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Design optimization and Modal Analysis of Cantilever I-section Beam For 0.5 ton capacity of Floor Mounted Jib Crane

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Abstract: **Today's industry demands versatile, efficient, and cost effective equipment while at the same time providing more flexibility along with significant savings through increased productivity. Most of the researcher's did work on static analysis of I-section for by using Analytical and Software basis. One of them ISMB-250 beam are taken for 0.5 ton capacity of Floor mounted jib crane. The purpose behind the design of I-section cantilever beam for 0.5 ton capacity of Floor mounted jib crane is that to reduce the cost, make more efficient and to provide more flexibility. Hence instead of ISMB-250 beam, ISMB-200 beam is selected for reducing cost, more efficient and more flexibility. To check behavior of ISMB200 beam for Floor mounted jib crane in the form of stresses & deformation using Analytical, ANSYS and Experimental (Strain gauge technique). The additional part in this project is Modal analysis. The purpose of modal analysis is to measure the dynamic vibration characteristics such as (Natural frequency, deflection) by using Analytical and Experimental (FFT Analyzer). The differences error between Analytical and FEA values of stress and deformation in the form of percentage are 18% and 8%. Similarly the stresses error difference between Analytical and Experimental are approximately near to 4%. The difference error between the natural frequencies is varying from 4% to 18%. From observing the below results, for lifting 500kg load the optimum I-section beam for Floor mounted jib crane is ISMB200. Due to this selected beam cost is reduced by 30% as compare to existing (ISMB250 beam). Future scope in this project is to perform harmonic response analysis using experimental and numerical.**

Keywords: **Design, structural analysis, modal analysis, strain gauge technique.**

I. INTRODUCTION

Now a day's material handling system manifests anisotropic behaviour. A jib crane can help to improve material handling efficiency and work flow. Serious consideration should be given to jib cranes for applications requiring repetitive lifting and transferring of loads within a fixed arc of rotation. Jib cranes are some type of lifting and hoisting machines used in the industry commonly. Their design and manufacturing processes are developing from day to day because of economic conditions and rivalry between companies. These factors enforce them to decrease the time used for the design and production of machines. For these machines, some classification studies were done in catalogues and books. Jib cranes are either attached to a building column or cantilever vertically from an independent floor mounted column. Floor mounted jib cranes is directly fixed on the floor without any support to keep it upright. To increase the lifting capacity and minimize the cost and time for building a ship and offshore structure, block lifting with multi-cranes becomes more and more important. There are different types of jib cranes, but we analysis of floor mounted jib crane In this study, designing of I- section cantilever beam of capacity 0.5 ton for floor mounted jib crane The jib crane requires at least two times Maintenance in a year. The current task has been taken to reduce the servicing required & to ensure smooth functioning without any breakdown. The important parameters are determined such as stresses, deflection, & lateral torsional buckling behaviour of regular I section cantilever beam of floor mounted jib crane subjected to uniformly distributed load(own-weight) & a concentrated of load at free end, is done by using static analysis. This project investigate stresses deflection &

natural frequency using Analytical, finite element analyses & experimental and compare results with each other till day many researcher studied on static analyses but in this project we studied static as well as modal analysis. The following crane design has been considered according to Indian standards. The detail classification of the crane is specified. Floor mounted jib crane of 0.9m span and having load carrying capacity of 500kg is selected for the analysis. The design factor for the stresses in the crane is based on the capacity plus 25% of the rated load for impact and 15% of the rated load for the weight of the hoist and trolley. Generally, this is used all along with the average yield stress of the material to find out the type of the design. This design provides a margin to allow for variations in material properties, operating conditions, and design assumptions. No crane should be supposed to ever, in any circumstance, be weighted beyond its rated capability. Jib crane have three degrees of freedom. They are vertical, radial, and rotary. However they cannot reach into corners. They are usually used where activity is localized.

Modal analysis is a process to determine the dynamic vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. It has become a major alternative to provide a helpful contribution in understanding control of many vibration phenomena which encountered in practice. In this work we compared the natural frequency for I- sectional beam using analytical and experimental approach. In engineering field vibration behavior of an element plays a key role without which it is incomplete. Resonance is a key aspect in dynamic analysis, which is the frequency of any system matches with the natural frequency of the system which may lead to catastrophes or system failure. Modal analysis has become a major alternative to provide a helpful contribution in understanding control of many vibration phenomena which encountered in practice. The main purpose of modal analysis is to obtain the data that will be used for other vibrational analysis. Practical applications the modal parameters are required to avoid resonance in structures affected by external periodic dynamic loads. There are various practical applications of modal analysis over various fields of science, engineering and technology like investigations related to aeronautical engineering, automobile engineering and mechanical engineering.

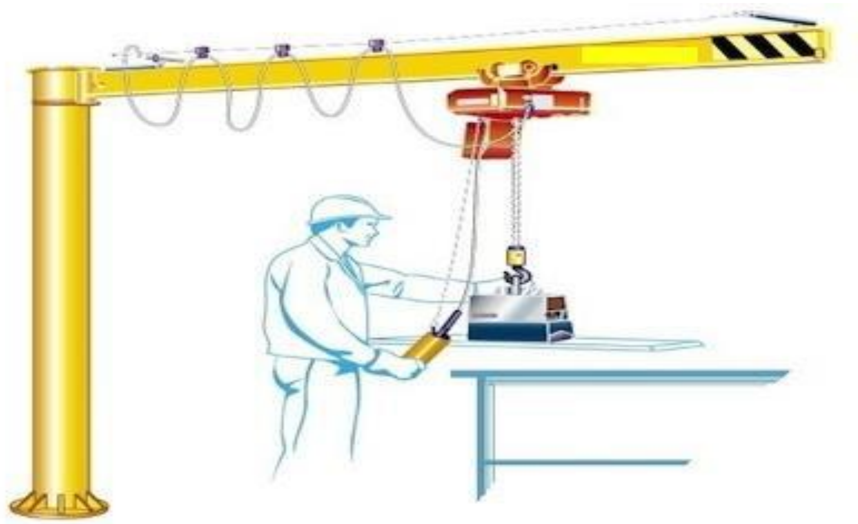


Fig 1: Floor Mounted Jib Crane.

