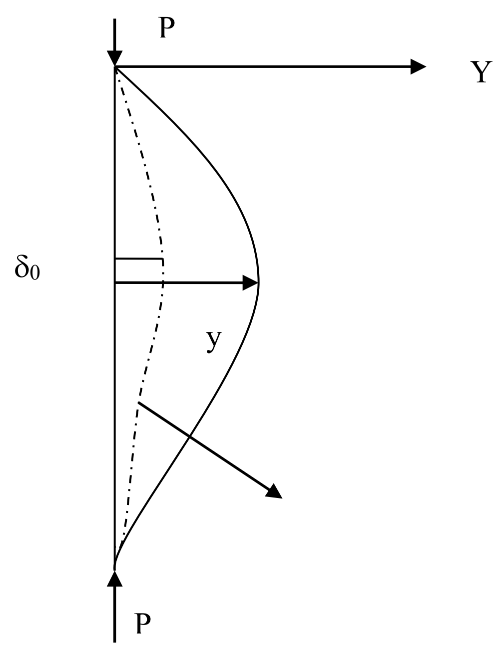
BUCKLING LOAD OF SLENDER ECCENTRIC COLUMNS AND CONSTRUCTION OF SOUTH WELL PLOT

Aim: Practical columns have some imperfections in the form of initial curvature and the buckling of loads of such struts is of real practical value. The experiment aims at measuring the buckling loads of columns and construction of South Well Plot. The imperfection amounts to initial curvature, which shows up in this plot.

Equipment: WAGNER beam set-up, hinged supports, load cells, Long column with initial curvature, mounted dial for deflection measurements.

Theory:

Consider a pin ended strut AB of length L, whose centroidal longitudinal axis is initially curved as shown in fig (1). Under the application of the end load P, the strut will have some additional lateral displacement y at any section.



In this case, Bending Moment at any point is proportional to the change in curvature of the column from its initial bent position y- δ_0. The equation for curvature of column is as follows.

$$M = P(y+y_0) = - EI\, d^2y/dx^2 \qquad \text{... (1)}$$

$P/EI = k^2$

$$\frac{d^2y}{dx^2} + k^2 y = -k^2 y_0 \qquad \text{... (2)}$$

Assuming initial curvature (y_o) to be sinusoidal, satisfying the equation

$$y_0 = \delta_0 \sin(\pi x / L) \qquad \text{... (3)}$$

Where δ_0 equals to the initial displacement at the centre of the strut

The general solution of this differential equation is

$$y = A\cos kx + B\sin kx + \frac{k^2\delta_0}{\pi^2/L^2 - k^2}\sin(\pi in(\pi \qquad \text{... (4)}$$

If the ends are hinged, for the end conditions, then:
y=0 at x =0 and x =L
This results in values of constants: A=B=0.
The resulting equation is as below:

$$y = \frac{k^2\delta_0}{\frac{\pi^2}{L^2} - k^2}\sin\left(\frac{\pi x}{L}\right) \qquad \text{... (5)}$$

Substituting Euler's buckling load (P_e)

$$P_e = \Pi^2 EI / L^2 \text{ and } k^2 = P/EI$$

$$y = \frac{\delta_0}{\frac{Pe}{P} - 1} \sin\left(\frac{\pi x}{L}\right) \quad \text{... (6)}$$

for x=L/2, at center of column, the deflection at center y_c

$$y_c = \frac{\delta_0}{\frac{Pe}{P} - 1} \quad \text{...(7)}$$

The value P_e represents the buckling load for perfectly straight strut. In the relation for deflection (y) , the additional lateral displacement of the strut, that the effect of end load P is to increase (y) by a factor $1/(p_e /p)-1$; shown by equation (6). When P approaches P_e, the additional displacement at mid length of the strut is expressed by eqn. (7).

The load deflection relationship of eqn. (5) is the basis of <u>*South Well plot*</u> technique for extrapolating for the elastic critical load from experimental measurement.

$$\frac{\delta}{\delta 0} = \frac{\frac{P}{Pe}}{1 - \frac{P}{Pe}} \quad \text{... (8)}$$

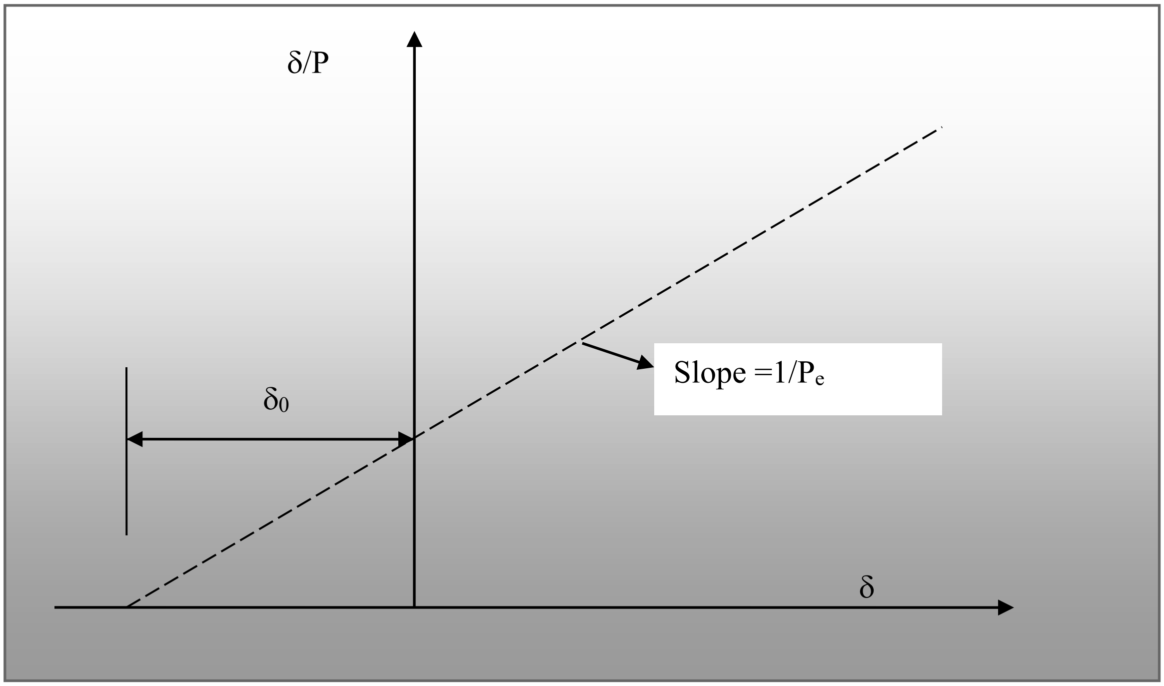
Rearranging the above equation we get

$$\delta = (P_e / P)\ \delta - \delta_0 \quad \text{...(9)}$$

or

$$\delta = (\delta / P)\ P_e - \delta_0$$

The linear relationship between δ/P and δ shown in figure below can be experimentally determined. Thus if a straight line is drawn which best fits the points determined from the experimental measurements of P and δ , the reciprocal of the slope of this line gives an estimate of the magnitude of δ_0 of the initial curvature that can be determined from the intercept on the horizontal axis.



