Strength of Materials

SYLLABUS

Module – I

1. Definition of stress, stress tensor, normal and shear stresses in axially loaded members.

Stress & Strain :- Stress-strain relationship, Hooke's law, Poisson's ratio, shear stress, shear strain, modulus of rigidity. Relationship between material properties of isotropic materials. Stress-strain diagram for uniaxial loading of ductile and brittle materials. Introduction to mechanical properties of metals-hardness,impact. **Composite Bars In Tension & Compression**:-Temperature stresses in composite rods – statically indeterminate problem. (10)

Module – II

2. **Two Dimensional State of Stress and Strain**: Principal stresses, principal strains and principal axes, calculation of principal stresses from principal strains.Stresses in thin cylinder and thin spherical shells under internal pressure , wire winding of thin cylinders.

3. **Torsion** of solid circular shafts, twisting moment, strength of solid and hollow circular shafts and strength of shafts in combined bending and twisting. (10)

Module – III

4. **Shear Force And Bending Moment Diagram**: For simple beams, support reactions for statically determinant beams, relationship between bending moment and shear force, shear force and bending moment diagrams.

5. **Pure bending** theory of initially straight beams, distribution of normal and shear stress, beams of two materials.

6. Deflection of beams by integration method and area moment method. (12)

Module – IV

7. Closed coiled helical springs.

8. **Buckling of columns :** Euler's theory of initially straight columns with various end conditions, Eccentric loading of columns. Columns with initial curvature. (8)

1

Text Books-:

1. Strength of materials by G. H. Ryder, Mc Millan India Ltd.,
2. Elements of Strength of Materials by S.P. Timoshenko and D.H. Young, East West Press Pvt. Ltd.,

Ref. Books:-

1. Introduction to solid mechanics by H. Shames, Prentice Hall India, New Delhi
2. Engineering mechanics of solid by E. P. Popov, Prentice Hall India, New Delhi
3. Engineering Physical Metallurgy, by Y. Lakhtin, MIR pub, Moscow.

Lecture 1

Stress

Stress is the internal resistance offered by the body to the external load applied to it per unit cross sectional area. Stresses are normal to the plane to which they act and are tensile or compressive in nature.
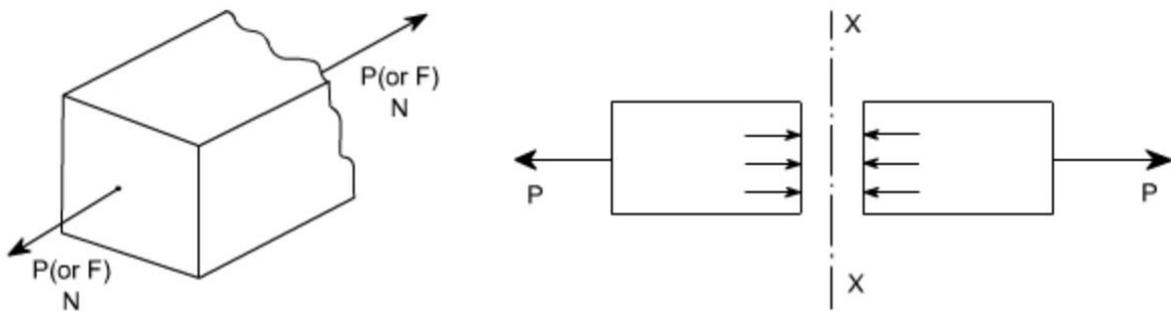
As we know that in mechanics of deformable solids, externally applied forces acts on a body and body suffers a deformation. From equilibrium point of view, this action should be opposed or reacted by internal forces which are set up within the particles of material due to cohesion. These internal forces give rise to a concept of stress. Consider a rectangular rod subjected to axial pull P. Let us imagine that the same rectangular bar is assumed to be cut into two halves at section *XX*. The each portion of this rectangular bar is in equilibrium under the action of load P and the internal forces acting at the section *XX* has been shown.

Now stress is defined as the force intensity or force per unit area. Here we use a symbol to represent the stress.

$$\frac{P}{A} -$$

Where A is the area of the *X−X* section

Here we are using an assumption that the total force or total load carried by the rectangular bar is uniformly distributed over its cross – section. But the stress distributions may be for from uniform, with local regions of high stress known as stress concentrations. If the force carried by a component is not uniformly distributed over its cross – sectional area, A, we must consider a small area, 'δA' which carries a small load 'δP', of the total force 'P', Then definition of stress is

$$\sigma = \frac{\delta F}{\delta A}$$

As a particular stress generally holds true only at a point, therefore it is defined mathematically as

$$\sigma = \lim_{\delta A \to 0} \frac{\delta F}{\delta A}$$

Units :

The basic units of stress in S.I units i.e. (International system) are N/m^2 (or Pa)

$MPa = 10^6 \ Pa$

$GPa = 10^9 \ Pa$

$KPa = 10^3 \ Pa$

Sometimes N/mm^2 units are also used, because this is an equivalent to MPa. While US customary unit is pound per square inch psi.

TYPES OF STRESSES : Only two basic stresses exists : (1) normal stress and (2) shear stress. Other stresses either are similar to these basic stresses or are a combination of this e.g. bending stress is a combination tensile, compressive and shear stresses. Torsional stress, as encountered in twisting of a shaft is a shearing stress. Let us define the normal stresses and shear stresses in the following sections.

Normal stresses : We have defined stress as force per unit area. If the stresses are normal to the areas concerned, then these are termed as normal stresses. The normal stresses are generally denoted by a Greek letter (σ)
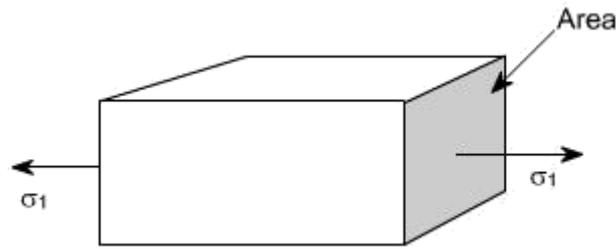
This is also known as uniaxial state of stress, because the stresses acts only in one direction however, such a state rarely exists, therefore we have biaxial and triaxial state of stresses where either the two mutually perpendicular normal stresses acts or three mutually perpendicular normal stresses acts as shown in the figures below :

(Biaxial state of stress)

(Triaxial state of stress)
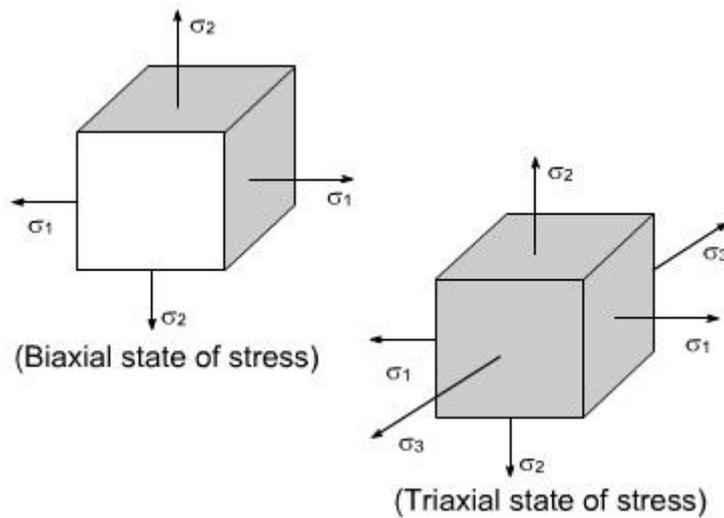
Tensile or compressive Stresses:

The normal stresses can be either tensile or compressive whether the stresses acts out of the area or into the area

(Tensile stress)

(Compressive stress)

8
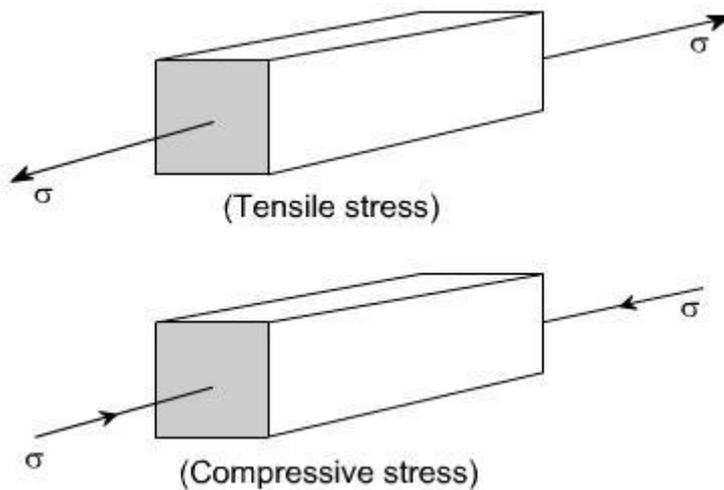
Bearing Stress: When one object presses against another, it is referred to a bearing stress (They are in fact the compressive stresses).

Bearing stresses at the contact surface
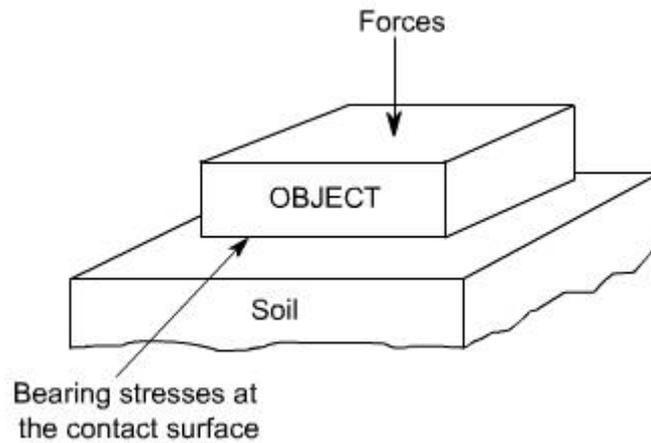
Sign convections for Normal stress

Direct stresses or normal stresses

- tensile +ve
- compressive –ve

Shear Stresses:

Let us consider now the situation, where the cross – sectional area of a block of material is subject to a distribution of forces which are parallel, rather than normal, to the area concerned. Such forces are associated with a shearing of the material, and are referred to as shear forces. The resulting stress is known as shear stress.
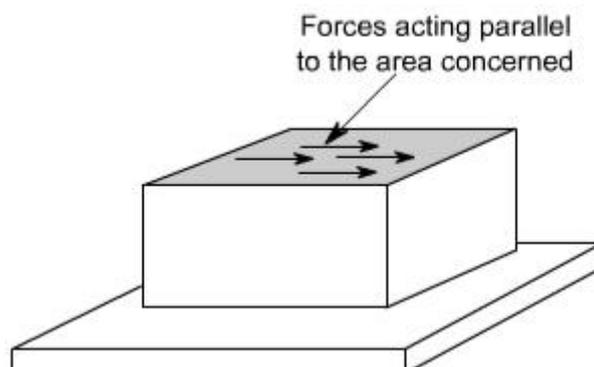
The resulting force intensities are known as shear stresses, the mean shear stress being equal to

$$\tau = \frac{P}{A}$$

Where P is the total force and A the area over which it acts. As we know that the particular stress generally holds good only at a point therefore we can define shear stress at a point as

$$\tau = \lim_{\delta A \to 0} \frac{\delta F}{\delta A}$$

The Greek symbol   (tau, suggesting tangential) is used to denote shear stress.

Complementary shear stresses:

The existence of shear stresses on any two sides of the element induces complementary shear stresses on the other two sides of the element to maintain equilibrium. As shown in the figure the shear stress in sides AB and CD induces a complimentary shear stress ' in sides AD and BC.
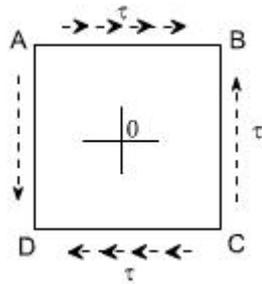
Sign convections for shear stresses:

- tending to turn the element C.W +ve.

- tending to turn the element C.C.W – ve.

Deformation of a Body due to Self Weight

Consider a bar AB hanging freely under its own weight as shown in the figure.
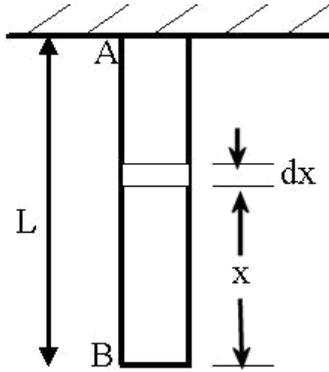
Let

L= length of the bar

A= cross-sectional area of the bar

E= Young's modulus of the bar material

w= specific weight of the bar material

Then deformation due to the self-weight of the bar is

$$L \quad \frac{WL}{2E}$$

Members in Uni – axial state of stress

Introduction: [For members subjected to uniaxial state of stress]

For a prismatic bar loaded in tension by an axial force P, the elongation of the bar can be determined as

$$\delta = \frac{PL}{AE} \quad \text{------- (1)}$$
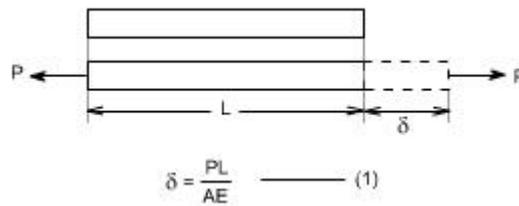
Suppose the bar is loaded at one or more intermediate positions, then equation (1) can be readily adapted to handle this situation, i.e. we can determine the axial force in each part of the bar i.e. parts AB, BC, CD, and calculate the elongation or shortening of each part separately, finally, these changes in lengths can be added algebraically to obtain the total charge in length of the entire bar.

11
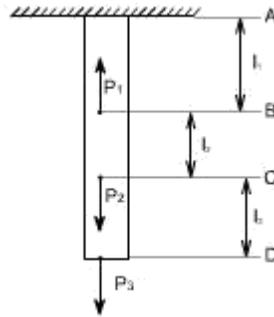
When either the axial force or the cross – sectional area varies continuosly along the axis of the bar, then equation (1) is no longer suitable. Instead, the elongation can be found by considering a deferential element of a bar and then the equation (1) becomes

$$d\delta = \frac{P_x \, dx}{E.A_x}$$

$$\delta = \int_0^l \frac{P_x \, dx}{E.A_x}$$

i.e. the axial force P_x and area of the cross – section A_x must be expressed as functions of x. If the expressions for P_x and A_x are not too complicated, the integral can be evaluated analytically, otherwise Numerical methods or techniques can be used to evaluate these integrals.

Principle of Superposition

The principle of superposition states that when there are numbers of loads are acting together on an elastic material, the resultant strain will be the sum of individual strains caused by each load acting separately.

Lecture 2: Numerical Problems on stress, shear stress in axially loaded members.

Example 1: Now let us for example take a case when the bar tapers uniformly from d at x = 0 to D at x = l
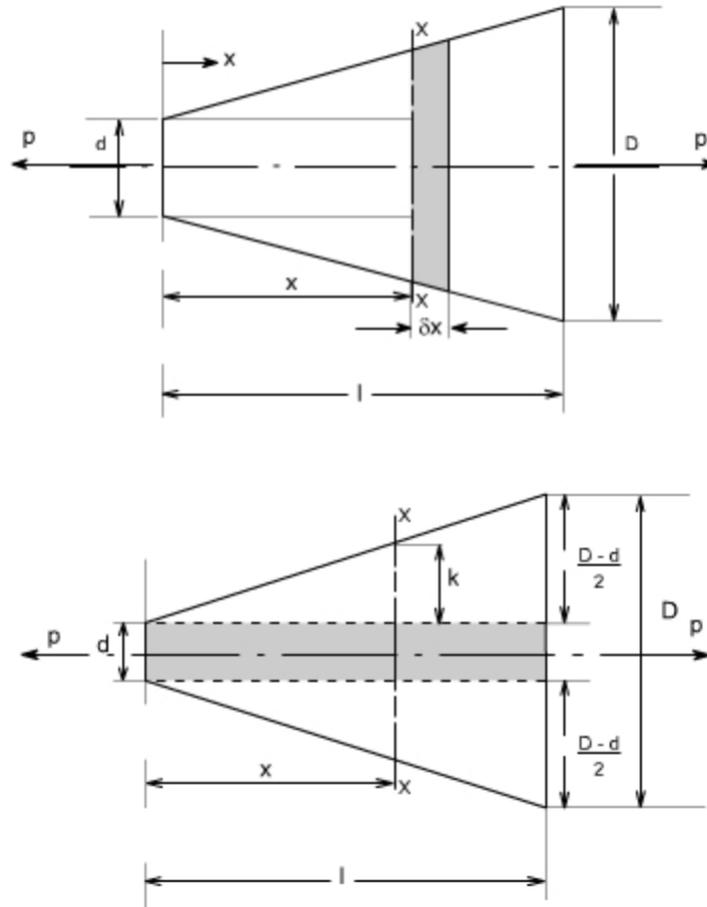
In order to compute the value of diameter of a bar at a chosen location let us determine the value of dimension k, from similar triangles

$$\frac{(D-d)/2}{l} = \frac{k}{x}$$

$$Thus, k = \frac{(D-d)x}{2l}$$

therefore, the diameter 'y' at the X-section

is or = d + 2k

$$y = d + \frac{(D-d)x}{l}$$

Hence the cross –section area at section X- X will be

$$A_x \text{ or a} = \frac{\pi}{4} y^2$$

$$= \frac{\pi}{4} \left[d + (D - d) \frac{x}{l} \right]^2$$

hence the total extension of the bar will be given by expression

$$= \frac{P}{E} \int_0^l \frac{\delta x}{a}$$

subsitituting the value of 'a' to get the
total extention of the bar

$$= \frac{\pi P}{4E} \int_0^l \frac{\delta x}{\left[d + (D - d) \frac{x}{l} \right]^2}$$

after carrying out the intergration we get

$$= -\frac{4.P.l}{\pi E} \left[\frac{1}{D} - \frac{1}{d} \right]$$

$$= \frac{4.P.l}{\pi E D.d}$$

hence the total strain in the bar $= \dfrac{4.P.l}{\pi E D.d}$

An interesting problem is to determine the shape of a bar which would have a uniform stress in it under the action of its own weight and a load P.

