Investigations of Partial Constrained Layer Damping treatment effect on Vibration Analysis

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Abstract— Constrained layer damping treatment is used to reduce structural vibration. Constrained-layer damping is a mechanical engineering technique for suppression of vibration. Typically a visco-elastic or other damping material is sandwiched between two sheets of stiff materials that lack sufficient damping by themselves. In the constrained layer damping technique the damping material is bonded to the structure similar to the free layer damping technique, then an another constraining layer having very high stiffness is constrained over the damping material.

Visco-elastic polymeric materials have been used to damp vibrations for many years. Energy is dissipated through relaxation processes in the long chain molecular networks when a polymeric material is subjected to vibrations. As there is a link between temperature and molecular motion the properties of visco-elastic materials vary with frequency and temperature.

In this project we will focus on partial constrained layer damping treatment. Experimental analysis is performed on full and partial constrained layer beam. It is observed from the analysis that the vibration response amplitude of Partial CLD beam is less as compare to full CLD beam.

Index Terms— [A] =Top Aluminum layer, [B] = Viscoelastic middle layer, [C] = Bottom Aluminums layer, [D] = Spacer, [ω] = frequency, [η] = Loss factor, [L1] = Length of spacer, [L2] = Length of Viscoelastic material, [L1+L2] = Total length.

1 Introduction

THE level of allowable vibrations in any structure is determined by many factors such like its effect to the strength of the structure and its separate elements, to the structural operability, structural health, human safety and excitation of mounted equipments. In the assessment of the strength characteristics of any structure, one of the most dangerous regimes in structural dynamic deformation is resonance, which is realized when the frequencies of the structure's natural oscillations coincide with a frequency of the external harmonic effects. In such loading regime, the amplitude levels of stresses and deformations in structure increase dramatically. Their correct and reliable theoretical assessment with the needed accuracy for practical purposes requires proper account of damping properties of structural materials caused by internal friction.

In the second half of the last century, a scientific field was born in mechanics related to the study of steady and unsteady interaction of acoustic waves with solid deformable bodies and thin-walled structural elements. Studies in this field continue attracting the attention of researchers with actuality, complexity and diversity of the phenomena inherent in the interactions among different physical nature. Related to this research direction, up to now, aero-hydro elasticity problems of thin-walled structures (in particular shells) covered in a number of monographs and literature reviews. However, these papers did not consider the questions of sound waves formation and theoretical study of sound insulation and sound absorption by various deformable bodies. These issues are not investigated by researchers up to now, although all handbooks or manuals, devoted to this issue and covering various multilayer structures, point out their good sound insulation and sound absorption properties [1, 2 etc.]. In practice, these properties have been studied mainly experimentally and the theoretical researches on these properties are mostly based on simplified equations of mechanics of multilayer structures since there is no theoretical basis developed to deal with these problems. As an example, noise in ground vehicles (i.e. automobiles) and aircraft structures is reduced by sticking some special coatings made from rubber-like materials on load bearing elements of construction, which are called by designers as sound isolation layers. However, these coatings in practice do not have any sound isolation properties, but having high damping properties, they reduce significantly the amplitudes of deformation and deflections in structural elements during their loading in resonance regimes, which lead, as a result, the reduction of sound pressure in structure's interior. Moreover, further will be shown in present study that using special coating materials with high damping ratios leads dramatic reductions in the level of cyclic stresses formed in structural elements and as a consequence dramatic increase in durability of structure.

A Problem Statement

In order to further verify the analytical model as well as to investigate various phenomena associated with partial damping, modal tests are carried out on a cantilever beam made of aluminium. Response is measured via an accelerometer near the root, and the beam is excited by a impact hammer at the free end.

Over a period of time the damping capacity of full constrained layer damping system reduces due to less shearing visco-elastic material. There is mainly extension of visco-elastic material near the curvature of beam during different bending modes. To overcome this problem the damping performance of the visco-elastic material in constrained layer damping treatment can be increased by creating artificial cutting of upper constrained layer and visco-elastic material which known as partial constrained layer damping treatment.