South Well Plot

Procedure:

1. Set up the two hinge supports on the WAGNER beam at the top and bottom supports. Fix the column in the supports.
2. Set the load reading to zero in load cell. Determine the center of column.
3. Set –up the deflection dial gage for reading the column deflections at the center of column. Set the deflection dial gage reading to zero.
4. Apply the vertical load in steps of 5 Kgs. each, in four steps (5Kg, 10Kg, 15Kg, and 20Kg) and record the deflections at each step of load.

Tabular Column:

Load (P) Kgs	Deflection (δ) mm	δ/P

Draw South Well Plot (δ/P vrs δ) .

Determine the slope and estimate P_e. Find out the initial deflection of column.

CAUTION: NEVER EXCEED 20 Kg OF LOAD ON THE COLUMN .

Example:1

A slender strut, 1800mm long, and of rectangular section 30mm x 12mm transmits a longitudinal load P acting at the centre of each end. The strut was slightly bent about its minor principal axis before loading. If the P is increased form 500N to 1500N, the deflection at the middle of the length increases by 4mm. Determine the amount of deflection before loading.

Find also the total deflection and the maximum stress when P is 2000 N.

Take $E = 2.15 \times 10^5$ N/mm^2.

$I = 1/12 \times 30 \times 12^3 = 4320$ mm^4

Let $P_e = \Pi^2 EI/L^2 = \Pi^2\, 2.15 \times 10^5 \times 4320/(1800 \times 1800) = 2830$ N

Let δ_1 be the central deflection when P =500 N and let δ_2 be the central deflection when P =1500 N

Substituting in $\delta_c = \dfrac{Pe}{Pe - P} x \delta_0$

$\delta_1 = 2830/(2830-500) \times \delta_0$ ...(1)

$\delta_2 = 2830/(2830-1500) \times \delta_0$ ...(2)

from eqns.1 & 2

$\boldsymbol{\delta_0 = 4.381}$ **mm**

P =2000 N $\delta c = 2830/(2830-2000) \times 4.381 = 14.94$ mm

A = 30 x 12 = 360mm^2 $Z = 1/6 \times 30 \times 12^2$

$Mc = P \times \delta c = 2000 \times 14.94 = 298880$ N-mm

$\sigma 0 = P/A = 2000/360$mm^2 $\sigma b = Mc/Z = 29880/720 = 41.5$ N/mm^2

$\sigma \max = \sigma 0 + \sigma b = 5.55 + 41.5 = 47.05$ N/mm^2

EXPERIMENT - 6

SHEAR FAILURE OF BOLTED AND RIVETED JOINTS

Aim: To determine the ultimate shear stress in a bolt

Equipment: WAGNER beam set-up, bolt for failure analysis, vernier caliper

Theory: Riveted and Bolted connections are common in structural assemblies. Following are modes of failure in riveted joints:

- Tension failure in a plate
- Shearing failure across one or more planes of rivet
- Bearing failure between plate and the rivet
- Plate shear or shear out failure in the plate

In a riveted joint, the rivets may themselves fail in shear. The tendency is to cut through the rivet across the section lying in the plane between the plates it connects.
If the load is transmitted through bearing between the plate and the shank of the rivet producing shear in the rivet, the rivet is said to be in shear.

When the load is transmitted by shear in only one section of the rivet, the rivet is said to be in single shear. When the loading of the rivet is such as to have the load transmitted in two shear planes, the rivet is said to be in double shear. When load is transmitted in more than two planes, the rivet is said to be in multiple shear.
Rivets and bolts subjected to both shear and axial tension shall be so proportioned that the calculated shear and axial tension do not exceed the allowable stresses τ_{uf} and σ_{tf} and the flowing expression does not exceed a specified value.

$$(\tau_{uf,cal} / \tau_{uf} + \sigma_{tf,cal} / \sigma_{tf}) \text{------------------------------------} (1)$$

Shearing failure of the rivet: In a riveted joint, the rivets may themselves fail in shear. The tendency is to cut through the rivet across the section lying in the plane between the plates it connects.

In analysing this possible manner of failure, one must always note whether a rivet acts in single shear or double shear. In the latter case, the two cross-sectional areas of the same rivet resist the applied force. The shearing stress is assumed to be *uniformly distributed* over the cross-section of the rivet.

Let P_{us} = pull required, per pitch length, for shear failure

f_s = ultimate shear strength of the rivet material

d = gross diameter of the bolt

Resisting area of the rivet section = $(\pi/4)\, d^2$ in single shear

and = $2 \times (\pi/4)\, d^2$ in double shear

$P_{us} = (\pi/4)\, d^2\ f_s$ for single shear

$P_{us} = 2 \times (\pi/4)\, d^2\ f_s$ for double shear

Procedure:

1. Set-up the WAGNER beam test set-up.
2. Note the diameter of the bolt.
3. Place a bolt in the slot.
4. Set the load cell to read zero.
5. Apply the load gradually.
6. Note the load reading at the point of shear failure of mild steel bold.

Calculate the ultimate shear stress from the formula below:

$$f_s = P_{us} / [2 \times (\pi/4)\, d^2] \quad \text{----------------------------------} (2)$$

CAUTION: NEVER EXCEED 3 TONS OF LOAD ON THE BEAM .

EXPERIMENT - 7

BENDING MODULUS OF A SANDWICH BEAM

Aim: To determine the Modulus in Bending of a cantilever sandwich beam

Equipment: Beam Test Set-up, load cells, Sandwich Beam of symmetrical section with strain gage installed. Strain measuring device.

Theory: Beams that are made of more than one material are called `composite beams`. Sandwich beam is one such example. It consists of following:

- Two thin layers of strong material, called faces, placed at top and bottom.
- Thick core, consisting of light weight, low strength material. The core simply serves as a filler or spacer. Sandwich construction is used where light weight combined with high strength and high stiffness is needed.

If both the materials are rigidly joined together, they will behave like a unit piece and the bending will take place about the combined axis. Such composite beams can be analysed by the same bending theory that is applicable for beam of single material.

On the other hand, if both the materials have been simply placed one above the other, they will bend about their respective axes. However, in both the cases, the total amount of moment of resistance will be equal to the sum of moments of resistance of individual sections.

The position of neutral axis may not be the centroid of the section. The criterion of strain compatibility has to be used, i.e. strain in two materials, at a given vertical distance from the neutral axis, has to be same.