Example 2: stresses in Non – Uniform bars

Consider a bar of varying cross section subjected to a tensile force P as shown below.
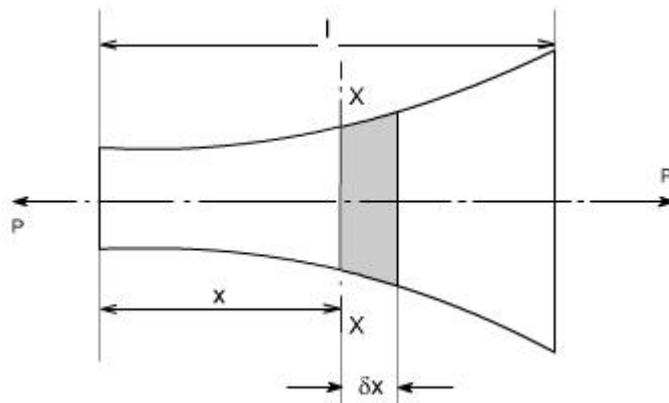
Let

a = cross sectional area of the bar at a chosen section XX

then

Stress $= p / a$

14

If E = Young's modulus of bar then the strain at the section XX can be calculated

= / E

Then the extension of the short element x. = .original length = / E. x

$$= \frac{P}{E} \frac{\delta x}{a}$$

Thus, the extension for the entire bar is

$$\delta = \int_0^l \frac{P}{E} \frac{\delta x}{a}$$

$$\text{or total extension} = \frac{P}{E} \int_0^l \frac{\delta x}{a}$$

let us consider such a bar as shown in the figure below:
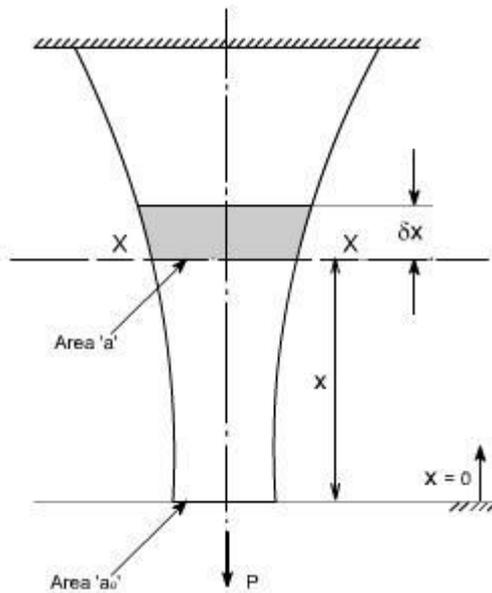
The weight of the bar being supported under section XX is

$$= \int_0^x \rho \, g \, a \, dx$$

where ρ is density of the bar.

thus the stress at XX is

$$\sigma = \frac{P + \int_0^x \rho \, g \, a \, dx}{a}$$

or $\sigma.a = P + \int_0^x \rho.g.a \, dx$

Differentiating the above equation with respect to x we get

$$\sigma.\frac{da}{dx} = \rho.g.a$$

$$\frac{da}{a} = \frac{\rho.g}{\sigma}.dx$$

intergrating the above equation we get

$$\int \frac{da}{a} = \int \frac{\rho.g}{\sigma} dx$$

$$\log_e a = \frac{\rho.g.x}{\sigma} + constant$$

In order to determine the constant of integration

let us apply the boundary conditions

at $x = 0$; $a = a_0$

thus, constant $= \log_e a_0$

or

$$\log_e a = \frac{\rho.g.x}{\sigma} + \log_e a_0$$

$$\log_e \left(\frac{a}{a_0} \right) = \frac{\rho.g.x}{\sigma}$$

or $\boxed{e^{\frac{\rho.g.x}{\sigma}} = \frac{a}{a_0}}$

also at $x = 0$

$$\sigma = \frac{P}{a_0}$$

Thus,

$$\frac{a}{a_0} = e^{\frac{\rho.g.x a_0}{P}}$$

16

Example 1: Calculate the overall change in length of the tapered rod as shown in figure below. It carries a tensile load of 10kN at the free end and at the step change in section a compressive load of 2 MN/m evenly distributed around a circle of 30 mm diameter take the value of E = 208 GN / m^2.

This problem may be solved using the procedure as discussed earlier in this section
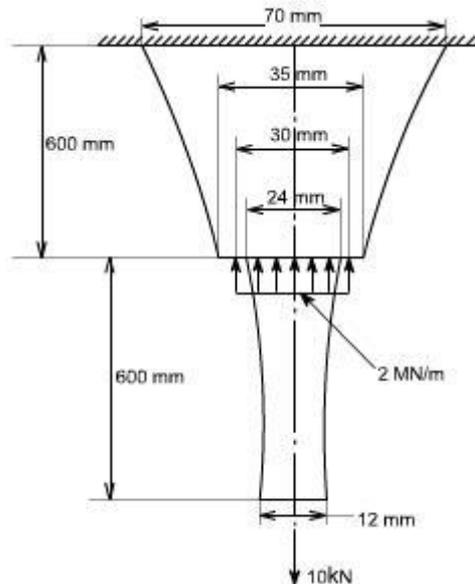
Example 2: A round bar, of length L, tapers uniformly from radius r_1 at one end to radius r_2 at the other. Show that the extension produced by a tensile axial load P

is $\dfrac{PL}{2\pi E r_1^2}$

If $r_2 = 2r_1$, compare this extension with that of a uniform cylindrical bar having a radius equal to the mean radius of the tapered bar.

Solution:

17
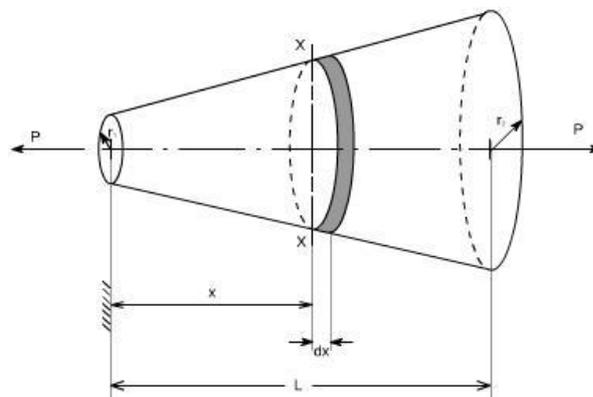
consider the above figure let r_1 be the radius at the smaller end. Then at a X crosssection XX located at a distance x from the smaller end, the value of radius is equal to

$$= r_1 + \frac{x}{L}(r_2 - r_1)$$

$$= r_1(1 + kx)$$

$$\text{where } k = \left(\frac{r_2 - r_1}{L}\right) \cdot \frac{1}{r_1}$$

$$\text{stress at section XX} = \frac{\text{load}}{\text{area}}$$

$$= \frac{P}{\pi r_1^2 (1 + kx)^2}$$

$$\text{hence strain at this section} = \frac{\text{stress}}{E}$$

$$= \frac{P}{E.\pi r_1^2 (1 + kx)^2}$$

Thus, for a small length dx of the bar at this section the extention is $\dfrac{P.dx}{E\pi r_1^2 (1 + kx)^2}$

Total extension of the bar can be found by integrating the above expression within the limits from x=0 to x=L

$$\text{Extension} = \int_0^L \frac{P.dx}{E.\pi r_1^2 (1 + k.x)^2}$$

$$= \frac{P}{E.\pi r_1^2} \int_0^L (1 + k.x)^{-2} dx$$

$$= \frac{P}{E.\pi r_1^2} \left[\frac{(1+kx)^{-1}}{-k}\right]_0^L$$

$$= \frac{P}{E.\pi r_1^2} \left[\frac{(1+kx)^{-1}}{-k} - \frac{1}{-k}\right]$$

$$= \frac{P}{E.\pi r_1^2.k} \left[1 - \frac{1}{1+kL}\right]$$

$$= \frac{PL}{E.\pi r_1^2 (1 + kL)}$$

$$\text{since } k = \frac{(r_2 - r_1)}{r_1 L}$$

$$\text{Thus, } 1 + kL = \frac{r_2}{r_1}$$

$$\text{Therefore, the extension} = \frac{PL}{\pi E r_1 r_2}$$

18

Comparing of extensions

For the case when $r_2 = 2.r_1$, the value of computed extension as above

becomes equal to $\dfrac{PL}{2\pi E r_1^2}$

The mean radius of taper bar

$= 1/2(\, r_1 + r_2\,)$

$= 1/2(\, r_1 + 2\, r_2\,)$

$= 3/2\, .r_1$

Therefore, the extension of uniform bar

= Orginal length . strain

$= L.\dfrac{\sigma}{E}$

$= \dfrac{L}{E}.\dfrac{P}{\pi\left(\frac{3}{2}r_1\right)^2}$

$= \dfrac{4PL}{9\pi E r_1^2}$

hence the

$$\dfrac{Extension\,of\,uniform}{Extension\,of\,tapered} = \left(\dfrac{4PL}{9\pi E r_1^2}\right)\Bigg/\dfrac{PL}{2\pi E r_1^2}$$

$$= \dfrac{8}{9}$$

Strain:

When a single force or a system force acts on a body, it undergoes some deformation. This deformation per unit length is known as strain. Mathematically strain may be defined as deformation per unit length. So,

Strain=Elongation/Original length

Or, $\dfrac{l}{L}$

Elasticity;

The property of material by virtue of which it returns to its original shape and size upon removal of load is known as elasticity.

Hooks Law

It states that within elastic limit stress is proportional to strain. Mathematically

E= $Strain\dfrac{Stress}{}$

Where E = Young's Modulus

Hooks law holds good equally for tension and compression.

Poisson's Ratio;

The ratio lateral strain to longitudinal strain produced by a single stress is known as Poisson's ratio. Symbol used for poisson's ratio is or $1/m$.

Modulus of Elasticity (or Young's Modulus)

Young's modulus is defined as the ratio of stress to strain within elastic limit.

Deformation of a body due to load acting on it

We know that young's modulus E= $Strain\dfrac{Stress}{}$,

Or, strain, $E\ \dfrac{P}{AE}$

Now, strain, $\dfrac{l}{l}$

So, deformation $\boxed{l \quad \dfrac{Pl}{AE}}$

Module 1

Lecture 4:   Numerical problems on Stress-strain relationship, Hooke's law, Poisson's ratio, shear stress

Module 1

Lecture 5: Shear strain, modulus of rigidity, bulk modulus. Relationship between material properties of isotropic materials.

Shear Strain

The distortion produced by shear stress on an element or rectangular block is shown in the figure. The shear strain or 'slide' is expressed by angle ϕ and it can be defined as the change in the right angle. It is measured in radians and is dimensionless in nature.
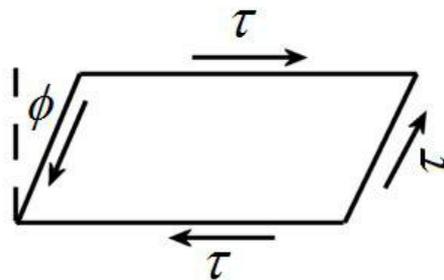
Modulus of Rigidity

For elastic materials it is found that shear stress is proportional to the shear strain within elastic limit. The ratio is called modulus rigidity. It is denoted by the symbol 'G' or 'C'.

$$G= \frac{shear\ stress}{shear\ strain}\ N/mm^2$$

Bulk modulus (K): It is defined as the ratio of uniform stress intensity to the volumetric strain. It is denoted by the symbol K.

$$K \frac{stress\ intensity}{volumetric\ strain} \quad v$$

Relation between elastic constants:

Elastic constants: These are the relations which determine the deformations produced by a given stress system acting on a particular material. These factors are constant within elastic limit, and known as modulus of elasticity E, modulus of rigidity G, Bulk modulus K and Poisson's ratio μ.

23

Relationship between modulus of elasticity (E) and bulk modulus (K):

$$E = 3K(1 - 2\mu)$$

Relationship between modulus of elasticity (E) and modulus of rigidity (G):

$$E = 2G(1 + \mu)$$

Relation among three elastic constants:

$$E = \frac{}{G \cdot 3K}$$

Stress-strain diagram for uniaxial loading of ductile and brittle materials.

Stress – Strain Relationship

Stress – strain diagram for mild steel

Standard specimen are used for the tension test.

There are two types of standard specimen's which are generally used for this purpose, which have been shown below:

Specimen I:

This specimen utilizes a circular X-section.

[specimen with circular X-section]
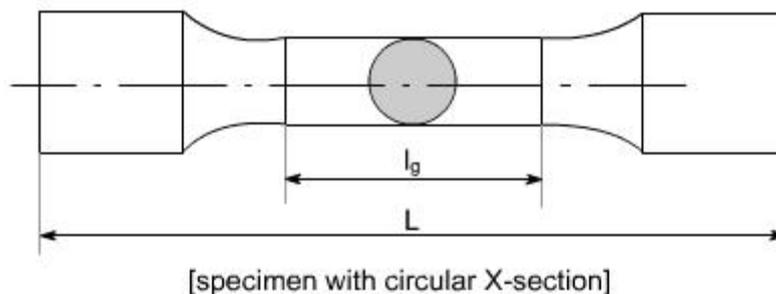
Specimen II:

This specimen utilizes a rectangular X-section.

[specimen with rectangular X-section]
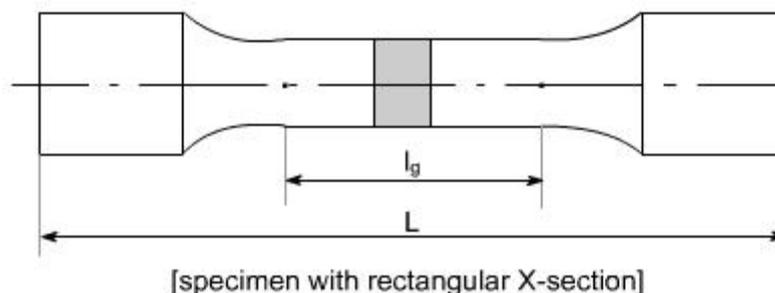
l_g = gauge length i.e. length of the specimen on which we want to determine the mechanical properties.The uniaxial tension test is carried out on tensile testing machine and the following steps are performed to conduct this test.

(i)   The ends of the specimen are secured in the grips of the testing machine.

(ii) There is a unit for applying a load to the specimen with a hydraulic or mechanical drive.

(iii) There must be some recording device by which you should be able to measure the final output in the form of Load or stress. So the testing machines are often equipped with the pendulum type lever, pressure gauge and hydraulic capsule and the stress Vs strain diagram is plotted which has the following shape.

A typical tensile test curve for the mild steel has been shown below
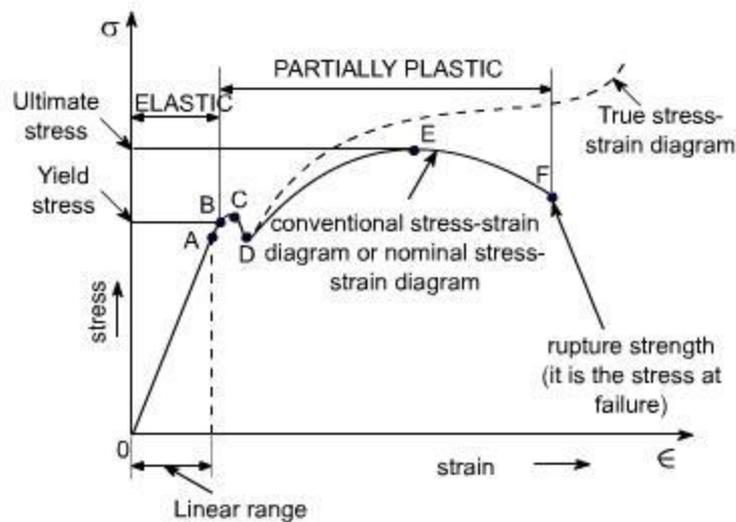
SALIENT POINTS OF THE GRAPH:

(A) So it is evident form the graph that the strain is proportional to strain or elongation is proportional to the load giving a st.line relationship. This law of proportionality is valid upto a point A.

or we can say that point A is some ultimate point when the linear nature of the graph ceases or there is a deviation from the linear nature. This point is known as **the limit of proportionality or the proportionality limit**.

(B) For a short period beyond the point A, the material may still be elastic in the sense that the deformations are completely recovered when the load is removed. The limiting point B is termed as **Elastic Limit** .

(C) and (D) - Beyond the elastic limit plastic deformation occurs and strains are not totally recoverable. There will be thus permanent deformation or permanent set

27

when load is removed. These two points are termed as upper and lower yield points respectively. The stress at the yield point is called the yield strength.

A study a stress – strain diagrams shows that the yield point is so near the proportional limit that for most purpose the two may be taken as one. However, it is much easier to locate the former. For material which do not posses a well define yield points, In order to find the yield point or yield strength, an offset method is applied.

In this method a line is drawn parallel to the straight line portion of initial stress diagram by off setting this by an amount equal to 0.2% of the strain as shown as below and this happens especially for the low carbon steel.
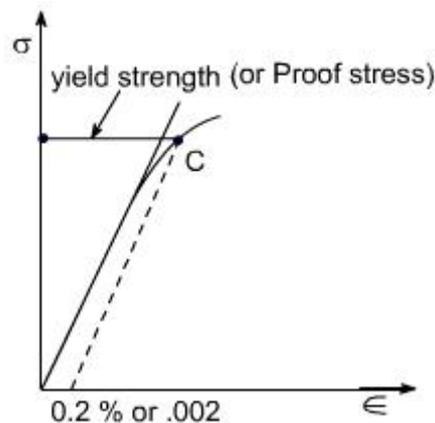
(E) A further increase in the load will cause marked deformation in the whole volume of the metal. The maximum load which the specimen can with stand without failure is called the load at the ultimate strength.

The highest point 'E' of the diagram corresponds to the ultimate strength of a material.

s_u = Stress which the specimen can with stand without failure & is known as Ultimate Strength or Tensile Strength.

s_u is equal to load at E divided by the original cross-sectional area of the bar.

(F) Beyond point E, the bar begins to forms neck. The load falling from the maximum until fracture occurs at F. Beyond point E, the cross-sectional area of the specimen begins to reduce rapidly over a relatively small length of bar and the bar is said to form a neck. This necking takes place whilst the load reduces, and fracture of the bar finally occurs at point F.

Nominal stress – Strain OR Conventional Stress – Strain diagrams:

Stresses are usually computed on the basis of the original area of the specimen; such stresses are often referred to as conventional or nominal stresses.

True stress – Strain Diagram:

Since when a material is subjected to a uniaxial load, some contraction or expansion always takes place. Thus, dividing the applied force by the corresponding actual area of the specimen at the same instant gives the so called true stress.

Percentage Elongation: 'd ':

The ductility of a material in tension can be characterized by its elongation and by the reduction in area at the cross section where fracture occurs.

It is the ratio of the extension in length of the specimen after fracture to its initial gauge length, expressed in percentage.

$$\delta = \frac{(l_1 - l_g)}{l_1} \times 100$$

l_l = gauge length of specimen after fracture(or the distance between the gage marks at fracture)

l_g = gauge length before fracture(i.e. initial gauge length)

For 50 mm gage length, steel may here a % elongation d of the order of 10% to 40%.

Ductile and Brittle Materials:

Based on this behaviour, the materials may be classified as ductile or brittle materials

Ductile Materials:

It we just examine the earlier tension curve one can notice that the extension of the materials over the plastic range is considerably in excess of that associated with elastic loading. The Capacity of materials to allow these large deformations or large extensions without failure is termed as ductility. The materials with high ductility are termed as ductile materials.

