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MODAL ANALYSIS OF ALUMINIUM & NATURAL RUBBER SANDWICHED BEAM

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Abstract— The composite materials consist of two or more different materials that are usually firmly bonded together at the interface. Many natural and Synthetic materials are of this nature, such as: reinforced rubber, filled polymers, GFRP (Glass Fiber Reinforcement Plastic), Nylon, aligned and chopped fiber composites, polycrystalline aggregates (metals), etc. In the present study the beam made of Rubber and carbon fiber sandwiched between aluminum plates are considered. The influence of the materials on the characteristics like natural frequencies, modes of vibration of structures has been investigated. The work presents the Computational modal analysis of a composite beam. In this work, the natural frequencies are measured using FFT analyzer.

Index Terms— composite materials, natural frequencies, modes, modal analysis, FEM

I. INTRODUCTION

VIBRATIONS are mechanical oscillations of physical objects such as a plate, particle, or body that is displaced from a position of equilibrium. The vibrations occur as the physical objects trade kinetic energy for potential energy. In structures, potential energy is elastic potential energy, or in other words, strain energy.

With no energy losses, a body that is given an initial amount of energy will retain that energy indefinitely. However, it can store the retained energy in multiple forms, such as strain energy and kinetic energy. The body will trade between energy types if the exchange satisfies the principle of least action, which says that the path an object takes between two fixed states will minimize the total energy of the system for a set of given equilibrium equations. That is to say that a structure will trade strain energy and kinetic energy only if it can do so in a way that does not increase the total energy of the system. This solution for the principle of least action is satisfied when the characteristic equation is solved. The characteristic equation is solved by the system's Eigen-values, which directly relate to physical properties such as frequency. The solutions to the system's characteristic equation are referred to as the system's modes, and they correlate directly to a particular frequency and shape function. Those particular frequencies are often called the modal frequencies or natural frequencies. Likewise, the shape function at the solution is often called the mode shape. Since these solutions do not require a forcing function, they are considered free or natural vibration.

Every object has a different response to forcing functions based on material properties, geometry, and boundary conditions. It is convenient to describe the vibrational response by a few main parameters: amplitude, mode shape, and frequency.

II. DAMPING

Damping is the resistance offered by a body to the motion of a vibratory system. The resistance may be applied by a liquid or solid internally or externally. The main advantage of providing damping in mechanical systems is just to control the amplitude of vibration so that the failure occurring because of resonance may be avoided.

Vibration damping plays an important role in machines and structures by improving performance and stability, reducing noise and increasing lifetime.

In mechanics, Damping may be realized using a dashpot. This device uses the viscous drag of a fluid, such as oil, to provide a resistance that is related linearly to velocity. The damping force F_C is expressed as follows:

$$F_C = -C\frac{dx}{dt}$$

Where C is the viscous damping coefficient, given in units of Newton seconds per meter (N s/m).

Generally, damped harmonic oscillators satisfy the second-order differential equation:

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$$\frac{d^2x}{dt^2}+2\zeta\omega_n\frac{dx}{dt}+\omega_n{}^2x=0$$

III. VIBRATION SUPPRESSION TECHNIQUES

In general, vibration suppression aims to reduce the system gain at and around one or more of the modal frequencies. This is done by dissipating energy at the desired frequency.

Vibration suppression can be achieved using two methods: passive damping and active damping. The passive damping method works by adding a viscoelastic material that is designed to dissipate energy in order to reduce vibration response. Passive vibration suppression is achieved using passive constrained layer damping treatments (PCLD). In this method, viscoelastic material is added between lamina in a composite layup in order to augment energy dissipation.
The active damping method works by adding piezoelectric actuators that can produce out of phase deformation in the piezoelectric actuator to affect the deflection of a structure. Active vibration control is achieved by using out of phase deflections for piezoelectric actuators. Control is typically achieved using feedback from a sensor to the piezoelectric actuator. Typical sensors are strain gages or a separate piece of piezoceramic used in the sensor configuration.

Piezoelectric materials are those that exhibit a relationship between their electrical state and mechanical strain. The unique properties of piezoelectric materials require different parameters to describe them. One such parameter is the electromechanical coupling factor, which describes the relation of stored mechanical energy to total stored energy with a piezoelectric material. Other parameters include the piezoelectric constants, which relate strain per electric field at constant stress, electric field per stress at constant stress, electric field at constant strain, stress at constant strain, and electric field per strain at constant stress. The most commonly used piezoelectric materials are piezoelectric ceramics. A very strong piezoelectric effect is present in lead ziconatetritanate, or PZT, which is a ceramic that is composed of lead, oxygen, and either titanium or zirconium. PZT comes in several different compositions, including PZT-2, PZT-4, PZT-5A, PZT-5H, and PZT-8.