A cantilever sandwich beam provided with strain gauge can be used to do experimental assessment of strains. The strain gauge is at a fixed position in the beam and load position can be varied. A strain gauge is mounted on a free surface, which in general, is in a state of plane stress where the state of stress is with regards to a specific xy rectangular rosette. Consider the three element rectangular rosette shown in fig-2, which provides normal strain components in three directions spaced at angles of 45^0. If an xy coordinate system is assumed to coincide with the gauge A and C then $\varepsilon x = \varepsilon_A$ and $\varepsilon y = \varepsilon_C$. Gauge B provides information necessary to determine γ_{xy}. Once εx, εy and γ_{xy} are known, then Hooke's law can be used to determine σx, σ_y, and τ_{xy}. However in this case the requirement is to determine Modulus in bending (E). an alternative approach is to measure deflection under lad condition

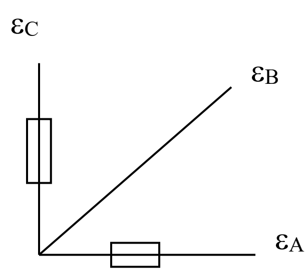


Fig-2

$$\sigma_x = \frac{E}{(1-\nu^2)}(\varepsilon_x + \nu\varepsilon_y); \quad \sigma_y = \frac{E}{(1-\nu^2)}(\varepsilon_y + \nu\varepsilon_x); \quad \tau_{xy} = \frac{E}{2(1+\nu\ )}\gamma_{xy}$$

$$\varepsilon x = \varepsilon_A; \quad \varepsilon_y = \varepsilon_C; \quad \gamma_{xy} = 2\,\varepsilon_B - \varepsilon_A - \varepsilon_C;$$

Sl.No	Load (N)	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}	$\nu=\varepsilon_y/\varepsilon_x$

Also, $\sigma_x = M/Z$, **Z =** $I/\ Y$ (1)

$\sigma_2 = (M - \sigma_1 Z_1) / Z_2$ (2)

$\sigma_2 = m\,\sigma_1$, $m = E_2 / E_1$ (3)

Suffix `1` & `2` refer two different sections of beam

Procedure: Mount the cantilever beam at the left support of beam test set-up. Connect the strain gauges wires with the strain measuring equipment. Use the following color codes

ε_A	Blue wires
ε_B	Green wires
ε_C	Red wires

Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with 2.5 Kg at some point and record the strains in three directions. Record the load value at the load cell.

Repeat the experiment with load value of 5 Kg. Compute the values of Poisson's ratio from:

$$\nu=\varepsilon_y/\varepsilon_x$$

Tabular Column:

Sl.No	Load (N)	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}	$\nu=\varepsilon_y/\varepsilon_x$

Find Modulus in bending through formulas above.

<u>CAUTION: NEVER EXCEED 15 Kg LOAD ON THE BEAM.</u>

EXPERIMENT - 8

VERIFICATION OF SUPERPOSITION THEOREM

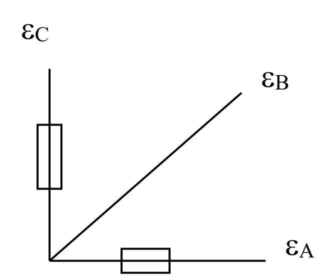
Aim: To verify the theorem of superposition

Equipment: Beam Test Set-Up with multiple loading capability (atleast two load points required), atleast two load cells, cantilever strain gauged beam, strain measuring equipment.

Theory:

Many times, a structural member is subjected to a number of forces acting not only at the ends, but also at the intermediate points along its length. Such a member can be analyzed by the application of the principal of superposition, the resulting strain will be equal to the algebraic sum of the strains caused by individual forces acting along the length of member.

The strain gauge is at a fixed position in the beam and load position can be varied. A strain gauge is mounted on a free surface, which in general, is in a state of plane stress where the state of stress is with regards to a specific xy rectangular rosette. Consider the three element rectangular rosette shown in fig-2, which provides normal strain components in three directions spaced at angles of 45^0. If an xy coordinate system is assumed to coincide with the gauge A and C then $\varepsilon x = \varepsilon_A$ and $\varepsilon y = \varepsilon c$. Gauge B provides information necessary to determine γ_{xy}.



$\varepsilon_x = \varepsilon_A$; $\varepsilon_y = \varepsilon_c$; $\gamma_{xy} = 2\,\varepsilon_B - \varepsilon_A - \varepsilon_c$

Sl.No	Load (N)	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}

Procedure:

1. Mount the cantilever beam at the left support of beam test set-up. Connect the strain gauges wires with the strain measuring equipment. Use the following color codes:

ε_A Blue wires

ε_B Green wires

ε_C Red wires

2. Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with 3.0 Kg at some point from vertical and record the strains in three directions. Record the load value at the load cell. Record the point of loading. **Remove this load.**

3. Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with load from horizontal direction. Apply, Load from right side end with load value of 2.0 Kg. , and record the strains in three directions. Record the load value. **Remove this load.**

4. Set the load cell to read `zero` value in the absence of load. Set the three strains to read `zero` in the absence of load. Now Load the beam with 3.0 Kg at <u>same point</u> from vertical as done earlier.
5. In addition load the beam with load from horizontal direction. Apply, Load from right side end with load value of 2.0 Kg.
6. Record the strains in three directions. Record the load values in the two load cells. Record the vertical load position.

Compute the values of γ_{xy} from the formula:

$$\gamma_{xy} = 2\,\varepsilon_B - \varepsilon_A - \varepsilon_c$$

<u>CAUTION: NEVER EXCEED 10 Kg LOAD ON THE BEAM.</u>

Tabular Column:

Table I: Strains due to Vertical Load

Sl.No	Vertical Load (N)	Load position mm	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}

Table II: Strains due to Horizontal Load

Sl.No	Horizontal Load (N)	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}

Table III: Strains due to combined loading

Sl.No	Vertical Load (N)	Horizontal load (N)	Vertical Load position mm	ε_A	ε_B	ε_C	ε_x	ε_y	γ_{xy}

<u>Note- The vertical load position should read same in Table I and Table III</u>
<u>Load values should read same in all Tables</u>

Add the values of strains in table I and table II and compare with values in Table III.

Verify the two values should be same, hence the proof of superposition theorem

Conclusion:

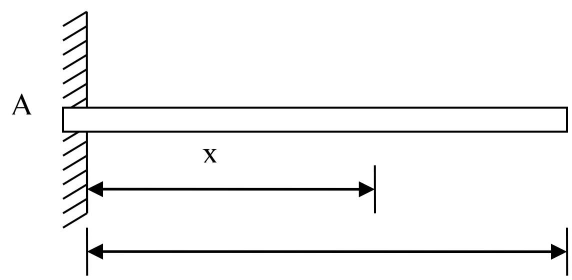
EXPERIMENT – 9

Determination of fundamental frequency of a cantilever beam and harmonics.