Brittle Materials:

A brittle material is one which exhibits a relatively small extensions or deformations to fracture, so that the partially plastic region of the tensile test graph is much reduced. This type of graph is shown by the cast iron or steels with high carbon contents or concrete.
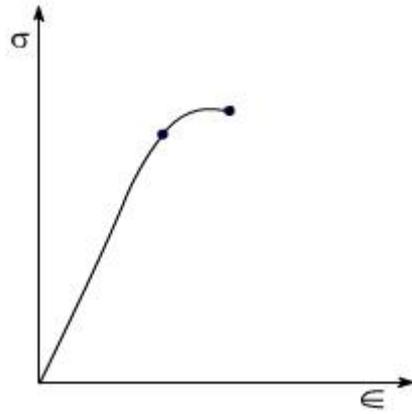
Module 1:

Lecture 8: Introduction to mechanical properties of metals-hardness, impact

Mechanical Properties of material:

Elasticity: Property of material by virtue of which it can regain its shape after removal of external load

Plasticity: Property of material by virtue of which, it will be in a state of permanent deformation even after removal of external load.

Ductility: Property of material by virtue of which, the material can be drawn into wires.

Hardness: Property of material by virtue of which the material will offer resistance to penetration or indentation.

Ball indentation Tests:

iThis method consists in pressing a hardened steel ball under a constant load P into a specially prepared flat surface on the test specimen as indicated in the figures below :
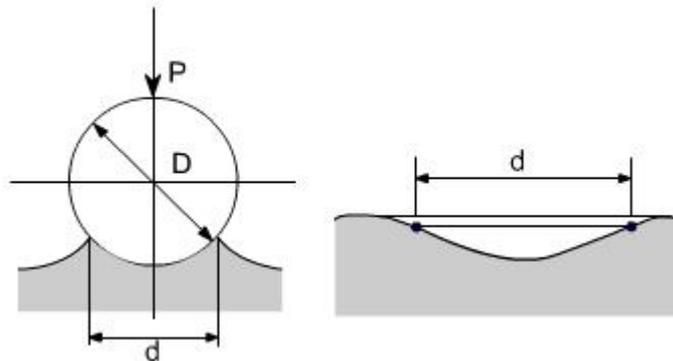
After removing the load an indentation remains on the surface of the test specimen. If area of the spherical surface in the indentation is denoted as F sq. mm. Brinell Hardness number is defined as :

BHN=P/F

F is expressed in terms of D and d

D = ball diameter

d = diametric of indentation and Brinell Hardness number is given by

$$BHN \quad \frac{2P}{D(D \quad \sqrt{D^2 \quad d^2})}$$

Then is there is also **Vicker's Hardness Number** in which the ball is of conical shape.

IMPACT STRENGTH

Static tension tests of the unnotched specimen's do not always reveal the susceptibility of metal to brittle fracture. This important factor is determined in impact tests. In impact tests we use the notched specimen's

this specimen is placed on its supports on anvil so that blow of the striker is opposite to the notch the impact strength is defined as the energy A, required to rupture the specimen,

Impact Strength = A / f

Where f = It is the cross – section area of the specimen in cm^2 at fracture & obviously at notch.

The impact strength is a complex characteristic which takes into account both toughness and strength of a material. The main purpose of notched – bar tests is to study the simultaneous effect of stress concentration and high velocity load application

Impact test are of the severest type and facilitate brittle friction. Impact strength values can not be as yet be used for design calculations but these tests as rule provided for in specifications for carbon & alloy steels.Futher, it may be noted that in impact tests fracture may be either brittle or ductile. In the case of brittle fracture, fracture occurs by separation and is not accompanied by noticeable plastic deformation as occurs in the case of ductile fracture.

Impact loads:

Considering a weight falling from a height h, on to a collar attached at the end as shown in the figure.

Let P= equivalent static or gradually applied load which will produce the same extension x as that of the impact load W

Neglecting loss of energy due to impact, we can have:

Loss of potential energy= gain of strain energy of the bar

$W(h + x) = \frac{1}{2}Px$

Now we have extension $x = \frac{Pl}{AE}$

Substituting the value of x in the above equation we have:

$W(h + \frac{Pl}{AE}) = \frac{1}{2}(P^2 l / AE)$

Solving the above equation we can have the following relation:

$P = W[1 + \sqrt{1 + 2hAE/Wl}\,]$

Important Case: for a particular case i.e. for h=0, for a suddenly applied load P=2W, i.e. the stress produced by a suddenly applied load is twice that of the static stress.

Numerical examples:

1.   Referring to the following figure let a mass of 100 kg fall 4cm on to a collar attached to a bar of steel 2cm diameter, 3m long. Find the maximum stress set up. Take E= 205,000 N/mm^2.

Applying the relation:

$\frac{P}{A} =$

$W[1 + \sqrt{1 + 2hAE/Wl}\,]/A$

$= \frac{981}{100}\left[1 + \sqrt{1 + \frac{2 \times 40 \times 100 \times 205{,}000}{981 \times 3 \times 1000}}\right]$

$= 134 \text{ M/mm}^2$

Composite Bars In Tension & Compression:-Temperature stresses in composite rods statically indeterminate problem.

Thermal stresses, Bars subjected to tension and Compression

Compound bar: In certain application it is necessary to use a combination of elements or bars made from different materials, each material performing a different function. In over head electric cables or Transmission Lines for example it is often convenient to carry the current in a set of copper wires surrounding steel wires. The later being designed to support the weight of the cable over large spans. Such a combination of materials is generally termed compound bars.

Consider therefore, a compound bar consisting of n members, each having a different length and cross sectional area and each being of a different material. Let all member have a common extension 'x' i.e. the load is positioned to produce the same extension in each member.
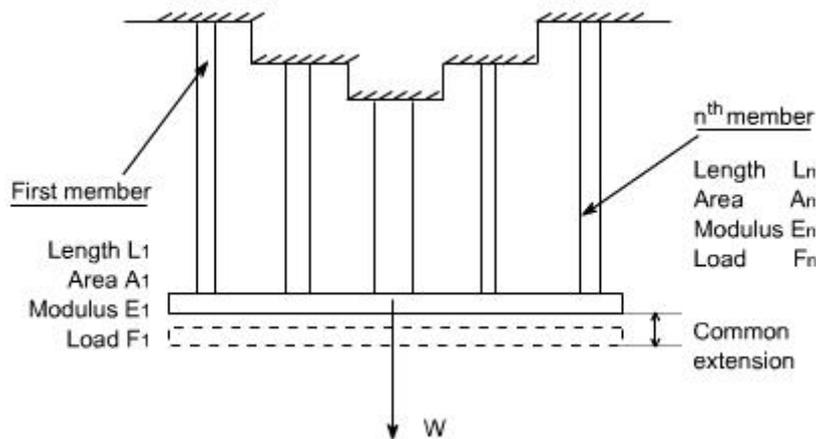
For the 'n' the members

$$\frac{stress}{strain} = E_n = \frac{F_n/A_n}{x_n/L_n}$$

$$= \frac{F_n \cdot L_n}{A_n \cdot x_n}$$

$$\text{or} \quad F_n = \frac{E_n \cdot A_n \cdot x_n}{L_n} = \frac{E_n \cdot A_n \cdot x}{L_n} \quad \dots(1)$$

Where F_n is the force in the nth member and A_n and L_n are its cross - sectional area and length.

Let W be the total load, the total load carried will be the sum of all loads for all the members.

$$W = \sum \frac{E_n.A_n.x}{L_n}$$

$$= x.\sum \frac{E_n.A_n}{L_n} \qquad \ldots\ldots(2)$$

From equation (1), force in member 1 is given as

$$F_1 = \frac{E_1.A_1.x}{L_1}$$

from equation (2)

$$x = \frac{W}{\sum \dfrac{E_n.A_n}{L_n}}$$

Thus, $F_1 = \dfrac{E_1.A_1}{L_1} . \dfrac{W}{\sum\left(\dfrac{E_n.A_n}{L_n}\right)}$

Therefore, each member carries a portion of the total load W proportional of EA / L value.

$$F_1 = \frac{\dfrac{E_1.A_1}{L_1}}{\sum \dfrac{E_n.A_n}{L_n}} . W$$

The above expression may be writen as

if the length of each individual member in same then, we may write $F_1 = \dfrac{E_1.A_1}{\sum E.A} . W$

Thus, the stress in member '1' may be determined as $\sigma_1 = F_1 / A_1$

Determination of common extension of compound bars: In order to determine the common extension of a compound bar it is convenient to consider it as a single bar of an imaginary material with an equivalent or combined modulus E_c.

Assumption: Here it is necessary to assume that both the extension and original lengths of the individual members of the compound bar are the same, the strains in all members will than be equal.

Total load on compound bar = $F_1 + F_2 + F_3 + \ldots\ldots + F_n$

where F_1 , F_2 ,….,etc are the loads in members 1,2 etc

But force = stress . area, therefore

$\sigma (A_1 + A_2 + \ldots\ldots + A_n) = \sigma_1 A_1 + \sigma_2 A_2 + \ldots\ldots + \sigma_n A_n$

Where σ is the stress in the equivalent single bar

Dividing throughout by the common strain .

$$\frac{\sigma}{\in}(A_1 + A_2 + ... + A_n) = \frac{\sigma_1}{\in}A_1 + \frac{\sigma_2}{\in}A_2 + ... \frac{\sigma_n}{\in}A_n$$

i.e $E_c(A_1 + A_2 + ... + A_n) = E_1A_1 + E_2A_2 + ...E_n.A_n$

or $E_c = \dfrac{E_1A_1 + E_2A_2 + ...E_n.A_n}{A_1 + A_2 + ... + A_n}$

or $E_c = \dfrac{\Sigma EA}{\Sigma A}$

with an external load W applied stress in the equivalent bar may be computed as

$$\text{stress} = \frac{W}{\Sigma A}$$

$$\text{strain in the equivalent bar} = \frac{x}{L} = \frac{W}{\Sigma A E_c}$$

$$\text{hence commen extension } x = \frac{W.L}{E_c \Sigma A}$$

The extension of any Length L is given by L t

(a) Original bar

Assume $\alpha_b > \alpha_a$

Steel
Brass
Steel

$\alpha_b. L . t$

A

$\alpha_a .L .t$

(b) Expanded position members free to expand inrepently

Steel   B   C
Brass
Steel

Extension of steel   Compression of brass

D

(c) Expanded position of the Compound bar

Steel
Brass
Steel
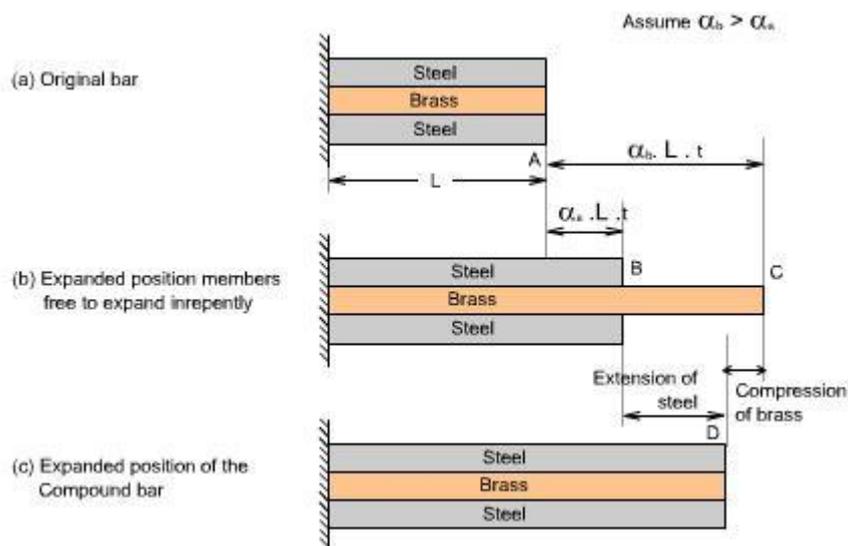
In general, changes in lengths due to thermal strains may be calculated form equation $_t$ = Lt, provided that the members are able to expand or contract freely, a situation that exists in statically determinates structures. As a consequence no stresses are generated in a statically determinate structure when one or more members undergo a uniform temperature change. If in a structure (or a compound bar), the free expansion or contraction is not allowed then the member becomes s statically indeterminate, which is just being discussed as an example of the compound bar and thermal stresses would be generated.

If the two materials are now rigidly joined as a compound bar and subjected to the same temp. rise, each materials will attempt to expand to its free length position but each will be affected by the movement of the other. The higher coefficient of expansion material (brass) will therefore, seek to pull the steel up to its free length position and conversely, the lower coefficient of expansion martial (steel) will try to hold the brass back. In practice a compromised is reached, the compound bar extending to the position shown in fig (c), resulting in an effective compression of the

brass from its free length position and an effective extension of steel from its free length position.

Two Dimensional State of Stress and Strain: Principal stresses. Numerical examples

Stresses on oblique plane: Till now we have dealt with either pure normal direct stress or pure shear stress. In many instances, however both direct and shear stresses acts and the resultant stress across any section will be neither normal nor tangential to the plane. A plane stse of stress is a 2 dimensional stae of stress in a sense that the stress components in one direction are all zero i.e

$\sigma_z = \tau_{yz} = \tau_{zx} = 0$

Examples of plane state of stress include plates and shells. Consider the general case of a bar under direct load F giving rise to a stress σ_y vertically
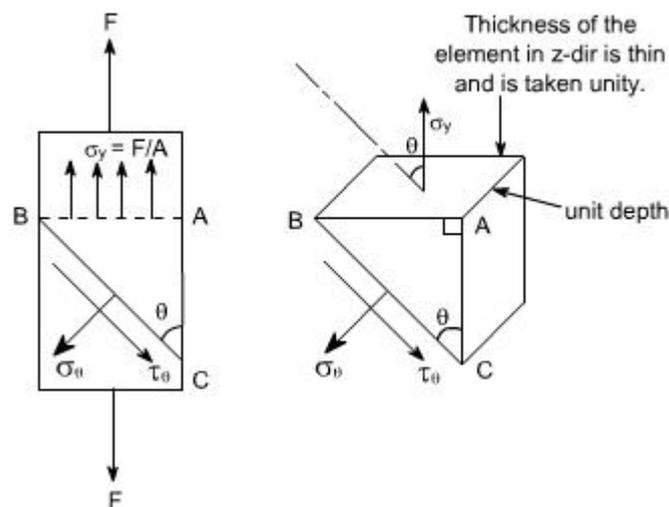
The stress acting at a point is represented by the stresses acting on the faces of the element enclosing the point. The stresses change with the inclination of the planes passing through that point i.e. the stress on the faces of the element vary as the angular position of the element changes. Let the block be of unit depth now considering the equilibrium of forces on the triangle portion ABC. Resolving forces perpendicular to BC, gives

σ_θ.BC.1 = σ_y sin θ. AB.1

but AB/BC = sin θ or AB = BC sin θ

Substituting this value in the above equation, we get

σ.BC.1 = σ_y sin θ. BC sin θ. 1 or σ_y sin^2 θ (1)

Now resolving the forces parallel to BC

τ.BC.1 = σ_y cos θ. AB sin. 1

again AB = BC cos θ

τ.BC.1 = σ_y cos θ. BC sin θ.1 or τ = σ_y sin θ cos θ

$$\frac{1}{2}. \sigma_y \sin 2\theta \qquad (2)$$

If θ = 90^0 the BC will be parallel to AB and τ = 0, i.e. there will be only direct stress or normal stress.

By examining the equations (1) and (2), the following conclusions may be drawn

(i) The value of direct stress σ is maximum and is equal to σ_y when v= 90^0.

(ii) The shear stress τ has a maximum value of 0.5 σ_y when θ = 45^0

<u>Material subjected to pure shear:</u>

Consider the element shown to which shear stresses have been applied to the sides AB and DC
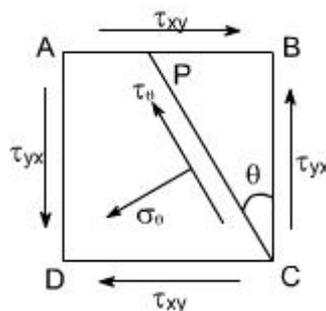
Complementary shear stresses of equal value but of opposite effect are then set up on the sides AD and BC in order to prevent the rotation of the element. Since the applied and complementary shear stresses are of equal value on the x and y planes. Therefore, they are both represented by the symbol τ_{xy}.

Now consider the equilibrium of portion of PBC
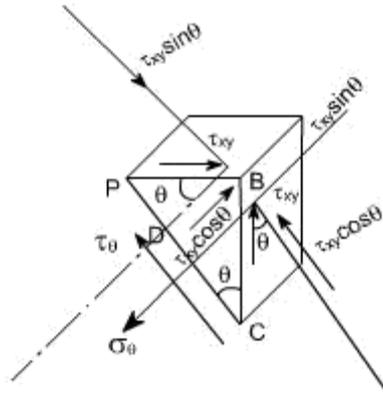
Assuming unit depth and resolving normal to PC or in the direction of

σ_θ.PC.1 = τ_{xy} .PB.cos θ .1+ τ_{xy} .BC.sin θ .1

= τ_{xy} .PB.cos θ + τ_{xy} .BC.sin θ

Now writing PB and BC in terms of PC so that it cancels out from the two sides PB/PC = sin θ BC/PC = cos θ

σ_θ.PC.1 = τ_{xy} .cos θ sin θ PC+ τ_{xy} .cos θ .sin θ .PC

= 2 τ_{xy} sin θ cos θ

Or,2 τ_{xy} sin 2θ (1)

Now resolving forces parallel to PC or in the direction of θ.then τ_{xy} PC.1 = τ_{xy}

. PB sin θ - τ_{xy} BC cos θ

-ve sign has been put because this component is in the same direction as that of θ.

again converting the various quantities in terms of PC we have

τ_{xy} PC. 1 = τ_{xy} . PB.sin^2 θ τ_{xy} - τ_{xy} PCcos^2 θ

= - τ_{xy} [cos^2 θ - sin^2 θ]

= - τ_{xy} cos2θ (2)

the negative sign means that the sense of θ is opposite to that of assumed one. Let us examine the equations (1) and (2) respectively

σ_θ = τ_{xy} sin2θ

The equation (1) represents that the maximum value of σ_θ is τ_{xy} when θ = 45^0.Let us take into consideration the equation (2) which states that

$= - _{xy} \cos 2$

It indicates that the maximum value of is $_{xy}$ when $= 0^0$ or 90^0. it has a value zero when $=$ 45^0.

From equation (1) it may be noticed that the normal component has maximum and minimum values of $+_{xy}$ (tension) and $_{xy}$(compression) on plane at $\pm 45^0$ to the applied shear and on these planes the tangential component is zero.