The problem under study consists of experimental analysis of effect of partial constrained layer damping treatment for reduction of structural vibration.

B Main Objectives

In this project we will focus on partial constrained layer damping treatment and full constrained layer damping treatment.

The objectives of the project are follows:

1. Study of Full Constrained Layer Damping Treatment.
2. Study of Partial Constrained Layer Damping Treatment.
3. Design and fabrication of CLD beam.
4. Experimental Vibration Analysis of Full Constrained Layer Damping Treatment.
5. Experimental Vibration Analysis of Partial Constrained Layer Damping Treatment.
6. Comparison of partial CLD treatment with Full CLD treatment.

2 LITERATURE REVIEW

Several scientific issues associated with damped beams have been analyzed by many investigators. This is evident from the studies described in two books on vibration damping by Nashif et al. [3] and Sun and Lu [4]. Much of the prior work has been limited to uniform free or constrained layer damping treatments.Among the earliest investigators, J.L.Crassidis derived a set of equations for free layer damping treatments. These equations are still being used for the standard ASTM test (also known as the J.L.Crassidis [9] beam test) to determine damping material properties provided an analytical solution to the problem of constrained layer treatments with limitation to beams of sinusoidal mode shapes. Rao [8] calculated natural frequencies and modal loss factors fora fully covered sandwich beam by solving a sixth order differential equation. Jean Marie [12]et al. incorporated both flexural and longitudinal shape functions in the eigen value problem using their Rayleigh-Ritz approach for a partially covered sandwich beam. More recently, we employed a similar approach to further extend the complex eigen solution theory for beams with multiple damping patches.

When the CL in PCLD techniques is replaced by or enhanced with actuators it is called ACLD. There has been much research in the field of ACLD in the past years. ACLD is described as a smart, fail safe and efficient vibration control method over a large frequency band, see for instance Li and Wang [1], Hau and Fung [5] and Liu and Wang [1]. But the performance is strongly dependent on the design variables (for instance the thicknesses, length, position and configuration). Liao and Wang [6] identified VEM parameter regions that will guarantee that the ACLD treatment can outperform both PCLD and plates where the piezo patch is directly mounted on the base plate. The best results can be achieved when the treatment is placed in an antinode, because the vibration of the plate is largest here. In a node the response of the plate cannot be sensed. There are two types of ACLD, direct and indirect (ACLDd and ACLDi respectively). With ACLDd the actuating piezo patch is bonded to the base plate by the VEM. With ACLDi the actuating piezo patch is bonded to the constraining layer, and the constraining layer is bonded to the base plate by a VEM layer. With both types of ACLD the shear strain in the VEM is increased and thus more energy can be dissipated.

In the article of Liu and Wang[1] they claim to reach a loss factors of maximal 1¡1:5 with ACLD (depending on the loss factor of the VEM layer and with very soft edge elements). In this article the ACLD model is enhanced with edge elements which connect the piezo constraining layer directly to the base layer in order to enhance the control authority of the piezo. The base plate is an aluminum beam 250mm long, 30 mm wide and 3 mm thick. The loss factor achieved with fairly stiff edge elements is much higher than the loss factor achieved without the edge elements or with soft edge elements. The loss factor achieved with the enhanced ACLD model can be even 2:5¡2:8 according to the article. One of the advantages of edge elements is that the optimal length of the treatment is half of the optimal length without edge elements, which can be an advantage when space constraints are an issue. One of the main advantages of ACLD with regard to systems where the piezo patch is directly mounted to the plate, Lio and Wang [1].

A. Different techniques of vibration control

1) Absorption
2) Structural Damping
3) Vibration Isolation
4) Use of barriers and enclosures

B. Types of damping

1) Free layer damping:

Damping material is applied to a surface via spray, roller or brush of which damping is to be achieved. This method is very useful for relatively thin structures. In these cases the applied damping material is often thicker than the structure itself.