Objective: To determine the lateral or transverse vibration of a cantilever beam when the beam is fixed at one end and free at the other end.

Introduction:

A beam which is cantilevered of span L with uniform mass w/g per unit run is shown in figure-1and fixed at one end. Assume a deflection function and obtain the first approximation for the fundamental frequency with the origin at the free end.



Cantilever beam

L is the span of the beam in mm
E is the Young's Modulus of the in N/mm^2
I $=bd^3/12$ is the Moment of Inertia of the beam mm^4
w = Weight per unit length
g = Acceleration due to gravity

The fundamental frequency of the cantilever beam
$\omega^2 = 12.39\ E\ I / wl^4$
The natural frequency for the transverse vibration of a uniform beam fixed at one end and free at the other end.
The four roots of the equation are:
$\beta 1\ L = 1.8751$
$\beta 2\ L = 4.6941$

$\beta 3$ L= 7.8548
$\beta 4$ L= 10.9955

$$\omega n = n^2 \pi^2 \sqrt{\frac{EI}{\rho A L^4}} \quad \text{.... (1)}$$

Example: Determine the three lowest natural frequency for the system shown in fig-1
Given m=10kg,E=200x10^9 N/m2; ρ =7800 kg/m3 ; A =2.6 x 10-3 m2 ; L=1m ; I= 4.7 x10^{-6} m4.

$$\omega n = 1^2 \pi^2 \sqrt{\frac{200x10^9 \, x4.7x10^{-6}}{7800x2.6x10^{-3} x1^4}} \quad =215.3\ \phi^2$$

$\phi 1$=1.423 ; $\phi 2$=4.113 ; $\phi 3$=7.192

$\omega 1$ = 486.1 rad/ s ; $\omega 2$ = 3642 rad/s ; $\omega 3$ = 1.114 x 10^4 rad/s.

EXPERIMENT-10

Frequency spectrum analysis for a cantilever beam

Objective:

To find first few natural frequencies of a cantilever by impact test and observe the Frequency spectrum analysis using FFT analyzer

Introduction:

A. Background:

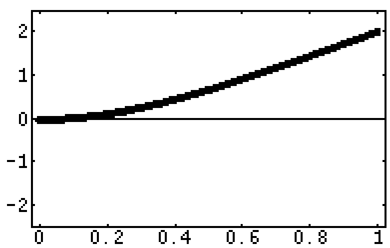
a. Beams and Cantilever Beams are structural members that have smaller dimensions of cross sections compared to its length (its axis) and are subjected to loads perpendicular to its axis; i.e. they are subjected to transverse loads. The whole beam deforms in the plane containing the axis and the transverse loads. We say that the beam bends. The beams are usually supported at both ends and they are termed differently depending on the support conditions. When one end of a beam is fixed, and the other free, it is called a Cantilever beam, or simply a Cantilever. When both end-supports re simple, the beam is called a Simply Supported Beam. If both ends of a beam are fixed, it is a Fixed-Fixed Beam or simply a Fixed Beam.

b. Physical systems that can be modelled as cantilever: The diving board on a swimming pool, the slab on a porch, wall mounted structures, overhanging booms of cranes, etc can be modelled as cantilever. These physical systems can be idealized with loss of some accuracy and generalization but ability and simplicity of analysis. The vibration characteristics of these systems can be very well understood by knowing the vibrations of its cantilever model.

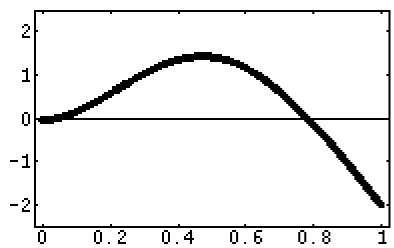
As explained in the general theory, the characteristics of natural vibration are extremely important in knowing the response of the systems to forced excitations.

In this experiment, we shall find out the natural frequencies of a cantilever from its response to harmonic support-excitation.

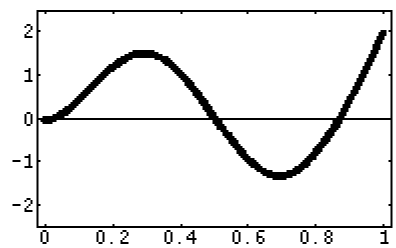
c. Natural Vibration of a Cantilever - Natural frequencies and mode shapes A Cantilever is a continuous system-its mass and elasticity are distributed all over its volume. It can be considered to have infinite very small masses connected by infinite very small springs resist the banding of the Cantilevers. Hence there are infinite degrees of freedom and infinite natural frequencies. And also, corresponding to every natural frequency, it has a particular shape of vibration, called Mode Shape. The lowest natural frequency is called Fundamental natural frequency and corresponding mode, fundamental mode or simply the first mode. Here are animations for first three modes of vibration of a Cantilever. Click on the figures to see the modes.



Mode - I



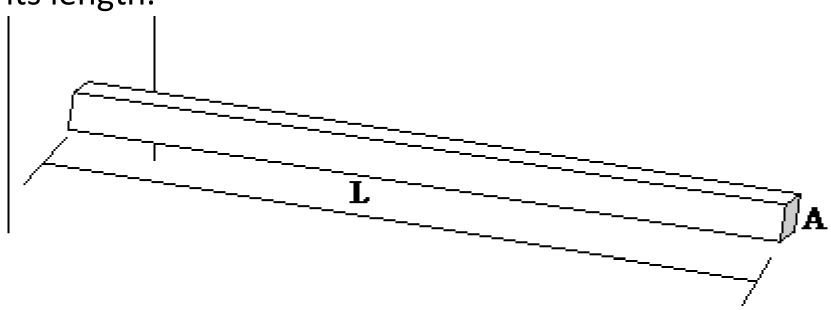
Mode - II



Mode - III

A cantilever of rectangular cross section bxd; Area of cross-section, A = bxd; and length

L is shown in the figure. Cross-sectional dimensions are small compared to its length.