Hence the system of pure shear stresses produces and equivalent direct stress system, one set compressive and one tensile each located at 45^0 to the original shear directions as depicted in the figure below:
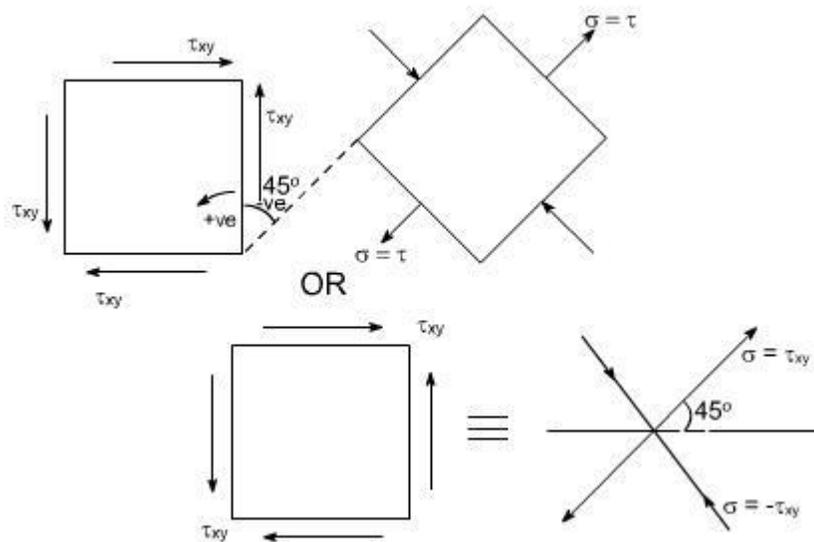
Thin cylinder and thin spherical shells under internal pressure and numerical examples. Wire winding of thin cylinders. Numerical examples.

Cylindrical Vessel with Hemispherical Ends:

Let us now consider the vessel with hemispherical ends. The wall thickness of the cylindrical and hemispherical portion is different. While the internal diameter of both the portions is assumed to be equal

Let the cylindrical vassal is subjected to an internal pressure p.
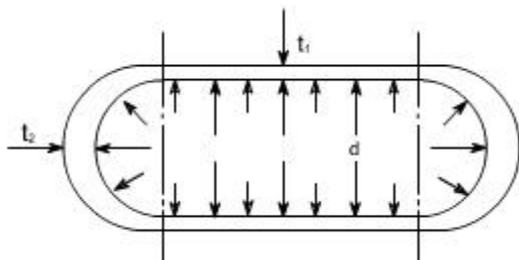
For the Cylindrical Portion

hoop or circumferential stress $= \sigma_{HC}$ 'c' here synifies the cylindrical portion.

$$= \frac{pd}{2t_1}$$

longitudnal stress $= \sigma_{LC}$

$$= \frac{pd}{4t_1}$$

hoop or circumferential strain $\in_2 = \dfrac{\sigma_{HC}}{E} - v\dfrac{\sigma_{LC}}{E} = \dfrac{pd}{4t_1 E}[2-v]$

or $\boxed{\in_2 = \dfrac{pd}{4t_1 E}[2-v]}$

For The Hemispherical Ends:
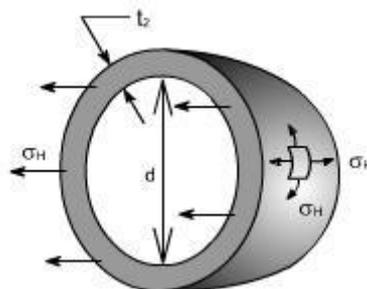
Because of the symmetry of the sphere the stresses set up owing to internal pressure will be two mutually perpendicular hoops or circumferential stresses of equal values. Again the radial stresses are neglected in comparison to the hoop stresses as with this cylinder having thickness to diametre less than1:20.

Consider the equilibrium of the half – sphere

Force on half-sphere owing to internal pressure = pressure x projected Area

= p. $d^2/4$

Resisting force $= \sigma_H . \pi.d.t_2$

$\therefore \qquad p.\dfrac{\pi d^2}{4} = \sigma_H . \pi d.t_2$

$\Rightarrow \sigma_H \text{ (for sphere)} = \dfrac{pd}{4t_2}$

similarly the hoop strain $= \dfrac{1}{E}\left[\sigma_H - v.\sigma_H\right] = \dfrac{\sigma_H}{E}[1 - v] = \dfrac{pd}{4t_2 E}[1 - v]$ or $\boxed{\epsilon_{2s} = \dfrac{pd}{4t_2 E}[1 - v]}$
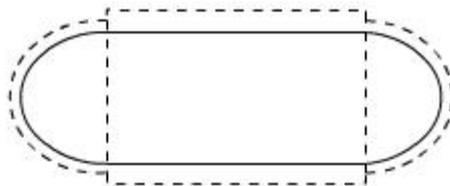
Fig – shown the (by way of dotted lines) the tendency, for the cylindrical portion and the spherical ends to expand by a different amount under the action of internal pressure. So owing to difference in stress, the two portions (i.e. cylindrical and spherical ends) expand by a different amount. This incompatibly of deformations causes a local bending and sheering stresses in the neighborhood of the joint. Since there must be physical continuity between the ends and the cylindrical portion, for this reason, properly curved ends must be used for pressure vessels.

Thus equating the two strains in order that there shall be no distortion of the junction

$\dfrac{pd}{4t_1 E}[2 - v] = \dfrac{pd}{4t_2 E}[1 - v]$ or $\dfrac{t_2}{t_1} = \dfrac{1 - v}{2 - v}$

But for general steel works $v = 0.3$, therefore, the thickness ratios becomes

$t_2 / t_1 = 0.7/1.7$ or

$t_1 \quad 2.4 t_2$

i.e. the thickness of the cylinder walls must be approximately 2.4 times that of the hemispheroid ends for no distortion of the junction to occur.

SUMMARY OF THE RESULTS : Let us summarise the derived results

(A) The stresses set up in the walls of a thin cylinder owing to an internal pressure p are :

(i) Circumferential or loop stress

σ_H = pd/2t

(ii) Longitudinal or axial stress

σ_L = pd/4t

Where d is the internal diametre and t is the wall thickness of the cylinder.

then

Longitudinal strain ϵ_L = 1 / E [σ_L – σ_H]

Hoop stain ϵ_H = 1 / E [σ_H − v σ_L]

(B) Change of internal volume of cylinder under pressure

$$= \frac{pd}{4tE}[5 - 4v]V$$

(C) Fro thin spheres circumferential or loop stress

$$\sigma_H = \frac{pd}{4t}$$

Thin rotating ring or cylinder

Consider a thin ring or cylinder as shown in Fig below subjected to a radial internal pressure p caused by the centrifugal effect of its own mass when rotating. The centrifugal effect on a unit length of the circumference is

p = m ω^2 r
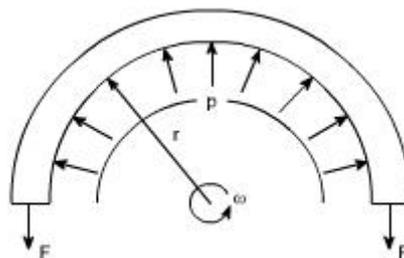
Fig 19.1: Thin ring rotating with constant angular velocity

Here the radial pressure 'p' is acting per unit length and is caused by the centrifugal effect if its own mass when rotating.

Thus considering the equilibrium of half the ring shown in the figure,

$2F = p \times 2r$ (assuming unit length), as $2r$ is the projected area

$F = pr$

Where F is the hoop tension set up owing to rotation.

The cylinder wall is assumed to be so thin that the centrifugal effect can be assumed constant across the wall thickness.

$F = $ mass x acceleration $= \mathbf{m\, \omega^2\, r \times r}$

This tension is transmitted through the complete circumference and therefore is resisted by the complete cross – sectional area.

hoop stress $= F/A = \mathbf{m\, \omega^2\, r^2 / A}$

Where A is the cross – sectional area of the ring.

Now with unit length assumed m/A is the mass of the material per unit volume, i.e. the density   .

hoop stress $_H = \quad \omega^2\, r^2$

Torsion of solid circular shafts

Torsion of circular shafts

Definition of Torsion: Consider a shaft rigidly clamped at one end and twisted at the other end by a torque T = F.d applied in a plane perpendicular to the axis of the bar such a shaft is said to be in torsion.
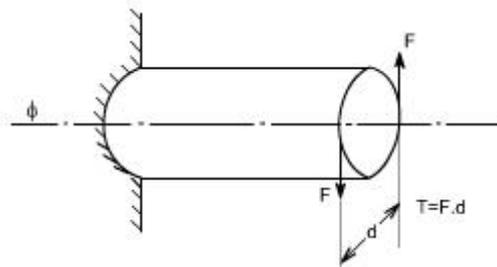
Effects of Torsion: The effects of a torsional load applied to a bar are

(i) To impart an angular displacement of one end cross – section with respect to the other end.

(ii) To setup shear stresses on any cross section of the bar perpendicular to its axis.

Assumption:

(i) The materiel is homogenous i.e of uniform elastic properties exists throughout the material.

(ii) The material is elastic, follows Hook's law, with shear stress proportional to shear strain.

(iii) The stress does not exceed the elastic limit.

(iv) The circular section remains circular

(v) Cross section remain plane.

(vi) Cross section rotate as if rigid i.e. every diameter rotates through the same angle.

Consider now the solid circular shaft of radius R subjected to a torque T at one end, the other end being fixed

Under the action of this torque a radial line at the free end of the shaft twists through an angle , point A moves to B, and AB subtends an angle ' ' at the fixed end. This is then the angle of distortion of the shaft i.e the shear strain.

Since angle in radius = arc / Radius

arc AB = R

= L   [since L and   also constitute the arc AB]

Thus,   = R / L   (1)

From the definition of Modulus of rigidity or Modulus of elasticity in shear

$$G = \frac{\text{shear stress}(\tau)}{\text{shear strain}(\gamma)}$$

where γ is the shear stress set up at radius R.

Then $\dfrac{\tau}{G} = \gamma$

Equating the equations (1) and (2) we get $\dfrac{R\theta}{L} = \dfrac{\tau}{G}$

$\dfrac{\tau}{R} = \dfrac{G\theta}{L}\left(= \dfrac{\tau'}{r}\right)$ where τ' is the shear stress at any radius r.

Stresses: Let us consider a small strip of radius r and thickness dr which is subjected to shear stress '.
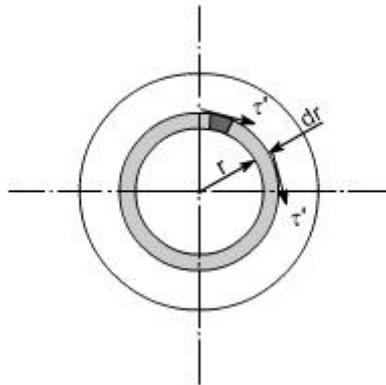
The force set up on each element

= stress x area

= ' x 2 r dr (approximately)

= ' . 2 r dr . r

= 2 ' . r². dr

$$T = \int_0^R 2\pi\tau'r^2\,dr$$

The total torque T on the section, will be the sum of all the contributions.

Since ' is a function of r, because it varies with radius so writing down   ' in terms of r from the equation (1).

68

$$\text{i.e } \tau' = \frac{G\theta.r}{L}$$

$$\text{we get } T = \int_0^R 2\pi \frac{G\theta}{L}.r^3 dr$$

$$T = \frac{2\pi G\theta}{L} \int_0^R r^3 dr$$

$$= \frac{2\pi G\theta}{L}. \left[\frac{R^4}{4}\right]_0^R$$

$$= \frac{G\theta}{L}. \frac{2\pi R^4}{4}$$

$$= \frac{G\theta}{L}. \frac{\pi R^4}{2}$$

$$= \frac{G\theta}{L}. \left[\frac{\pi d^4}{32}\right] \quad \text{now substituting } R = d/2$$

$$= \frac{G\theta}{L}. J$$

$$\text{since } \frac{\pi d^4}{32} = J \text{ the polar moment of inertia}$$

$$\text{or } \frac{T}{J} = \frac{G\theta}{L} \qquad \qquad \text{......(2)}$$

if we combine the equation no.(1) and (2) we get $\boxed{\dfrac{T}{J} = \dfrac{\tau'}{r} = \dfrac{G.\theta}{L}}$

Where

T = applied external Torque, which is constant over Length L;

J = Polar moment of Inertia

$$= \frac{\pi d^4}{32} \text{ for solid shaft}$$

$$= \frac{\pi(D^4 - d^4)}{32} \text{ for a hollow shaft.}$$

[D = Outside diameter ; d = inside diameter]

G = Modules of rigidity (or Modulus of elasticity in shear)

= It is the angle of twist in radians on a length L.

Tensional Stiffness: The tensional stiffness k is defined as the torque per radius twist

i.e, k = T / = GJ / L

Power Transmitted by a shaft : If T is the applied Torque and is the angular velocity of the shaft, then the

power transmitted by the shaft is

$$P = T.\omega = \frac{2\pi NT}{60} = \frac{2\pi NT}{60.10^3} \text{ kw}$$

$$\text{where } N = rpm$$

TORSION OF HOLLOW SHAFTS:

From the torsion of solid shafts of circular x – section , it is seen that only the

material at the outer surface of the shaft can be stressed to the limit assigned as an

allowable working stresses. All of the material within the shaft will work at a lower stress and is not being used to full capacity. Thus, in these cases where the weight reduction is important, it is advantageous to use hollow shafts. In discussing the torsion of hollow shafts the same assumptions will be made as in the case of a solid shaft. The general torsion equation as we have applied in the case of torsion of solid shaft will hold good

$$\frac{T}{J} = \frac{\tau}{r} = \frac{G.\theta}{l}$$

For the hollow shaft

$$J = \frac{\pi(D_0{}^4 - d_i{}^4)}{32}$$ where D_0 = Outside diameter

d_i = Inside diameter

Let $d_i = \frac{1}{2}.D_0$

$$\tau_{max^m}\Big|_{solid} = \frac{16T}{\pi D_0{}^3} \qquad (1)$$

$$\tau_{max^m}\Big|_{hollow} = \frac{T.D_0/2}{\frac{\pi}{32}(D_0{}^4 - d_i{}^4)}$$

$$= \frac{16T.D_0}{\pi D_0{}^4\left[1 - (d_i/D_0)^4\right]}$$

$$= \frac{16T}{\pi D_0{}^3\left[1 - (1/2)^4\right]} = 1.066.\frac{16T}{\pi D_0{}^3} \qquad (2)$$

Hence by examining the equation (1) and (2) it may be seen that the $_{max}{}^m$ in the case of hollow shaft is 6.6% larger than in the case of a solid shaft having the same outside diameter.

Reduction in weight:

Considering a solid and hollow shafts of the same length 'l' and density ' ' with $d_i = 1/2$ D_o

70
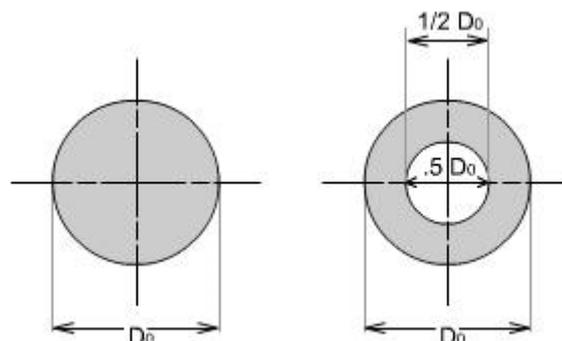
Weight of hollow shaft

$$= \left[\frac{\pi D_0^2}{4} - \frac{\pi (D_0/2)^2}{4} \right] l \times \rho$$

$$= \left[\frac{\pi D_0^2}{4} - \frac{\pi D_0^2}{16} \right] l \times \rho$$

$$= \frac{\pi D_0^2}{4} [1 - 1/4] l \times \rho$$

$$= 0.75 \frac{\pi D_0^2}{4} l \times \rho$$

Weight of solid shaft $= \frac{\pi D_0^2}{4} \cdot l \cdot \rho$

Reduction in weight $= (1 - 0.75) \frac{\pi D_0^2}{4} l \times \rho$

$$= 0.25 \frac{\pi D_0^2}{4} l \times \rho$$

Hence the reduction in weight would be just 25%.

Illustrative Examples :

Problem 1

A stepped solid circular shaft is built in at its ends and subjected to an externally applied torque. T_0 at the shoulder as shown in the figure. Determine the angle of rotation $_0$ of the shoulder section where T_0 is applied ?
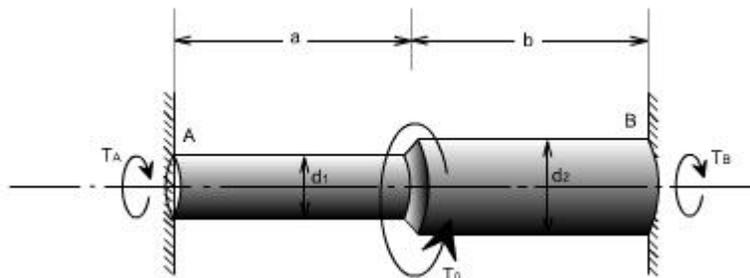
Concept of Shear Force and Bending moment in beams:

When the beam is loaded in some arbitrarily manner, the internal forces and moments are developed and the terms shear force and bending moments come into pictures which are helpful to analyze the beams further. Let us define these terms
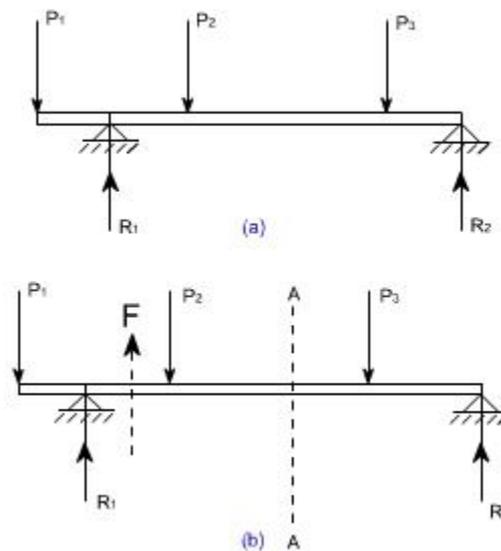
Fig 1

Now let us consider the beam as shown in fig 1(a) which is supporting the loads P_1, P_2, P_3 and is simply supported at two points creating the reactions R_1 and R_2 respectively. Now let us assume that the beam is to divided into or imagined to be cut into two portions at a section AA. Now let us assume that the resultant of loads and reactions to the left of AA is 'F' vertically upwards, and since the entire beam is to remain in equilibrium, thus the resultant of forces to the right of AA must also be F, acting downwards. This forces 'F' is as a shear force. The shearing force at any x-section of a beam represents the tendency for the portion of the beam to one side of the section to slide or shear laterally relative to the other portion.

Therefore, now we are in a position to define the shear force 'F' to as follows:

At any x-section of a beam, the shear force 'F' is the algebraic sum of all the lateral components of the forces acting on either side of the x-section.

Sign Convention for Shear Force:

The usual sign conventions to be followed for the shear forces have been illustrated in figures 2 and 3.

A

F

The resultant force which is in upward direction and is towards the L.H.S of the X-section is +ve Shear Force

The resultant force which is in the downward direction and is towards the R.H.S of the X-section is +ve Shear Force.

