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Damage Detection Using Modal Strain Energy Method in Honeycomb Sandwich Beams with Multiple Delaminations

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Abstract

The modal strain energy variation of a cantilever sandwich beam (consisting of top and bottom face plates with honeycomb core in between) with multiple delaminations or debonds embedded between the face-layer laminates and the honeycomb core, from that of a sandwich beam without any delamination is studied herein using free vibration analysis. The influence of size and location of delaminations on the natural frequency is investigated for sandwich beams with multiple delaminations. It is observed that modal strain energy plot can give the location and extent of damage or debond more accurately than a mode shape plot. It is also seen that the delaminations reduce the natural frequency and modal strain energy and change the modeshape. The effect of second debond is not significant if it is located near to the free end and far away from the first debond.

Keywords: *Delamination, Debond, Honeycomb Sandwich beams, Free Vibration, Natural frequency, Mode Shape, Modal Strain Energy, Modal Strain Energy Change Ratio(MSECR)*

1. Introduction

The composite materials are being increasingly used in different fields of structural engineering replacing the metallic materials. Sandwich structures which is a class of composites have significant applications in aerospace, automotive, marine and other areas. This is due to the fact that composite materials possess high specific strength and stiffness combined with thermal and acoustic insulation. A sandwich beam is made of two skin surfaces separated by a core, and this core is glued to the skins with strong-bond adhesives. The common type of damage seen in sandwich beam is the delamination of skin from the core part. This debonding reduces the stiffness of the structure and makes it easier to buckle under compressive loads. In order to manage this problem, it is necessary to understand the critical failure modes of honeycomb sandwich components and assess the level of damage that can be tolerated in these components. In space applications we have to ensure that the components used are of zero defects. Thus damage detection techniques have great importance in this scenario. They increase safety, extent serviceability and reduce maintenance costs. Since NDT methods are time consuming and labour intensive for large structures, model based damage identification techniques are generally used to monitor changes in structural dynamic characteristics or changes in the dynamic response of structures. Vibration based methods are a new approach for damage detection and are more globally sensitive to damage than localized methods such as ultrasonic and thermography methods. Combined with modal analysis, these techniques provide local as well as global damage information. The modal strain energy based technique, which is the latest method of vibration based damage detection techniques, can serve as an effective damage detection tool, which demand computationally efficient damage models of sandwich structures. The present study deals with the modal analysis of the honeycomb sandwich beam with single-face multiple delaminations.

J.T.S Wang and J.A.Gibby [2] investigated the free vibrations of beams containing split regions and established a simple and systematic procedure based on a consistent formulation for determining the natural frequencies and their corresponding mode shapes for the split beams. Hyeung-Yun Kim and Woonbong Hwang [1] conducted studies on the effect of debonding or delamination (embedded between the face layer laminates and the honeycomb core) on natural frequencies and frequency response functions of honeycomb sandwich beams both theoretically and experimentally and the results were compared. J.R. Banerjee [3] studied the free vibration of three-layered symmetric sandwich beams using the dynamic stiffness method. The method was found to be superior and useful over finite element method in many circumstances due to the fact that it accounts for an infinite number of degrees of freedom of a vibrating structure. Shishir Kr. Sahu [7] performed experimental and numerical study on vibration of industry-driven ‘woven’ fiber Glass/Epoxy composite plates with different size of delamination and boundary conditions (B.C). Nisha.A.S and Saraswathy.B [6] presented the analytical formulation for free vibration of sandwich beams consisting of two different layers and is split along the interface with single and multiple through-width delaminations. Manoj Kumar [5] carried out the experimental validation of a vibration-based damage identification method based on changes in modal strain energies before and after occurrence of damage for a composite sandwich beam. The accuracy and sensitivity of this method for predicting location, extent and type of damage in a composite sandwich beam were examined. The unique contribution of this method is the capability to predict damage extents for interactive damage modes. K. R. Pradeep [4] conducted a study on the modal strain energy change ratio(MSECR) for the damage identification in honeycomb sandwich structures by modal analysis using ANSYSTM finite element package for accurate simulation of undamaged and damaged sandwich plate configurations. The study was limited to single through-width symmetric delamination. But multiple delaminations further weaken a sandwich beam and cause a shift in natural frequencies. An increased number of delaminations also considerably complicate the problem and hence no analytical solution has been studied in detail so far in the case of a honeycomb sandwich beam with multiple delaminations. So the present study deals with the modal analysis of the honeycomb sandwich beam with single face-multiple delamination using the finite element package ANSYS(Mechanical APDL) for the detection of damage extent and location using modal strain energy plots.

2. Finite Element Analysis

Since the reliability of the finite element package ANSYS for the modal analysis of delaminated sandwich beams has already accepted from the previous studies using the software, ANSYS(Mechanical APDL) is used in the present study for modal analysis.The finite element models used for the present study are generated using a meshing application developed in Visual Basic. A single keypoint is created at the ends of the delamination and two keypoints are created anywhere along the delamination using this program.The volumes above the delaminated region are defined using one of the two keypoints and the volumes below are defined using the other keypoint. This ensures no connectivity between the volumes above and below the delaminated region and thus it behaves as a delaminated beam. A honeycomb sandwich beam of length 630 mm, width 96 mm, thickness of the core 40mm and thickness of the top and bottom facing 0.5 mm as shown in Fig.1 is modelled. The honeycomb core is of aluminium and the skin is of carbon-epoxy. Intact beam and beam with double debonds are analysed for cantilever boundary condition. Element used for the skin and core is SHELL 63 whose geometry is shown in Fig.2.

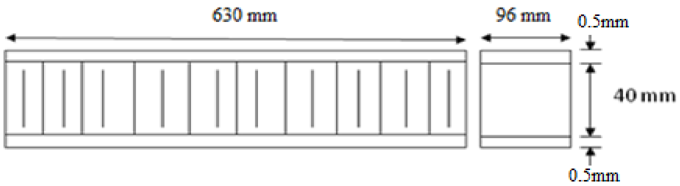

Fig.1: Dimensions of the sandwich beam

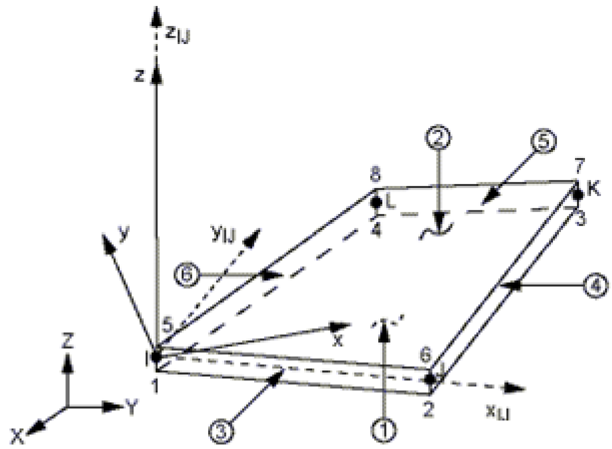

Fig.2: Geometry of shell63

Table 1 & Table 2 shows the properties of skin and core used in the modal analysis.

Table 1: Properties of Carbon Epoxy skin

Modulus of elasticity Exx	294.3 GPa
Modulus of elasticity Eyy=Ezz	5.957 GPa
Transverse shear modulus Gxx=Gyy=Gzz	4.896 GPa
Poison's ratio	0.346
Density	1700 kg/m^3

Table 2: Properties of Honey comb core (Aluminium)

Poisson's ratio	0.3
Density	2700kg/m^3
Modulus of elasticity	70GPa
Cell size	6mm (inscribed circle diameter)
Cell wall thickness	2.54 x 10^{-5} m

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The 3D model, used for the present analysis is given in Fig.3.

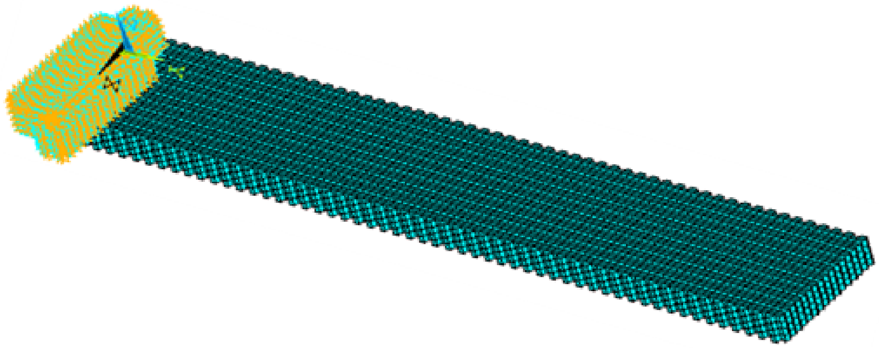

Fig.3: 3D Model of honeycomb cantilever beam.

The specimens used in the modal analysis are shown in Fig.4 to Fig.7.The location, size and distance between the delaminations are varying in each cases.

