**Experiment 3**

**The Beam**

**Ali Almoslim**

**EGME-306A**

**Mo-8:30a.m**

**ABSTRACT**

The main objectives of this experiment are to determine the stress, deflection, and the strain of a supported beam under load. Also to experimentally verify the beam stress and flexure formulas. The experiment was accomplished by using the machine which applies a load to a supported beam and measure the deflection and the strain of it. The moment of inertia was 0.0845 and deflection of 0.0083%.

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**INTRODUCTION AND THEORY**

Structural members are usually designed to carry tensile, compressive, or transverse loads. A member which carries load transversely to its length is called a beam. In this experiment, a beam will be symmetrically loaded as shown in Fig. III-1(a), where *P* is the applied load. Note that at any cross section of the beam there will be a shear force *V* (Fig. III-1(b)) and moment *M* (Fig. III-1c). Also, in the central part of the beam (between the loads *P*/2) *V* is zero and *M* has its maximum constant value. Notice the sign convention of a positive moment, *M*, causing a negative (downward) deflection, *y*.

If in this part a small slice *EFGH* of the beam is imagined to be cut out, as shown, then it is clear that the external applied moment, *M*, must be balanced by internal forces (stresses) at the sections (faces) *EF* and *GH*. For *M* applied as shown in Fig. III-2(a), these forces would be compressive near the top, *EG*, and tensile near the bottom, *FH*. Since the beam material is considered elastic, these forces would deform the beam such that the length EG would tend to become shorter, and *FH* would tend to become longer. The first fundamental assumption of the beam theory can be stated as follows:

“Sections, or cuts, which are plane (flat) before deformation, remain plane after deformation.”

Thus, under this assumption, the parallel and plane section EF and GH will deform into plane sections E’F’ and G’H’ which will intersect at point O, as shown in Fig. III-2(b). Since E’F’ and G’H’ are no longer parallel, they can be thought of as being sections of a circle at some radial distance from O. Convince yourself of this by drawing a square on an eraser and observe its shape when you bend the eraser. Since the forces near E’G’ are compresiive, and those near F’H’ are tensile, there must be some radial distance *r* where the forces are neither compressive nor tensile, but zero. This axis, N-N, is called the neutral axis. Notice that N-N is not assumed to lie in the center of the beam.

Consider an arc of distance +η, from the neutral axis, or distance *r* + η from O (Fig. III-2(b)). At this radius, the length of arc is *l*’=(*r* + η) Δθ. As shown in Fig. III-2(a), the length of the arc was *l* before the deformation. This length is also equal to *r*Δθ (because at N-N there are no forces to change the length). Thus, the strain at distance +η from the neutral axis can be found by:

(III-1)

In other words, the axial strain is proportional to the distance from the neutral axis. It is remarked that this strain is positive, because positive η was taken on the tensile side of N-N in Fig. III- 2(b). Had η been taken in the opposite direction, then the strain would have been negative, as appropriate for the compressive side.

The second fundamental assumption is that Hooke’s Law applies both in tension and compression with the same modulus of Elasticity. Thus, from Eqs. (I-3) and (III-1),

(III-2)

If *c* is the maximum distance from the neutral axis (largest positive or negative value of η), then

the maximum stress (compressive or tensile) is given by σm = *Ec*/*r*, and Eq. (III-2) can also be written as

(III-3)

That is, the stress at a section EF or GH, due to applied moment *M*, varies linearly from zero at

the neutral axis to some maximum value σm (positive or negative) when η = *c*. To obtain the

beam stress formula, it remains to define where the neutral axis is located, and to relate σm to *M*.

To locate the neutral axis, it is observed that the tensile and compressive forces on a section are equal to the stress times a differential element of area, as shown in Fig. III-2(c). For static equilibrium, the sum (or integral) of all these internal forces must be zero. That is,

where, the integrals are over the whole cross-sectional area. Thus, it is seen that the neutral axis is located such that the first moment of area about it is zero; that is, the neutral axis passes through the centroid of the cross-sectional area. In Fig. III-2(c), a rectangular area was used for illustration; however, any shape of vertically symmetric cross-sectional area is valid for the area integral.

In a similar fashion, the moment due to all the forces is the sum (or integral) of the forces times their moment arms about the neutral axis, and this must be equal to the external applied moment.