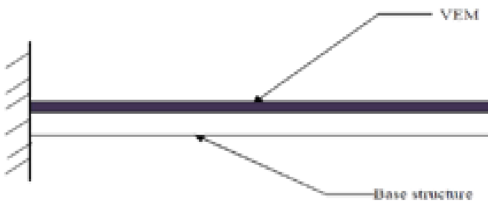

Fig. 1: Free layer damping

2) Constrained layer damping:

In the constrained layer damping technique the damping material is bonded to the structure similar to the free layer damping technique, then an another constraining layer having very high stiffness is constrained over the damping material .

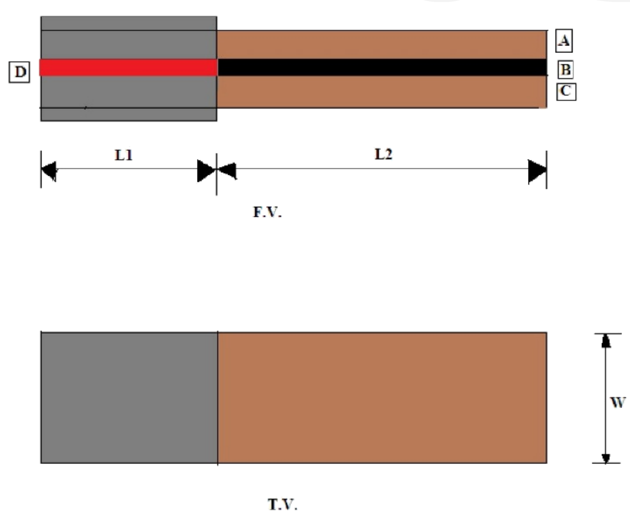

Fig. 2: Constrained layer damping

3) Partial Constrained Damping:

Partial damping means upper constraining layer and damping visco-elastic material is discontinuous (i.e. not full layer of it) it is separated by some distance.

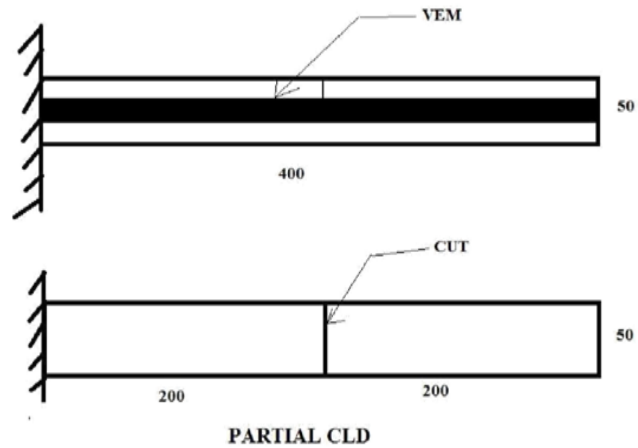

Fig. 3: Partial Constrained layer damping

3 EXPERIMENTAL INVESTIGATION

A. Attach the accelerometer on the vibrating surface whose modal parameters are to be estimated.

B. Now, impact the impact hammer on to the beam to produce excitation of the plate.

C. Repeat the procedure at different points on the beam, this is done to increase the accuracy of output.

D. Connect the accelerometer to the FFT Analyzer system with the help of the cables.

E. Connect the impact hammer to the FFT Analyzer system with the help of the cables.

F. Connect the FFT Analyzer system to the computer with the help of USB port of the computer

G. Observe the FRF (Frequency Response Function) curve generated in the RT Pro software in the computer.

4 INSTRUMENTATION

A. FFT(Fast Fourier Transform) Analyzer :

FFT Spectrum analyzers take a time varying input signal, like you would see on an oscilloscope trace, and compute its frequency spectrum. Fourier's theorem states that any waveform in the time domain can be represented by the weighted sum of sine and cosines. The FFT spectrum analyzer samples the input signal, computes the magnitude of its sine and cosine components, and displays the spectrum of these measured frequency components.