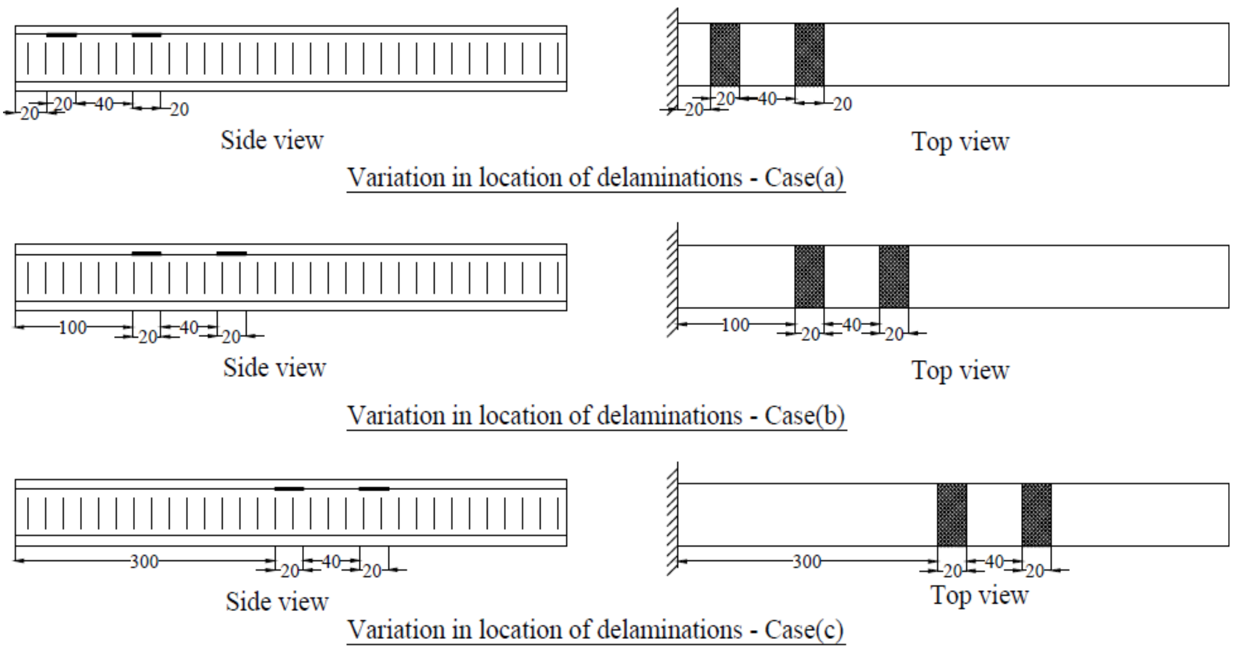

Fig.4: Schematic diagram of specimens with variation in location of delaminations

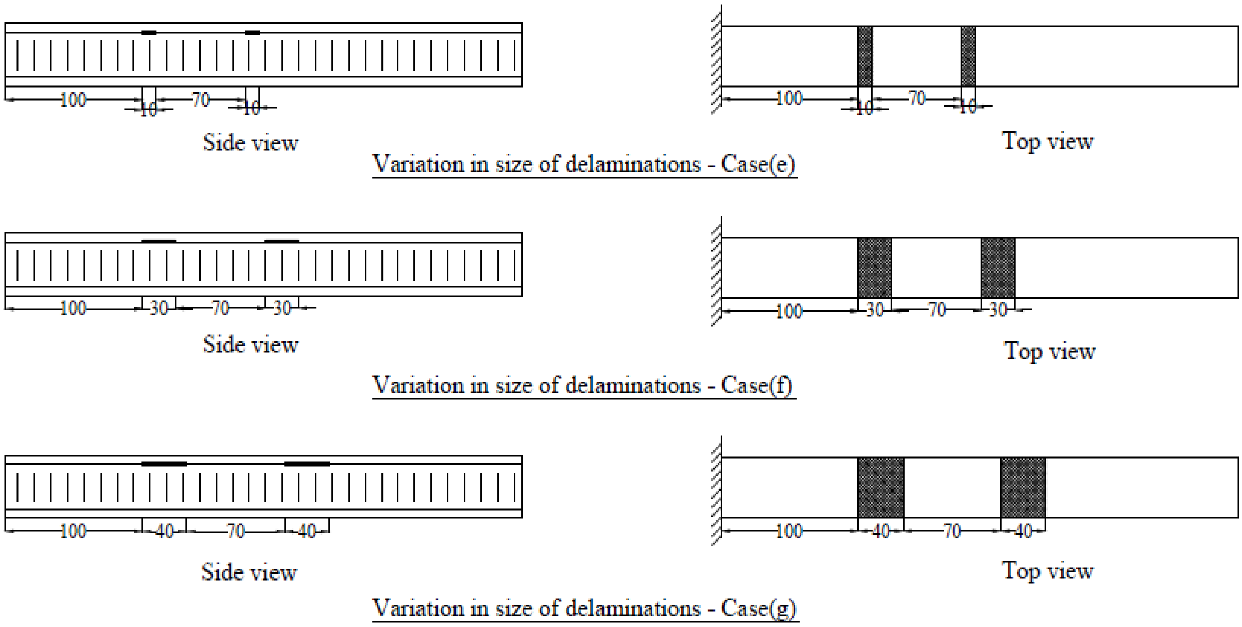

Fig.5: Schematic diagram of specimens with variation in size of delaminations

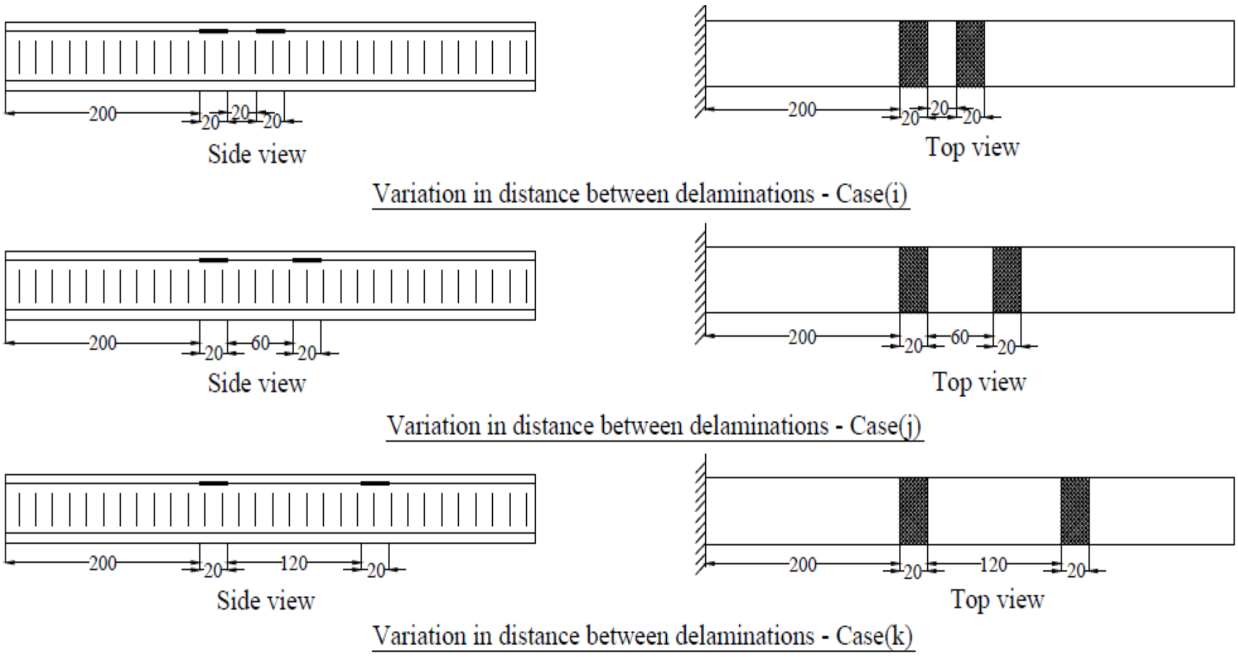

Fig.6: Schematic diagram of specimens with variation in distance between delaminations