IV. PROBLEM STATEMENT

Vibrations in structures have become increasingly problematic in low margin design, where structures are built to have a minimum amount of material or weight. Frequently, solutions have poor modal characteristics due to other constraints on the system, such as geometry or weight.

V. OBJECTIVE

To obtain the natural frequencies of beams of different sandwiched materials.

Table 1. CAD Model

Sr. No.	Specimen Name	Dimensions of Specimen		
		Length	**Width**	**Thickness**
1	A30NR	450	50	3

Note:
1. All dimensions are in mm
2. “A” denotes Aluminum
3."NR" denotes Natural Rubber

VI. FINITE ELEMENT METHOD

Finite element method has become a very powerful tool for a wide range of engineering problems. Applications range from deformation and stress analysis of automotive, aircraft, building and bridge structures.

In this method of analysis, a complex region defining a continuum is discredited into simple geometric shapes called finite elements. The element material property and the governing relationships are considered over these elements and expressed in terms of unknown values at nodes. An assembly process, duly considering the loading and constraints, results in the set of equations. Solution of this equation gives an approximate behavior of the continuum.

VII. FINITE ELEMENTS APPROACHES

There are two different Finite Element approaches to analyze structures, namely-
A. Force method.
B. Displacement.

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A. Force method:

The member forces are the basic unknowns in the system of equations.

B. Displacement method:

The nodal displacements are the basic unknowns in the system of equations. The analysis has been done using the concept of the Finite Element Method (FFM). The fundamental concept of finite element method is that, any continuous quantity such as temperature, pressure and displacement can be approximate by a discrete model. There are many problems where analytical solutions are difficult or impossible to obtain. In such cases FEM provides an approximate and a relatively easy solution. Finite element method becomes more powerful when combined with the rapid processing capabilities of computers.

The basic idea of FEM is to discretize the entire structure into small elements. Each element is defined by the nodes or grids and the nodes serve as a link between two elements. Then the continuous quantity is approximated over each element by a polynomial equation. This gives a system of equations, which is solved by using matrix techniques to get the values of the desired quantities.

VIII. MESHING OF SANDWICHED BEAM

The meshing of sandwich beam is done using solid elements. Hex elements are used to mesh the beam and the meshing quality and connectivity are maintained as per meshing standards.

Then the material properties are applied as shown in the below table.

Table 2. Material Properties

Sr. No.	Type of Material	Young's Modulus (GPa)	Shea Modulu s (GPa)	Density (Kg/m3)	Poisson' s Ratio
1	Aluminium	71	27.3	2700	0.33
2	Nat. Rubber	0.0015 4	0.005	950	0.45

The beam is now treated as cantilever beam by applying hinge at one end of the beam. The Modal analysis on the beam is carried out for getting natural frequencies of the beam.

A. Meshing on model

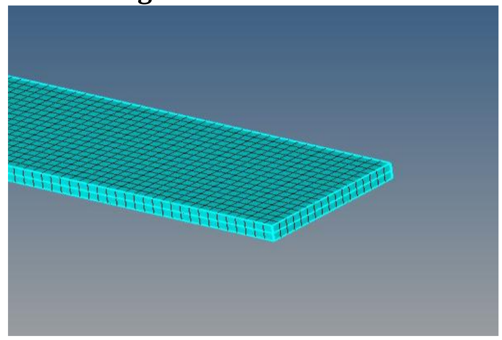

Fig. 1. Hexmesh on the sandwich beam

B. Application of boundary conditions

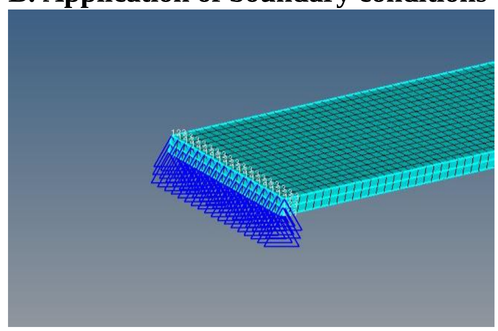

Fig. 2. Boundary conditions

The hinge support is given by constraining degrees of freedom as shown in the above figure