A
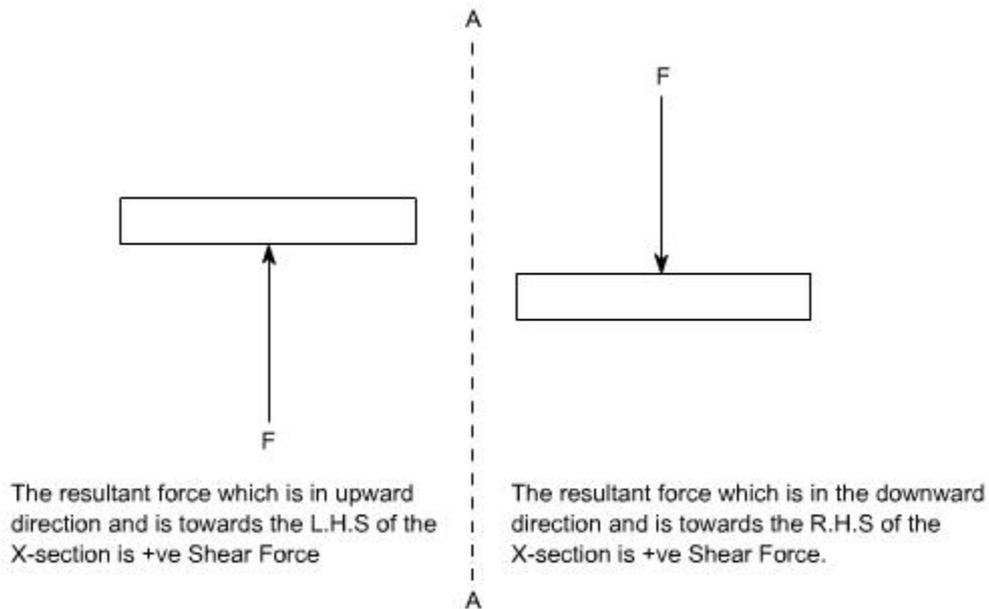
Fig 2: Positive Shear Force

A

F

The resultant force which are in the downward direction and is on the L.H.S of the X-section is -ve Shear Force.

The resultant force which are in upward direction and is on the R.H.S of the X-section is -ve Shear Force.

A
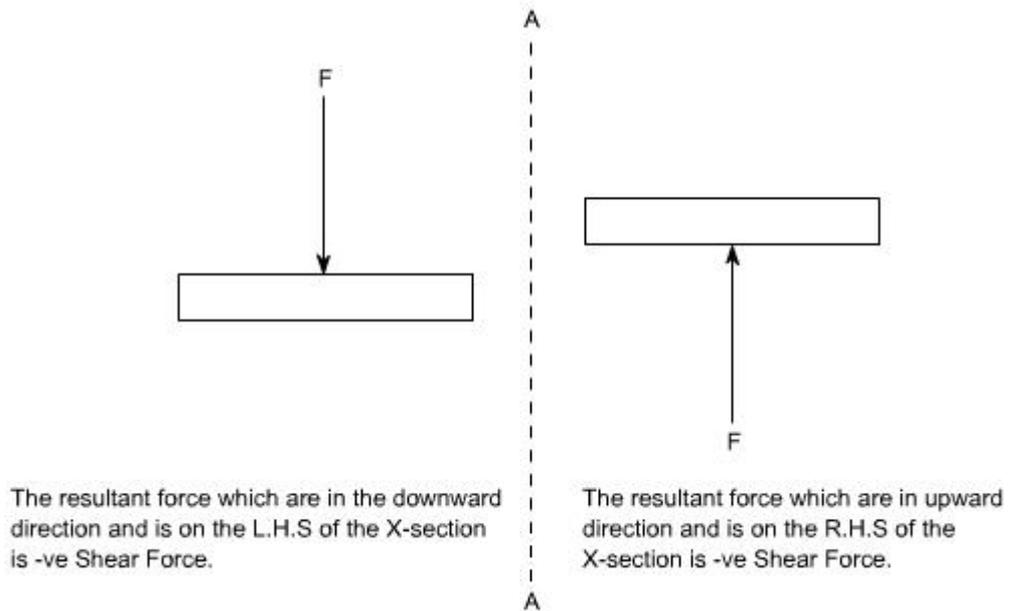
Fig 3: Negative Shear Force

Bending Moment:
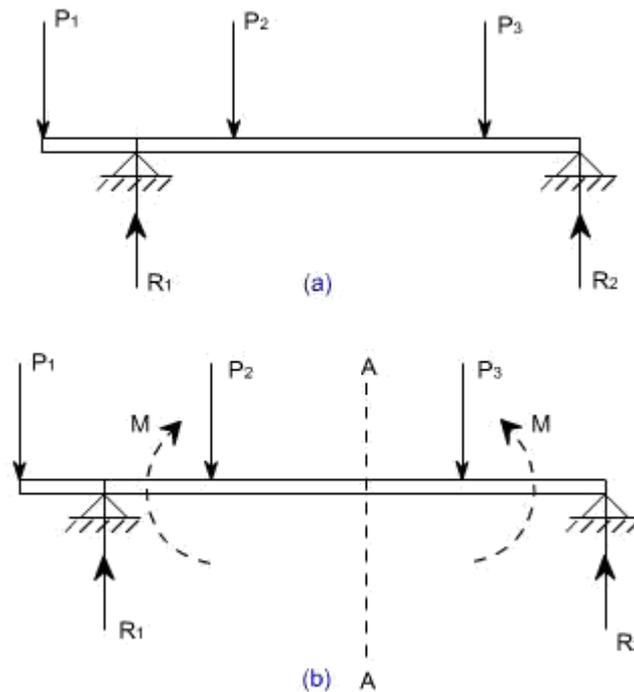
Fig 4

Let us again consider the beam which is simply supported at the two prints, carrying loads P_1, P_2 and P_3 and having the reactions R_1 and R_2 at the supports Fig 4. Now, let us imagine that the beam is cut into two potions at the x-section AA. In a similar manner, as done for the case of shear force, if we say that the resultant moment about the section AA of all the loads and reactions to the left of the x-section at AA is M in C.W direction, then moment of forces to the right of x-section AA must be 'M' in C.C.W.

Then 'M' is called as the Bending moment and is abbreviated as B.M. Now one can define the bending moment to be simply as the algebraic sum of the moments about an x-section of all the forces acting on either side of the section

Sign Conventions for the Bending Moment:

For the bending moment, following sign conventions may be adopted as indicated in Fig 5 and Fig 6.

74
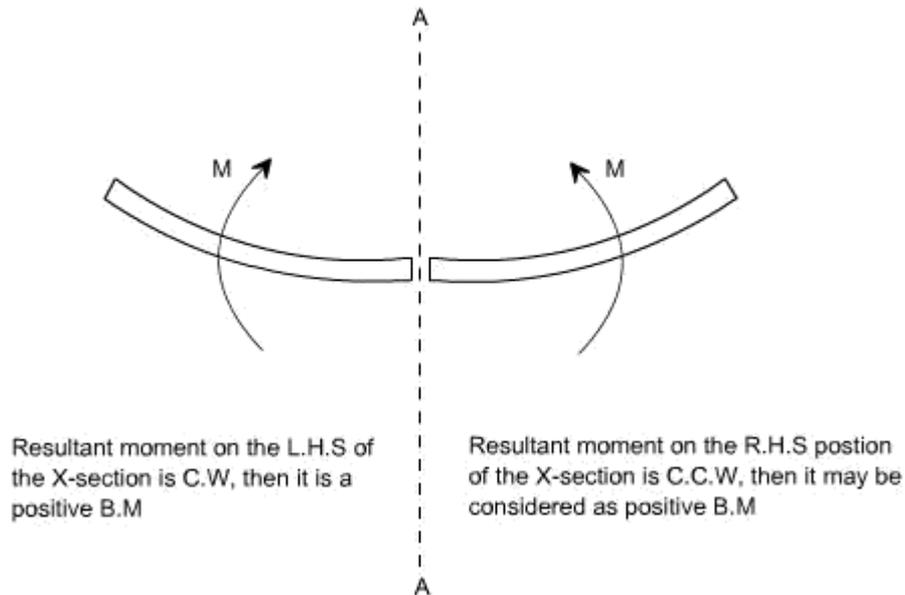
Fig 5: Positive Bending Moment
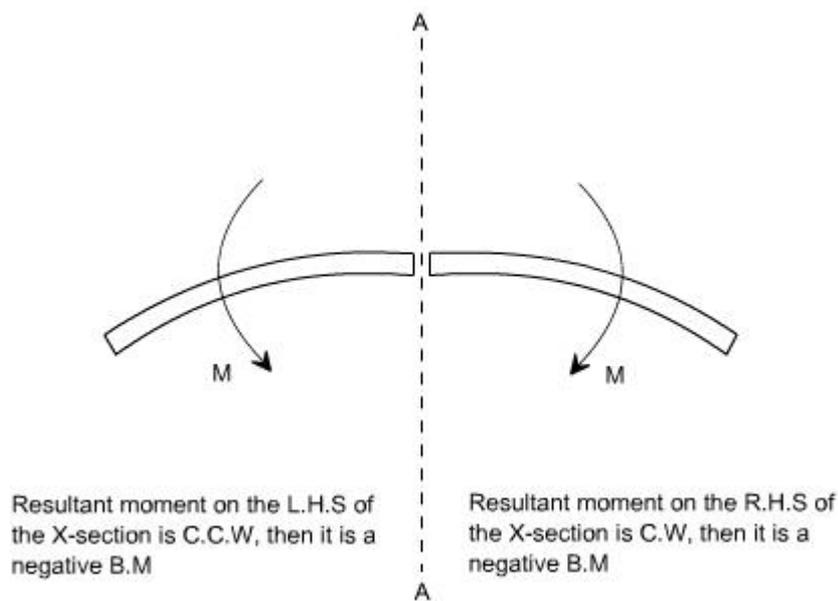
Fig 6: Negative Bending Moment

Some times, the terms 'Sagging' and Hogging are generally used for the positive and negative bending moments respectively.

Bending Moment and Shear Force Diagrams:

The diagrams which illustrate the variations in B.M and S.F values along the length of the beam for any fixed loading conditions would be helpful to analyze the beam further.

Thus, a shear force diagram is a graphical plot, which depicts how the internal shear force 'F' varies along the length of beam. If x dentotes the length of the beam, then F is function x i.e. F(x).

Similarly a bending moment diagram is a graphical plot which depicts how the internal bending moment 'M' varies along the length of the beam. Again M is a function x i.e. M(x).

Basic Relationship Between The Rate of Loading, Shear Force and Bending Moment:

The construction of the shear force diagram and bending moment diagrams is greatly simplified if the relationship among load, shear force and bending moment is established.

Let us consider a simply supported beam AB carrying a uniformly distributed load w/length. Let us imagine to cut a short slice of length dx cut out from this loaded beam at distance 'x' from the origin '0'.
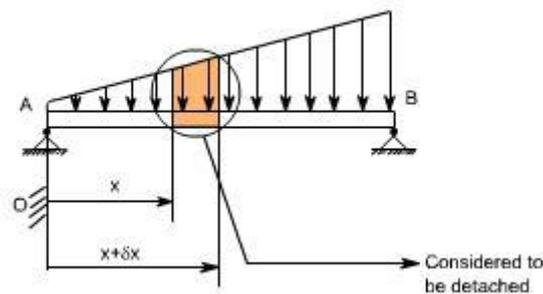
Let us detach this portion of the beam and draw its free body diagram.
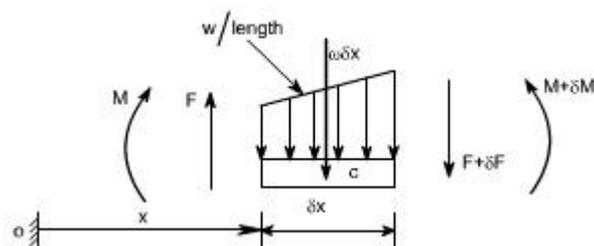
The forces acting on the free body diagram of the detached portion of this loaded beam are the following

• The shearing force F and F+ δF at the section x and x + δx respectively.

• The bending moment at the sections x and x + δx be M and M + dM respectively.

• Force due to external loading, if 'w' is the mean rate of loading per unit length then the total loading on this slice of length δx is w. δx, which is approximately acting through the centre 'c'. If the loading is assumed to be uniformly distributed then it would pass exactly through the centre 'c'.

This small element must be in equilibrium under the action of these forces and couples.

Now let us take the moments at the point 'c'. Such that

$$M + F.\frac{\delta x}{2} + (F + \delta F).\frac{\delta x}{2} = M + \delta M$$

$$\Rightarrow F.\frac{\delta x}{2} + (F + \delta F).\frac{\delta x}{2} = \delta M$$

$$\Rightarrow F.\frac{\delta x}{2} + F.\frac{\delta x}{2} + \delta F.\frac{\delta x}{2} = \delta M \ [\text{Neglecting the product of}$$
$$\delta F \text{ and } \delta x \text{ being small quantities}]$$

$$\Rightarrow F.\delta x = \delta M$$

$$\Rightarrow F = \frac{\delta M}{\delta x}$$

Under the limits $\delta x \to 0$

$$\boxed{F = \frac{dM}{dx}} \qquad \cdots\cdots\cdots \quad (1)$$

Resolving the forces vertically we get

$$w.\delta x + (F + \delta F) = F$$

$$\Rightarrow w = -\frac{\delta F}{\delta x}$$

Under the limits $\delta x \to 0$

$$\Rightarrow w = -\frac{dF}{dx} \ \text{or} \ -\frac{d}{dx}(\frac{dM}{dx})$$

$$\boxed{w = -\frac{dF}{dx} = -\frac{d^2M}{dx^2}} \qquad \cdots\cdots\cdots (2)$$

Conclusions: From the above relations,the following important conclusions may be drawn

• From Equation (1), the area of the shear force diagram between any two points, from the basic calculus is the bending moment diagram

$$M = \int F.dx$$

• The slope of bending moment diagram is the shear force, thus

$$F = \frac{dM}{dx}$$

Thus, if F=0; the slope of the bending moment diagram is zero and the bending moment is therefore constant.'

• The maximum or minimum Bending moment occurs where $\frac{dM}{dx} = 0$.

The slope of the shear force diagram is equal to the magnitude of the intensity of the distributed loading at any position along the beam. The –ve sign is as a consequence of our particular choice of sign conventions

Procedure for drawing shear force and bending moment diagram:

Preamble:

The advantage of plotting a variation of shear force F and bending moment M in a beam as a function of 'x' measured from one end of the beam is that it becomes easier to determine the maximum absolute value of shear force and bending moment.

Further, the determination of value of M as a function of 'x' becomes of paramount importance so as to determine the value of deflection of beam subjected to a given loading.

Construction of shear force and bending moment diagrams:

A shear force diagram can be constructed from the loading diagram of the beam. In order to draw this, first the reactions must be determined always. Then the vertical components of forces and reactions are successively summed from the left end of the beam to preserve the mathematical sign conventions adopted. The shear at a section is simply equal to the sum of all the vertical forces to the left of the section.

When the successive summation process is used, the shear force diagram should end up with the previously calculated shear (reaction at right end of the beam. No shear force acts through the beam just beyond the last vertical force or reaction. If the shear force diagram closes in this fashion, then it gives an important check on mathematical calculations.

The bending moment diagram is obtained by proceeding continuously along the length of beam from the left hand end and summing up the areas of shear force diagrams giving due regard to sign. The process of obtaining the moment diagram from the shear force diagram by summation is exactly the same as that for drawing shear force diagram from load diagram.

It may also be observed that a constant shear force produces a uniform change in the bending moment, resulting in straight line in the moment diagram. If no shear force exists along a certain portion of a beam, then it indicates that there is no change in moment takes place. It may also further observe that dm/dx= F therefore, from the fundamental theorem of calculus the maximum or minimum moment occurs where the shear is zero. In order to check the validity of the bending moment diagram, the terminal conditions for the moment must be satisfied. If the end is free or pinned, the computed sum must be equal to zero. If the end is built in, the moment computed by the summation must be equal to the one calculated initially for the reaction. These conditions must always be satisfied.

Illustrative problems:

In the following sections some illustrative problems have been discussed so as to illustrate the procedure for drawing the shear force and bending moment diagrams

1. A cantilever of length carries a concentrated load 'W' at its free

end. Draw shear force and bending moment.

Solution:

At a section a distance x from free end consider the forces to the left, then F = -W (for all values of x) -ve sign means the shear force to the left of the x-section are in downward direction and therefore negative

Taking moments about the section gives (obviously to the left of the section)

M = -Wx (-ve sign means that the moment on the left hand side of the portion is in the anticlockwise direction and is therefore taken as –ve according to the sign convention) so that the maximum bending moment occurs at the fixed end i.e. M = -W l

From equilibrium consideration, the fixing moment applied at the fixed end is Wl and the reaction is W. the shear force and bending moment are shown as,
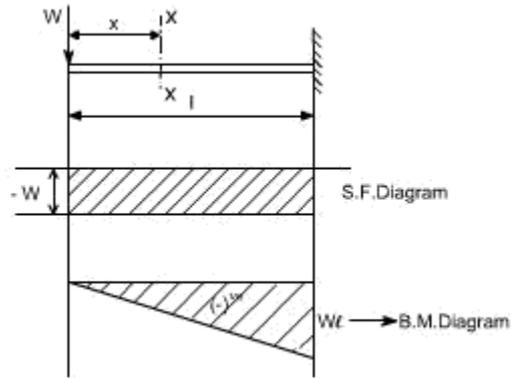
2. Simply supported beam subjected to a central load (i.e. load acting at the mid-way)
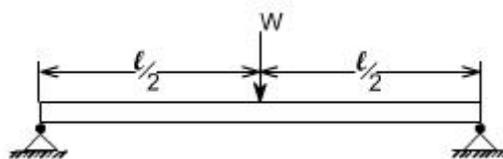
By symmetry the reactions at the two supports would be W/2 and W/2. now consider any section X-X from the left end then, the beam is under the action of following forces.
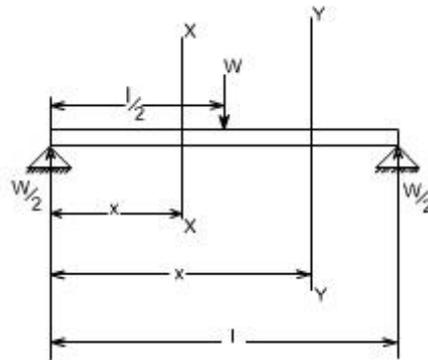
.So the shear force at any X-section would be = W/2 [Which is constant upto x < l/2]

If we consider another section Y-Y which is beyond l/2 then

$$S.F_{Y-Y} = \frac{W}{2} - W = \frac{-W}{2}$$ for all values greater =

l/2 Hence S.F diagram can be plotted as,
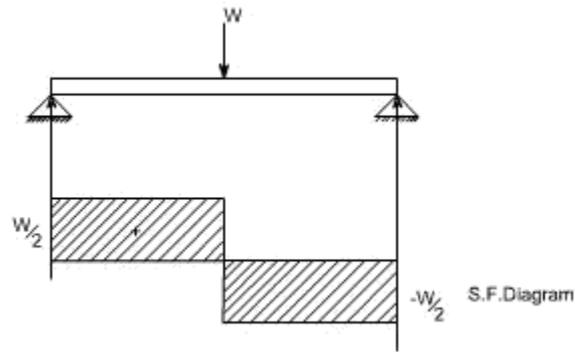
S.F.Diagram

.For B.M diagram:

If we just take the moments to the left of the cross-section,

$$B.M_{X-X} = \frac{W}{2} x \text{ for x lies between } 0 \text{ and } l/2$$

$$B.M_{at\,x=\frac{l}{2}} = \frac{W}{2} \frac{l}{2} \text{ i.e B.M at } x = 0$$

$$= \frac{Wl}{4}$$

$$B.M_{Y-Y} = \frac{W}{2} x - W\left(x - \frac{l}{2}\right)$$

Again

$$= \frac{W}{2} x - Wx + \frac{Wl}{2}$$

$$= -\frac{W}{2} x + \frac{Wl}{2}$$

$$B.M_{at\,x=l} = -\frac{Wl}{2} + \frac{Wl}{2}$$

$$= 0$$

Which when plotted will give a straight relation i.e.
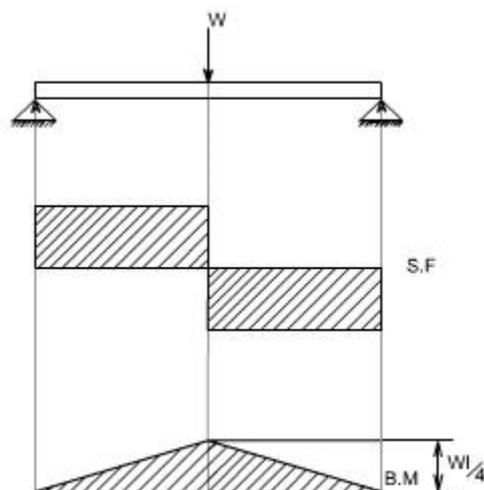
It may be observed that at the point of application of load there is an abrupt change in the shear force, at this point the B.M is maximum.