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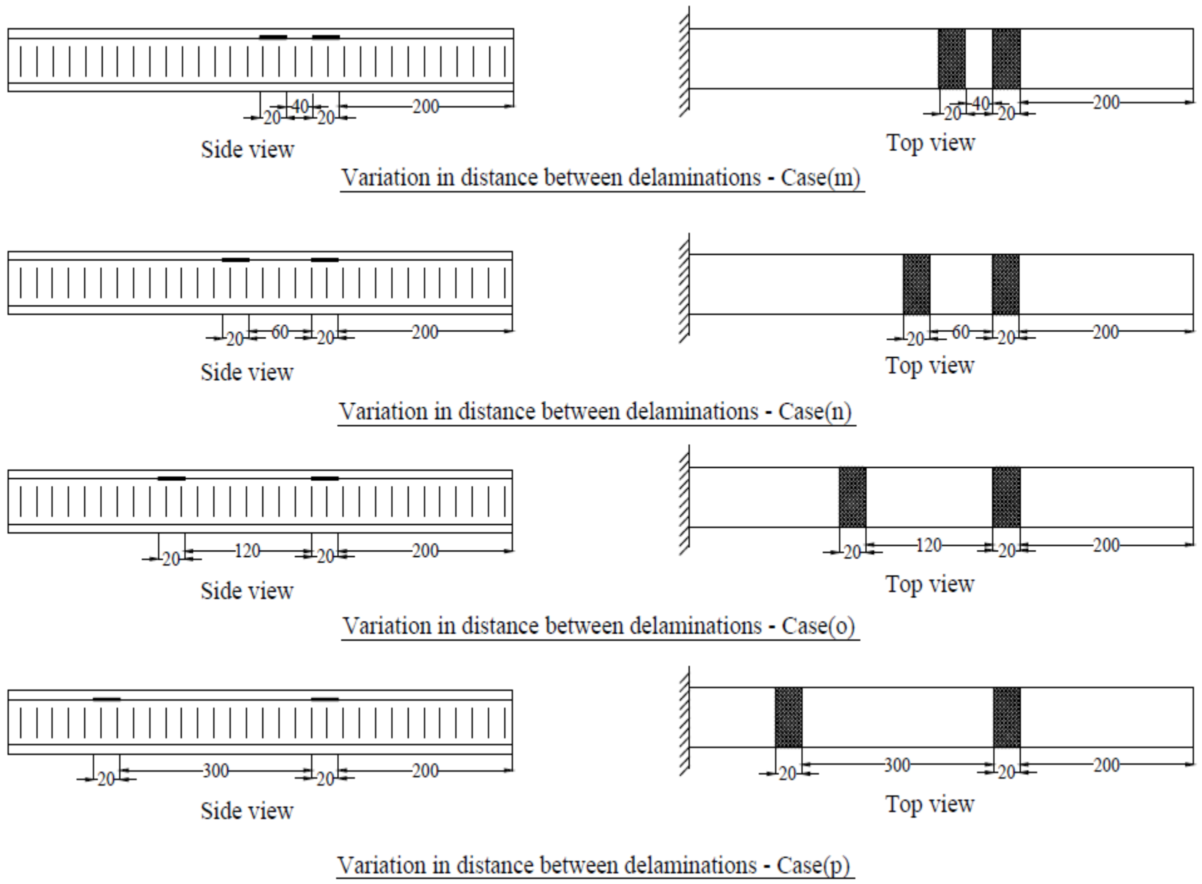

Fig.7: Schematic diagram of specimens with variation in distance between delaminations

3. Results and Discussions

Modal analysis has been carried out and extracted first four natural frequencies, but first and second mode frequencies are considered in the study since their effect is maximum and it is found that the results can be applicable to higher modes of vibration. It is noted from the results presented in Table 3 that there is a decreasing trend in frequencies of the damaged honeycomb sandwich beam due to reduction in stiffness. The maximum extent of delamination applied in the study is 6.35% of the total length of the specimen which can be considered as a relevant case of delamination since such minor debonds are difficult to be detected by obsolete vibration-based damage identification methods and thus require modal strain energy method. Therefore the modal strain energy plots prepared are for delaminations of size less than 50mm.Fig.8 to Fig.11 displays the accuracy obtained in detecting location of debond while using modal strain energy plot than a mode shape plot.

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Table 3: Natural frequencies of various specimens from modal analysis

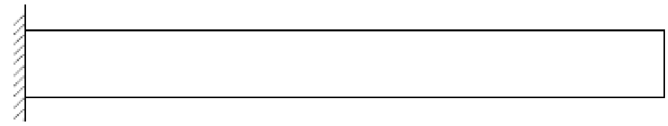
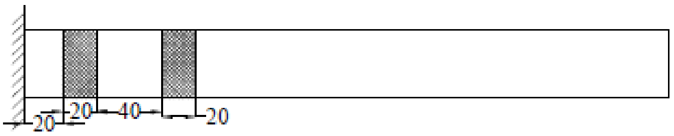
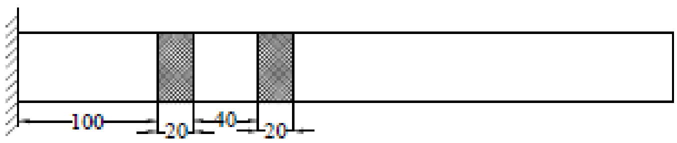
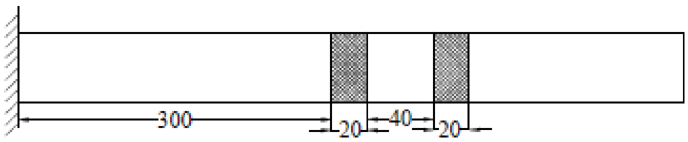
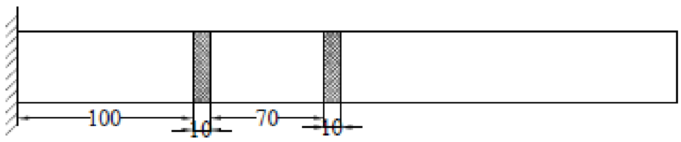
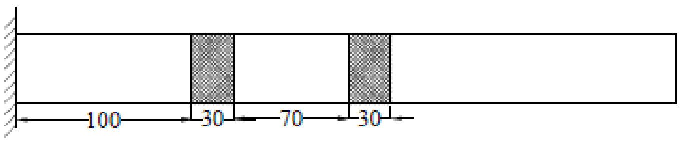
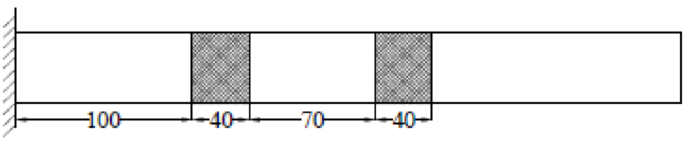
Specimen	First mode frequency(Hz)	Second mode frequency(Hz)
Intact Sandwich beam 	74.327	442.89
A 	73.936	438.96
B 	73.993	439.50
C 	74.263	441.13
E 	74.172	441.74
F 	73.534	428.43
G 	73.746	434.26

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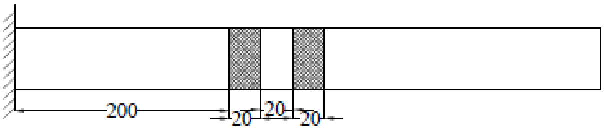
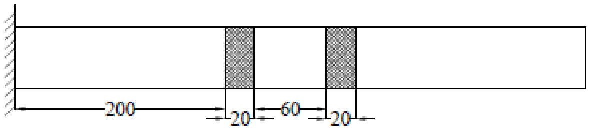
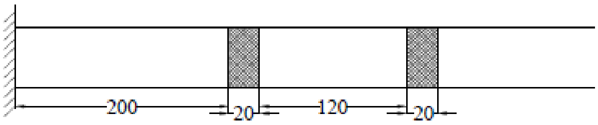
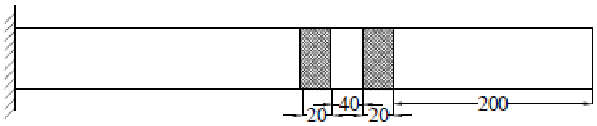
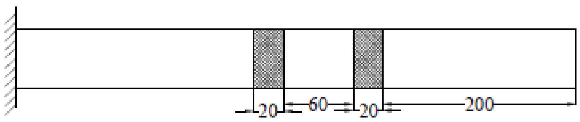
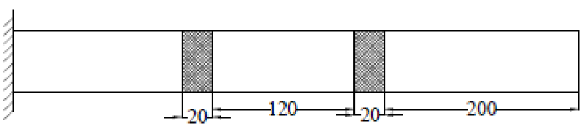
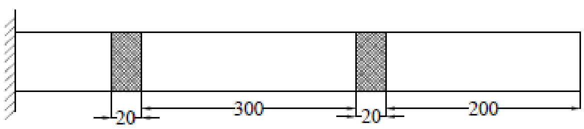
Table 3: Natural frequencies of various specimens from modal analysis

Specimen	First mode frequency(Hz)	Second mode frequency(Hz)
I 	74.127	440.55
J 	74.160	440.84
K 	74.174	440.60
M 	74.293	441.20
N 	74.277	441.16
O 	74.262	441.41
P 	74.080	439.57

The displacement values of the specimen are plotted from which the strain energy values are calculated by using the formula,

$$U = \frac{1}{2} \int_0^l EI \left(\frac{\partial^2 w}{\partial x^2}\right)^2 dx$$

(1)

Where, U = Modal strain energy

EI = Flexural rigidity of the sandwich beam

w = Displacement of the sandwich beam under free vibration

Modal strain energy change ratio,

$$(\text{MSECR}) = \frac{\text{Modal strain energy of intact beam} - \text{modal strain energy of debonded beam}}{\text{modal strain energy of intact beam}}$$

Comparison of delaminated sandwich beam with sound sandwich beam considering the first mode results are shown in following figures.