The Model is created by applying material and properties

C. Meshing and boundary condition application on model A30NR

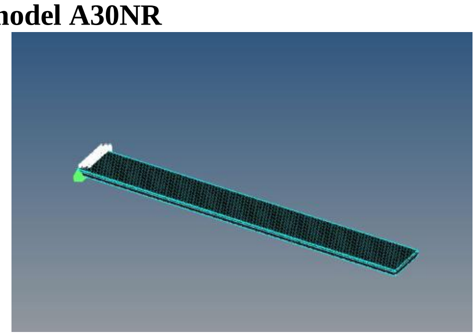

Fig. 3. Meshing and boundary Conditions

Table. 3. Mesh details

Type of element	Hex
Element size	1.5
No of elements	69300
No of nodes	71638

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IX. MODE SHAPES OBTAINED FROM ANALYSIS OF A30NR

A. Mode 1

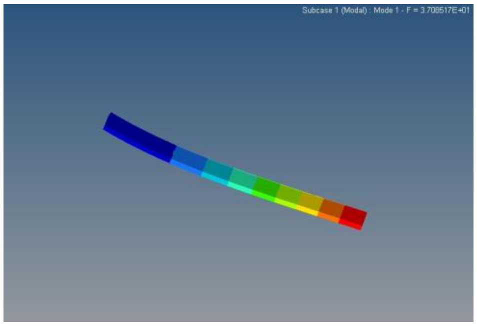

Fig. 4. 1st mode shape of the beam

Above figure shows 1st mode shape of the beam with frequency as 37.08Hz

B. Mode 2

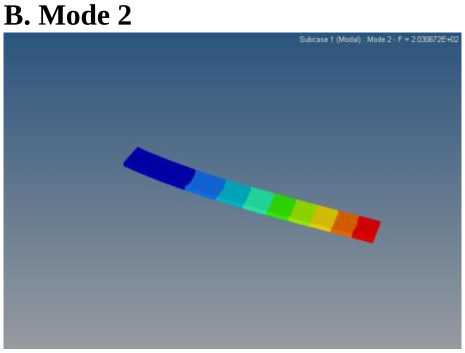

Fig.5. 2nd mode shape of the beam

Above figure shows 2ndmode shape of the beam with frequency as 203.06Hz

C. Mode 3

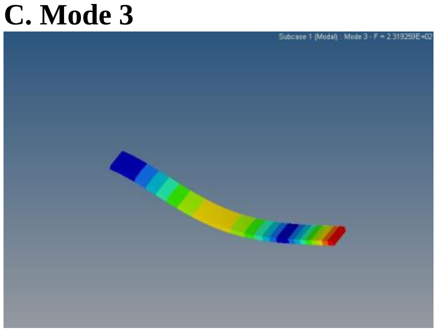

Fig. 6. 3rd mode shape of the beam

Above figure shows 3rd mode shape of the beam with frequency as 231.9Hz

D. Mode 4

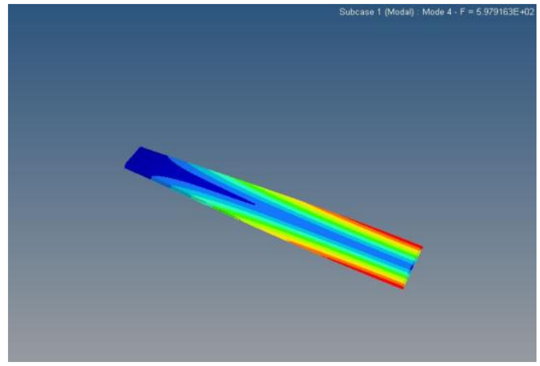

Fig. 7. 4th mode shape of the beam

Above figure shows 4th mode shape of the beam with frequency as 597.9Hz

E. Mode 5

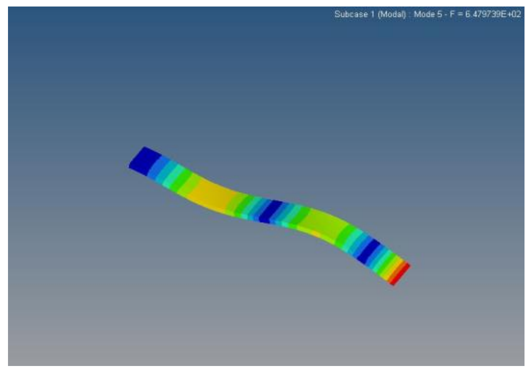

Fig. 8. 5th mode shape of the beam

Above figure shows 5th mode shape of the beam with frequency as 647.9Hz

F. Mode 6

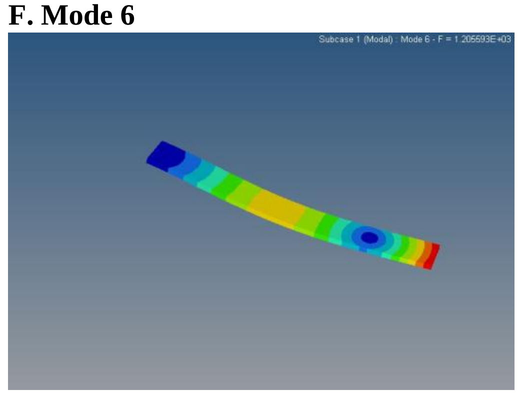

Fig. 9. 6th mode shape of the beam

Above figure shows 6th mode shape of the beam with frequency as 1205.5Hz

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X. TESTING

On a structure dynamic loading can vary from recurring cyclic loading of the same repeated magnitude, such as a unbalanced motor which is turning at a specified number of revolutions per minute on a structure (for example), to the other extreme of a short time, intense, nonrecurring load, termed shock or impact loading, such as a bird striking an aircraft component during flight. A continuous infinity of dynamic loads exists between these extremes of harmonic oscillation and impact. Associated mode shapes. Mathematically, there are infinity of natural frequencies and mode shapes in a continuous structure.

Dynamic loading can vary from intense, nonrecurring load known as shock load such as bird striking aero plane to recurring cyclic loading of magnitude which repeats itself such as unbalanced motors rotating at particular R.P.M. Any structures amplitude may rapidly grows with time if its frequency of oscillation matches its natural frequency.