3. A cantilever beam subjected to U.d.L, draw S.F and B.M diagram.
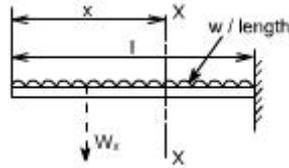
Here the cantilever beam is subjected to a uniformly distributed load whose intensity is given w / length.

Consider any cross-section XX which is at a distance of x from the free end. If we just take the resultant of all the forces on the left of the X-section, then

S.F$_{xx}$ = -Wx for all values of 'x'. ---------- **(1)**

S.F$_{xx}$ = 0

S.F$_{xx \; at \; x=1}$ = -Wl

So if we just plot the equation No. (1), then it will give a straight line relation. Bending Moment at X-X is obtained by treating the load to the left of X-X as a concentrated load of the same value acting through the centre of gravity.

Therefore, the bending moment at any cross-section X-X is

$$B.M_{X-X} = - \; W \; x \; \frac{x}{2}$$
$$= - \; W \frac{x^2}{2}$$

The above equation is a quadratic in x, when B.M is plotted against x this will produces a parabolic variation.

The extreme values of this would be at x = 0 and x = l

$$B.M_{at \; x \; = \; 1} = - \frac{Wl^2}{2}$$
$$= \frac{Wl}{2} \; - Wx$$

Hence S.F and B.M diagram can be plotted as follows:
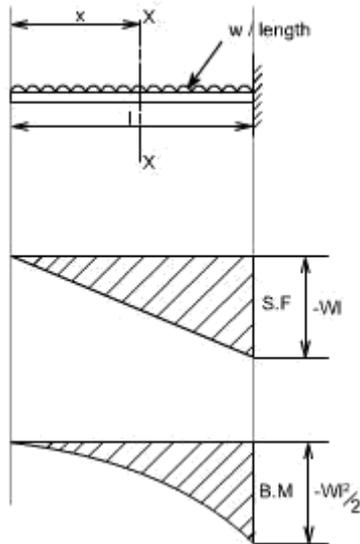
4. Simply supported beam subjected to a uniformly distributed load [U.D.L].
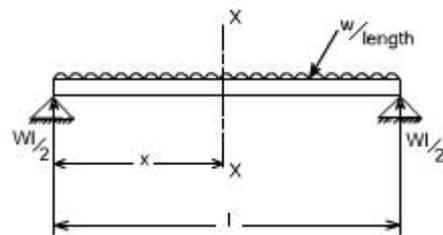
The total load carried by the span would be

= intensity of loading x length

= w x l

By symmetry the reactions at the end supports are each wl/2

If x is the distance of the section considered from the left hand end of the beam.

S.F at any X-section X-X is

$$= \frac{Wl}{2} - Wx$$

$$= W\left(\frac{l}{2} - x\right)$$

Giving a straight relation, having a slope equal to the rate of loading or intensity of the loading.

$$S.F_{\text{at } x=0} = \frac{wl}{2} - wx$$

so at

$$S.F_{\text{at } x=\frac{l}{2}} = 0 \text{ hence the S.F is zero at the centre}$$

$$S.F_{\text{at } x=l} = -\frac{Wl}{2}$$

The bending moment at the section x is found by treating the distributed load as acting at its centre of gravity, which at a distance of x/2 from the section
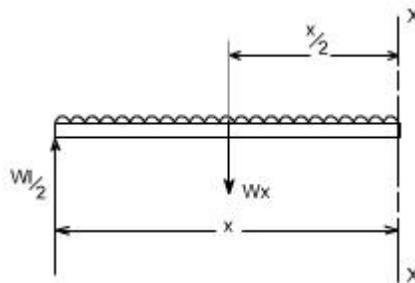
$$B.M_{X-X} = \frac{Wl}{2}x - Wx.\frac{x}{2}$$

so the

$$= W.\frac{x}{2}(l - 2) \quad ....(2)$$

$$B.M_{\text{at } x=0} = 0$$

$$B.M_{\text{at } x=l} = 0$$

$$B.M\Big|_{\text{at } x=l} = -\frac{Wl^2}{8}$$

So the equation (2) when plotted against x gives rise to a parabolic curve and the shear force and bending moment can be drawn in the following way will appear as follows:
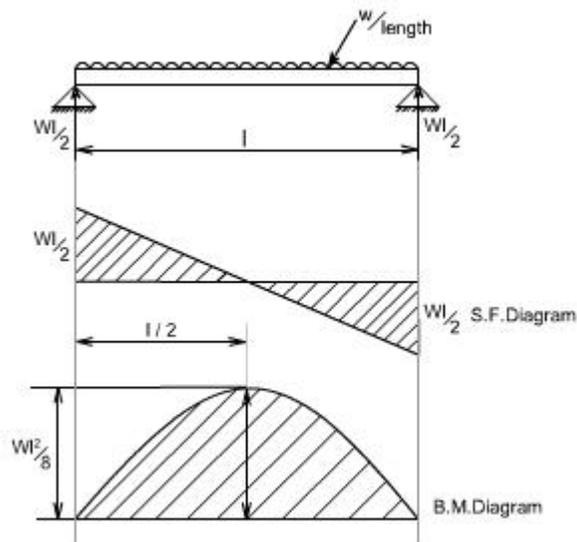
Bending

Loading restrictions:

As we are aware of the fact internal reactions developed on any cross-section of a beam may consists of a resultant normal force, a resultant shear force and a resultant couple. In order to ensure that the bending effects alone are investigated, we shall put a constraint on the loading such that the resultant normal and the resultant shear forces are zero on any cross-section perpendicular to the longitudinal axis of the member,

That means F = 0

since $\dfrac{dM}{dX} = F = 0$ or M = constant.

Thus, the zero shear force means that the bending moment is constant or the bending is same at every cross-section of the beam. Such a situation may be visualized or envisaged when the beam or some portion of the beam, as been loaded only by pure couples at its ends. It must be recalled that the couples are assumed to be loaded in the plane of symmetry.
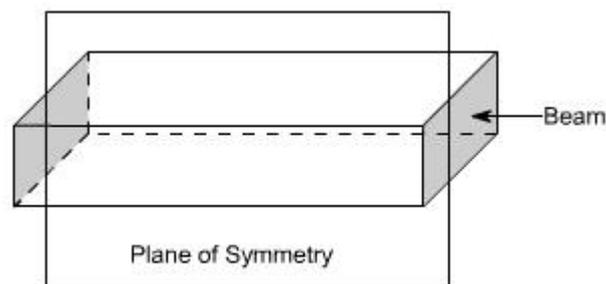
Fig (1)
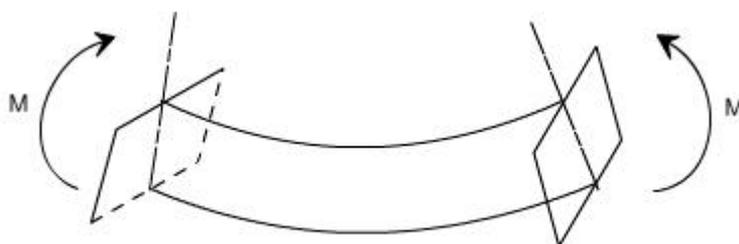
Fig (2)

When a member is loaded in such a fashion it is said to be in **pure bending.** The examples of pure bending have been indicated in EX 1and EX 2 as shown below :

EX .2

zero S.F

S.F.D

Constant B.M

B.M.D

EX. 1

M

SFD

BMD
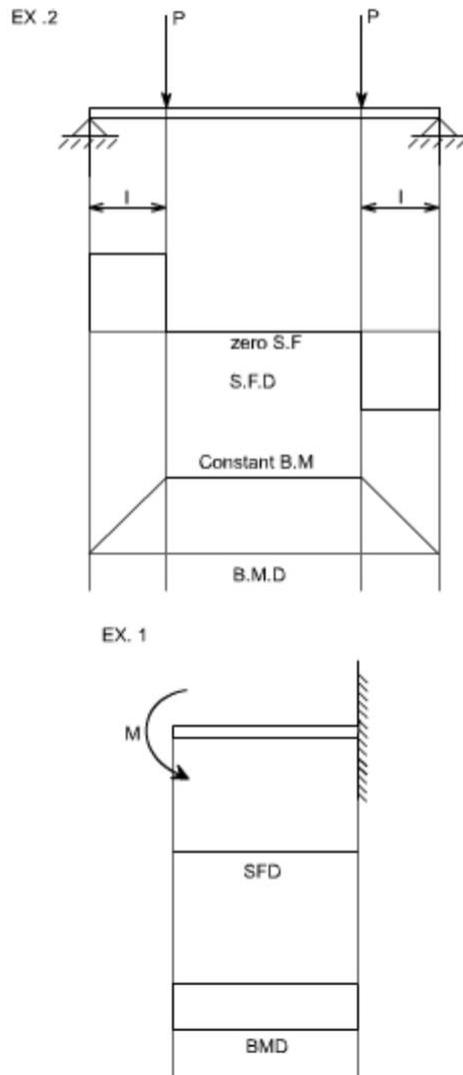
When a beam is subjected to pure bending are loaded by the couples at the ends, certain cross-section gets deformed and we shall have to make out the conclusion that,

1. Plane sections originally perpendicular to longitudinal axis of the beam remain plane and perpendicular to the longitudinal axis even after bending , i.e. the cross-section A'E', B'F' (refer Fig 1(a)) do not get warped or curved.

2. In the deformed section, the planes of this cross-section have a common intersection i.e. any time originally parallel to the longitudinal axis of the beam becomes an arc of circle.
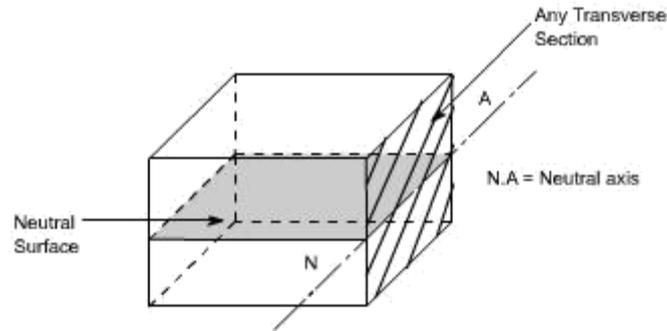
We know that when a beam is under bending the fibres at the top will be lengthened while at the bottom will be shortened provided the bending moment M acts at the ends. In between these there are some fibres which remain unchanged in length that is they are not strained, that is they do not carry any stress. The plane containing such fibres is called neutral surface.

The line of intersection between the neutral surface and the transverse exploratory section is called the neutral axisNeutral axis **(N A)** .

Bending Stresses in Beams or Derivation of Elastic Flexural formula :

In order to compute the value of bending stresses developed in a loaded beam, let us consider the two cross-sections of a beam**HE** and **GF** , originally parallel as shown in fig 1(a).when the beam is to bend it is assumed that these sections remain parallel i.e.**H'E'** and **G'F'** , the final position of the sections, are still straight lines, they then subtend some angle .

Consider now fiber AB in the material, at adistance y from the N.A, when the beam bends this will stretch to A'B'

Therefore,

$$\text{strain in fibre } AB = \frac{\text{change in length}}{\text{orginal length}}$$

$$= \frac{A'B' - AB}{AB}$$

But $AB = CD$ and $CD = C'D'$

refer to fig1(a) and fig1(b)

$$\therefore \text{ strain } = \frac{A'B' - C'D'}{C'D'}$$

Since CD and C'D' are on the neutral axis and it is assumed that the Stress on the neutral axis zero. Therefore, there won't be any strain on the neutral axis

87

$$= \frac{(R + y)\theta - R\theta}{R\theta} = \frac{R\theta + y\theta - R\theta}{R\theta} = \frac{y}{R}$$

However $\frac{stress}{strain} = E$ where E = Young's Modulus of elasticity

Therefore, equating the two strains as obtained from the two relations i.e,

$$\frac{\sigma}{E} = \frac{y}{R} \text{ or } \frac{\sigma}{y} = \frac{E}{R} \qquad \ldots\ldots\ldots(1)$$
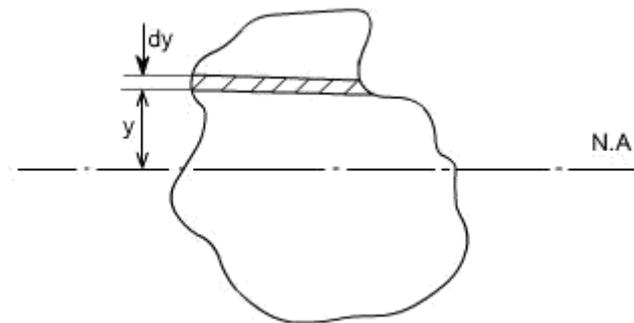
Consider any arbitrary a cross-section of beam, as shown above now the strain on a fibre at a distance 'y' from the N.A, is given by the expression

$$\sigma = \frac{E}{R} y$$

if the shaded strip is of area 'dA'
then the force on the strip is

$$F = \sigma \, \delta A = \frac{E}{R} y \, \delta A$$

Moment about the neutral axis would be $= F.y = \frac{E}{R} y^2 \delta A$

The toatl moment for the whole
cross-section is therefore equal to

$$M = \sum \frac{E}{R} y^2 \, \delta A = \frac{E}{R} \sum y^2 \delta A$$

Now the term $\sum y^2 \delta A$ is the property of the material and is called as a second moment of area of the cross-section and is denoted by a symbol I.
Therefore

$$M = \frac{E}{R} I \qquad \ldots\ldots(2)$$

combining equation 1 and 2 we get

$$\boxed{\frac{\sigma}{y} = \frac{M}{I} = \frac{E}{R}}$$

This equation is known as the Bending Theory Equation. The above proof has involved the assumption of pure bending without any shear force being present.

Therefore this termed as the pure bending equation. This equation gives distribution of stresses which are normal to cross-section i.e. in x-direction.

Section Modulus:

From simple bending theory equation, the maximum stress obtained in any cross-section is given as

$$\sigma_{max}{}_m = \frac{M}{I} y_{max}{}_m$$

For any given allowable stress the maximum moment which can be accepted by a particular shape of cross-section is therefore

$$M = \frac{I}{y_{max}{}_m} \sigma_{max}{}_m$$

For ready comparison of the strength of various beam cross-section this relationship is some times written in the form

$$M = Z \sigma_{max}{}_m \quad \text{where } Z = \frac{I}{y_{max}{}_m}$$ **Is termed as section modulus**

The higher value of Z for a particular cross-section, the higher the bending moment which it can withstand for a given maximum stress.

Theorems to determine second moment of area: There are two theorems which are helpful to determine the value of second moment of area, which is required to be used while solving the simple bending theory equation.

Second Moment of Area :

Taking an analogy from the mass moment of inertia, the second moment of area is defined as the summation of areas times the distance squared from a fixed axis. (This property arised while we were driving bending theory equation). This is also known as the moment of inertia. An alternative name given to this is second moment of area, because the first moment being the sum of areas times their distance from a

given axis and the second moment being the square of the distance or $\int y^2 \, dA$.
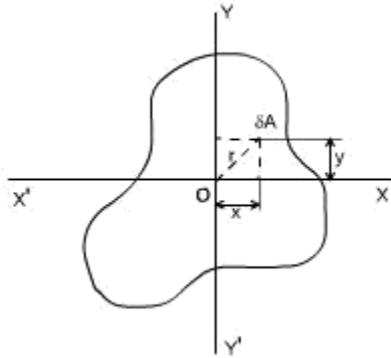
Consider any cross-section having small element of area d A then by the definition

I_x(Mass Moment of Inertia about x-axis) = $\int y^2\, dA$ and I_y(Mass Moment of Inertia about y-axis) = $\int x^2\, dA$

Now the moment of inertia about an axis through 'O' and perpendicular to the plane of figure is called the polar moment of inertia. (The polar moment of inertia is also the area moment of inertia).

i.e,

J = polar moment of inertia

$$= \int r^2\, dA$$
$$= \int (x^2 + y^2)\, dA$$
$$= \int x^2\, dA + \int y^2\, dA$$
$$= I_X + I_Y$$

or $J = I_X + I_Y$ .... (1)

The relation (1) is known as the **perpendicular axis theorem** and may be stated as follows:

The sum of the Moment of Inertia about any two axes in the plane is equal to the moment of inertia about an axis perpendicular to the plane, the three axes being concurrent, i.e, the three axes exist together.