Thus,

(III-4)

If *I* is defined as the second moment of area about the neutral axis, commonly called the moment

of inertia,

(III-5)

then Eq. (III-4) can be written as:

(III-6)

where *Z* = *I*/*c* is the section modulus, which depends only on the cross-sectional geometry of the beam. Equation (III-6) is the beam stress equation which relates the maximum (compressive or tensile) stress to the applied moment. Notice its similarity to Equation (I-1), the stress equation for uniaxial tension. It is understood, of course, that σm is the maximum bending stress at a particular location, *x*, along the beam. In general, both σm and *M* are functions of *x*, and are related by Eq. (III-6).

The remaining question about the beam concerns its degree of deformation, or flexure. That is, how is the radius of curvature, *r*, related to the moment *M* (or load *P*)? From calculus, it can be shown that the curvature of a function *y*(*x*) is given by

Thus, if *x* is the distance along the beam, *y* will be the deflection as indicated in Fig. III-1(a). For most beams of practical interest, this deflection will be small, so that the slope *dy*/*dx* will be very

small compared to 1. Hence, a very good approximation is

But, since σm = *Ec*/*r* = *Mc*/*I*, there results the differential equation of the elastic curve:

(III-7)

To obtain the elastic curve of the beam, *y*(*x*), and the maximum deflection, *ym*, it is necessary to integrate Eq. (III-7) using the moment function *M*(*x*) in Fig. III-1(c). Thus, using *M*(*x*) = *Px*/2 for

0 ≤ *x* ≤ *a* and *M*(*x*) = *Pa*/2 for *a* ≤ *x* ≤ *a* + *b*, it is found that

and that the maximum deflection at *x* = *a* + *b*/2 is

(III-8)

In particular, for *a* = *b* = *L/*3,

(III-9)

**PROCEDURES**

This experiment will be applied on a 1018 steel beam ,which has *E* = 30x10\*6 psi, using the MTS testing machine. Because the beam is not perfectly symmetric we had to measure the cross-sectional dimension carefully and the location of the loading points. After that, we placed the beam in the testing machine and aligned the 12-inch black marks on the beam with roller supports of the lower fixture. We had to make sure the beam is centered on the lower support with strain gauge facing down. Then we went to the computer and entered the *TestWork 4* software. When prompted, we had to make sure the name field under login says “306A\_Lab” then clicked OK to login then under the Open Method dialog, we had to select“exp-3 4 Point Flex Mod X” after that we had to select the Motor Reset button right corner by clicking on it. We zeroed the load by right clicking on the load cell icon and selecting zero channel. Then, we had position the upper bending fixture over the beam using the handset after enabling the handset by pressing the unlock button, then slowly lowering the crosshead using the down arrow until the fixture is touching the beam. After that, we used the thumb wheel of the handset to lower the fixture onto the beam. We watched for the load reading increase when the upper fixture makes contact. Then, we slowly lifted the fixture until a very slight preload of approximately 0.2 lb is applied. Then we locked the handset. Then we connected the strain gauge wire to the #1 strain channel of the grey DAQ box, data acquisition is conducted by the *LabVIEW* software. We took the magnetic base holding the dial indicator, and positioned it with the dial indicator in the center of the beam on the bottom side. We made sure the dial indicator is not touching the strain gauge, after that we locked the MTS frame by activating the magnetic base then zeroed the dial indicator. Next, we started the *LabVIEW* software by double click the icon on the desktop, selecting “Open” and double clicking “exp2&3-Strain Mod-15 LV7.1” then we pressed the white arrow to start the strain gauge acquisition then we zeroed the strain. After that we pressed the green arrow on the *TestWorks* 4 GUI. We loaded the beam up to 1000 lb for each in 100 lb and recorded the data. After reaching 1000 lb we repeated the experiment after making sure it zeroed out just to assure accuracy of our work.

**SUMMARY OF IMPORTANT RESULTS**

**I**

In graph I. we can notice that it’s a deflection vs load diagram as titled. we have the theoretical values and the experimental ones. As we can see from the graph the is a noticeable difference between the two values as the two lines get different slops.

II

As we can see we, graph II is about the strees vs load diagram, where the load is the x-axis and the stress is the y-axis which is represented in a straight line as shown.



