ANLYTICAL AND EXPERIMENTAL ANALYSIS OF MULTILAYERED SANDWICH STRUCTURES WITH VISCOELASTIC CORE.

Mohit D. Pathak[1] Dr. Gopal E. Chaudhari[2s]

[1]M. Tech Student, [2]Professor

Department of Mechanical Engineering Department

[1,2]J. T. Mahajan College of Engineering, Faizpur, Maharashtra, India.

1. ABSTRACT

This work represents vibration analysis of a viscoelastic sandwich beam. Three layered viscoelastic sandwich beam combination of metal as well as non-metal is modeled using CATIA software. Different specimens have been modeled by varying the core layers (Natural Rubber and Neoprene Rubber) and face layers. The Natural frequencies are obtained for various models using different face layers and boundary conditions. Experimentation was also conducted in order to verify the results obtained from finite element analysis. The results clearly show that there is decrease in natural frequency of beams modeled with Neoprene as a core layer compared to the rubber for the fixed-free boundary conditions.

Index Terms:- Multilayered Sandwich Structures, Vibration Analysis, Viscoelastic Core, Neoprene Rubber, Natural Rubber.

2. INTRODUCTION

The airborne and structure borne noise and vibration occurs frequently in systems. The traditional passive control methods that include use of absorbers, barriers, mufflers, silencers, etc are for airborne noise. For systems with constant excitation frequency, modification of system's stiffness or mass reduces the unwanted vibrations as these parameters alter the resonant frequencies. A Viscoelastic material exhibits both viscous fluid and elastic solid material characteristics. There are mainly two methods of treatment of viscoelastic material viz., constrained layer and unconstrained layer or free layer treatment. In a sandwich structure generally the bending loads are carried by the force couple formed by the face sheets and the shear loads are carried by the lightweight core material. Depending on the functional requirements, sandwich structure utilizes the constrained layer treatment method to obtain the best properties out from all layers.

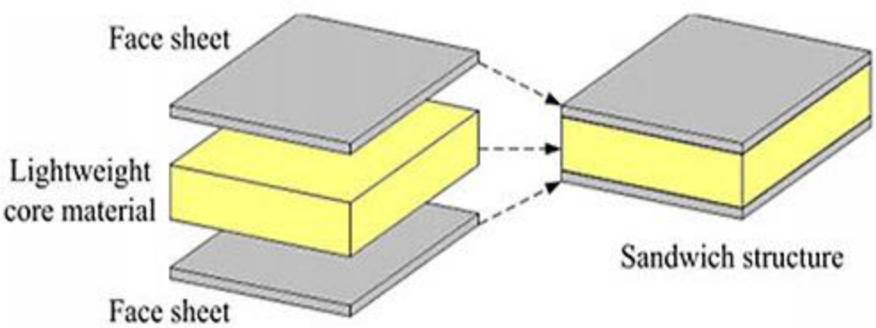


Fig.2.1 Main components of sandwich structures

The material choice in sandwich structures depends upon the need of employment such as high strength, high temperature resistivity, surface finish etc. In recent times the number of available cores has increased enormously due to the introduction of more competitive cellular plastics. Combining options of face sheet materials with different core materials give the new ideas to be integrated with a wide range of applications.

Top layer: Fiber Reinforced Polymer FRP (Nonmetallic), Metal like Aluminium, Mild Steel
Core layer: Viscoelastic Material (Rubber, Neoprene)
Bottom layer: Fiber Reinforced Polymer FRP, Metallic (Aluminium/ Mild Steel)

3. PROBLEM STATEMENT

Sandwich beams which are the answer to many structural problems demanding self-control and flexible characteristics involving mechanical and thermal stresses. The technological implications of his class of beams are immense, as they are especially useful in remote operations, expensive space operations subjected to extreme thermo-mechanical loadings, aerospace skins, protective shields, components in reactor vessels, machine tools, and medical applications, to name only a few. The beams have characteristics such as thermo-electro-mechanical coupling, functionality, intelligence, and gradation at micro and Nano scales. The reliability and integrity of these systems are the main challenges before us. They can be customized to operate under varying conditions covering the whole spectrum of electro-thermo-mechanical conditions. The conditions can vary across a wide range of temperature, magnetic & electric fields, pressure and mechanical load, and or a combination of two or many. Experimental investigations of both these systems & beams although possible, are prohibitively expensive, and therefore must be complemented with simulations and theoretical analyses.