II. PROBLEM DEFINITION

To find out optimum I-type beam for 0.5 ton capacity of FMJC. To check behavior of optimum I –type beam i.e. ISMB-200 for Floor mounted jib crane in the form of stresses & deformation using Analytical, ANSYS and Experimental (Strain gauge technique). Also to check the additional parts in this project are dynamic vibration characteristics of I-section beam i.e. the natural frequency and deformation by Analytical and Experimental approach.

III. OBJECTIVES OF THE STUDY

- To reduce cost of structure, make more efficient and flexible.
- To check behavior of optimum I-beam (i.e. stresses, deformation and natural frequencies).
- To compare analytical, Numerical and experimental result.

IV. ANALYTICAL DESIGN

Performance characteristics of the Floor Mounted Jib Crane are depending on its movement

Work requirement:

Capacity: 0.5 ton, Rotation:270^0, Height of lift: 1000mm, Span length: 900mm

Support: Floor

Selection of I-section: Type: Indian Standard Medium Beam (ISMB)

Material: Structural Steel

A]. Existing I-beam design for 0.5 ton capacity of FMJC.

Table 1: Specification of ISMB250 (from Indian std. catalogue)

Designation	Depth (mm)	Width (mm)	Web thick. (mm)	Flange thick. (mm)	Area (mm^2)
ISMB250	250	125	6.9	12.5	4750

Mass per unit length: 37.4Kg/m

Self-Weight: 32kg

Table 2: Properties of Structural Steel

Properties	**Value**
Density	7850 (kg/m^3)
Poisson ratio	0.3
Young's modulus	2.1×10^5 (MPa)
Tensile Ultimate Strength	460 (MPa)
Compressive Yield strength	250 (MPa)
Tensile Yield strength	250 (MPa)

Actual load carried by the beam: Taken Approximately.

- Lifted load: 500kg
- Hoist load: 150kg
- Dead load: 315kg
- Total load: 315+500+150= 965kg

1. Stress at free end.

I_{xx}= $51.31\text{x}10^6 mm^4$

M= total load x length

According to flexural formula,

$$\frac{\sigma b}{y} = \frac{M}{I} \ldots\ldots\ldots\ldots [1]$$

$$\sigma b = 20.756MPa \ldots\ldots\ldots.l = 900mm$$

$$\sigma b = 115.316MPa \ldots\ldots\ldots\ l = 5000mm$$

$$\sigma b < Syt \ldots\ldots\ldots (ISO\ Std.)$$

2. Stress at mid point.

$$\sigma b = 10.378MPa$$

$$\sigma b = 57.65MPa$$

3. Deflection at free end.

$$\delta 1 = \frac{WL^3}{3EI} = 0.2334mm \ldots\ldots\ldots\ldots l = 900mm$$

$$\delta 2 = \frac{WL^3}{3EI} = 38.21mm \ldots\ldots\ldots\ldots . l = 5000mm$$

4. Flexural Rigidity(EI)= 10.775×10^{12} Nmm^2

5. Factor of Safety:

$$FOS = \frac{Syt}{\sigma b} = 2.06$$

B] Selected Beam- ISMB200

Material: Structural Steel

Table 3: Specification of ISMB200 (from Indian std. catalogue)

Designation	Depth (mm)	Width (mm)	Web thick. (mm)	Flange thick. (mm)	Area (mm^2)
ISMB200	200	100	5.9	10.8	3080

Properties	**Value**
Density	7850 (kg/m^3)
Poisson ratio	0.3
Young's modulus	2.1×10^5 (MPa)
Tensile Ultimate Strength	460 (MPa)
Compressive Yield strength	250 (MPa)
Tensile Yield strength	250 (MPa)

Mass per unit length: 25.4Kg/m

Self-Weight: 21kg

Capacity: 0.5 ton

Rotation:270^0, Height of lift: 1000mm, Span length: 900mm, Support: Floor

Selection of I-section: Type: Indian Standard Medium Beam (ISMB)

Trolley connected hoist model and weight CBSG(CBSP)010- 40kg

Exact Load calculation for 0.5ton capacity of FMJC:

Capacity-0.5ton

Lifted load= $\frac{500x9.81}{1000}$= 4.9KN

Impacting lifted load is 25% of lifted load.

Impacting LL=0.25*4.9= 1.225KN

Total lifted load= 4.9+1.225= 6.125KN

Trolley load= $\frac{40x9.81}{1000}$= 0.389KN

Trolley load due to dead load factor=1.2*0.389=0.468KN

Total Trolley load=0.468+0.389=0.857KN

Dead load=25.4Kg/m

For 5m length of span of mass =5*25.4=127Kg

Total dead load= 127*9.81=1.245KN

Total load=6.125+0.857+1.245=8.227KN=8227N

- **Static Analysis:**

1. Stress at free end.

I_{xx} = 22.2049x10^6mm^4, Syt=250MPa

$$\sigma b = 34.35MPa \dots\dots\dots. l = 900mm$$

$$\sigma b = 165.47MPa \dots\dots\dots. l = 5000mm$$

$$\sigma b < Syt \dots\dots\dots (ISO\ Std.)$$

2. Stress at mid point.

$$\sigma b = 17.17MPa$$

$$\sigma b = 82.73MPa$$

3. Deflection at free end.

$$\delta 1 = \frac{WL^3}{3EI} = 0.428mm \dots\dots\dots\dots l = 900mm$$

$$\delta 2 = \frac{WL^3}{3EI} = 48.841mm \dots\dots\dots\dots. l = 5000mm$$

4. Flexural Rigidity(EI)= 5.42x10^{12}Nmm^2

5. Factor of Safety:

$$\text{FOS} = \frac{Syt}{\sigma b} = 1.51$$

Similar procedure has follow for other two beams i.e. ISMB225 and ISMB175. Its values are shown in below table.