CIRCULAR SECTION :

For a circular x-section, the polar moment of inertia may be computed in the following manner
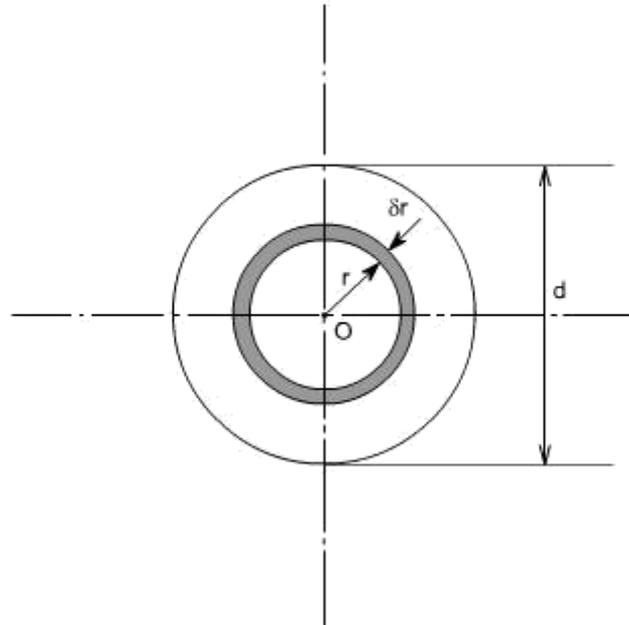
Consider any circular strip of thickness r located at a radius 'r'.

Than the area of the circular strip would be dA = 2 r. r

$$J = \int r^2 dA$$

Taking the limits of intergration from 0 to d/2

$$J = \int_0^{\frac{d}{2}} r^2 2\pi r \, \delta r$$

$$= 2\pi \int_0^{\frac{d}{2}} r^3 \delta r$$

$$J = 2\pi \left[\frac{r^4}{4} \right]_0^{\frac{d}{2}} = \frac{\pi d^4}{32}$$

however, by perpendicular axis theorem

$$J = I_x + I_y$$

But for the circular cross-section, the I_x and I_y are both equal being moment of inertia about a diameter

$$I_{dia} = \frac{1}{2} J$$

$$I_{dia} = \frac{\pi d^4}{64}$$

for a hollow circular section of diameter D and d,
the values of J and I are defined as

$$J = \frac{\pi \left(D^4 - d^4 \right)}{32}$$

Thus $$I = \frac{\pi \left(D^4 - d^4 \right)}{64}$$

91

Parallel Axis Theorem:

The moment of inertia about any axis is equal to the moment of inertia about a parallel axis through the centroid plus the area times the square of the distance between the axes.
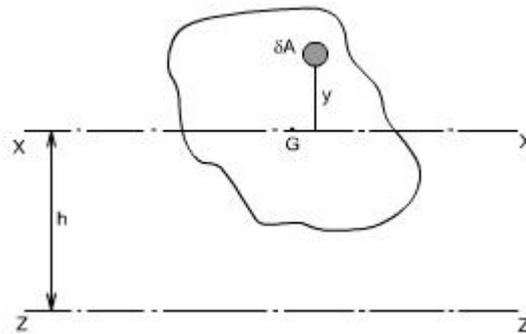
If 'ZZ' is any axis in the plane of cross-section and 'XX' is a parallel axis through the centroid G, of the cross-section, then

$I_z = \int (y+h)^2 \, dA$ by definition (moment of inertia about an axis ZZ)

$\quad = \int \left(+ 2yh + h^2 \right) dA$

$\quad = \int y^2 dA + h^2 \int dA + 2h \int y dA$

$\qquad\qquad$ Since $\int y dA = 0$

$\quad = \int y^2 dA + h^2 \int dA$

$\quad = \int y^2 dA + h^2 A$

$I_z = I_x + Ah^2 \quad I_x = I_G$ (since cross-section axes also pass through G)

$\qquad\qquad$ Where A = Total area of the section

Rectangular Section:

For a rectangular x-section of the beam, the second moment of area may be computed as below :
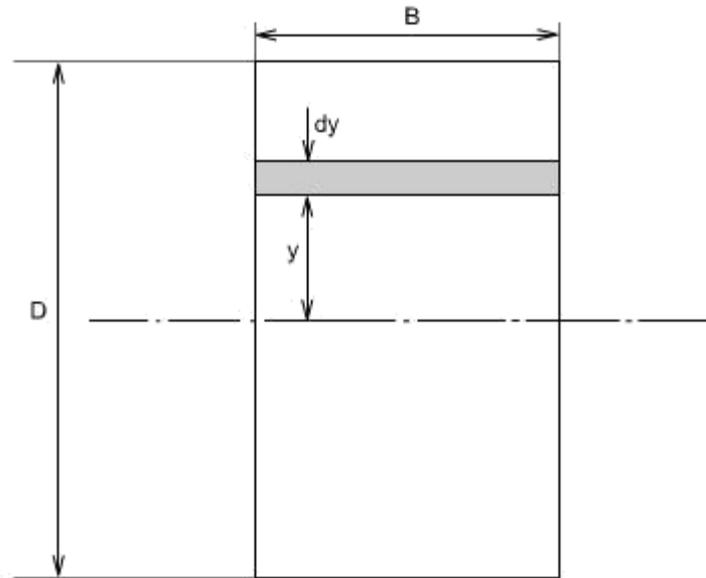
Consider the rectangular beam cross-section as shown above and an element of area **dA** , thickness **dy** , breadth **B** located at a distance **y** from the neutral axis, which by symmetry passes through the centre of section. The second moment of area **I** as defined earlier would be

$$I_{N.A} = \int y^2 \, dA$$

Thus, for the rectangular section the second moment of area about the neutral axis i.e., an axis through the centre is given by

$$I_{N.A} = \int_{-\frac{D}{2}}^{\frac{D}{2}} y^2 \, (B \, dy)$$

$$= B \int_{-\frac{D}{2}}^{\frac{D}{2}} y^2 \, dy$$

$$= B \left[\frac{y^3}{3} \right]_{\frac{D}{2}}^{-\frac{D}{2}}$$

$$= \frac{B}{3} \left[\frac{D^3}{8} - \left(\frac{-D^3}{8} \right) \right]$$

$$= \frac{B}{3} \left[\frac{D^3}{8} + \frac{D^3}{8} \right]$$

$$I_{N.A} = \frac{BD^3}{12}$$

Similarly, the second moment of area of the rectangular section about an axis through the lower edge of the section would be found using the same procedure but with integral limits of **0** to **D** .

Therefore
$$I = B\left[\frac{y^3}{3}\right]_0^D = \frac{BD^3}{3}$$

These standards formulas prove very convenient in the determination of I_{NA} for build up sections which can be conveniently divided into rectangles. For instance if we just want to find out the Moment of Inertia of an I - section, then we can use the above relation.
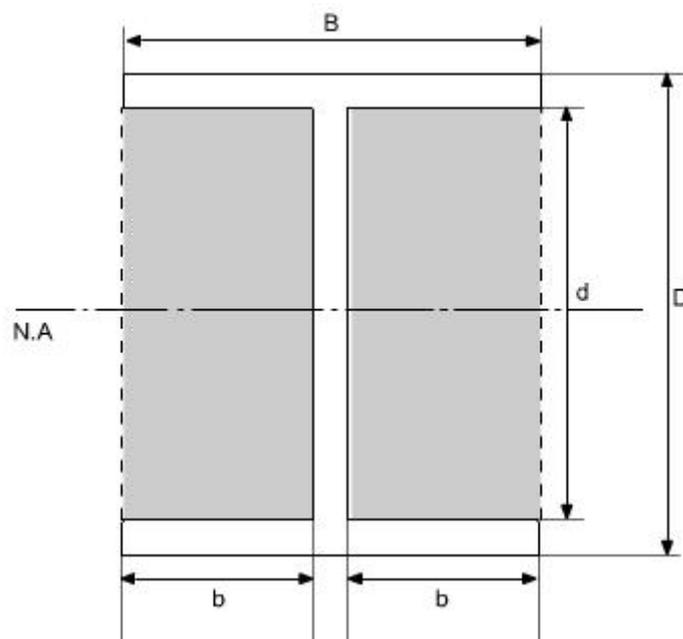
$$I_{N.A} = I_{of\ dotted\ rectangle} - I_{of\ shaded\ portion}$$

$$\therefore I_{N.A} = \frac{BD^3}{12} - 2\left(\frac{bd^3}{12}\right)$$

$$I_{N.A} = \frac{BD^3}{12} - \frac{bd^3}{6}$$

Use of Flexure Formula:

Illustrative Problems:

An I - section girder, 200mm wide by 300 mm depth flange and web of thickness is 20 mm is used as simply supported beam for a span of 7 m. The girder carries a distributed load of 5 KN /m and a concentrated load of 20 KN at mid-span. Determine the

 (i). The second moment of area of the cross-section of the girder

94

(ii). The maximum stress set up.

Solution:

The second moment of area of the cross-section can be determained as follows : For sections with symmetry about the neutral axis, use can be made of standard I value for a rectangle about an axis through centroid i.e. (bd 3)/12. The section can thus be divided into convenient rectangles for each of which the neutral axis passes through the centroid. Example in the case enclosing the girder by a rectangle

$$I_{girder} = I_{rectangle} - I_{shaded\ portion}$$

$$= \left[\frac{200 \times 300^3}{12}\right] 10^{-12} - 2 \left[\frac{90 \times 260^3}{12}\right] 10^{-12}$$

$$= (4.5 - 2.64\ )10^{-4}$$

$$= 1.86 \times 10^{-4}\ m^4$$

The maximum stress may be found from the simple bending theory by equation

$$\frac{\sigma}{y} = \frac{M}{I} = \frac{E}{R}$$

i.e.

$$\sigma_{max^m} = \frac{M_{max^m}}{I}\ y_{max^m}$$
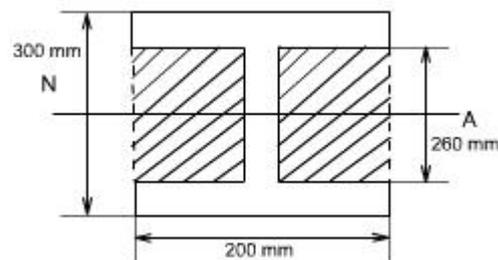
Computation of Bending Moment:

In this case the loading of the beam is of two types

(a) Uniformly distributed load

(b) Concentrated Load

In order to obtain the maximum bending moment the technique will be to consider each loading on the beam separately and get the bending moment due to it as if no other forces acting on the structure and then superimpose the two results.
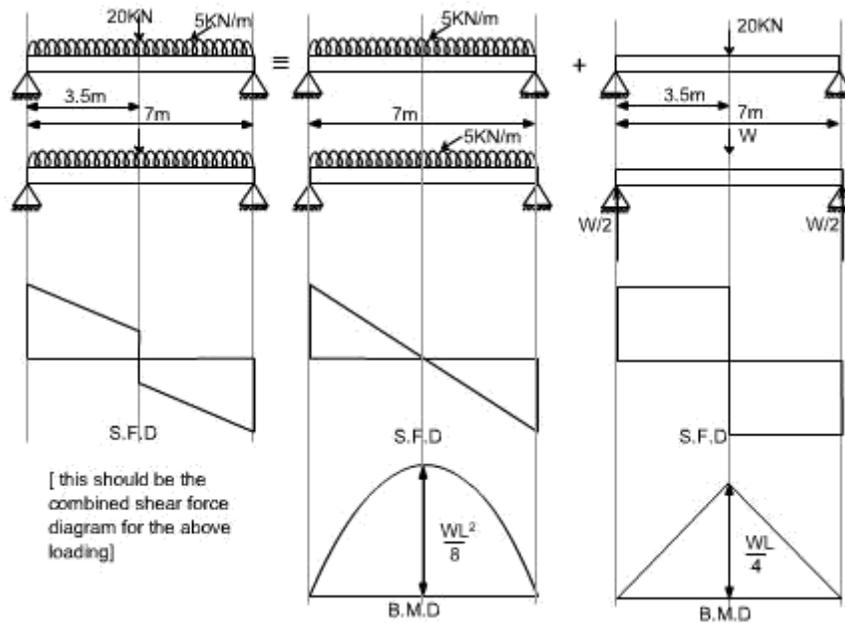
Hence

$$M_{max^m} = \frac{wL}{4} + \frac{wL^2}{8}$$

$$= \frac{20 \times 10^3 \times 7}{4} + \frac{5 \times 10^3 \times 7^2}{8}$$

$$= (35.0 + 30.63) 10^3$$

$$= 65.63 \, k \, Nm$$

$$\sigma_{max^m} = \frac{M_{max^m}}{I} \, y_{max^m}$$

$$= \frac{65.63 \times 10^3 \times 150 \times 10^3}{1.06 \times 10^{-4}}$$

$$\sigma_{max^m} = 51.8 \, MN/m^2$$

Shearing Stresses in Beams

All the theory which has been discussed earlier, while we discussed the bending stresses in beams was for the case of pure bending i.e. constant bending moment acts along the entire length of the beam.

Deflection of Beams

Deflection of Beams

Introduction:

In all practical engineering applications, when we use the different components, normally we have to operate them within the certain limits i.e. the constraints are placed on the performance and behavior of the components. For instance we say that the particular component is supposed to operate within this value of stress and the deflection of the component should not exceed beyond a particular value.

In some problems the maximum stress however, may not be a strict or severe condition but there may be the deflection which is the more rigid condition under operation. It is obvious therefore to study the methods by which we can predict the deflection of members under lateral loads or transverse loads, since it is this form of loading which will generally produce the greatest deflection of beams.

Assumption: The following assumptions are undertaken in order to derive a differential equation of elastic curve for the loaded beam

1. Stress is proportional to strain i.e. hooks law applies. Thus, the equation is valid only for beams that are not stressed beyond the elastic limit.

2. The curvature is always small.

3. Any deflection resulting from the shear deformation of the material or shear stresses is neglected.

It can be shown that the deflections due to shear deformations are usually small and hence can be ignored.
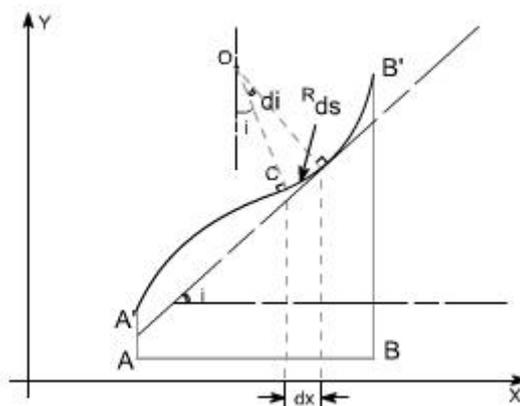
Consider a beam AB which is initially straight and horizontal when unloaded. If under the action of loads the beam deflect to a position A'B' under load or infact we say that the axis of the beam bends to a shape A'B'. It is customary to call A'B' the curved axis of the beam as the elastic line or deflection curve.

In the case of a beam bent by transverse loads acting in a plane of symmetry, the bending moment M varies along the length of the beam and we represent the variation of bending moment in B.M diagram. Futher, it is assumed that the simple bending theory equation holds good.

$$\frac{\sigma}{y} = \frac{M}{I} = \frac{E}{R}$$

If we look at the elastic line or the deflection curve, this is obvious that the curvature at every point is different; hence the slope is different at different points.

To express the deflected shape of the beam in rectangular co-ordinates let us take two axes x and y, x-axis coincide with the original straight axis of the beam and the y – axis shows the deflection.

Futher,let us consider an element ds of the deflected beam. At the ends of this element let us construct the normal which intersect at point O denoting the angle between these two normal be di

But for the deflected shape of the beam the slope i at any point C is defined,

$$\tan i = \frac{dy}{dx} \quad \dots\dots(1) \quad \text{or} \quad i = \frac{dy}{dx} \text{ Assuming } \tan i = i$$

Futher

$$ds = R \, di$$

however,

$$ds = dx \text{ [usually for small curvature]}$$

Hence

$$ds = dx = R \, di$$

or $\boxed{\dfrac{di}{dx} = \dfrac{1}{R}}$

substituting the value of i, one get

$$\frac{d}{dx}\left(\frac{dy}{dx}\right) = \frac{1}{R} \text{ or } \frac{d^2 y}{dx^2} = \frac{1}{R}$$

From the simple bending theory

$$\frac{M}{I} = \frac{E}{R} \text{ or } M = \frac{EI}{R}$$

so the basic differential equation governing the deflection of beams is

$$\boxed{M = EI \frac{d^2 y}{dx^2}}$$

This is the differential equation of the elastic line for a beam subjected to bending in the plane of symmetry. Its solution y = f(x) defines the shape of the elastic line or the deflection curve as it is frequently called.

Relationship between shear force, bending moment and deflection: The relationship among shear force, bending moment and deflection of the beam may be obtained as

Differentiating the equation as derived

$$\frac{dM}{dx} = EI \frac{d^3 y}{dx^3} \quad \text{Recalling} \ \frac{dM}{dx} = F$$

Thus,

$$F = EI \frac{d^3 y}{dx^3}$$

Therefore, the above expression represents the shear force whereas rate of intensity of loading can also be found out by differentiating the expression for shear force

$$\text{i.e} \ w = -\frac{dF}{dx}$$

$$w = -EI \frac{d^4 y}{dx^4}$$

Therefore if 'y' is the deflection of the loaded beam, then the following important relations can be arrived at

$$\boxed{\text{slope} = \frac{dy}{dx}}$$

$$\boxed{B.M = EI \frac{d^2 y}{dx^2}}$$

$$\boxed{\text{Shear force} = EI \frac{d^3 y}{dx^3}}$$

$$\boxed{\text{load distribution} = EI \frac{d^4 y}{dx^4}}$$

Methods for finding the deflection: The deflection of the loaded beam can be obtained various methods.The one of the method for finding the deflection of the beam is the direct integration method, i.e. the method using the differential equation which we have derived.

Direct integration method: The governing differential equation is defined as

$$M = EI\frac{d^2y}{dx^2} \quad \text{or} \quad \frac{M}{EI} = \frac{d^2y}{dx^2}$$

on integrating one get,

$$\frac{dy}{dx} = \int \frac{M}{EI}dx + A \text{----} \text{ this equation gives the slope}$$

of the loaded beam.

Integrate once again to get the deflection.

$$y = \int\int \frac{M}{EI}dx + Ax + B$$

Where A and B are constants of integration to be evaluated from the known conditions of slope and deflections for the particular value of x.

