Damping Evaluation of Conventional and Composite Plates using Different Structures

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***Abstract*—Sandwich composites are widely used in aerospace and automobile structure and panels for their excellent stiffness and low weight. Composite materials exhibit a wide range of mechanical properties and a complex structural behavior. Fiber reinforced polymer composites (FRP)as compared to conventional materials have properties such as light weight, high specific strength and stiffness which are applicable to many engineering applications such as automobile structures, body panel, aerospace manufacturing parts, snowboards etc. Damping is an important aspect which is related to behavior of composite structures andwhich results in dissipation of energy in a vibration system. In this present work Frequency response functions are obtained for different composites with two boundary conditions and required frequencies with mode shapes.In order to obtain a frequency response function considering viscoelastic damping properties, the study is based on modal analysis of laminated sandwich plate, stiffened plate and Honeycomb cored sandwich plate in the frequency domain. Sandwich structure layered plates are formed of these three materials: Aluminium (Al), Mild Steel (MS) and Fiber Reinforced Plastic (FRP) and thus frequency response function graphs are plotted for these sandwich structure plates and the results are compared. All the work is done inANSYS 14 Software and experimental validation with the Fast Fourier Transform analyzer.**

Keywords— Composite plates of different structures, CATIA and ANSYS Software, Modal analysis using FFT Analyzer and half power band width method.

I. INTRODUCTION

Composite structures, beams, plates and shells are common in many sectors of the automotive and aircraft industries. Automobile manufacturers with the use of composites are induced to reduce weight and with the customer demand have fuel economy and high performance at low cost. Use of such structures is now being considered for naval applications because of their potential for improved strength to weight ratio and resistance to harsh environments. As composite materials are widely used in many fields such as car body panel and structures so there is a need for accurate prediction so that they can be designed against the failure due to various types of dynamic loads. In the past few years the problem of health monitoring and fault detection of structuresis greater concern.Subsequently, these methods of fault detection are based on the comparison of the vibrant responseof the healthy structure with the active response of the deserted structure. Each of these has its own aspects and problems that has an effect on the results of the fault detection. Innovations inaircraft ,design of motor vehicle application and light-weight sandwich structures have formed the basis for the development of honeycomb structured panels and their advantages are low weight, good stiffness with great structural strength. Due to anti-shock properties, honeycomb structures are used today as shock-absorbing layers both in automobile and in sports gear productions. They are suited for design and structural applications as a result of their minimum ratio of weight to load-bearing and bending strength. Additionally this composite material which consists of a honeycomb core and external facing plates are adapted according to the requirements with regard to strength and choices of materials made and the good properties of these materials are being considered and valued. Due to the drawbacks of vibrations, it is important to eliminate or reduce the vibration i.e. provide damping for this purpose the vibration analysis is to be done. In analysis of vibration we can find out the response of vibration in the forms of natural frequencies and displacements. Transverse shear effect and damping loss factor increases with the increase in the thickness of the skins and core of the sandwich.In this present work area of vibration analysis, we have taken plates of different materials and formed composites plate, stiffen plate and viscoelastic honeycomb core composite plate.The conventional and FRP materials are combined to form composites and modal analysis is done using FFT analyser and Finite Element Method using ANSYS Software.Sonatural frequencies, amplitude of composite materials were found out through experimentation and fem software to avoid resonance condition and the results were compared.

II. LITERATURE REVIEW

K. K. Viswanathan*et al.*[1] explained the free vibration of layered circular cylindrical shell with cross-ply walls including shear deformation theory by using spline method. The choice of thismethod was due to the possibility that a chain of lower order approximation could yield greater accuracy than a global higher order approximation and successfully tested the spline collocation method over a two point boundary value problem with a cubic spline. Then the problem was solved for a frequency parameter using an eigen solution technique, to obtain as many frequencies as required starting from the least. The mode shapes were constructed from the spline coefficients and were computed from the eigenvectors.

Li Jun *et al.*[2]presented comparison of first-order shear deformation theory and the dynamic analysis of generally layered composite beams based on higher-order shear deformation theory. Formulated an exact dynamic stiffness matrix for a generally laminated beam based on third-order shear deformation theory which accounted for a parabolic variation of the transverse shear strain through beam thickness. An exact analytical method which is called the dynamic stiffness matrix method was used to determine the free vibration characteristics of generally layered composite beams. The numerical results were compared with the existing solutions whenever possible to demonstrate and validate the method.Walter Lacarbonara*et al.*[3]explained general higher-order approach development for the case of multi-layered laminates with arbitrary fiber orientation. Investigated lay-up geometries and span-to-thickness ratios where thickness-wise variable shear elongations and deformations become truly important. Exemplary results were obtained with the theory for reference cases and compared with 3D elasticity theory or other existing higher-order theories.

K. Kantha Rao *et al.*[4] has done design and analysis of lifting surface of aerospace with honeycomb core. Lifting surfaces were essentially designed to take up bending loads due to lift. Bending stresses were found to be maximum at the top and bottom surfaces and low stresses at the middle. Honeycomb panel construction suits the requirement where top and bottom skin takes the bending load. Analysis was carried out for the specimen level using three point bending test for understanding the bending behavior of honeycomb sandwich panels.