Table 4: Stress & Deformation of various types of beam

Designation	Stress (MPa)	Deflection (mm)	FOS
ISMB250	a) 20.76 (L=900) b)115.31 (L=5000)	a) 0.233 b) 38.21	2.01
ISMB225	a) 27.42(L=900) b)138.134(L=5000)	a) 0.323 b) 54.25	1.78
ISMB200	a) 34.35 (L=900) b) 165.47 (L=5000)	a) 0.424 b) 48.84	1.51
ISMB175	a) 56.586(L=900) b) 303.15(L=5000)	a) 0.82 b)138.48	0.82

Factor of safety is generally taken in the range of 1.5-3 for every industrial machine or equipments. But the calculated fos is in the standard range. Calculated value of bending stress for 5000m length of beam is 165.47MPa, this getting value is less than the permissible tensile yield strength (250MPa). Hence from the above calculation, we conclude that the ISMB200 beam is safe.

- **Analytical Optimization Method**

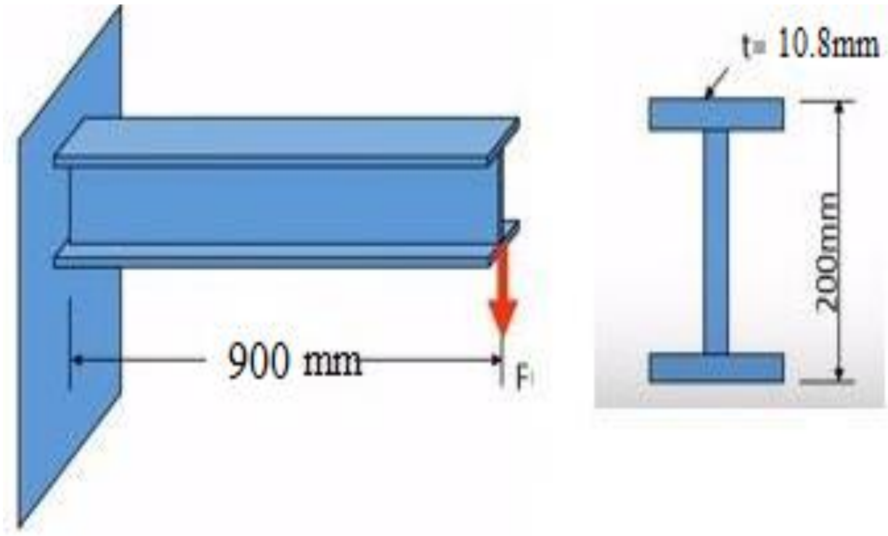


Fig 2: Cantilever beam.

The objective of the problem is to minimize the weight, cost or equivalently the cross-sectional area.

$$f(x) = \rho l(2(BxTf) + (TwxH))$$

where *B* and *H* are width and height of the cross section, and Tw and Tf are thickness of web and flange respectively, and the design variables are x $\rho l(2(BxTf) + (TwxH))$.

Three constraints are considered. The first constraint is that the maximum stress at the fixed end of the cantilever beam is less than the yield strength S_{yt} = 250Mpa.

$$g^1(x) = \frac{MY}{I} - Syt \leq 0$$

Where M= Total load x Length, I=22.20x10^6mm^4, Syt=250Mpa

The second constraint is that the tip displacement does not exceed an allowable value δmax,

$$\delta max = \frac{1}{10} the\ span\ length$$

$$g^2(x) = \frac{wl^3}{3EI} - \delta max \leq 0$$

$$g^2(x) = -1.16mm$$

Third constraint is the maximum shear stress does not exceed an allowable shear stress τ

$$g^3(x) = \frac{MY}{2I} - \tau_{\max} \leq 0$$

$$g^3(x) = -36.05MPa$$

According to max shear stress theory

$$\tau_{\max} = \frac{Syt}{2Fos} = 0.5*Syt=125MPa$$

Hence from the above calculation, we conclude that the ISMB200 beam is optimum beam for 0.5ton capacity of floor mounted jib crane.

- **Numerical Analysis:**

Bending stress, total deformation, mode shapes and natural frequency of a 0.5 ton capacity of cantilever floor mounted jib crane taken by ANSYS Workbench 14.5 Software