Let us consider its natural vibration in vertical plane, perpendicular to its length L. Let I be the second moment of the area of cross section about neutral axis perpendicular to the plane of vibration;

$$I = \frac{1}{12} bd^3$$

Let E be the modulus of elasticity of the material from which the cantilever is made. For steel value E is taken as E = 210GPa (210x109 N/m^2) and for Aluminium, it is 70 GPa Let ρ be the density of the material; for steel, ρ = 7800 kg/m^3; for Aluminium, ρ = kg/m^3

d. Equation of Motion Once disturbed from its position of equilibrium and left to its own, the cantilever will vibrate naturally; it will perform natural vibration. From theory, we know that the vibration of a cantilever is governed by the equation

$$EI\frac{\partial^4 v}{\partial x^4}+\rho A\frac{\partial^2 v}{\partial t^2}=0$$

at x = 0 (i.e. at fixed end): Deflection v(0,t) = 0 and Slope dv(0,t)/dx = 0, at all t and at x = L (i.e. at free end), Bending moment

$$EI\frac{\partial^2 v}{\partial x^2}=0$$

and Shear force

$$EI\frac{\partial^3 v}{\partial x^3}=0$$

The initial condition is v(x,0) = 0 As both, boundary conditions and initial conditions, are specified, the problem is said to be of mixed initial value and boundary value problem. For small amplitudes of vibration of the cantilever, the motion can be assumed to be harmonic and we can write this equation in terms of amplitude of vibration as a function of x alone. The equation is as follows:

$$\frac{d^4V(x)}{dx^4}-\frac{\rho A\omega^2}{EI}V(x)=0$$

and Shear force

$$\frac{d^4V(x)}{dx^4}-\lambda V(x)=0$$

There are infinite sets of V(x) and λ which together satisfy the above equation. Such problems are called Eigenvalue problems and the solutions are called eigenvalues λi, and eigenvectors V(x)i. V(x) is function of x that shows shape of the cantilever (Amplitudes of vibration at different values of x) corresponding to the respective frequencies of natural vibration λi. The shape of cantilever vibrating with certain natural frequency is called mode shape of cantilever for that frequency. Three of them were shown in figures earlier.

e. Impact test, excited modes and natural frequencies

When a cantilever is given an impact at some point, it is set into vibration. In general, the cantilever will not vibrate in any one single natural mode with corresponding single natural frequency of vibration. Rather, number of modes will participate in its vibration depending on the point of impact with corresponding natural frequencies as components of the periodic vibration. In an impact test, an accelerator is fitted at some point on the cantilever and the cantilever is hit with an impact hammer giving an impulse to the cantilever. FFT analysis of the signal received from the load cell fitted at the tip of the impact hammer reveals that it contains all frequencies over a range. Similarly, the FFT analysis of the signal will also reveal that it contains all those frequencies but amplitude of vibration corresponding to the natural frequencies will be high. Exactly this phenomenon is used to identify the natural frequencies of the cantilever by impact test. Ratio of signals received from the accelerometer and that of the impact hammer is taken in frequency domain which is called FRF (Frequency Response Function). The accelerometer is fixed at one point on the cantilever and impulse is given at predetermined points with the impulse hammer. Corresponding FRFs are computed using the software of modal analysis. Using techniques of curve fitting, Modal Identification Function is generated that shows peaks at the natural frequencies with the selected range of frequencies. And thus the natural frequencies of the cantilever are found by the impact test.

Vibratory systems around us

Here are some examples of physical systems where the vibrations are prominent and can be observed easily. In musical instruments the vibrations are intentional. The parts of musical instruments are designed so that they generate sounds that are pleasant to listen. In many cases the vibrations are unwanted and we try to minimize them.

1. A chandelier hanging from ceiling oscillates to and fro following an initial disturbance; maybe due to a breeze of air.

2. The oscillations of the chandelier at cathedral of Pisa, Italy, were studied by the famous scientist Galileo Galilee.
3. A load attached at end of a wire-rope of a crane oscillates to and fro due to initial disturbance; maybe due to sudden stopping of carriage of the crane while revolving about the vertical axis.
4. The pendulum used in clock of olden days used to oscillate to and fro once every second. i.e. it had a period of oscillation of one second.
5. String of a guitar, when plucked and left to its own, vibrates and makes a musical sound. It comes to rest after a while; the vibrations die out. Similarly, the diaphragm of a table vibrates when hit and left to its own. It also comes to rest after some time.

All these are examples of vibratory systems that are set into vibration following an initial disturbance. All these systems have three components: mass, due to which the system possesses inertia; elasticity, due to which potential energy can be stored; and components that dissipate energy causing the vibratory motion to be damped which bring them to rest after some time.

Vibration or vibratory systems are classified in number of ways. Some of the classifications are given below:

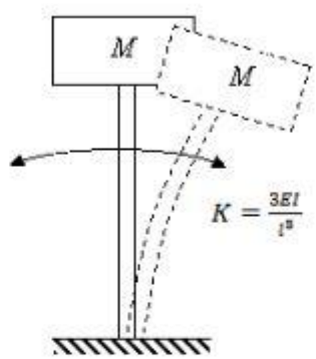
Free and forced vibration - A free vibration occurs due to initial displacement or velocity, or both, applied to the system only initially. There is no external force acting on the system when the system is vibrating. A forced vibration occurs when the system vibrates in response to external force applied continuously. When the force applied is periodic, i.e. it repeats itself after a fixed interval of time, the forced vibration is called periodic. If the periodic force and hence the resulting vibration varies sinusoidally with respect to time, the vibration is called harmonic. If the force is not periodic, the forced vibration is called aperiodic or random.

Damped and undamped vibration - When the vibratory system has elements that offer resistance to motion, energy is continuously dissipated and the free vibrations of such systems come to halt after some time. This is called damped vibration and such systems are called damped systems. Forced vibration of a damped system continues as long as the force acts but some of the work done by the external force is lost in overcoming the resistance offered by the damping elements. Systems without damping elements are called undamped systems and their vibrations are called undamped vibrations. All systems in nature have some or the other damping element and their natural vibrations are damped. Hence they come to rest after some time following free vibrations. Nevertheless, we study the vibration of undamped systems because the concepts developed in studying them are useful in analyzing and understanding the phenomena occurring in vibration of damped as well as complicated systems. When the force of resistance offered by a damping element is proportional to velocity of mass of the system, it is termed as viscous damping and the damping element is called a viscous damper. If the force of resistance has a constant value, it is termed as Coulomb damping. Damping due to dry friction shows this kind of behavior. Coulomb damping can occur when the system has components rubbing over each other. There are other types of damping also which shall be discussed later.

Degrees of freedom : The vibratory systems are classified as single-degree-of-freedom systems, Multi-degree-of-freedom-systems or continuous systems. The number of degrees of freedom corresponds to the number of independent co-ordinates required to completely describe the motion of the system. In fact, it is the sum of the possible ways each mass can move independently of other masses. The translation of a mass along the three axes, X, Y and Z, and the three rotations about each of these axes constitute possible ways of motion of a mass. Many times, many of these six motions of a mass are restricted and a mass can have one or two degrees of freedom, i.e. only translation or translation and rotation of a single mass about any one of the axes.

Linear and Non-linear Vibrations: Vibration is said to be linear if the damping force is proportional to velocity, inertia force is proportional to mass, and restoring force is proportional to displacement. If any of this proportionality is not satisfied, the system is said to be non-linear.