4. OBJECTIVE

The objective of this study is to understand the mechanical behavior and failure mechanisms of sandwich structures with viscoelastic material and fiber reinforced polymer (FRP) face sheets.

1) To Formulate of governing differential equation of motion of Sandwich Euler-Bernoulli beam.
2) Free vibration analysis of Sandwich beam with different boundary conditions.
3) The effects viscoelastic material on the fundamental frequency and mode shapes
4) To validate the numerical results with experimental work for real life application.

5. MATERIAL PROPERTIES

Table 5.1 Material properties of sandwich beam for face and core layers

Type of material	Young's Modulus E (GPa)	Shear Modulus G (GPa)	Density in Kg/m^3	Poisson's Ratio ν
Aluminium	70	27.3	2766	0.33
Mild Steel	200	76.9	7850	0.30
FRP	2	0.5	1700	0.30
Rubber	0.00154	0.005	950	0.45
Neoprene	0.0008154	0.000273	960	0.49

6. ANALYTICAL APPROACH

6.1 Simple Beam Theory

By using Euler's Bernoulli beam theory,

$$\frac{\partial^2 w}{\partial t^2}+\frac{EI}{\rho}\ \frac{\partial^4 w}{\partial x^4}=0 \qquad (1)$$

To find the response of the system one may use the variable separation method by using the following equation.

$$w(x,t)=\varphi(x)q(t) \qquad (2)$$

$\varphi(x)$ is known as the mode shape of the system and $q(t)$ is known as the time modulation. Now equation (1) reduces to

$$\varphi(x)\frac{\partial^2 q(t)}{\partial t^2}+\frac{EI}{\rho}\frac{\partial^4\varphi(x)}{\partial x^4}q(t) \qquad (3)$$

or

$$-\frac{EI}{\rho}\left(\frac{1}{\varphi(x)}\right)\left(\frac{\partial^4\varphi(x)}{\partial x^4}\right)=\frac{1}{q}\left(\frac{\partial^2 q}{\partial t^2}\right) \qquad (4)$$

Since the left side of equation (4) is independent of time t and the right side is independent of x the equality holds for all values of t and x. Hence each side must be a constant. As the right side term equals to a constant implies that the acceleration $\left(\frac{\partial^2 q}{\partial t^2}\right)$ is Proportional to displacement $q(t)$.

$$-\frac{EI}{\rho}\left(\frac{1}{\varphi(x)}\right)\left(\frac{\partial^4\varphi(x)}{\partial x^4}\right)=\frac{1}{q}\left(\frac{\partial^2 q}{\partial t^2}\right)=-\omega^2 \qquad (5)$$

Hence,

$$\frac{d^2q}{dt^2}+\omega^2 q=0$$

$$\frac{\partial^4\varphi(x)}{\partial x^4}-\frac{\rho\omega^2}{EI}\varphi(x)=0 \qquad (7)$$

Taking, $\beta^4=\frac{\rho\omega^2}{EI}$

The above equation can be written as

$$\frac{\partial^4\varphi(x)}{\partial x^4}-\beta^4\ \varphi(x)=0 \qquad (8)$$

The solution of equation (6) and (8) can be given by

$$q(t)=C_1\sin\omega t+C_2\cos\omega t \qquad (9)$$

$$\varphi(x) = A\sinh(\beta x) + B\cosh(\beta x) + C\sin(\beta x) + D\cos(\beta x) \quad (10)$$

Hence,

$$w(x,t) = (A\,sinh\,\beta x + Bcosh\,\beta x + Csin\,\beta x\, + Dcos\,\beta x)(C_1\,sin\,\omega t + \,C_2\,cos\omega t) \quad (11)$$