Fig. 4: FFT Analyzer

B. Impact Hammer:

Impact Hammers have been designed to excite and measure impact forces on small to medium structures such as engine blocks, car frames and automotive components. An accelerometer (or laser velocity transducer) is used to measure the response of the structure. By using a multichannel FFT analyzer, the frequency response function and mode shapes of the test structure can then be derived. Contrary to using an electro-dynamic exciter, an impact hammer does not apply additional mass loading to the test object and it provides a very portable solution for excitation.

Fig. 5: Impact Hammer

C. Accelerometer:

Accelerometers are specially designed for the measurement of whole-body vibration. It consists of a tri-axial accelerometer housed in a semi-rigid Nitrile rubber disc and complies with ISO standards. It can be placed under a seated person, on a vibrating surface with a suitable weight on top, or strapped onto the body. It detects vibration in directions along the body, back-to-front and side-to-side. It also includes transducer electronic data sheet (TEDS), which contains sensors and application-specific information, including frequency response. The built-in accelerometer is mounted inside the rubber pad by means of a clip facilitating easy removal, calibration, and subsequent remounting.

Fig. 6: Accelerometer

5 RESULTS

A. Plane Undamped Beam :

Table No.1 shows the frequency response analysis result of undamped beam is shown. From undamped beam FRF(Frequency Response Curve)curve, the vibration response amplitude (dB) and natural frequency (Hz) corresponding to each bending mode are obtained.

Mode No.	Hz	dB
1	8	4
2	55	29
3	166	34
4	334	32.33

Table No.1

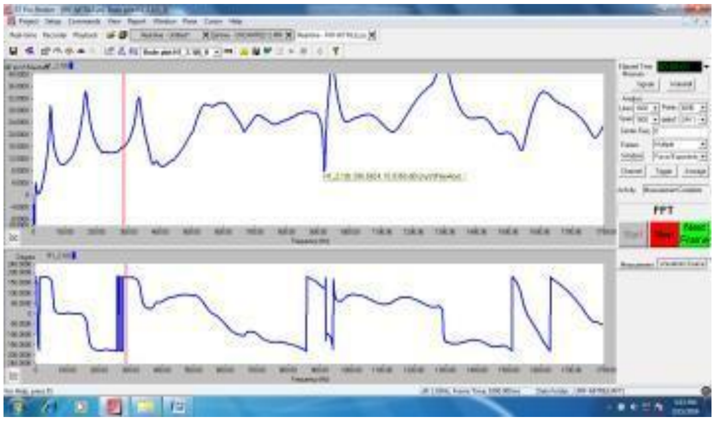

Fig. 7: Frequency response function curve of plane undamped beam

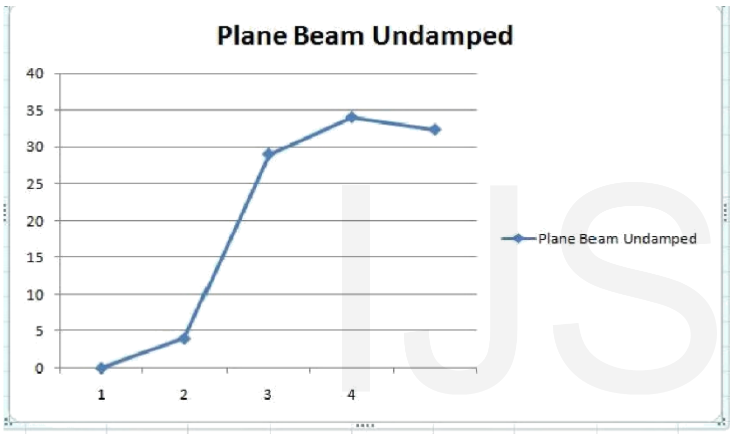

Fig. 8: Graph of mode number Vs. amplitude for Plane undamped beam

B. Beam using Butyl as VEM:

Table No.2 & 3 shows the frequency response analysis result of undamped beam is shown. From undamped beam FRF curve, the vibration response amplitude (dB) and natural frequency (Hz) corresponding to each bending mode are obtained.