Structure can be overstressed which leads to its failure or due to large oscillations amplitude may be limited at large value which further leads to fatigue damages.

Time dependent loading should be compared with natural frequency to ensure structural integrity of any structure. These two frequencies should be considerably different. While designing structure over deflecting and overstressing should be taken care of and resonances should be avoided.

A. Modal Testing

In modal testing, FRF measurements are usually made under controlled conditions, where the test structure is artificially excited by using either an impact hammer, or one or more shakers driven by broadband signals. A multi-channel FFT analyzer is then used to make FRF measurements between input and output DOF pairs on the test structure.

B. Exciting modes with impact testing

With the ability to compute FRF measurements in an FFT analyzer, impact testing was developed during the late1970's, and has become the most popular modal testing method used today. Impact testing is a fast, convenient, and low cost way of finding the modes of machines and structures.

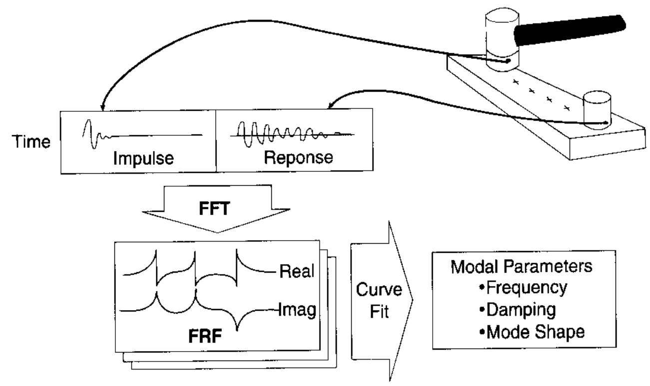

Fig. 10. Schematic testing set-up

C. Impact Testing.

Impact testing is depicted in Figure 9. The following equipment is required to perform an impact test,

1. An ***impact hammer*** with a load cell attached to its head to measure the input force.
2. An ***accelerometer*** to measure the response acceleration at a fixed point & direction.
3. A 2 or 4 channel ***FFT analyzer*** to compute FRFs.
4. ***Post-processing modal software*** for identifying modal parameters and displaying the mode shapes in animation.

A wide variety of structures and machines can be impact tested. Of course, different sized hammers are required to provide the appropriate impact force, depending on the size of the structure; small hammers for small structures, large hammers for large structures.

D. Modal Frequency as Peak Frequency

The ***frequency of a resonance peak*** in the FRF is used as the modal frequency. This peak frequency, which is also dependent on the frequency resolution of the measurements, is not exactly equal to the modal frequency but is a close approximation, especially for lightly damped structures. The resonance peak should appear ***at the same frequency in almostevery FRF measurement***. It won't appear in those measurements corresponding to nodal lines (zero magnitude) of the mode shape.