Solving engineering problems: Analytical methods are usually applied to models of actual systems. We carry out experiments on models if physical systems are not available for testing. While preparing such models, we exclude superfluous details of the system but include all essential and important features of the actual system. While doing so, we idealize and approximate important behaviour of the system without affecting much the accuracy in predicting the behaviour. The system model so developed provides ease of application of analytical and experimental techniques. Once a satisfactory model is developed, laws of Physics can be applied which give a set of mathematical equations relating the properties and variables of the system. Such a set of mathematical equations is called mathematical model of the system. Solving the set of equations (or a single mathematical equation) provides expression for the system variable in terms of location and time. We call this as 'solution' of the problem. As an illustration of the concepts described above, see the example given below.

Problem statement	To find natural frequency of oscillation of the tower in the direction perpendicular to the vertical axis (i.e. natural frequency of transverse oscillation of the tower) and position of the head at any given time.	
Physical model of the system	 	The physical model for the system under consideration can be as shown in This is the simplest model. Only the mass of the building at the top is considered and it is considered to be concentrated at one point. The mass of the vertical pillar supporting the buiding is neglected and is considered to be a cantilever offering only elesticity. Thus it becomes a single degree of freedom system with single mass and only one way of motion of the mass: translation in direction perpendicular to the vertical axis of the building. Further we

		assume that the amplitude of this motion to be small. The equivalent stiffness of the cantilever is given by $K = \frac{3EI}{l^3}$

. The symbols carry their usual meanings.

Mathematical model

Using Newton's second law of motion, the equation of motion of the mass is written as

$$M\frac{d^2x}{dt^2} + Kx = 0$$

The first term is the inertia force which is equal to mass multiplied by acceleration and the second term is the spring force given by stiffness of the spring multiplied by its elongation or compression. The differential equation is a mathematical model of the system.

General Solution

The solution to the above differential equation is given by

$$x = A\sin\omega_n t + B\cos\omega_n t$$

A and B are constants that depend the initial conditions, i.e. the displacement and velocity of the mass when we started measuring our time.

These are known as initial conditions.

ω_n is the natural frequency in radians per second and is given by

$$\omega_n = \sqrt{\frac{K}{M}}$$

Particular solution obtained from the initial conditions

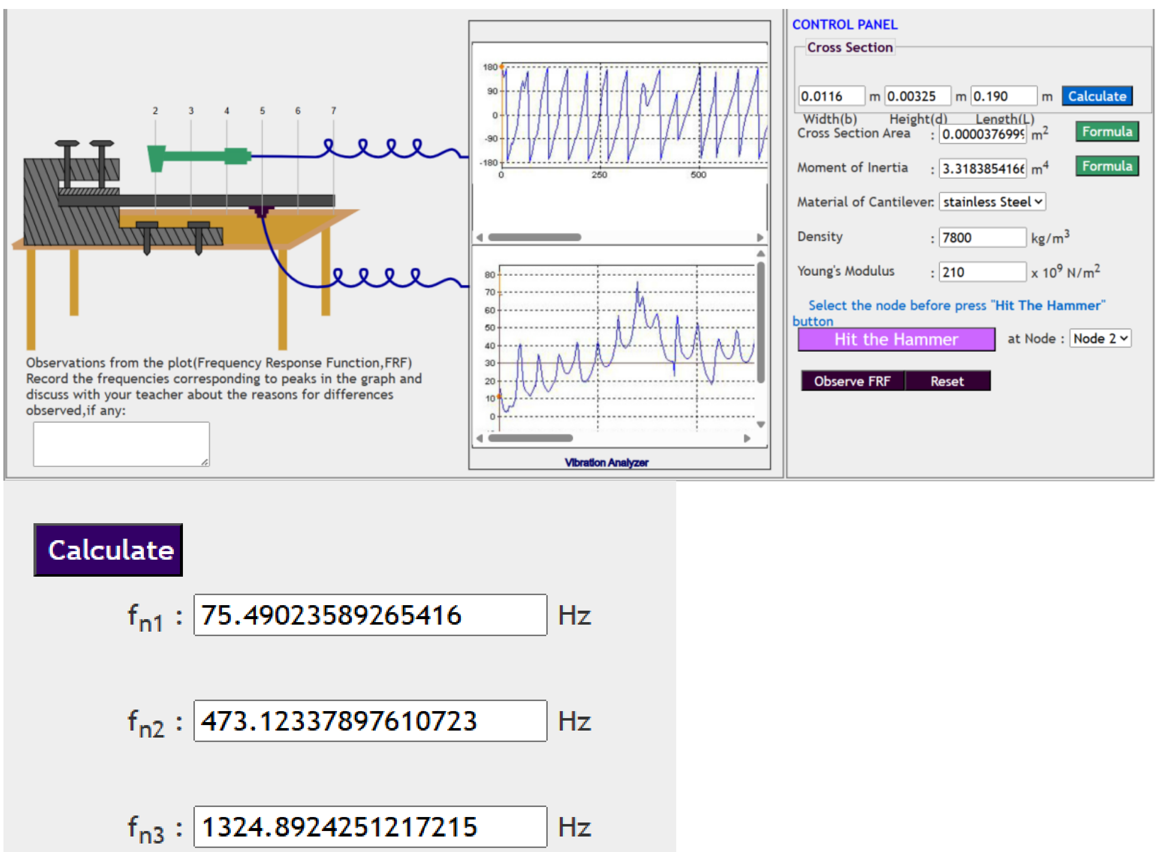
Substituting the initial conditions in above expression, we can obtain the values of A and B. Thus if X_0 and V_0 are the initial displacement and velocity, respectively, given to the mass, the above expression will yield values of A and B as $A = \frac{V_0}{\omega_n}$ and

$$B = X_0$$

Now the expression for x becomes

$$x = \frac{V_0}{\omega_n} \sin\omega_n t + X_0 \cos \omega_n t$$

And we can obtain the value of x at any time t from this expression.
Thus, we have obtained the expressions for natural frequency and position of the head at any given time 't' and the problem stated by the problem statement is solved.



https://va-coep.vlabs.ac.in/exp/cantilever-modal-analysis/simulation.html

use this link for simulation

EXPERIMENT-11

Vibration induced structural damage studies

Aim:

To create the model of a cantilever, generate the mesh with default global mesh control settings and find six natural frequencies and their respective mode shapes. The material used is Structural Steel.

The following steps are required to complete this Experiment:

a. Create a new project.
b. Create the model.
c. Generate the mesh
d. Specify the boundary conditions.
e Solve the analysis.
f. Retrieve the analysis results.
g. Play the animation.
h. Save the model.

Creating a New Project

1. Start ANSYS Workbench session and then add the Modal analysis system to the Project Schematic window.

2. In the Project Schematic window, rename the Modal analysis system to as Cantilever.

3. Choose the Save button from the Standard toolbar; the Save As dialog box is displayed. Save

4. Create a new folder with the name c10 at the location C: ANSYS_WB. Open the c10 folder and then create another folder in it with the name Tut01.