In case of cantilever beam the boundary conditions are :
At left end i.e.,

$$at\ x = 0 \quad w(x,t) = 0 \quad (displacement = 0)$$

$$\frac{\partial w\,(x,t)}{\partial x} = 0 \quad (\,Slope = 0)$$

At the free end i.e.,

$$at\ x = L \quad \frac{\partial^2 w\,(x,t)}{\partial x^2} = 0 \quad (Bending\ Moment = 0)$$

$$\frac{\partial^3 w\,(x,t)}{\partial x^3} = 0 \quad (\,Shear\ Force = 0)$$

$$\text{At } x = 0 \quad \varphi(x) = 0 \text{ and } \frac{\partial\varphi(x)}{\partial x} = 0 \quad (12)$$

$$\text{At } x = L \quad \frac{\partial^2\varphi(x)}{\partial x^2} = 0 \text{ and } \frac{\partial^3\varphi(x,t)}{\partial x^3} = 0 \quad (13)$$

Substituting these boundary conditions in the general solution,

$$\phi(x) = A\cosh\beta x + B\sinh\beta x + C\cos\beta x + D\sin\beta x \quad (14)$$

From equation (12)

$$A = -C\ and\ B = -D \quad (15)$$

$$\phi(x) = A(\cosh\beta x - \cos\beta x) + B(\sinh\beta x - \sin\beta x) \quad (16)$$

From (13 and 16) one may have

$$\frac{A}{B} = -\frac{\sinh\beta l + \sin\beta l}{\cosh\beta l + \cos\beta l} = -\frac{\cosh\beta l + \cos\beta l}{\sinh\beta l - \sin\beta l} \quad (17)$$

or,

$$cos\beta l\ cosh\beta l = -1 \quad (18)$$

and the root of the equation is, $\beta l = \frac{(2n-1)\pi}{2}$

Hence one may solve the frequency equation $\cos\beta l\ \cosh\beta l = -1$ to obtain frequencies of different modes. For the first two modes the values of βl are calculated as 1.875, 4.694, 7.85, 10.99 and 14.13.

Table 6.1 Value of βl

Mode	For Fixed Free Condition
I	1.875
II	4.694
III	7.855
IV	10.99

7. EXPERIMENAL ANALYSIS

The complete experimental setup for vibration analysis is shown in figure.5.1. In order to obtain natural frequencies of plates. The values of natural frequencies obtained by exciting the handle bar using Impact Hammer shown in figure 5.1 and measuring the response by an Accelerometer which was connected to FFT Analyzer.

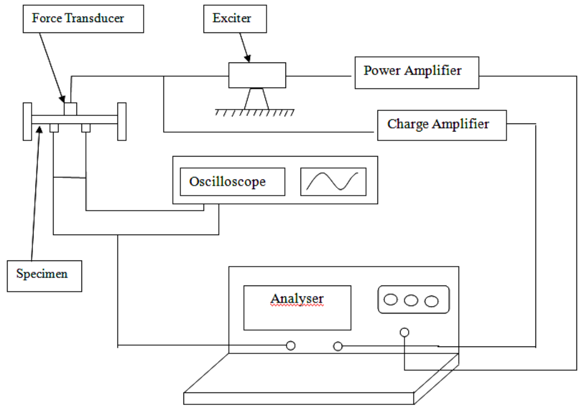


Figure 7.1 Schematic Diagram of Experimental Setup

It is a simple beam with a certain density and strength and dimension (450x31x3) mm^3. Total nine specimens were created in configuration given below.

Specimen 1: Aluminium – Aluminium- Aluminium
Specimen 2: Aluminium- Natural Rubber-Aluminium
Specimen 3: Aluminium- Neoprene Rubber-Aluminium
Specimen 4: Mild Steel – Mild Steel- Mild Steel
Specimen 5: Mild Steel- Natural Rubber-Mild Steel
Specimen 6: Mild Steel- Neoprene Rubber-Mild Steel
Specimen 7: FRP – FRP- FRP
Specimen 8: FRP- Natural Rubber-FRP
Specimen 9: FRP- Neoprene Rubber-FRP