1) Butyl Full Constrained:

Mode No.	Hz	dB
1	13.65	2.44
2	65	18.04
3	167.36	21.57
4	320	22.34

Table No.2

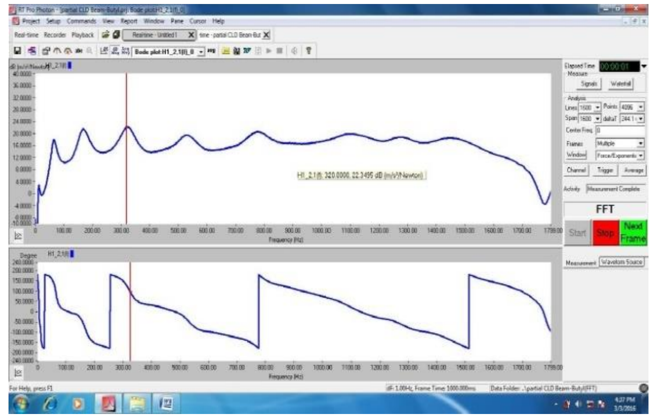

Fig.9: Frequency response function curve of Butyl (Full)

2) Butyl Partial Constrained:

Mode No.	Hz	dB
1	9	-4.4
2	54	13.23
3	166	17.19
4	308	15.43

Table No.3

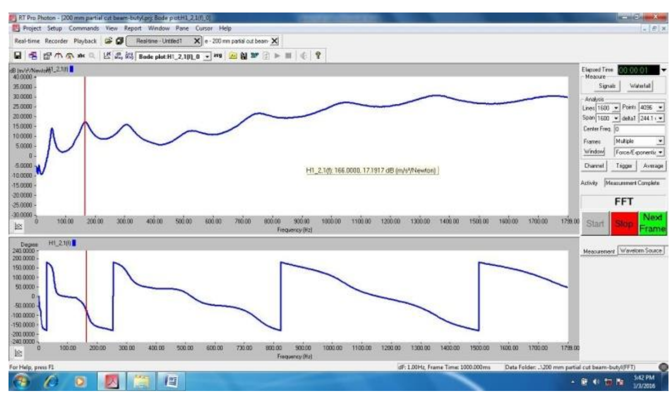

Fig.10: Frequency response function curve of Butyl (Partial)

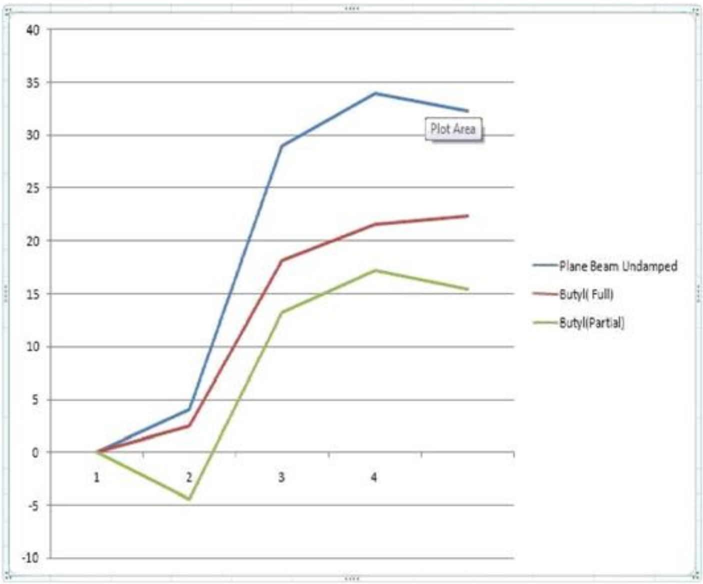

Fig. 11: Graph of mode number Vs. amplitude for Butyl VEM

C. *Beam using Neoprene as VEM*:

Table No.4 & 5 shows the frequency response analysis result of undamped beam is shown. From undamped beam FRF curve, the vibration response amplitude (dB) and natural frequency (Hz) corresponding to each bending mode are obtained.