**SAMPLE CALCULATION AND ERROR ANALYSIS**

To find the  use the following equation:

Using the position points, calculate the neutral axis, but it is important to remember that this value isn’t the maximum neutral axis, which is what is needed. In that case, simply subtract total distance by the calculated neutral axis.

The moment of Inertia is calculated by the following formula:

Manually determined moment of Inertia

To find the section modulus we use the formula:

Where I is the calculated moment of Inertia and c is the distance from the neutral axis

To find the maximum bending stress we start with the following equation:

Where,

P (maximum load) = 997.3 lb

a (distance away from reference) = 4 in

Providing the applied moment of,

Now the maximum bending stress can be found using Eq. (1)

The deflection is determined by the following formula:

Or

Now, Eq. (9) is used to find the deflection of the crossbeam given the maximum pressure, moment of inertia, and distance between loads on the beam. Using the previous maximum load, initial length of 12 in, initial modulus elasticity and calculated moment of Inertia, the maximum deflection comes out to be…

The Stress and Deflection percent error were calculated using the results obtained by the load applied by the MTS machine and the dial indicator and a thermometer.

*Stress (*)  % Error = x 100% = x 100% =  1.20%

*Deflection*  % Error = x 100% = x 100% = .0083 %

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  | Ai | Yi | Ay | D | AD^2 | I |
| 1 | 0.261 | 0.1295 | 0.034 | 0.3875 | 0.039 | 0.00146 |
| 2 | 0.116 | 0.524 | 0.061 | 0.007 | 5.7x10^-6 | 2.7x10-3 |
| 3 | 0.253  0.63 | 0.915 | 0.231  0.326 | 0.398 | 0.4 | 1.36x10^-6 |

D1= 0.517 – 0.1295 = 0.3875 in Y=∑yA/∑A = 0.326/0.63 = 0.517

D2= 0.524 – 0.517 = 0.007 in

D3= 0.915 – 0.517= 0.398 in

I1= (bh^3)/12 = (1.007(0.259)^3)/12= 1.46x10^-3 in ^4, I2= 2.714x10^-3 in^4, I3= 1.36x10^-3 in^4

Moment = 0.039 + 5.7x10^-6 + 0.04 + 1.4x10^-3 + 2.714x10^-3 + 1.36x10^-3 = 0.0845 in^4

**Error Analysis**

Human error is a common one that could happen in any experiments, in this example, human error might happened in the setup of the equipment. Another possibility is that we didn’t get the write readings. Also, the machine could be old and not accurate enough. Last one is that the sample we tested might not manufactured properly. So these reasons explain why we got 1.20% error for stress and 0.0083% for deflection.

**DISCUSSION AND CONCLUSION**

After finishing the experiment and calculating the data and graph them,we figured out many features of the beam. Whenever the load increases, the deflection and the stress increase too. We can use the strain to find the theoretical stress and we can use moment, moment of inertia, and the neutral axis to find the experimental stress. And about the deflection, we used the dial indicator to find it directly from the experiment for the experimental and we used load, length, modulus of elasticity, and the moment of inertia to find the theoretical data. The graph clearly shows that the load and deflection and load and stress have a directly proportional relationship.

**REFRENCES**

[1] CSUF EGME 306A Lab Manual.

**Appendix**

|  |  |  |
| --- | --- | --- |
| deflection(in) | Max Stress at B | |
| 0.0009 | 1225.223179 | |
| 0.002 | 2450.446358 | |
| 0.0033 | 3663.417306 | |
| 0.0046 | 4888.640485 | |
| 0.0059 | 6113.863664 | |
| 0.0073 | 7351.339075 | |
| 0.0086 | 8539.805559 | |
| 0.0099 | 9814.037665 | |
| 0.0103 | 11049.06263 | |
| 0.0107 | 12217.92554 | |
| Distance of Neutral Axis | | Moment of Inertia | | Applied Moment | Modulus of Elasticity |  |
| 0.5179 | | 0.0845397 | | 200 | 30000000 |  |
|  | |  | | 400 |  |  |
|  | |  | | 598 |  |  |
|  | |  | | 798 |  |  |
|  | |  | | 998 |  |  |
|  | |  | | 1200 |  |  |
|  | |  | | 1394 |  |  |
|  | |  | | 1602 |  |  |
|  | |  | | 1803.6 |  |  |
|  | |  | | 1994.4 |  |  |