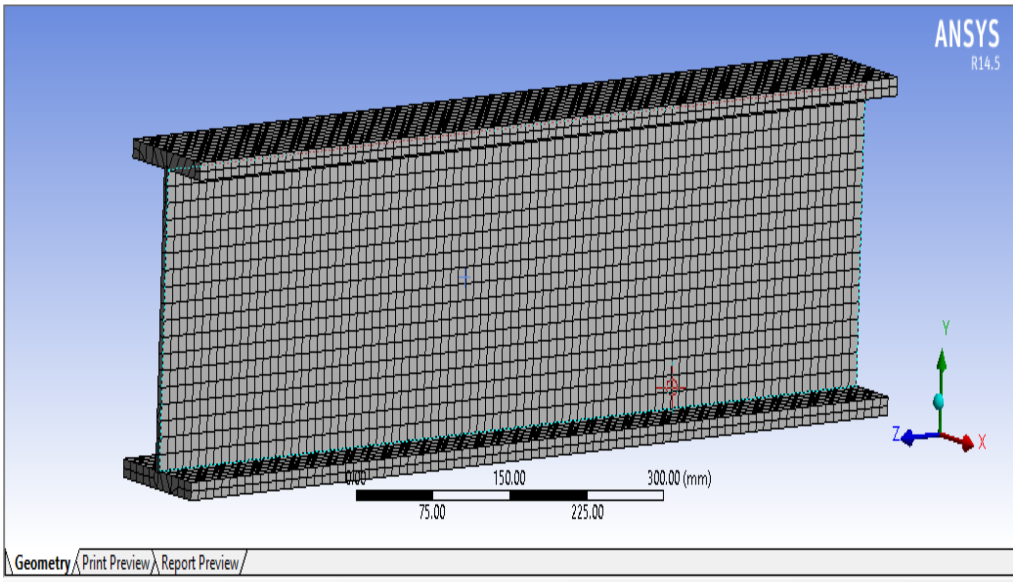


Fig 3: Meshing I-section model

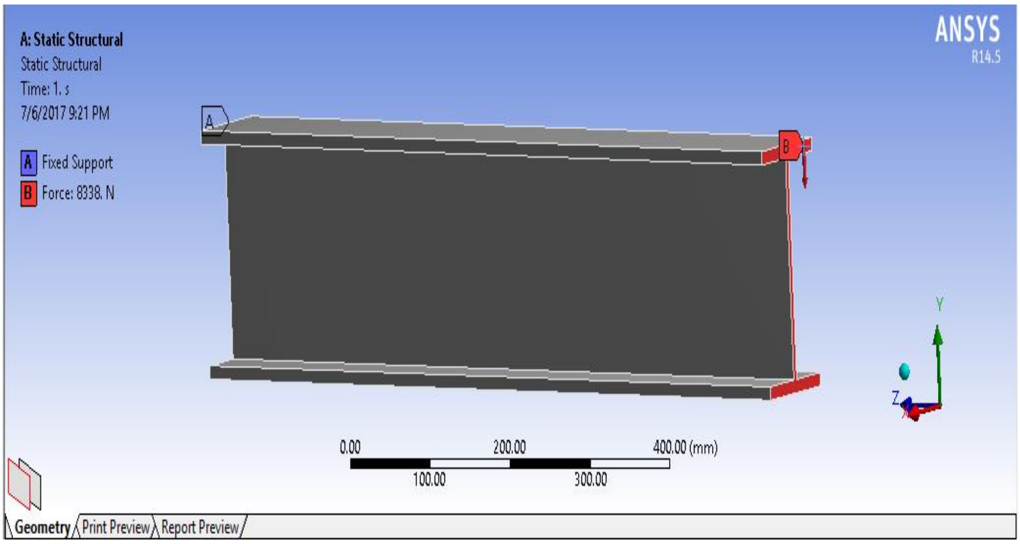


Fig 4: Applying Boundary and loading condition.

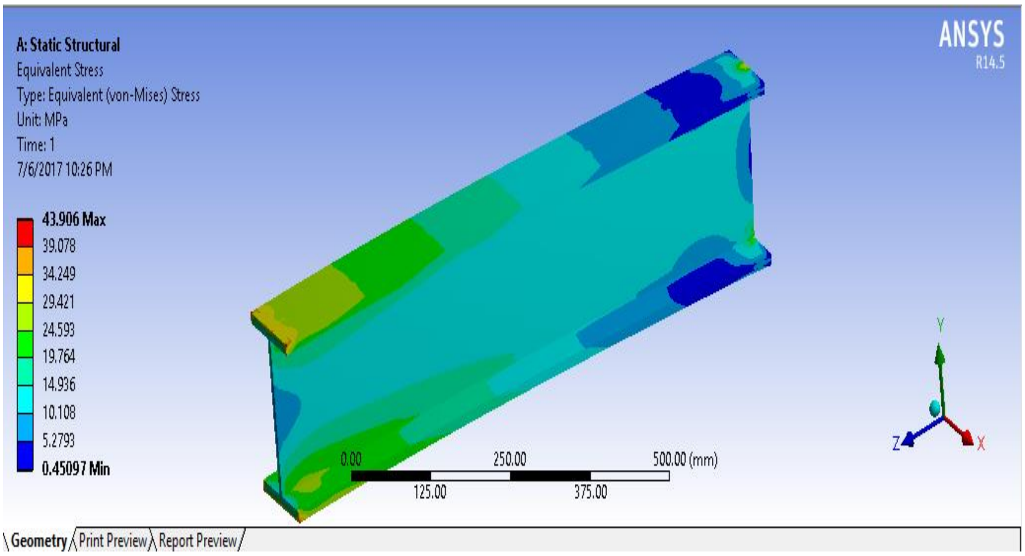


Fig 5: Von-Misses stress at load free end.