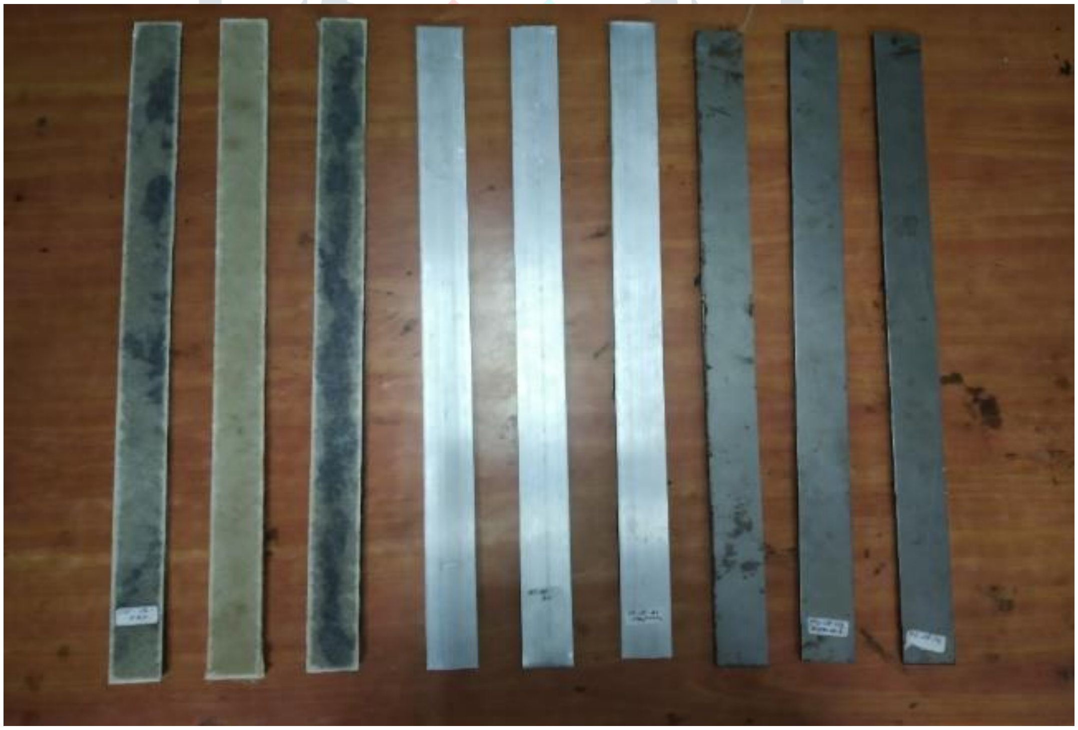


Figure 7.2 Specimens of FRP, AL, MS with Viscoelastic Layer

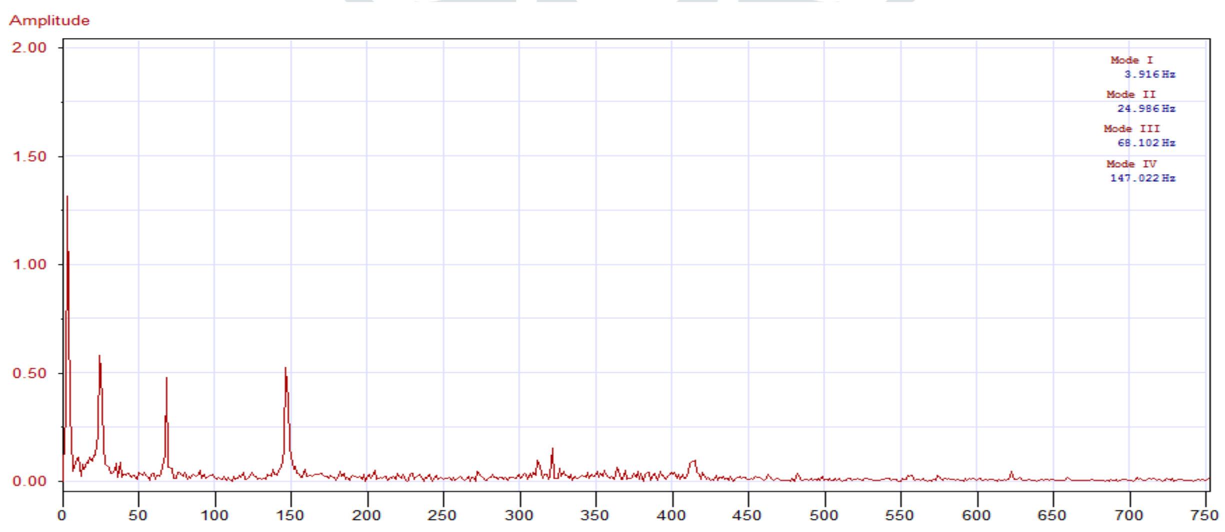
8. RESULT AND OUTCOME

Table no. 8.1 Natural Frequencies of First Four Mode of Sandwich Beam with Fixed Free Condition (Cantilever Condition) using Experimental Approach