5. In this folder, save the project with the name c10_ansWB_tut01.
Creating the model

After creating the project, you now need to work in the Design Modeler to create the model.

1. Double click on the geometry cell; the Design Modeler window along with the ANSYS Workbench dialog box is displayed.

2. Select the Millimeter radio button in the ANSYS Workbench dialog box and then choose the OK button to specify millimeter as the unit for creating the sketch.

3. In the Design Modeler window, select XY Plane from the Tree Outline to specify it as the sketching plane. Next, orient the view normal to the viewing direction.

4. Invoke the Sketching mode. Next, create a rectangle and then dimension

5. Extrude the sketch to a depth of 150 mm.

Generating the Mesh for the Model

Now, you need to generate the mesh of the model.

1. Double-click on the Model cell in the Cantilever analysis system and wait for some time; the Mechanical window is displayed. Also, you will notice that in the Outline window, the Mesh node is displayed in the Tree Outline with a yellow thunderbolt attached to it.

2. Click on Mesh in the Tree Outline; the Details of "Mesh" window is displayed.

3. In the Details of "Mesh" window, expand the Sizing node, if not already expanded.

4.In the Sizing node in the Details of "Mesh" window, enter 2.5 in the Element Size edit box.

5. Right-click on Mesh in the Tree Outline and then choose the Preview > Surface Mesh from the shortcut menu displayed; the preview of the mesh for the model is displayed.

6. Choose the Generate Mesh tool from the Mesh drop-down in the Mesh Generate Mesh contextual toolbar; the mesh is generated,

Setting the Boundary Conditions

1. After the mesh is generated, you need to set the boundary conditions under which the analysis is to be performed.

2. Right-click on Modal node in the Tree Outline and then choose Insert > Fixed Support from the shortcut menu displayed; Fixed Support with a question symbol is added under the Modal node in the Tree Outline. Also, the Details of "Fixed Support" window is displayed.

3. In the Details of "Fixed Support" window, click on the Geometry cell to display the Apply and Cancel buttons, if not already displayed.

4. Select the side face of the model,

5. Next, choose the Apply button from the geometry selection box in the Details of "Fixed Support" window; Fixed support is applied to the selected face.

Solving the Modal Analysis

After specifying the boundary conditions in the Mechanical window, you need to set the variables to define the results and solve the analysis.

1. Select Analysis Settings under the Modal node in the Tree Outline; the Details of "Analysis

2.Settings" window is displayed.

3.In the Details of "Analysis Settings" window, expand the Options node, if it is not already expanded. Enter 6 in the Max Modes to Find edit box, if not already specified by default. Also make sure that No is selected in the Limit Search to Range drop-down list,

4. Right-click on the Solution node in the Tree Outline and then choose the Solve option from the shortcut menu displayed; the analysis is solved.

5 Select the Solution node in the Tree Outline; the Graph and Tabular Data windows are displayed,

Retrieving Analysis Results After the analysis is solved,

1. Right-click in the Graph window, a shortcut menu is displayed.

2. Choose Select All from this shortcut menu to select all the data available in the Graph window,

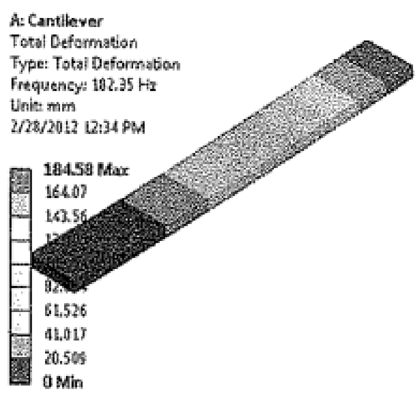
3. After the columns in the Graph window are selected, right-click again to display a shortcut menu. Graph Messages Animatinn10 Frames Message Graph

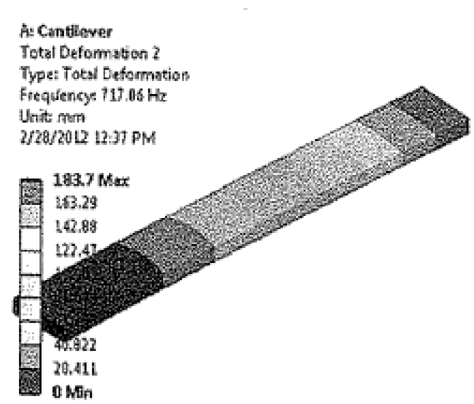
4 Choosing the Create Mode Shape Results from the shortcut menu displayed Choose the Create Mode The number of modes under the Solution node depend upon the value specified in the Max Modes to Find edit box in the Details of "Analysis Settings" window.

5. Right-click on the Solution node again and then choose the Evaluate All Results from the shortcut menu displayed; all the six results are ready to be viewed.

6. Select Total Deformation under the Solution node in the Tree Outline; the first mode is displayed in the Graphics screen, as shown in Figure 10-21.

7. Select Total Deformation 2 under the Solution node; the second mode shape is displayed in the Graphics screen,. The corresponding Legends of the mode shapes are also displayed in the figures.





EXPERIMENT-12

Fault detection and de-lamination studies in composite plate

Aim

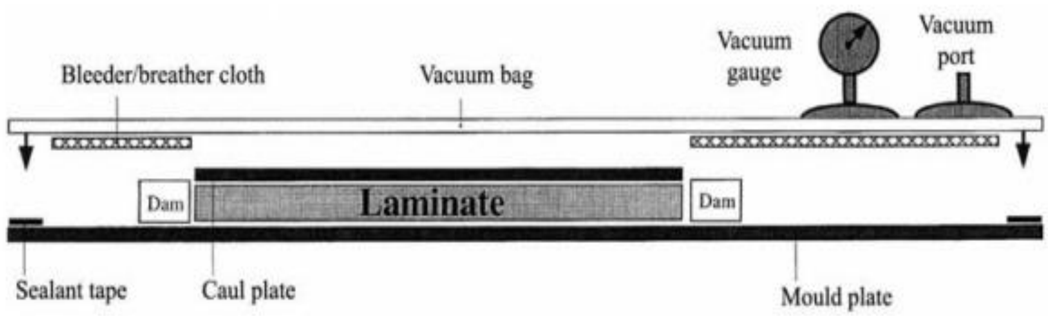
To conduct delamination test and understand modes of failure

Introduction

The goals for any fracture toughness test are to obtain consistent results, to simulate in-situ cracking and to give a fracture toughness value that can reliably characterize tested materials for structural applications. In the case of composite materials all these requirements predominantly depend on the successful simulation of a starter crack. The starter crack must represent a site in a structure where crack initiation is most likely to occur. These sites are usually voids and pre-existing cracks that exist due to manufacturing errors and/or fatigue loading during the service life of the component. Since composites are multi-phase materials successful simulation of a starter crack is not always a straightforward task, and various standards exist to represent this. Three major standardization organizations: the American Society for Testing and Materials (ASTM), European Structural Integrity Society (ESIS) and Japanese Industrial Standards Group (JIS), developed and adopted similar standards for mode I