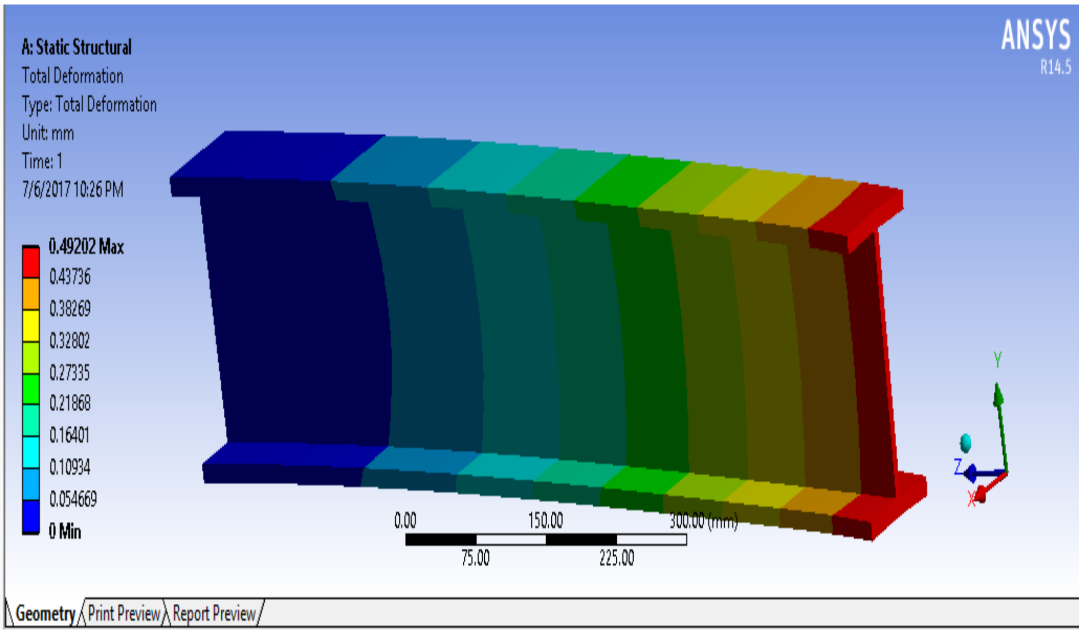


Fig 6: Deformation at free end.

- **Experimental Stress Analysis:**

Strain gauges are used as sensors in many systems to measure forces, moments, and the deformations of structures and materials. This experiment deals with measuring the strain in a cantilever beam through the use of four resistance strain gages; two mounted on top of the beam and two mounted below. A static load will be incremented at different locations along the beam to produce measurable strains.

Table 5: Strain gauge Specification:

Type	FLA-5-11
Gauge Length	5mm
Gauge Factor	2.13 ±1%
Gauge resistance	120±0.3Ω
Temp. Compensation	11

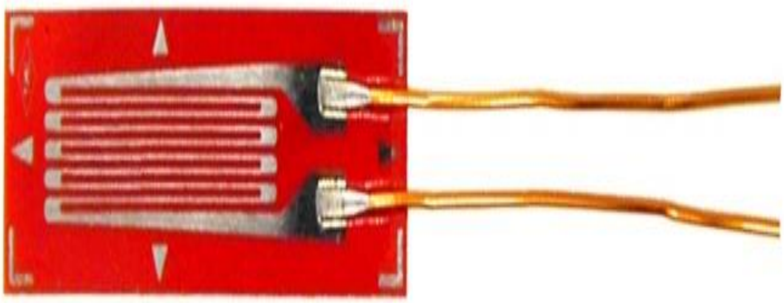


Fig 7: Bonded metal foil strain gauges.

Fig 8: Experimental set up for strain measurement.

Steps for Experimental Analysis:

- One end of the beam fixed with the help of C-Clamp to the seating stool as shown in above fig.
- Strain gauge paste at fixed end.
- Strain gauge wire connects to high performance portable TDS-530 digital data logger.
- Weight hang to the free end of the beam
- Take the reading from digital 30 channel data logger.
- The result is displayed on the screen of TDS-530 Data logger.

Table 6: Experimental Strains and Stresses

Load (N)	Experimental Strain (μ)	Experimental stress(N/mm^2)
50	2	0.42
100	3	0.63
150	4	0.84
200	5	1.05
8238	166	35.75

V. MODAL ANALYSES

➢ **Analytical natural frequencies for optimum beam.**

Theoritical Circular frequncies of I-section cantilever beam has determined by using following formulaes,

$$\omega = (\beta nl)^2 \sqrt{\frac{EI}{\rho A l^4}} \quad \ldots\ldots\ldots\ldots\ldots[2]$$

(βnl) = constant relative to vibration bound condition

ω = circular frequency.

Relation between natural frquency and circular frequency is,

$$fn = \frac{\omega}{2\pi} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots[3]$$

fn = Natural frequency

Where: $\beta_1 l$=1.875, $\beta_2 l$=4.694, and ; For $n \geq 3$ $\beta nl = (2n-1)\pi/2$