Sandwich Pattern	Natural Frequency Hz			
	Mode I	Mode II	Mode III	Mode IV
Al-Al-Al	13.149	74.526	223.067	429.209
Al-Ru-Al	10.004	38.440	92.857	162.161
Al-Ne-Al	**3.344**	**29.486**	**70.711**	**132.473**
MS-MS-MS	12.446	74.398	223.236	419.408
MS-Ru-MS	9.991	29.415	79.405	153.433
MS-Ne-MS	**3.916**	**24.986**	**68.102**	**147.022**
FRP-FRP-FRP	11.591	58.561	184.889	356.112
FRP-Ru-FRP	8.990	34.592	86.263	140.417
FRP-Ne-FRP	**5.174**	**24.262**	**64.701**	**97.391**

Graph 8.1 Cantilever Al-Ne-Al

Graph 8.2 Cantilever St-Ne-St



Graph 8.3 Cantilever FRP-Ne-FRP

The natural frequency for Al-Al-Al specimen is very high. If we replace core material, then the natural frequency is going to be decrease.

For FRP-Ne-FRP specimen, natural frequency is well lower as compare to other cases.

9. CONCLUSION

From this work,The results obtained from the modal analysis clearly show that natural frequency for the same mode. From the results one can infer that damping characteristics for neoprene viscoelastic material has significant effect when compared with the rubber viscoelastic material Results show that the viscoelastic constrained layer damping treatment has a great significance in controlling the vibration of structures like beams, plates, etc.

10. REFERENCES AND BOOKS

1. Mergen H. Ghayesh, “Asymmetric viscoelastic nonlinear vibrations of imperfect AFG beams”, Applied Acoustics 154 (2019), PP.121–128
2. M. Bilassea, E.M. Daya, L. Azrar, “Linear and nonlinear vibrations analysis of viscoelastic sandwich beams”, Journal of Sound and Vibration 329 (2010), PP. 4950–4969
3. M. Latifi, M. Kharazi, H.R. Ovesy, “Effect of integral viscoelastic core on the nonlinear dynamic behavior of composite sandwich beams with rectangular cross sections”, International Journal of Mechanical Sciences 123 (2017), PP. 141–150
4. Jon Garcia Barruetabena, Fernando Cortes, “Finite elements analysis of the vibrational response of an adhesively bonded beam”, Engineering Structures 171 (2018), PP. 94–104
5. N. Jacques, E.M. Daya, M. Potier-Ferry, “Nonlinear vibration of viscoelastic sandwich beams by the harmonic balance and finite element methods”, Journal of Sound and Vibration 329 (2010), PP. 4251–4265
6. M. Meunier, R.A. Shenoi, “Forced response of FRP sandwich panels with viscoelastic materials”, Journal of Sound and Vibration 263 (2003), PP.131–151
7. Dvir Elmalich, Oded Rabin Ovitch, “On the effect of inter-laminar contact on the dynamics of locally delaminated FRP strengthened walls”, International Journal of Non Linear Mechanics 77(2015), PP. 141–157
8. P. Bangarubabu, K. Kishore Kumar and Y. Krishna, “Damping Effect of Viscoelastic Materials on Sandwich Beams”, International Conference on Trends in Industrial and Mechanical Engineering (ICTIME'2012) March 24-25, 2012
9. M. R. Doddamani, S. M. Kulkarni, “Dynamic response of fly ash reinforced functionally graded rubber composite sandwiches - a Taguchi approach” International Journal of Engineering, Science and Technology Volume 3, No. 1, 2011, PP. 166-182
10. Dr.P.S.Senthil Kumar1, Karthik. K, “Vibration Damping Characteristics of Hybrid Polymer Matrix Composite”, International Journal of Mechanical & Mechatronics Engineering, Volume 15 No.01, Feb.2015

BOOKS

11. Dr. V.P.Singh, “*Mechanical Vibration*”, Dhanpat Rai & Co. Ltd.,Delhi, pp. 416-448.
12. John Case, Lord Chilver and Carl T.F. Ross, “*Strength of Materials and Structures*”, John Wiley & Sons, 4th Edition, pp. 295-338