Materials and specimen preparation

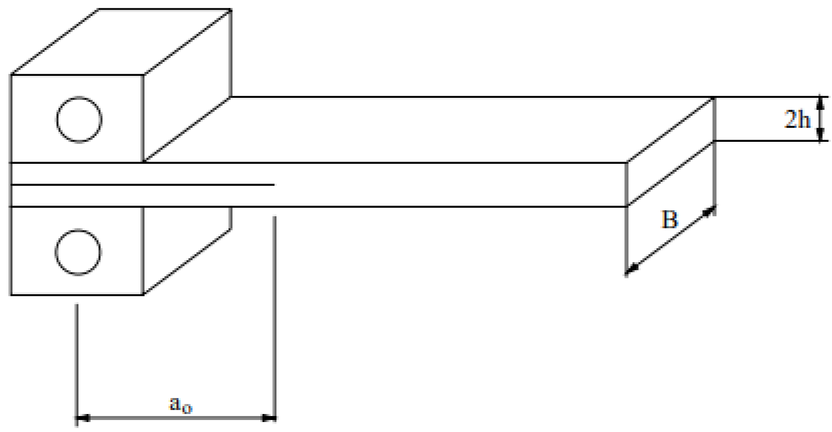
The materials used in this part of the study were Dow Derakane 8084 vinyl-ester resin with unidirectional E-glass (Colan AR106) fiber reinforcement. The fibres were obtained as a woven fabric, and unidirectional fiber plies were made by removing all weft yarns, except two end yarns, which were used for binding. The linear density of the fibers in the warp yarn was 1.2 g/m, so the areal weight of the unidirectional ply was 354 g/m^2. Vinyl-ester resin was mixed with additives which serve to control the speed of the resin solidification process, and thus obtain sufficient time for the manufacturing process. For this purpose 1.5% methyl-ethyl-ketone peroxide (MEKP) was used as a ’promoter’ (supplied VE was pre-promoted by the manufacturer with 0.3% of cobalt naphthenate) together with 0.2% of 2,4pentanedione (2,4P) used as a ’retarder’ of the chemical reaction. Each laminate was fabricated by hand in a wet lay-up. Alternate layers of liquid resin and fiber plies were placed inside a dam on a flat mould plate. At the end of the lay up procedure, a caul plate was placed on top of the laminate to insure uniform thickness. The vacuum bagging technique was than applied to cure the laminate under atmospheric pressure and at room temperature. The manufacturing procedure is illustrated in Figure After an initial room temperature cure in the vacuum bag, each laminate was post-cured at 90◦C for 4 hours, in order to obtain uniform properties for the laminate and complete the curing process.



3.3 Experimental procedures

3.3.1 Mode I testing

Two different test procedures were followed for this study. The first one was as reported in the ASTM and ESIS standards [97, 98], and it required specimens with a foil starting defect and without any pre-cracks. The second one followed a slightly modified JIS standard [99, 100], where specimens had to be pre-cracked. Pre-cracking was performed under fatigue mode I loading until the pre-crack length was between 2 and 5mm, and during this process, the cyclic load applied was kept at around 80% of the load required to initiate a crack in the double cantilever beam (DCB) specimen under static loading. This level of loading was found to be sufficient to produce the desired pre-crack length over an acceptable period of time, without causing undesirable premature fracture. A frequency range of 0.5-0.8 Hz was used with an amplitude between 2 and 4mm. Both the pre-cracking and testing were conducted in displacement control on an Instron 4505 Universal Testing Machine, and the typical DCB specimen geometry is shown in figure



The length of the specimens were 120mm and the width and thickness were 20mm and 5mm, respectively. Loading was applied via the aluminum blocks at the end of the specimens and the cross-head speed was 1mm/min. Crack propagation was monitored using 1 and 5mm guide marks on the side of specimen, and a load-displacement plot provided data for each crack length used for the calculation of GIc, using the experimental compliance method (Berry's method) [98], with the standard correction factors for large displacements being used. The strain energy release rate is given by the expression:

$$G_{Ic} = \frac{nP\delta}{2Ba}$$

where P is the applied load, δ is displacement, B is average width of specimen, a is measured crack length and n is a slope factor calculated from the logarithmic plot of crack length a versus compliance C, using the assumption that the compliance is given by the expression:

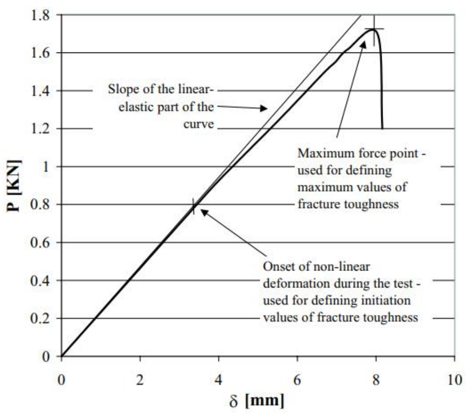
$$\mathbf{C} = K \cdot \mathbf{a}^n \; ; where \; n \leq 3$$

Tabular column:

Sl.no	Load in N	Displacement in mm	Stress in N/mm^2	Strain

Results:

Plot the cure of load vs diagram and stress vs strain



VIVA QUESTIONS

1. Explain Castigliano`s theorem and its verification through any experimental set-up
2. Explain Maxwell`s reciprocal theorem and its verification though experiment
3. What is Betti`s theorem
4. Explain South well plot
5. What are various modes of failures of riveted joints
6. What is the value of maximum loading allowed in the existing experimental beam test set-up
7. What is the value of maximum loading allowed in the existing Wagner beam set-up
8. What is bending modulus
9. What is a sandwich beam
10. What is strain compatibility
11. Explain unit load method for determining the deflection of beams
12. How Young's modulus and Poisson's ratio can be determined from beam test set-up
13. Explain Superposition theorem
14. What are mode shapes and types of modes in vibrations
15. What is a Rosette strain gauge?
16. Explain unsymmetrical bending of beams
17. What is the purpose of Stiffeners, what is the purpose of Longirons
18. What stresses are taken up by the web and the flange of the spars.
19. Explain how bending and torsion is taken up by the wing structure
20. What is a torsion box
21. Explain how bending and torsion is taken up by the fuselage structure
22. What is shear center and how it can be determined through experimental set-up
23. What is Flexibility matrix
24. What is Power Spectral Density
25. What are Principal stress axes
26. What is simple stress, simple strain
27. What is plain stress, plain strain
28. What is a stress Tensor
29. What is a Wagner Beam test set-up? What are Wagner assumptions?
30. Explain the Energy methods used for structural analysis.