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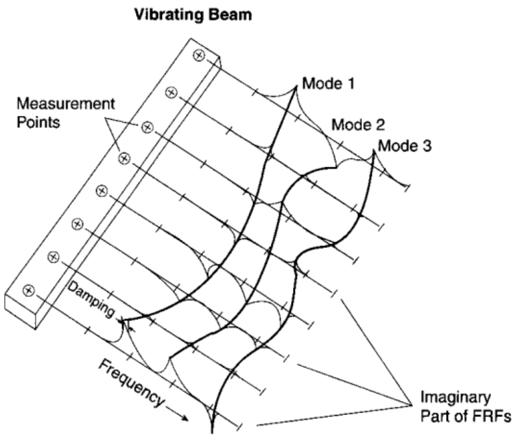

Fig. 11. Curve Fitting FRF Measurements.

The testing of the sandwich beam is carried out by clamping at one end as cantilever beam. The hammering test is carried out as stated above. The figure shows the testing set up.

Fig. 12. Specimen held as cantilever beam

The FFT analyser is connected to a sensor which reads the response of the system. The hammer test is initiated by slightly hammering the free end of the cantilever beam. Then the response of the system is recorded by the FFT analyser at three points which are marked on the beams.

Fig. 13. Hammer test

Fig. 14. FFT analyzer

The response of the system is stored in the FFT analyzer which are further connected to data acquisition system to generate respective graphs.

XI. RESULTS AND DISCUSSION

The results are obtained by various readings one after another at same point on the specimen. The different values of the FFT analyzer readings are listed in the table. The nearest value to the FEM results received is considered for validation.

The natural frequencies obtained from the experimental analysis are as follows

For A30NR

A. Reading at point 1

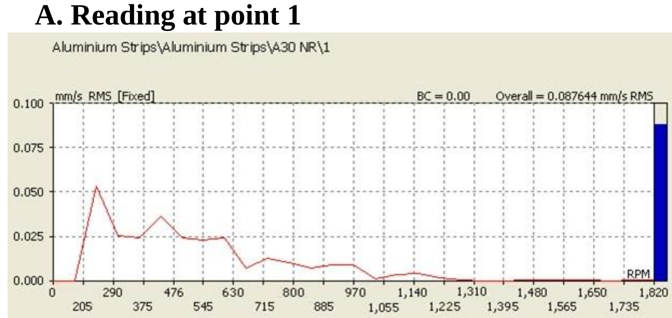

Fig. 15. Reading at point 1

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B. Reading at point 2

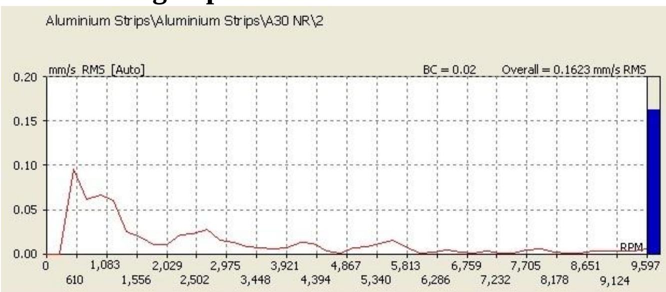

Fig. 16. Reading at point 2

C. Reading at point 3

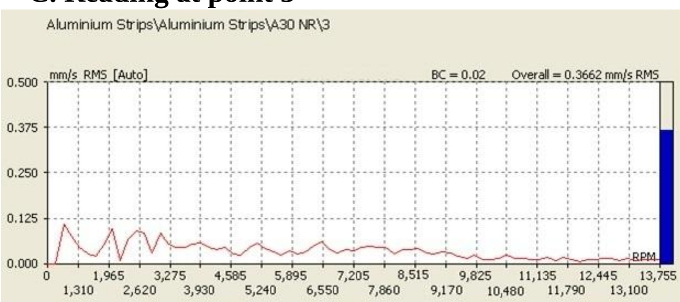

Fig. 17. Reading at point 3

D. Result Comparison :

Table. 4. Comparison between FEM and Experimental Results

Modes	FEM results	Experimental results	% deviation
Mode 1	231	250	8.22%
Mode 2	597	600	0.5%
Mode 3	647	655	1.23%

XII. CONCLUSION

CAD model of Composite Beam is modeled and FEA analysis is carried out for the vibration response. The Composite beam model is analyzed for design safety. Natural frequency of Composite beam is extracted and the result were interpreted and compared study is made. Fabrication of the Composite beam is done and is tested using FFT analyzer. Results of FEA and experimentation are compared and validated, which shows acceptable deviations.

Hence the objective is achieved.

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