Illustrative examples : let us consider few illustrative examples to have a familiarty with the direct integration method

Case 1: Cantilever Beam with Concentrated Load at the end:- A cantilever beam is subjected to a concentrated load W at the free end, it is required to determine the deflection of the beam
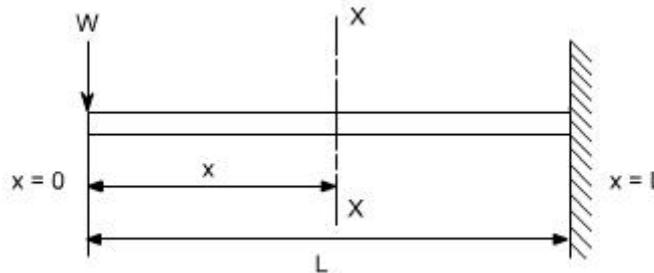
In order to solve this problem, consider any X-section X-X located at a distance x from the left end or the reference, and write down the expressions for the shear force abd the bending moment

$$S.F\big|_{x-x} = -W$$

$$B.M\big|_{x-x} = -W.x$$

Therefore $M\big|_{x-x} = -W.x$

the governing equation $\dfrac{M}{EI} = \dfrac{d^2y}{dx^2}$

substituting the value of M interms of x then integrating the equation one get

$$\frac{M}{EI} = \frac{d^2y}{dx^2}$$

$$\frac{d^2y}{dx^2} = -\frac{Wx}{EI}$$

$$\int \frac{d^2y}{dx^2} = \int -\frac{Wx}{EI}\,dx$$

$$\frac{dy}{dx} = -\frac{Wx^2}{2EI} + A$$

Integrating once more,

$$\int \frac{dy}{dx} = \int -\frac{Wx^2}{2EI}\,dx + \int A\,dx$$

$$y = -\frac{Wx^3}{6EI} + Ax + B$$

The constants A and B are required to be found out by utilizing the boundary

conditions as defined below

i.e at x= L ; y= 0-------------------   (1)

at x = L ; dy/dx = 0   -------------------   (2)

Utilizing the second condition, the value of constant A is obtained as

$$A = \frac{WL^2}{2EI}$$

While employing the first condition yields

$$y = -\frac{WL^3}{6EI} + AL + B$$

$$B = \frac{WL^3}{6EI} - AL$$

$$= \frac{WL^3}{6EI} - \frac{WL^3}{2EI}$$

$$= \frac{WL^3 - 3WL^3}{6EI} = -\frac{2WL^3}{6EI}$$

$$B = -\frac{WL^3}{3EI}$$

Substituting the values of A and B we get

$$y = \frac{1}{EI}\left[-\frac{Wx^3}{6EI} + \frac{WL^2x}{2EI} - \frac{WL^3}{3EI}\right]$$

The slope as well as the deflection would be maximum at the free end hence putting x=0 we get,

$$\boxed{y_{max} = -\frac{WL^3}{3EI}}$$

$$\boxed{(Slope)_{max}m = +\frac{wL^2}{2EI}}$$

Case 2: A Cantilever with Uniformly distributed Loads:- In this case the cantilever beam is subjected to U.d.l with rate of intensity varying w / length.The same procedure can also be adopted in this case

102
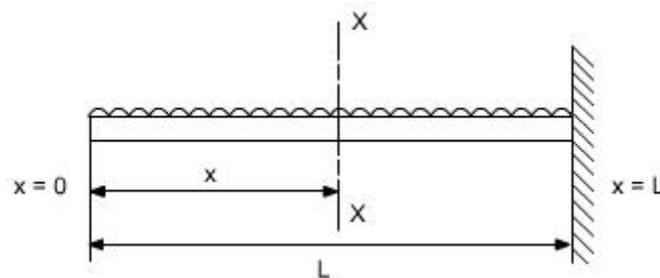
$$S.F\big|_{x-x} = -w$$

$$B.M\big|_{x-x} = -w.x.\frac{x}{2} = w\left(\frac{x^2}{2}\right)$$

$$\frac{M}{EI} = \frac{d^2y}{dx^2}$$

$$\frac{d^2y}{dx^2} = -\frac{wx^2}{2EI}$$

$$\int \frac{d^2y}{dx^2} = \int -\frac{wx^2}{2EI}\,dx$$

$$\frac{dy}{dx} = -\frac{wx^3}{6EI} + A$$

$$\int \frac{dy}{dx} = \int -\frac{wx^3}{6EI}\,dx + \int A\,dx$$

$$y = -\frac{wx^4}{24EI} + Ax + B$$

Boundary conditions relevant to the problem are as follows:

1. At x = L; y = 0

2. At x= L; dy/dx = 0

The second boundary conditions yields

$$A = +\frac{wx^3}{6EI}$$

whereas the first boundary conditions yields

$$B = \frac{wL^4}{24EI} - \frac{wL^4}{6EI}$$

$$B = -\frac{wL^4}{8EI}$$

$$\text{Thus, } y = \frac{1}{EI}\left[-\frac{wx^4}{24} + \frac{wL^3 x}{6} - \frac{wL^4}{8}\right]$$

So y_{max}m will be at x = 0

$$\boxed{y_{max}m = -\frac{wL^4}{8EI}}$$

$$\boxed{\left(\frac{dy}{dx}\right)_{max}m = \frac{-wL^3}{6EI}}$$

Case 3: Simply Supported beam with uniformly distributed Loads:- In this case a simply supported beam is subjected to a uniformly distributed load whose rate of intensity varies as w / length.

103
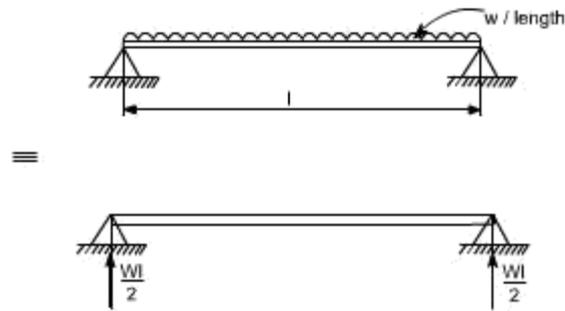
In order to write down the expression for bending moment consider any cross-section at distance of x metre from left end support.
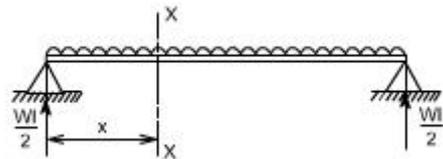
$$S.F|_{x-x} = w\left(\frac{l}{2}\right) - w.x$$

$$B.M|_{x-x} = w.\left(\frac{l}{2}\right).x - w.x.\left(\frac{x}{2}\right)$$

$$= \frac{wl.x}{2} - \frac{wx^2}{2}$$

The differential equation which gives the elastic curve for the deflected beam is

$$\frac{d^2y}{dx^2} = \frac{M}{EI} = \frac{1}{EI}\left[\frac{wl.x}{2} - \frac{wx^2}{2}\right]$$

$$\frac{dy}{dx} = \int \frac{wlx}{2EI}dx - \int \frac{wx^2}{2EI}dx + A$$

$$= \frac{wlx^2}{4EI} - \frac{wx^3}{6EI} + A$$

Integrating, once more one gets

$$y = \frac{wlx^3}{12EI} - \frac{wx^4}{24EI} + A.x + B \qquad\qquad ----- (1)$$

Boundary conditions which are relevant in this case are that the deflection at each support must be zero.

i.e. at x = 0; y = 0 : at x = l; y = 0

let us apply these two boundary conditions on equation (1) because the boundary conditions are on y, This yields B = 0.

104

$$0 = \frac{wl^4}{12EI} - \frac{wl^4}{24EI} + A.l$$

$$A = -\frac{wl^3}{24EI}$$

So the equation which gives the deflection curve is

$$y = \frac{1}{EI}\left[\frac{wLx^3}{12} - \frac{wx^4}{24} - \frac{wL^3 x}{24}\right]$$

Futher

In this case the maximum deflection will occur at the centre of the beam where x = L/2

[i.e. at the position where the load is being applied].So if we substitute the value of x = L/2

Then $$y_{max}{}^m = \frac{1}{EI}\left[\frac{wL}{12}\left(\frac{L^3}{8}\right) - \frac{w}{24}\left(\frac{L^4}{16}\right) - \frac{wL^3}{24}\left(\frac{L}{2}\right)\right]$$

$$\boxed{y_{max}{}^m = -\frac{5wL^4}{384EI}}$$

Conclusions

(i) The value of the slope at the position where the deflection is maximum would be zero.

(ii) Thevalue of maximum deflection would be at the centre i.e. at x = L/2.

The final equation which is governs the deflection of the loaded beam in this case is

$$y = \frac{1}{EI}\left[\frac{wLx^3}{12} - \frac{wx^4}{24} - \frac{wL^3 x}{24}\right]$$

By successive differentiation one can find the relations for slope, bending moment, shear force and rate of loading.

Deflection (y)

$$yEI = \left[\frac{wLx^3}{12} - \frac{wx^4}{24} - \frac{wL^3 x}{24}\right]$$

$$\frac{-5WL^4}{384EI}$$
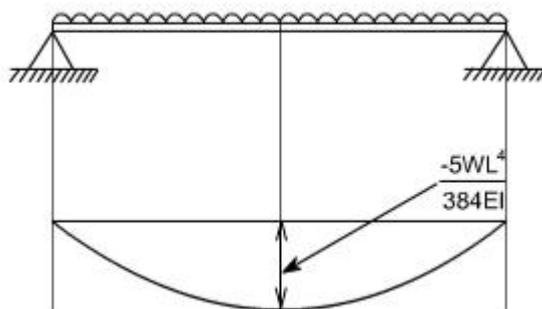
Slope (dy/dx)

$$EI.\frac{dy}{dx} = \left[\frac{3wLx^2}{12} - \frac{4wx^3}{24} - \frac{wL^3}{24}\right]$$

$$\frac{-WL^3}{24} \qquad \frac{+WL^3}{24}$$

3rd degree Polynomial
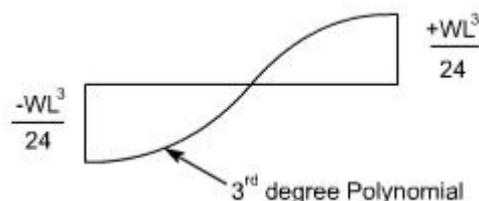
Bending Moment   So the bending moment diagram would

105

$$\frac{d^2y}{dx^2} = \frac{1}{EI}\left[\frac{wLx}{2} - \frac{wx^2}{2}\right]$$

be
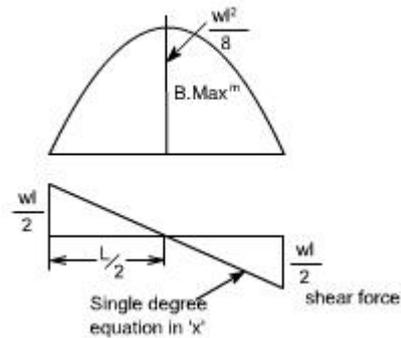
Shear Force

Shear force is obtained by

taking

third derivative.

$$EI\frac{d^3y}{dx^3} = \frac{wL}{2} - w.x$$

Rate of intensity of loading

$$EI\frac{d^4y}{dx^4} = -w$$

Case 4: The direct integration method may become more involved if the expression for entire beam is not valid for the entire beam.Let us consider a deflection of a simply supported beam which is subjected to a concentrated load W acting at a distance 'a' from the left end.
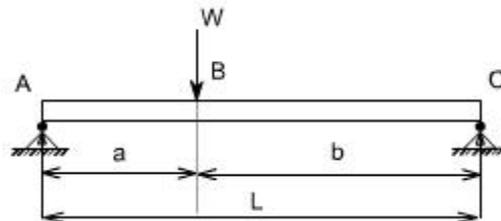
Let R_1 & R_2 be the reactions then,
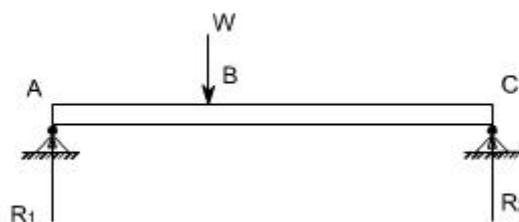
B.M for the portion AB

$M|_{AB} = R_1.x \quad 0 \le x \le a$

B.M for the portion BC

$M|_{BC} = R_1.x - W(x - a) \quad a \le x \le l$

so the differential equation for the two cases would be,

$$EI\frac{d^2y}{dx^2} = R_1 x$$

$$EI\frac{d^2y}{dx^2} = R_1 x - W(x - a)$$

These two equations can be integrated in the usual way to find 'y' but this will result in four constants of integration two for each equation. To evaluate the four constants of integration, four independent boundary conditions will be needed since the deflection of each support must be zero, hence the boundary conditions (a) and (b) can be realized.

Further, since the deflection curve is smooth, the deflection equations for the same slope and deflection at the point of application of load i.e. at x = a. Therefore four conditions required to evaluate these constants may be defined as follows:

(a) at x = 0; y = 0 in the portion AB i.e. $0 \le x \le a$

(b) at x = l; y = 0 in the portion BC i.e. $a \le x \le l$

(c) at x = a; dy/dx, the slope is same for both portion

(d) at x = a; y, the deflection is same for both

portion By symmetry, the reaction R_1 is obtained as

$$R_1 = \frac{Wb}{a+b}$$

Hence,

$$EI\frac{d^2y}{dx^2} = \frac{Wb}{(a+b)} x \qquad\qquad 0 \le x \le a \text{ --------(1)}$$

$$EI\frac{d^2y}{dx^2} = \frac{Wb}{(a+b)} x - W(x-a) \qquad a \le x \le l \text{ ---------(2)}$$

integrating (1) and (2) we get,

$$EI\frac{dy}{dx} = \frac{Wb}{2(a+b)} x^2 + k_1 \qquad\qquad 0 \le x \le a \text{ -------(3)}$$

$$EI\frac{dy}{dx} = \frac{Wb}{2(a+b)} x^2 - \frac{W(x-a)^2}{2} + k_2 \qquad a \le x \le l \text{ -------(4)}$$

Using condition (c) in equation (3) and (4) shows that these constants should be equal, hence letting

$K_1 = K_2 = K$

Hence

$$EI\frac{dy}{dx} = \frac{Wb}{2(a+b)}x^2 + k \qquad\qquad 0 \le x \le a\text{------}(3)$$

$$EI\frac{dy}{dx} = \frac{Wb}{2(a+b)}x^2 - \frac{W(x-a)^2}{2} + k \qquad a \le x \le l\text{-------}(4)$$

Integrating agian equation (3) and (4) we get

$$EIy = \frac{Wb}{6(a+b)}x^3 + kx + k_3 \qquad\qquad 0 \le x \le a\text{------}(5)$$

$$EIy = \frac{Wb}{6(a+b)}x^3 - \frac{W(x-a)^3}{6} + kx + k_4 \qquad a \le x \le l\text{------}(6)$$

Utilizing condition (a) in equation (5) yields

$$k_3 = 0$$

Utilizing condition (b) in equation (6) yields

$$0 = \frac{Wb}{6(a+b)}l^3 - \frac{W(l-a)^3}{6} + kl + k_4$$

$$k_4 = -\frac{Wb}{6(a+b)}l^3 + \frac{W(l-a)^3}{6} - kl$$

But $a+b = l$,

Thus,

$$k_4 = -\frac{Wb(a+b)^2}{6} + \frac{Wb^3}{6} - k(a+b)$$

Now lastly k_3 is found out using condition (d) in equation (5) and equation (6), the condition (d) is that,

At x = a; y; the deflection is the same for both portion

Therefore $y\big|_{\text{from equation 5}} = y\big|_{\text{from equation 6}}$

or

$$\frac{Wb}{6(a+b)}x^3 + kx + k_3 = \frac{Wb}{6(a+b)}x^3 - \frac{W(x-a)^3}{6} + kx + k_4$$

$$\frac{Wb}{6(a+b)}a^3 + ka + k_3 = \frac{Wb}{6(a+b)}a^3 - \frac{W(a-a)^3}{6} + ka + k_4$$

Thus, $k_4 = 0$;

OR

$$k_4 = -\frac{Wb(a+b)^2}{6} + \frac{Wb^3}{6} - k(a+b) = 0$$

$$k(a+b) = -\frac{Wb(a+b)^2}{6} + \frac{Wb^3}{6}$$

$$k = -\frac{Wb(a+b)}{6} + \frac{Wb^3}{6(a+b)}$$

so the deflection equations for each portion of the beam are

$$Ely = \frac{Wb}{6(a+b)}x^3 + kx + k_3$$

$$Ely = \frac{Wbx^3}{6(a+b)} - \frac{Wb(a+b)x}{6} + \frac{Wb^3 x}{6(a+b)} \qquad \text{----for } 0 \le x \le a \text{-----} (7)$$

and for other portion

$$Ely = \frac{Wb}{6(a+b)}x^3 - \frac{W(x-a)^3}{6} + kx + k_4$$

Substituting the value of 'k' in the above equation

$$Ely = \frac{Wbx^3}{6(a+b)} - \frac{W(x-a)^3}{6} - \frac{Wb(a+b)x}{6} + \frac{Wb^3 x}{6(a+b)} \qquad \text{For } \textbf{for } a \le x \le l \text{-----} (8)$$

so either of the equation (7) or (8) may be used to find the deflection at $x = a$

hence substituting $x = a$ in either of the equation we get

$$Y\big|_{x=a} = -\frac{Wa^2 b^2}{3EI(a+b)}$$

OR if $a = b = l/2$

$$\boxed{Y_{\text{max}^m} = -\frac{WL^3}{48EI}}$$

<u>ALTERNATE METHOD:</u> There is also an alternative way to attempt this problem in a more simpler way. Let us considering the origin at the point of application of the load,
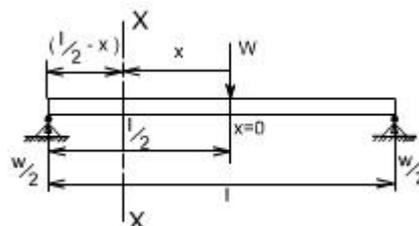
$$S.F|_{xx} = \frac{W}{2}$$

$$B.M|_{xx} = \frac{W}{2}\left(\frac{l}{2} - x\right)$$

substituting the value of M in the governing equation for the deflection

$$\frac{d^2y}{dx^2} = \frac{\frac{W}{2}\left(\frac{l}{2} - x\right)}{EI}$$

$$\frac{dy}{dx} = \frac{1}{EI}\left[\frac{WLx}{4} - \frac{Wx^2}{4}\right] + A$$

$$y = \frac{1}{EI}\left[\frac{WLx^2}{8} - \frac{Wx^2}{12}\right] + Ax + B$$

Boundary conditions relevant for this case are as follows

(i) at x = 0; dy/dx=

0 hence, A = 0

(ii) at x = l/2; y = 0 (because now l / 2 is on the left end or right end support since we have taken the origin at the centre)

Thus,

$$0 = \left[\frac{WL^3}{32} - \frac{WL^3}{96} + B\right]$$

$$B = -\frac{WL^3}{48}$$

Hence he equation which governs the deflection would be

$$y = \frac{1}{EI}\left[\frac{WLx^2}{8} - \frac{Wx^3}{12} - \frac{WL^3}{48}\right]$$

Hence

$$Y_{max^m}\big|_{at\,x=0} = -\frac{WL^3}{48EI} \quad \text{At the centre}$$

$$\left(\frac{dy}{dx}\right)_{max^m}\bigg|_{at\,x=\pm\frac{L}{2}} = \pm\frac{WL^2}{16EI} \quad \text{At the ends}$$

Hence the integration method may be bit cumbersome in some of the case. Another limitation of the method would be that if the beam is of non uniform cross section,
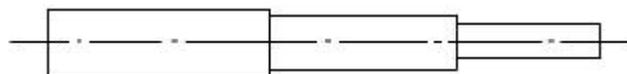
i.e. it is having different cross-section then this method also fails.

So there are other methods by which we find the deflection like

1. Macaulay's method in which we can write the different equation for bending moment for different sections.

2. Area moment methods