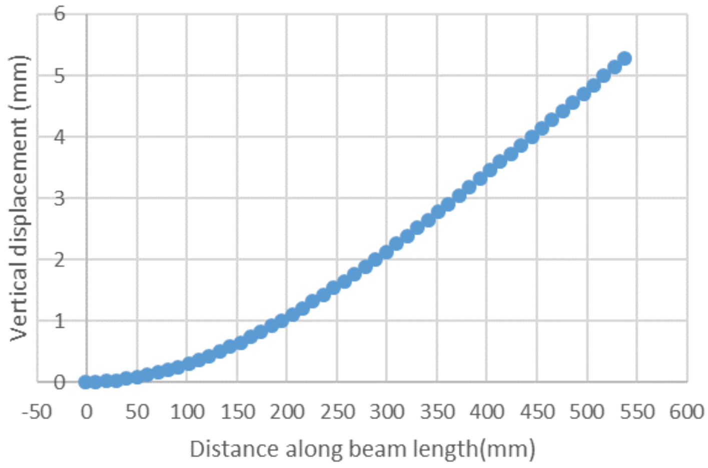


Fig.8: First mode shape of sound sandwich beam specimen

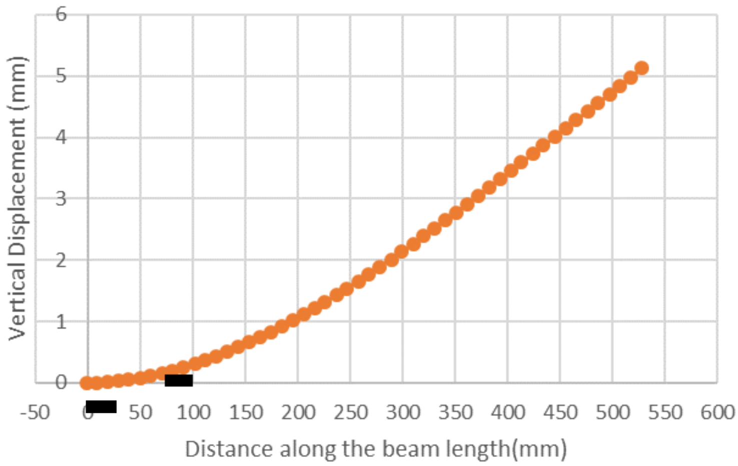


Fig.9: First mode shape of sandwich beam specimen A

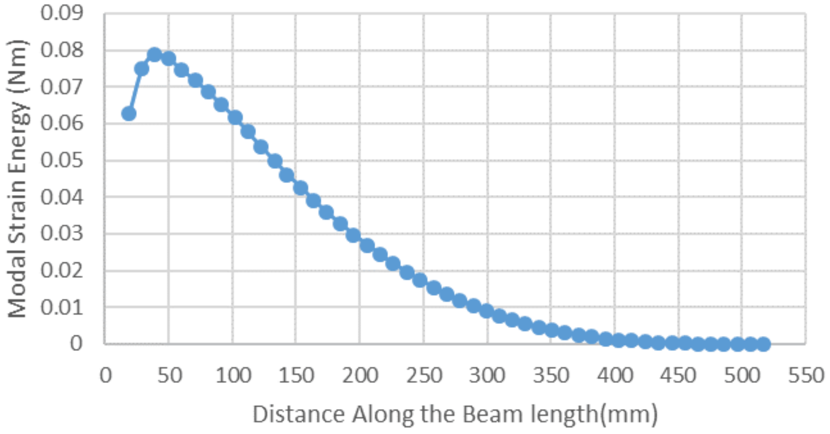


Fig.10: Modal Strain Energy for first mode shape of sound sandwich beam specimen

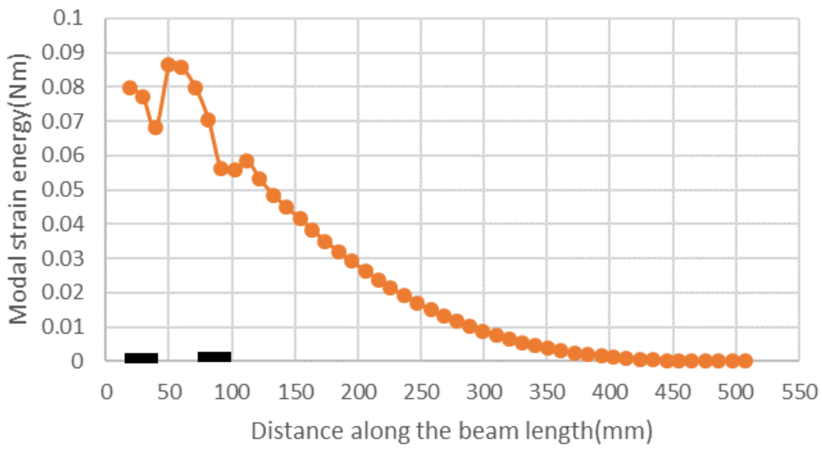


Fig.11: Modal Strain Energy for first mode shape of sandwich beam specimen A

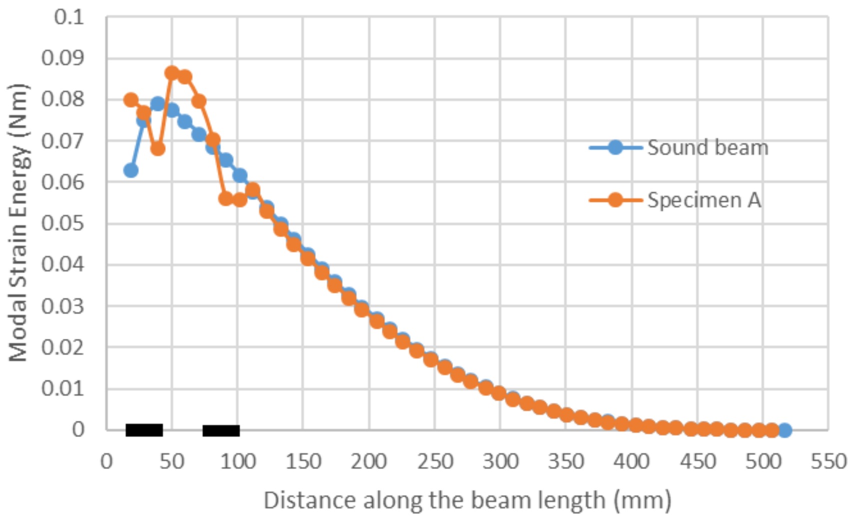


Fig.12: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen A with that of sound beam specimen

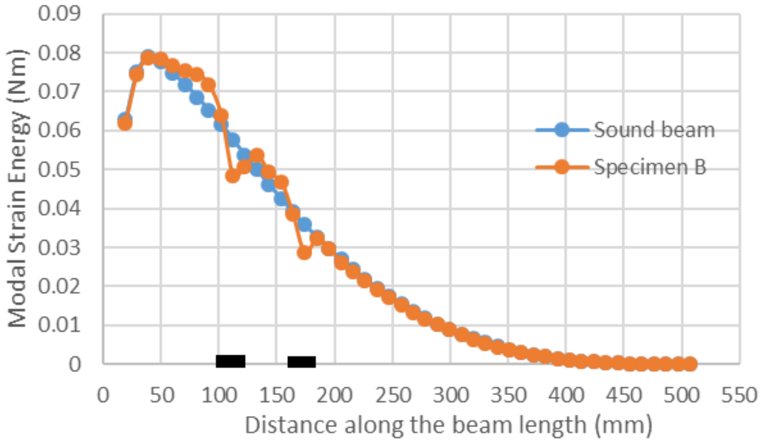


Fig.13: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen B with that of sound beam specimen

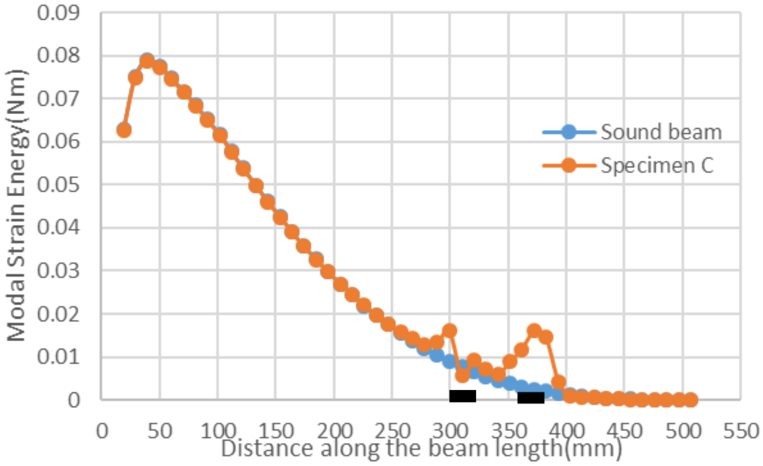


Fig.14: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen C with that of sound beam specimen