1. Natural frequency at mode first

Ƒn1= 15.86Hz………… β1l=1.875

2. Natural frequency at mode second

Ƒn2= 101.39Hz……... β2l=4.694

3. Natural frequency at mode Third

Ƒn3= 253.79Hz……… β3l=7.855

4. Natural frequency at mode Forth

Ƒn4=542.26Hz………. β4l=10.995

5. Natural frequency at mode Fifth

Ƒn5=822.036Hz………. β5l=14.137

- **Experimental Modal Analysis:**

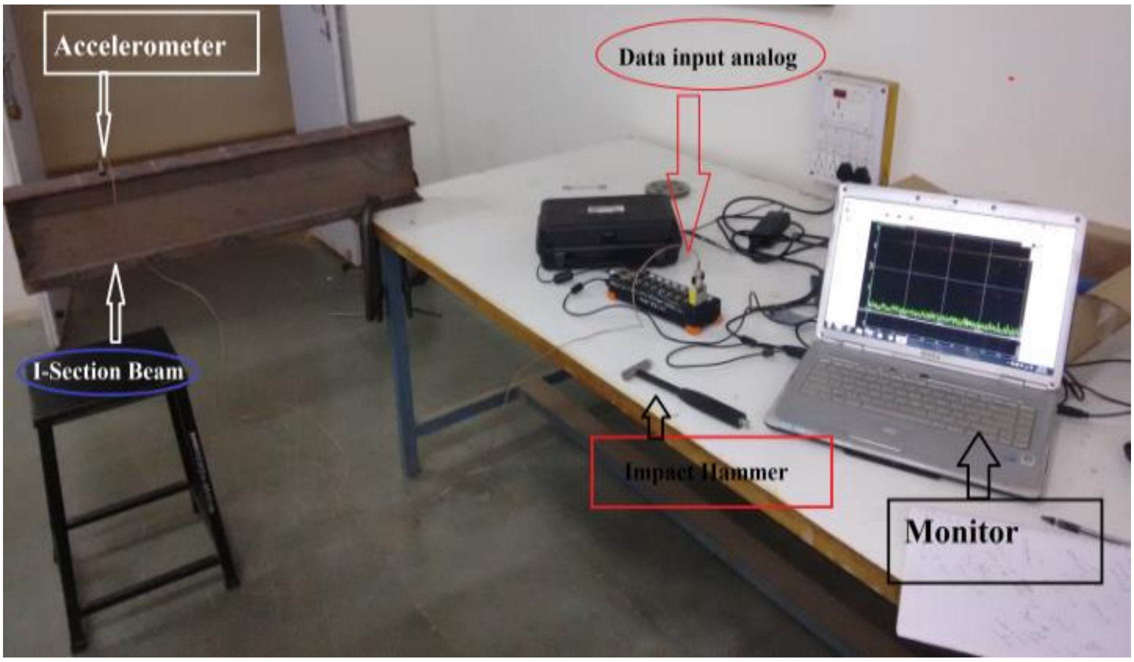


Fig 9: Experimental Set up.

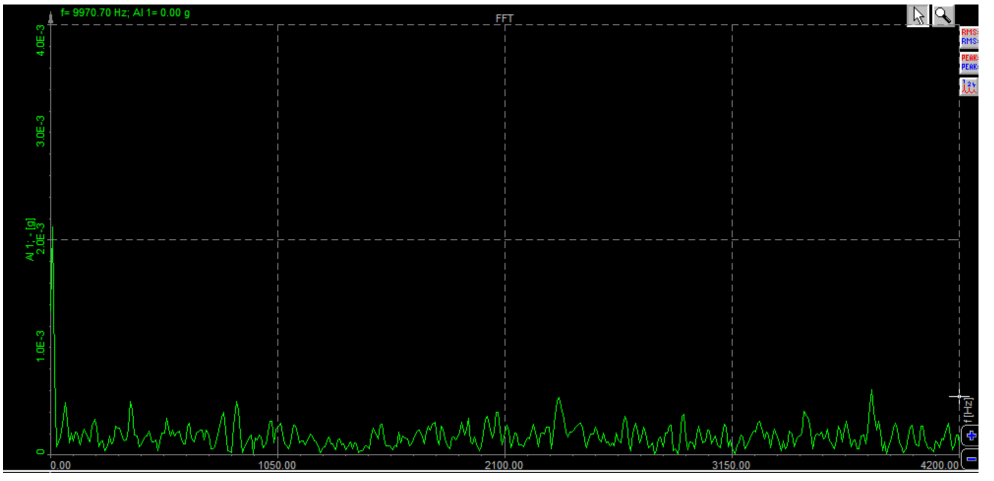
Steps for Experimental Analysis:

- One end of the beam fixed with the help of C-Clamp on the Table as shown in above fig.
- Accelerometer is connected to beam and Data input analog, the output is given to Computer.
- Power is given to PC and FFT.
- Open the DEWESoft7 in PC.
- Fixed the Accelerometer at the free end of beam and applied the load by using Impact hammer at different modes. Uni-axial IEPE Accelerometer is used.
- The result is displayed on the PC.

➢ **Experimental Results by FFT:**

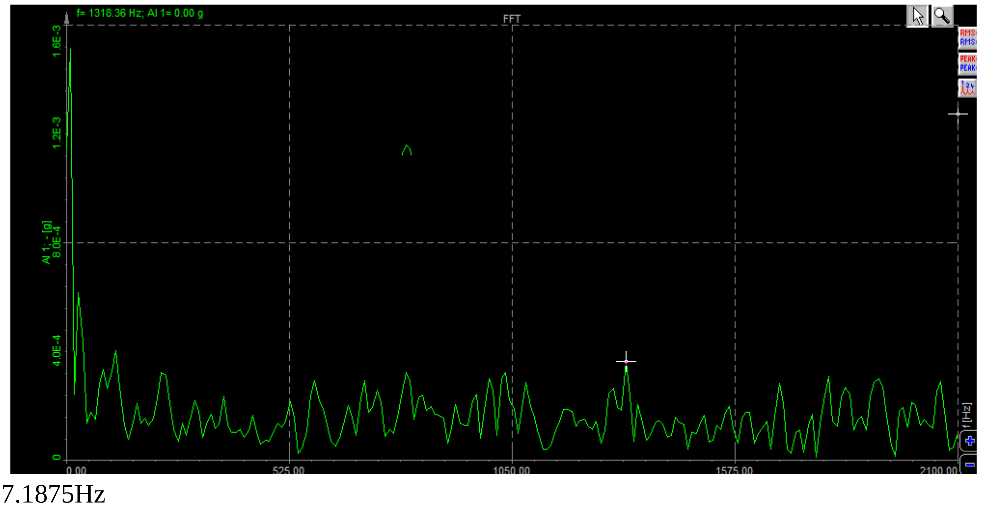
Material: Structural Steel

For Mode Shape1:



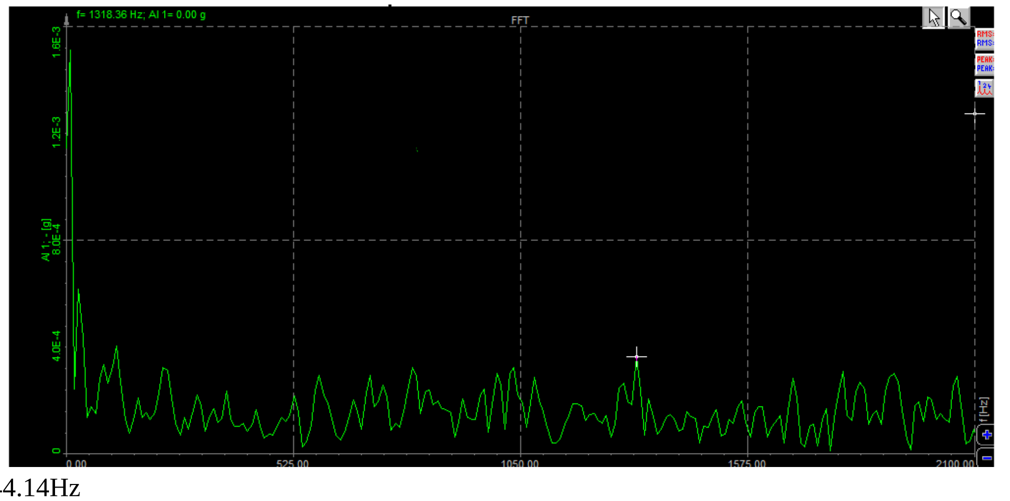
Frequency = 19.531Hz
Acceleration=0.001007g =0.0098m/s^2

For Mode shape2:



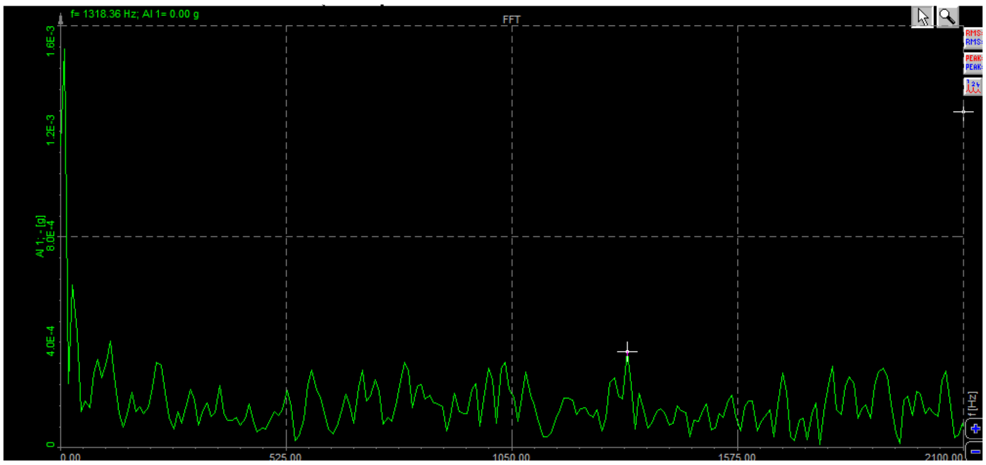
Frequency= 117.1875Hz
Acceleration=0.000404g=0.00396m/s^2

For Mode shape3:



Frequency=244.14Hz
Acceleration=0.000325g= 0.00318m/s^2

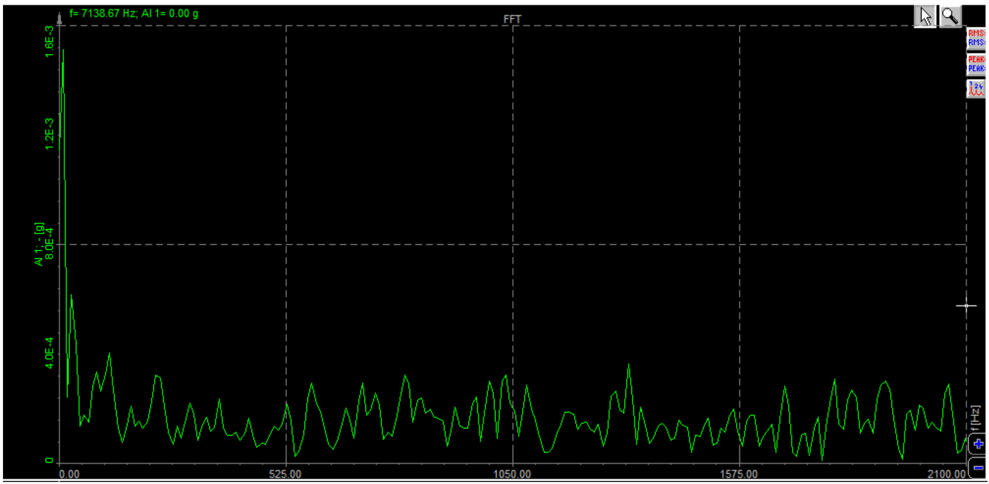
For mode shape4:



Frequency=585.937Hz

Acceleration=0.000292g= 0.00286m/s^2

For Mode shape5:



Frequency=859.78

Acceleration=0.000496g=0.00486m/s^2

- **Results by Analytical, Numerical and Experimental:**

Table 7: Comparison of stress and deformation

Parameter	Analytical	FEA	Experimental
Stress (MPa)	34.35	41.59	35.75
Deformation (mm)	0.428	0.4733	0.435

Table 8: Comparison of frequency

Mode	Analytical Frequency (Hz)	Experimental Frequency (Hz)	% of Error
1	14.86	19.531	19%
2	101.39	117.187	12%
3	253.79	244.14	4%
4	542.36	585.937	7%
5	822.06	859.375	4%