1) Neoprene Full Constrained

Mode No.	Hz	dB
1	14	3.94
2	65	17.18
3	168	21.47
4	317	20.03

Table No.: 4

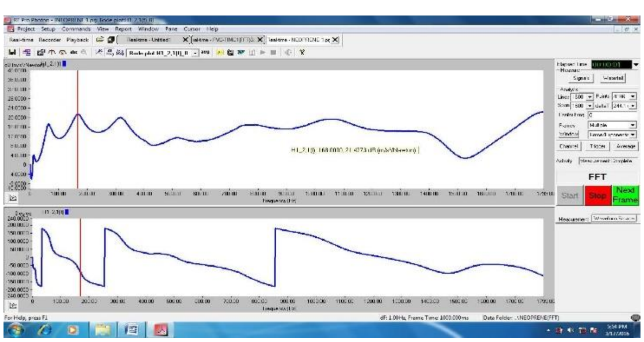

Fig.12: Frequency response function curve of Neoprene (Full)

2) Neoprene Partial Constrained

Mode No.	Hz	dB
1	9	-3.8
2	56	14.19
3	175	16.31
4	330	17.40

Table No.: 5

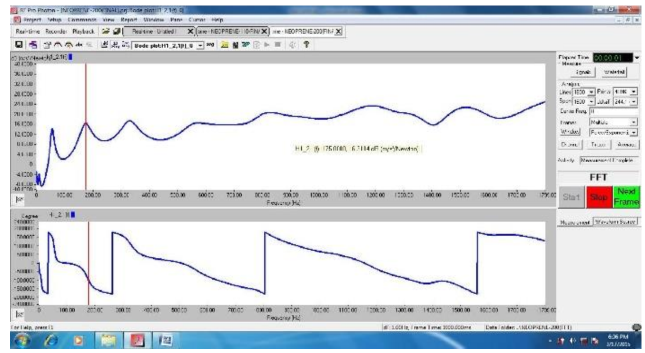

Fig.13: Frequency response function curve of Neoprene (Partial)

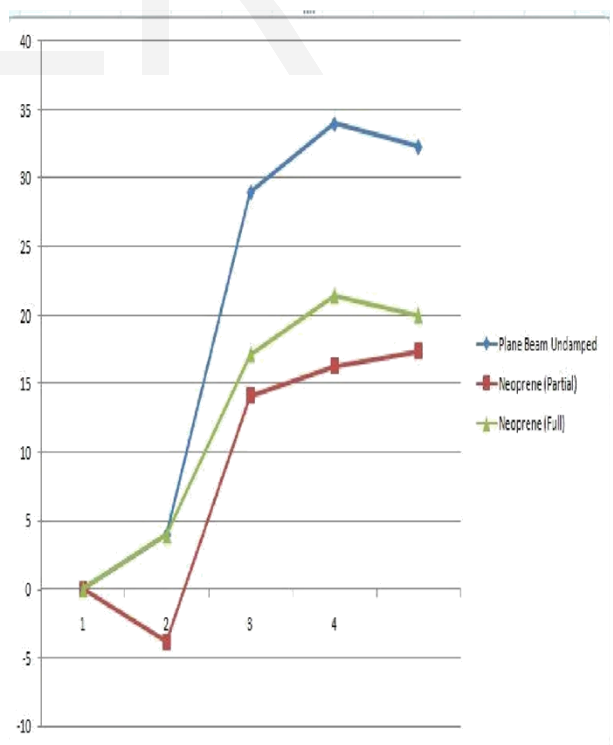

Fig. 14: Graph of mode number Vs. amplitude for Neoprene VEM

D. Beam using Styrene Butadiene Rubber (SBR) as VEM:

Table No.6 & 7 shows the frequency response analysis result of undamped beam is shown. From undamped beam FRF curve, the vibration response amplitude (dB) and natural frequency (Hz) corresponding to each bending mode are obtained.