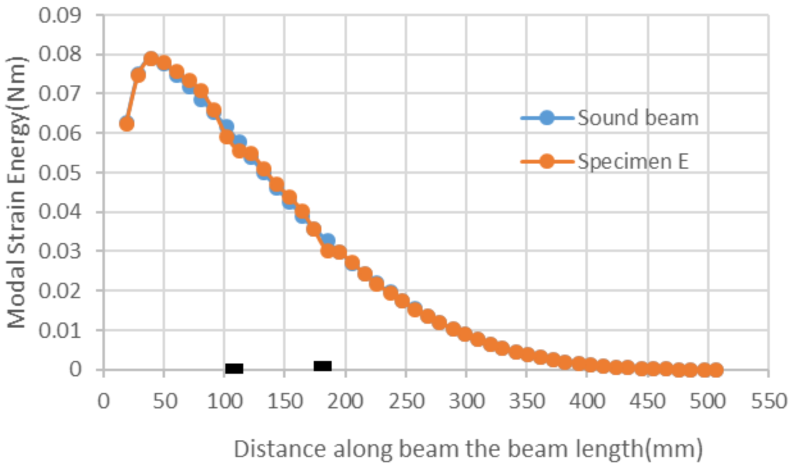


Fig.15: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen E with that of sound beam specimen

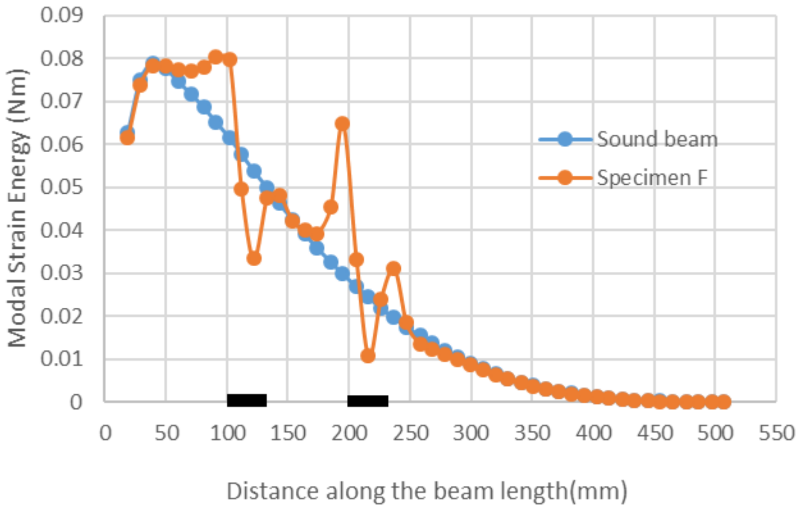


Fig.16: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen F with that of sound beam specimen

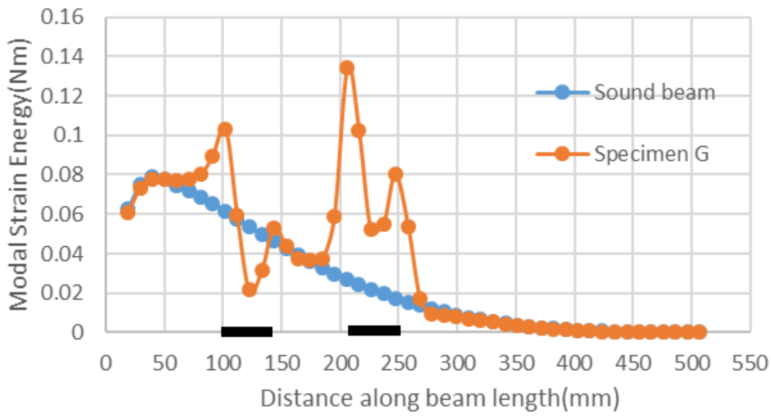


Fig.17: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen G with that of sound beam specimen

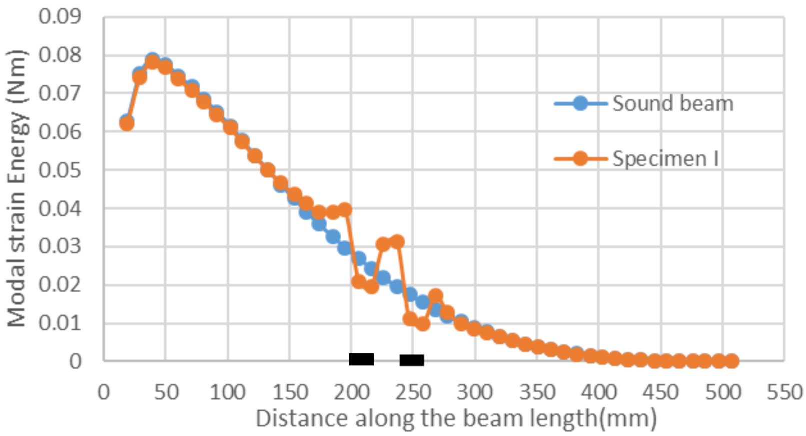


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Fig.18: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen I with that of sound beam specimen

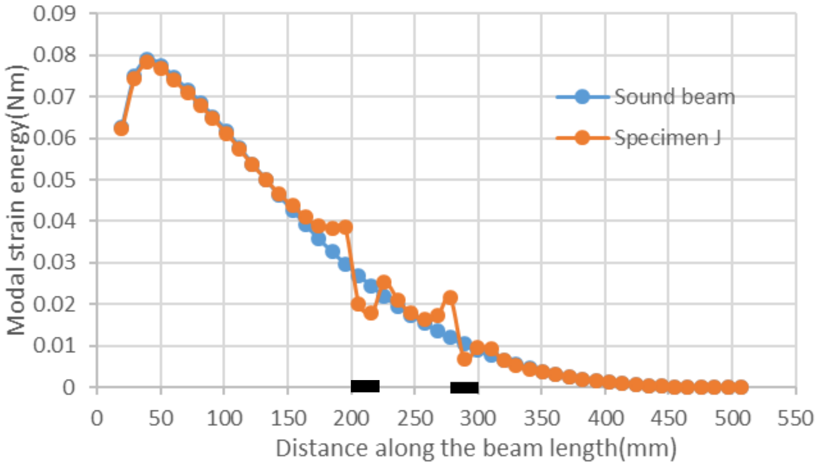

Fig.19: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen J with that of sound beam specimen

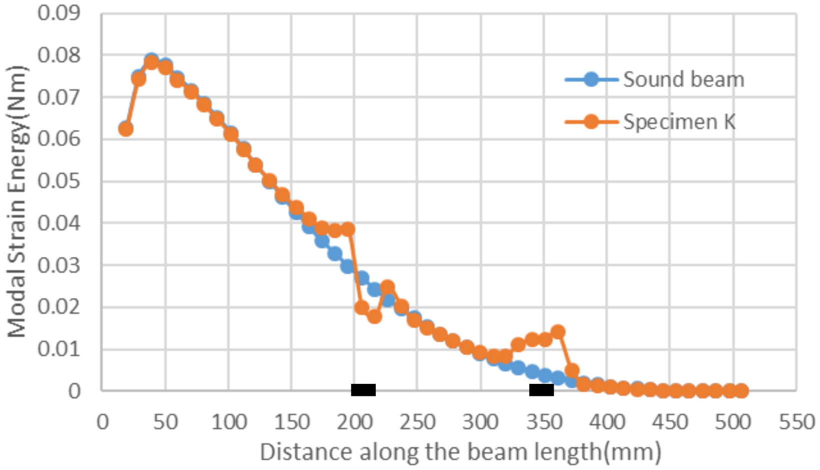

Fig.20: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen K with that of sound beam specimen