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- **Graphical Results:**

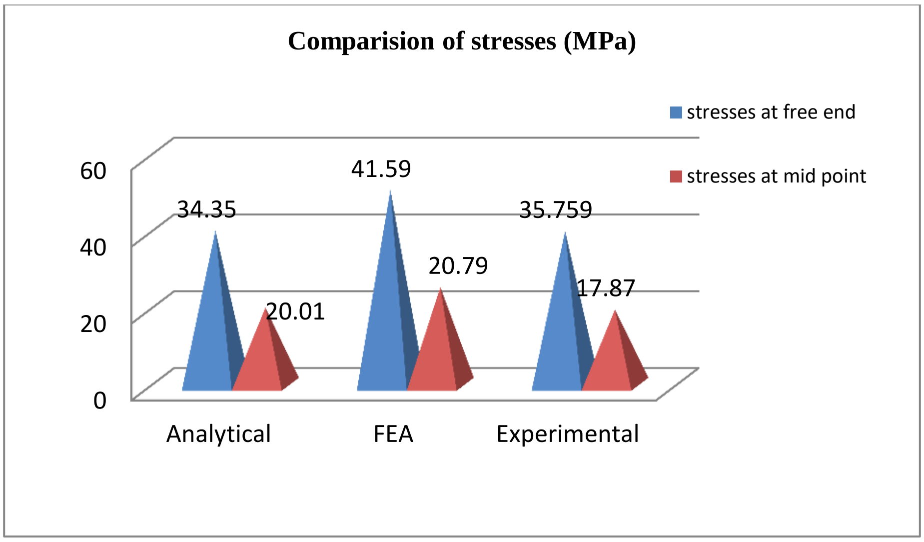

Fig 10: Stress Analysis

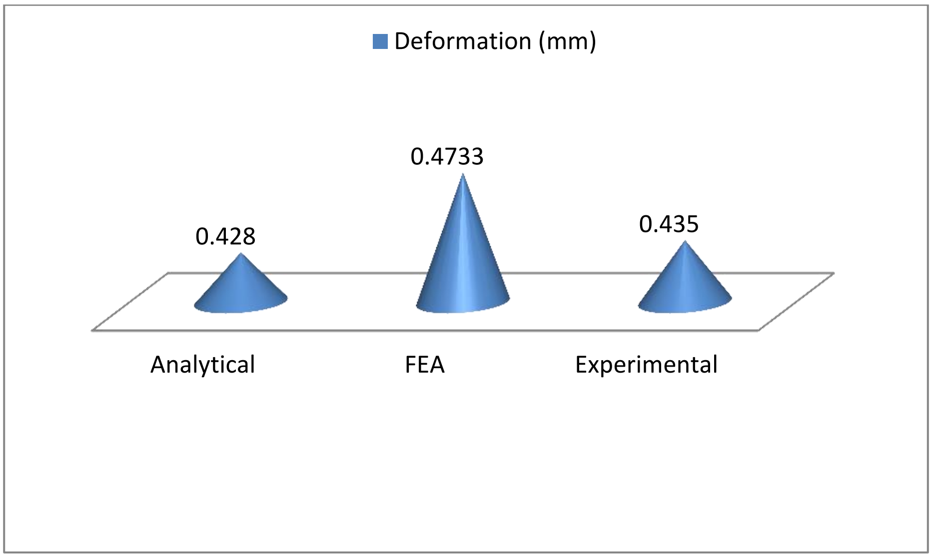

Fig 11: Deformation Analysis

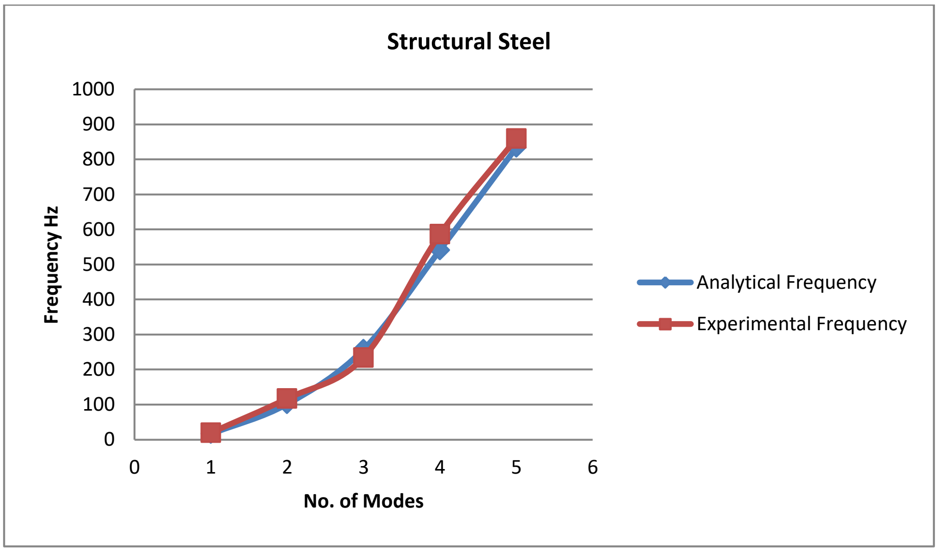

Fig 12: Natural Frequency Analysis

VI. CONCLUSION

The stress, deformation and natural frequency are determined by using Analytical, Numerical and Experimental approach..

1) Comparison of analytical and Numerical (ANSYS) values equivalent stress when load lifting 500kg, at free end point is 34.35MPa and 41.59MPa for length of boom is 900mm and the error between these values are 18%
2) Analytical Deformation value and Software deformation value, when load applied at free end of beam for length of beam 900mm is 0.428mm and 0.473mm and error is that 9%.
3) The difference between Analytical and Experimental stress varies from 4%.
4) The difference between Analytical and Experimental Natural frequencies varies from 4%-18%.
5) If number of mode increases, then frequency as well as acceleration of beam also increases.
6) As the Distance from Fixed end increases, frequency also increases.
7) If stress increases, reduced life of span.
8) By observing the above results, for lifting 500kg load we are taken ISMB200 beam instead of ISMB250 for FMJC.
9) Cost reduced by 30% as compare to existing beam.

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