1) SBR Full Constrained

Mode No.	Hz	dB
1	14	1.1
2	72	14.34
3	185	17.56
4	349	17.13

Table No.: 6

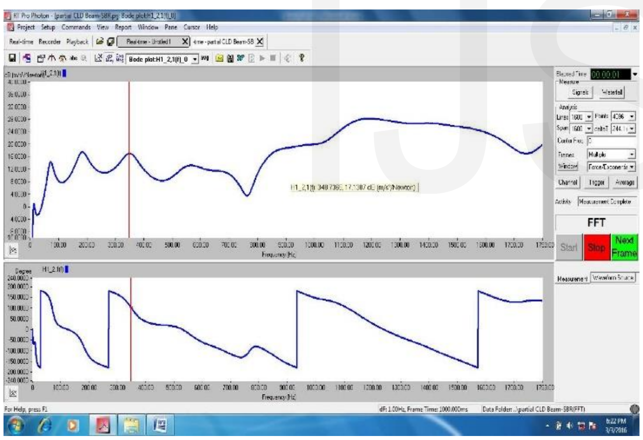

Fig.15: Frequency response function curve of SBR (Full)

2) SBR Partial Constrained

Mode No.	Hz	dB
1	10	-8.12
2	55	11.29
3	172	12.96
4	329.19	13.99

Table No.: 7

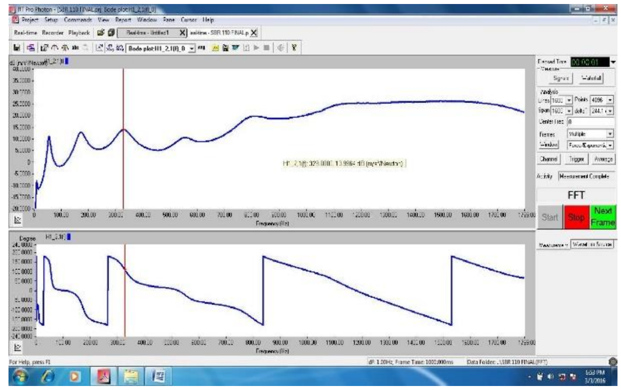

Fig.16: Frequency response function curve of SBR (Partial)

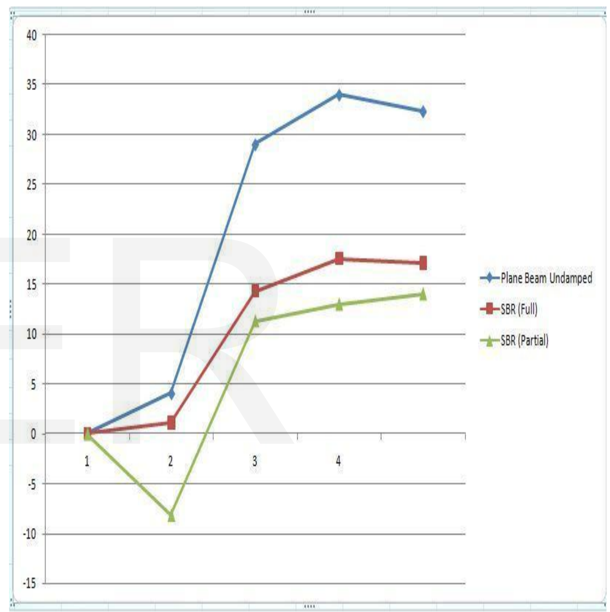

Fig. 17: Graph of mode number Vs. amplitude for SBR VEM

6 CONCLUSION

From the experimental analysis it is observed that the vibration response Amplitude of full CLD beam and Partial CLD beam with is less as compare to undamped beam. It is also observed that the vibration response Amplitude of Partial CLD beam is less as compare to full CLD beam. It is due to more shear region in partial CLD treatment causes more heat dissipation occurs which lead to reduction of vibration energy of beam. Hence partial CLD treatment is more useful than full CLD treatment. Its applications are found in many parts of Aerospace and Automotive vehicle.

7 ACKNOLEDGEMENTS

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