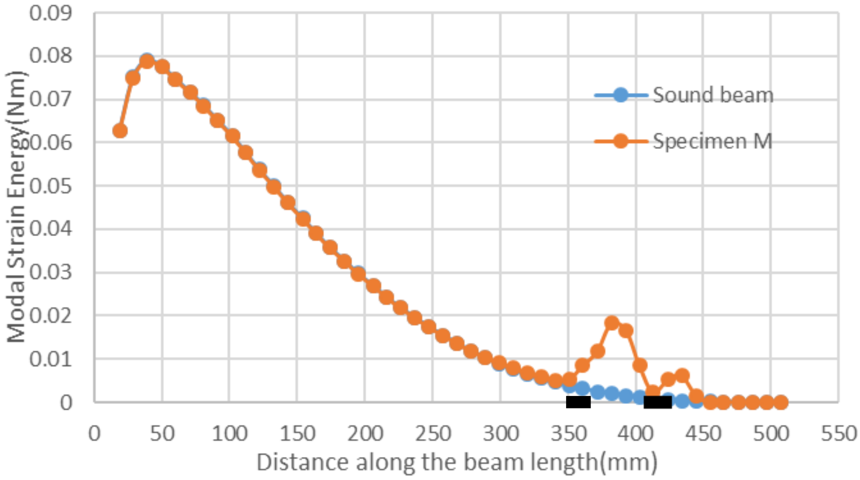

Fig.21: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen M with that of sound beam specimen

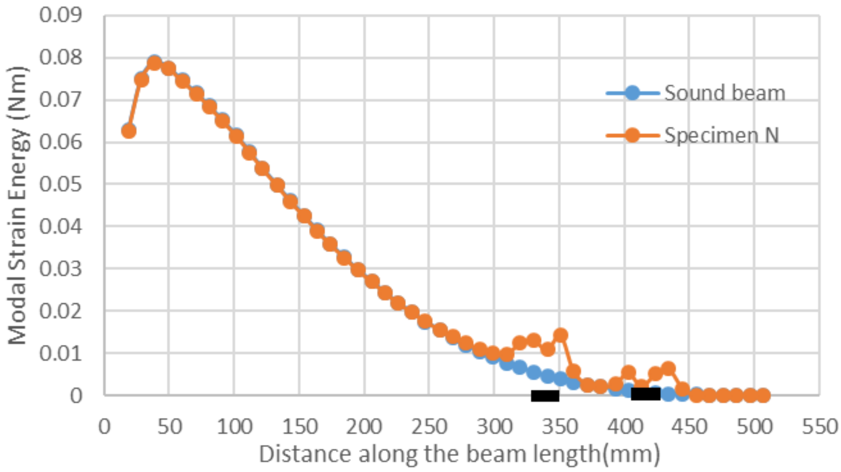

Fig.22: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen N with that of sound beam specimen

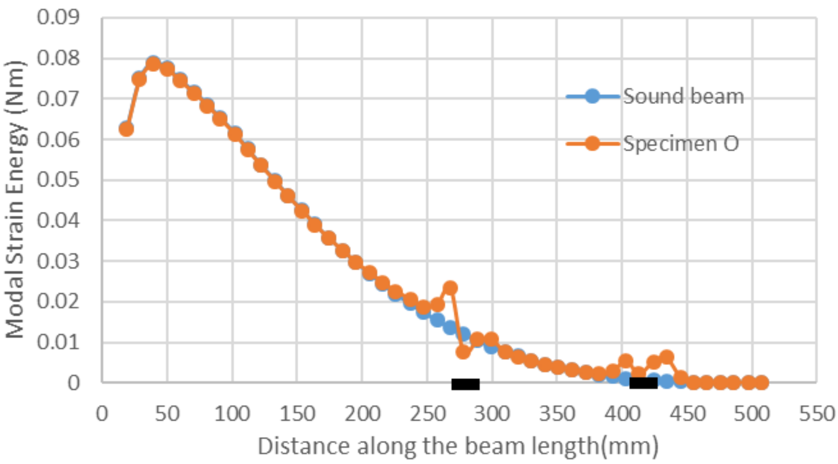

Fig.23: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen O with that of sound beam specimen

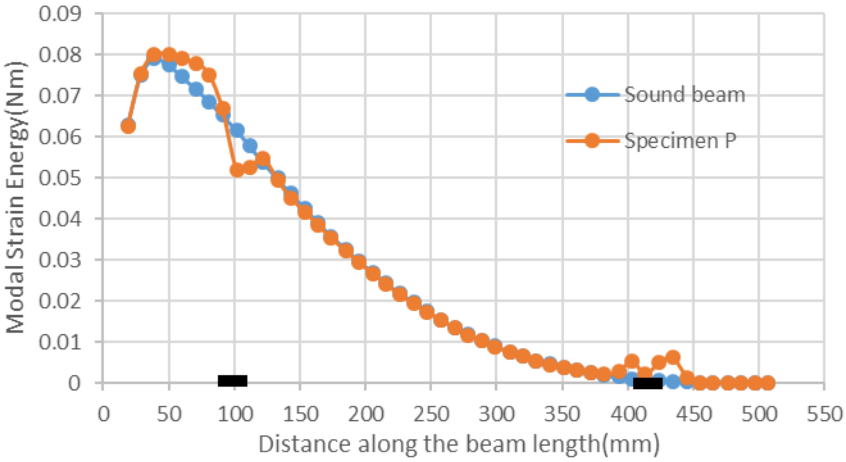

Fig.24: Comparison of modal strain energy plot for first mode shape of sandwich beam specimen P with that of sound beam specimen

A decrease in strain energy is seen at the location of debond but an irregularity in the modal strain energy variation is seen beyond 50% of the length of the specimen from the fixity by observing Fig.12 to Fig.24.A sudden increase in modal strain energy is seen just prior to the location of debond in some cases where the debond size is comparatively large.

The variation in strain energy of the different cases of delaminated sandwich beams from that of sound sandwich beams for higher modes can be obtained from modal strain energy change ratio (MSECR) plot as shown in the following figures. Fig. 25 to Fig.37 shows the MSECR plots for the second mode.

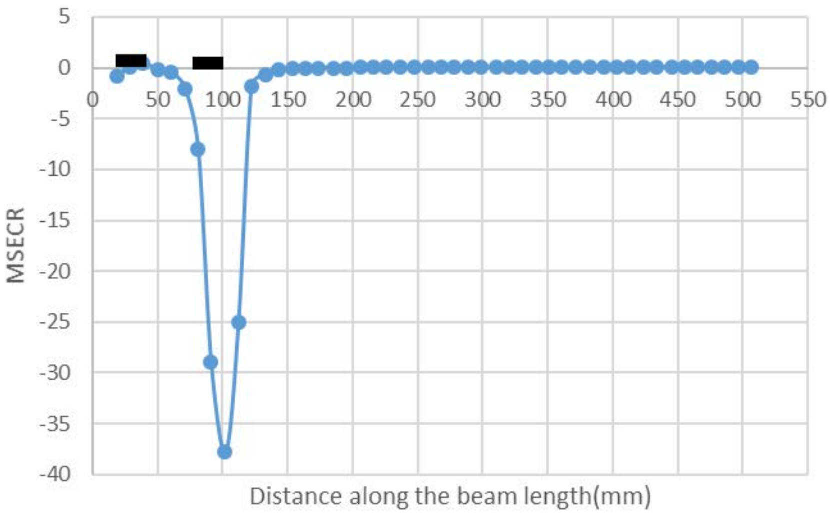

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Fig.25: MSECR plot of Specimen A

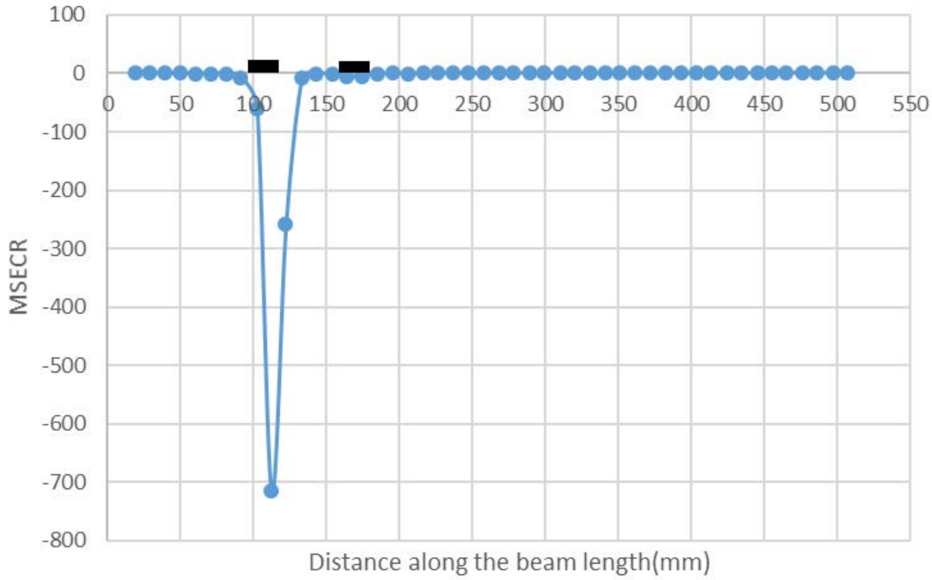

Fig.26: MSECR plot of Specimen B

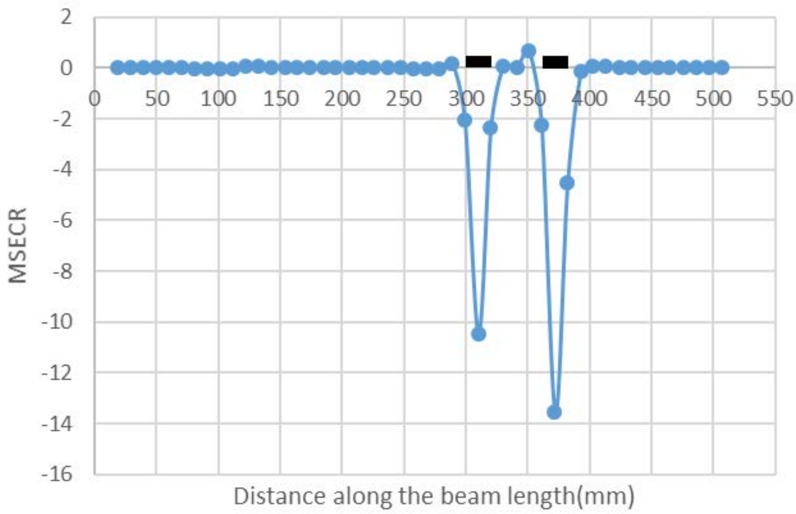

Fig.27: MSECR plot of Specimen C

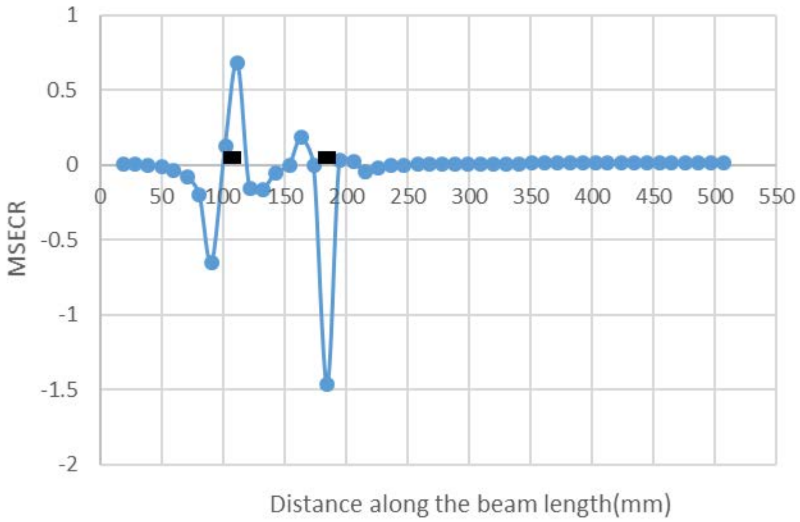

Fig.28: MSECR plot of Specimen E

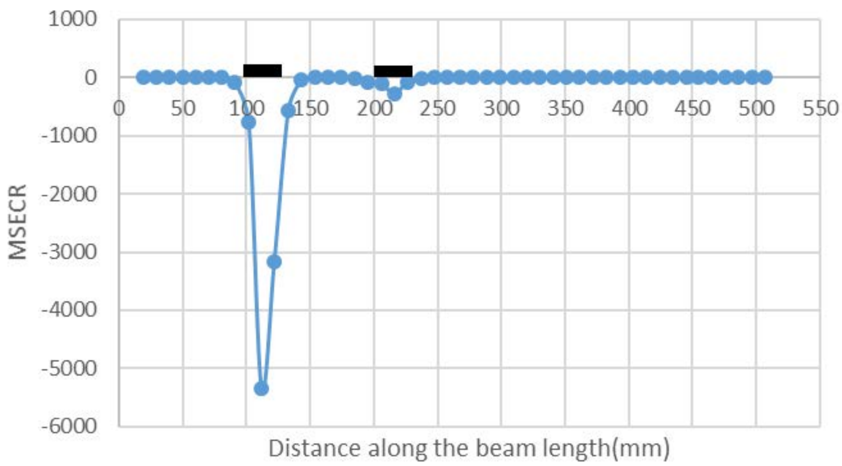

Fig.29: MSECR plot of Specimen F

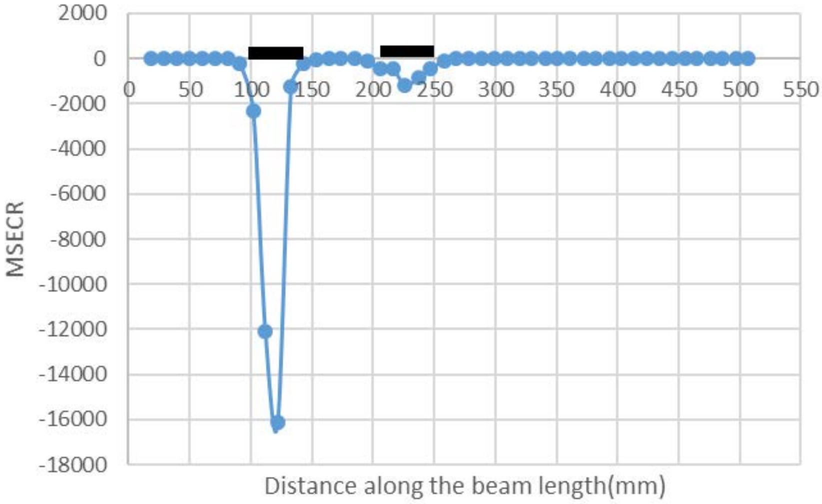

Fig.30: MSECR plot of Specimen G

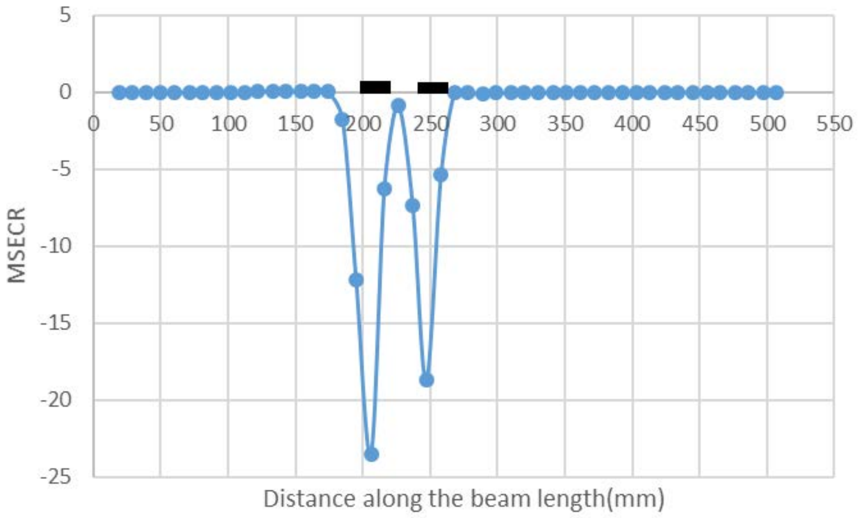

Fig.31: MSECR plot of Specimen I

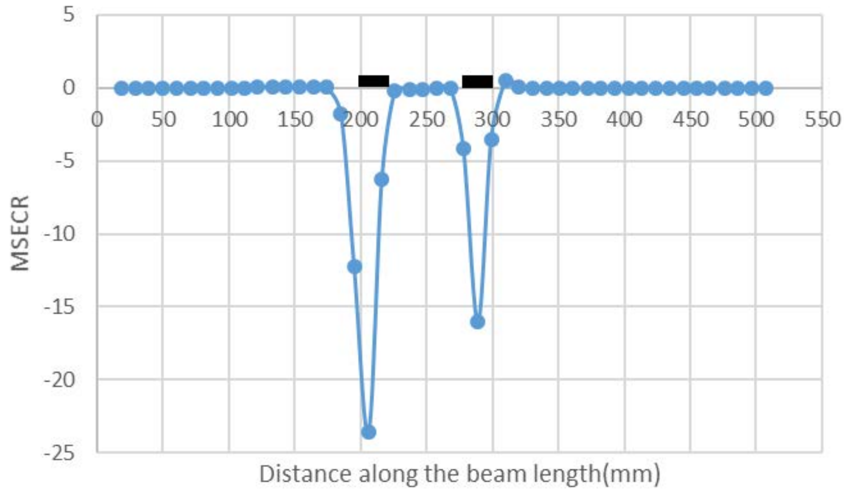

Fig.32: MSECR plot of Specimen J

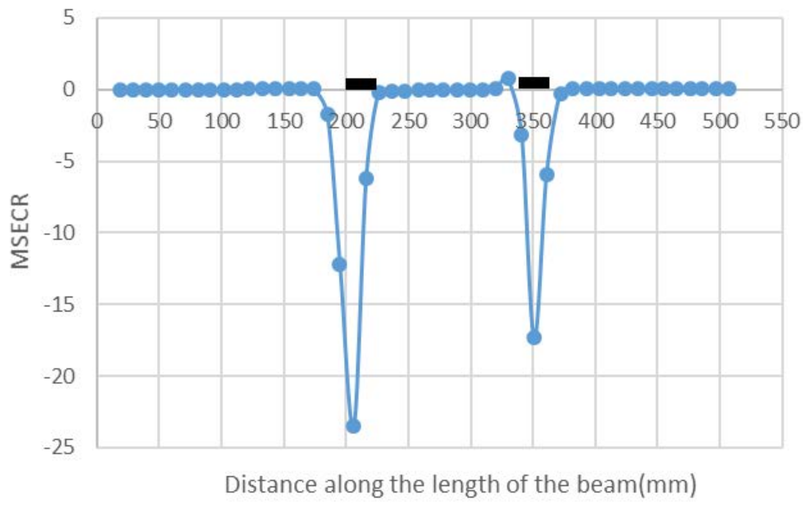

Fig.33: MSECR plot of Specimen K

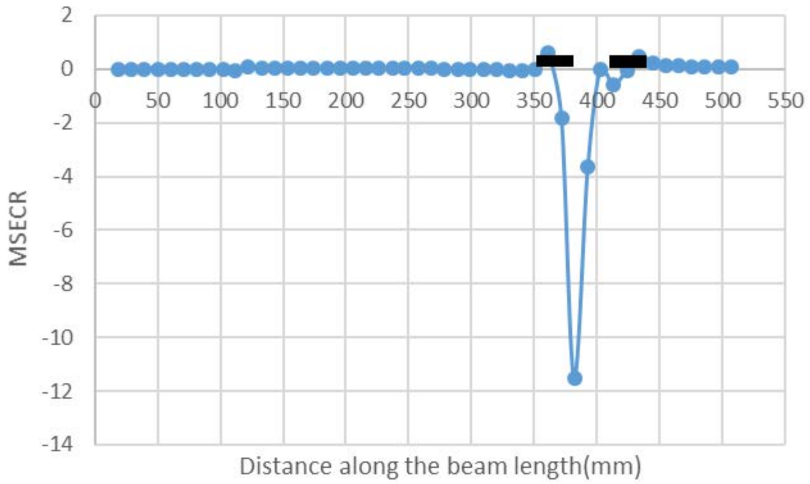

Fig.34: MSECR plot of Specimen M

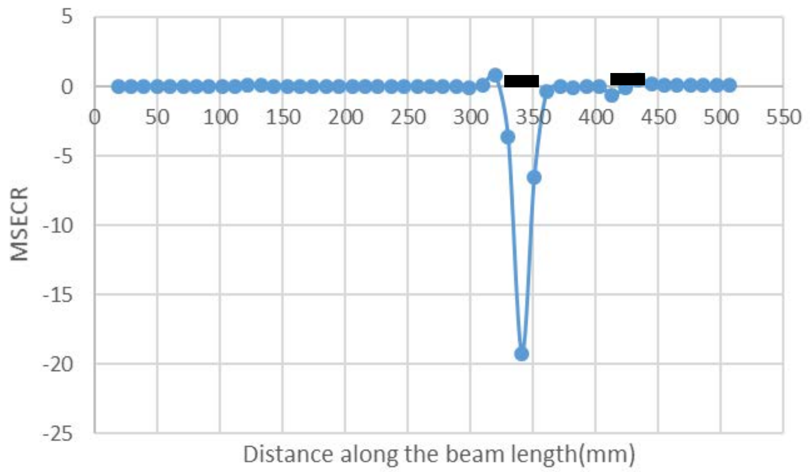

Fig.35: MSECR plot of Specimen N

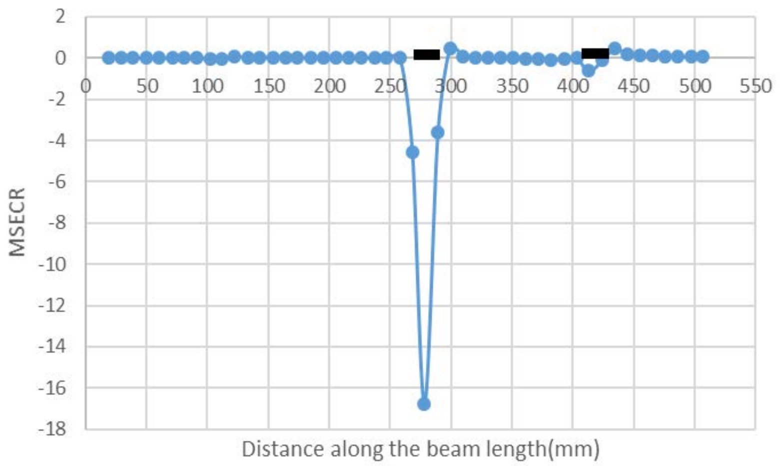

Fig.36: MSECR plot of Specimen O

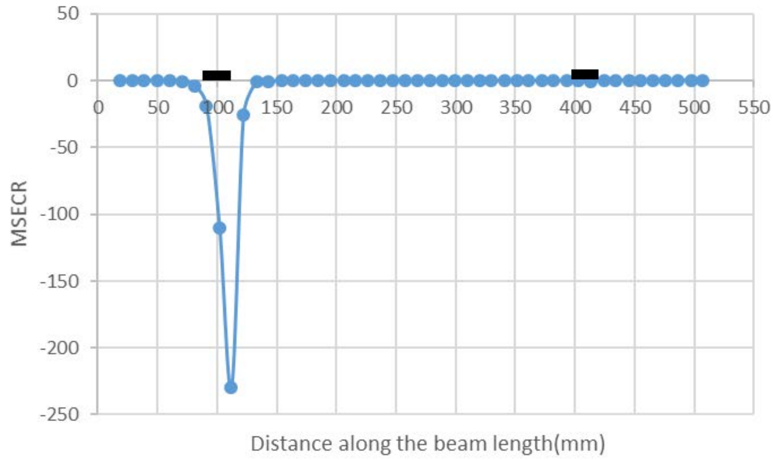

Fig.37: MSECR plot of Specimen P

The MSECR value corresponding to both of the debonds is found larger when they are very close and the value for the second debond from fixity diminishes when it is located far away from the first debond. But a debond very near to fixity is having a negligible effect on MSECR.

4. Concluding Remarks

The modal analysis of the honeycomb sandwich beam containing multiple delaminations on the interface between the top-skin and the core performed to extract the natural frequency and modal strain energies is presented in this paper. The influence of delamination size and location on the natural frequency and modal strain energy is also investigated. Although the effect of delamination is studied on first two modes of vibration, the results obtained are also applicable to higher modes of vibration. The study proposes the use of modal strain energy and the modal strain energy change ratio (MSECR) as in indicator of damage and for detecting the damage location and size. Applicability of the MSECR concept for damage detection in the honeycomb sandwich structures is examined by introducing debonds in the sandwich beam. Reduction trend is noticed in natural frequencies of debonded sandwich structures due to reduction in stiffness but is not sufficient to locate the damage. Thus the modal strain energy and MSECR helps to indicate the damage from the measure mode shape. It is observed that the presence of a second debond does not produce a considerable reduction in natural frequencies compared to that of a beam with single debond. It is seen that modal strain energy plot gives a better idea of debond than a mode shape plot. The influence of debonds on strain energy is well felt when it is located within a length less than 50% of the total length of the specimen from fixity. The effect of a second debond is felt only when it is very close to the first one. It is clear that as the spacing between two similar debonds is greater than the length of the debond, there is no reasonable change in frequency values and mode shapes and the effect on modal strain energy is also minimum. Hence for identifying the exact location and extent of multiple debonds also, the modal strain energy plots and MSECR plots can be effectively used.

5. References

[1] Hyeung-Yun Kim ,Woonbong Hwang, "Effect of delamination on natural frequencies and frequency response functions of honeycomb sandwich beams" , *Composite structures* 55 (2002),51-62

[2] J.T.S Wang, J.A.Gibby, "Vibrations of split beams", *Journal of Sound and Vibration* 84(4)(1982), 491-502

[3] J.R.Banerjee, "Free vibration of sandwich beams using the dynamic stiffness method", *Computers and Structures* 81(2003),1915-1922

[4] K. R. Pradeep, "Modal strain energy change ratio for the damage identification in honeycomb sandwich structures" , *Canadian Journal of Basic and Applied Sciences* 02(01) 2014, 10-24

[5] Manojkumar, "Experimental validation of modal strain energies based damage identification method for a composite sandwich beam", *Composites Science and Technology* 69(2009),1635-1643

[6] Nisha.A.S, Saraswathy.B, "Dynamic Analysis of Delaminated Sandwich Composites", *International Journal of Modern Engineering Research (IJMER)* 3(1) 2013, 172-17

[7] Shishir Kr. Sahu, "Experimental and numerical study on vibration of industry-driven 'woven' fiber Glass/Epoxy composite plates with different size of delamination and boundary conditions