SharayuU.Ratnaparkhi *et al.*[5] introduced Glass/ Epoxy laminated composite plates which were manufactured and modal testing was done for free-free boundary condition by using Data Acquisition System. Frequency Response Functions are obtained by FFT. Quantitative results were presented to show the effects of different parameters like aspect ratio and fibre orientation. DOE data, FRF results and simulation results were compared. Free-Free boundary condition was carried out for analysing the effect of factors such as no. of layers, fiber orientation angle and aspect ratio on the natural frequency. D. Biswas and M C. Ray *et al.* [6] introduced geometrically nonlinear vibrations of rotating laminated composite beams using the vertically reinforced 1-3 PZC materials. The substrate beam was considered thin hence, Von-Karman type nonlinear strain–displacement relations and first order shear deformation theory (FSDT) were used to derive the coupled electromechanical nonlinear FE model. K Lavanya*et al.*[7] studied glass fibre composite fabrication in 0^0 orientation and basic properties were evaluated experimentally.Damping ratio was calculated using half-power band widthmethod from the FRF obtained in vibration test. Numericalanalysis was also carried out for the same specimens using finiteelement analysis. Comparison of the numerical results by the experimental results for 0^0 orientations were made to investigatethe accuracy of the finite element model. Further effect of fibreorientation and viscoelastic material on damping ratio wasstudied. Satish.N*et al.*[8] summarized finite element models on damping treatment. Different fiber system and also study of the influence of boundary conditions on the modal damping of laminated composites was carried out. Classical modal testing of bi-woven laminates under various boundary conditions and analysis of the damping was carried out. Results obtained from both laminates were compared and it was observed that damping properties of graphite is more dominant than glass laminates.

B.S.Ben*et al.*[9] introduced viscoelastic damping which appeared as the prioritizing mechanism in undamaged polymer composites vibrating at small amplitudes. An ultrasonic based hybrid method using combined frequency response and finite element was developed to measure the damping properties of composite material. Method employed combined finite element and frequency response for finding the damping characteristics of composite materials, which were used in high frequency applications.Tests were conducted with scan view plus software as virtual controller using ultrasonic pulse generator.

S. P. Parida, R. R. Dash *et al.*[10] studied the effect of variation of fiber orientations and width on natural frequency for different boundary conditions of the beam. Mathematical modeling for free vibration analysis of Timoshenko composite beam was proposed. The problem for the analysis was defined and validation of FEA analysis done. Effect of different fiber orientation and geometric ratio for different boundary condition on natural frequency were investigated in a detail. The effect of geometric ratio on the natural frequency was studied and presented. K. Senthil Kumar *et al.* [11] presented the study of fiber length and content on mechanical strength and dynamic characteristics of sisal fiber polyester composite (SFPC) and banana fiber polyester composite (BFPC) and its effects using impulse hammer technique. Correlation between their dynamic characteristics and mechanical properties was done with the help of interfacial observations based on microscopy analysis. Experimental investigation was carried out on free vibration characteristics of short sisal fiber (SFPC) and banana fiber (BFPC) polyester composites. Influence of fiber length and weight percentage on mechanical properties and free vibration characteristics were analyzed. Natural frequencies and associated modal damping value of the composite laminates were obtained by carrying out the experimental modal analysis. N. VijayaSai*et al.*[12] objective of the work was to determine the damping factor and mode shapes for a cantilevered rectangular plate of hybrid sisal-bagasse fabric reinforced epoxy composite with fiber orientation [+90°/+45°/0°/-45°/-90°] using a Fast Fourier Technique (FFT) based spectrum analyzer.P. Ganesa and S. Thirumavalavan*et al.* [13] experimental study was carried out to investigate the factors measured for Kevlar 49 fiber reinforced mechanical properties experimentally such as tensile strength and composite.Results showed that the tensile strength decreased measured in the experiments described were important with the increasing of glass fiber loadings which were for successful design of dynamically loaded components attributed to the absence of adhesion between made of such a material.

Tiangui Ye *et al.*[14]studied the first-order shear deformation shell theory based on the general shell equations which employed to include the effects of rotary inertias and shear deformation.The rotation and displacement of the open shells, regardless of boundary conditions was invariantly expressed as Chebyshev polynomials of first kind in both directions. Thereby, all the Chebyshev expanded coefficients were treated equally and independently as the generalized coordinates and solved directly by using the Rayleigh–Ritz procedure. Different combinations of classical boundary conditions (e.g. completely free, shear-diaphragm restrained, simply supported and clamped) and uniform elastic restraints as well as their combinations were considered in the investigation. Parametric studies were also undertaken, giving insight into the effects of elastic restraint parameter, fiber orientation, layer number, conical angle as well as subtended angle on the vibration frequencies of the composite open shells.

Masaki Kameyama and Masahiro Arai *et al.*[15]explained the effect of laminate configuration on the damping characteristics which was investigated for symmetrically laminated plates. To examine the effect of laminate configuration, the concept of specific damping capacity was also introduced and the damping characteristics were represented on the lamination parameter plane, instead of using composite fiber orientation angle. The laminate configurations for the symmetrically laminated plates with maximum damping subjected to the constraints on the natural frequencies were determined by using differential evaluation in which lamination parameters were used as design variables,instead of ply fiber orientation angles and ply thicknesses.

P. Aumjaud *et al.*[16] showed the location and orientation of DSLJ damper which is optimised using two different approaches on a honeycomb-cored sandwich plate;(i) a simple and quick parametric optimisation based on the strain distribution of the mode shape of each structure considered, and (ii) a complex and computationally demanding multi-objective evolutionary algorithm, namely the Adaptive Indicator-Based Evolutionary Algorithm (IBEA).The assessments of the efficacy of two different optimizationapproaches to this problem were parametric optimisation and the Adaptive Indicator-Based EvolutionaryAlgorithm (IBEA) which were also studied. Both approaches were used to increase the damping while minimising the additionalmass of the damping inserts applied to the structure.

Riccardo Vescovini*et al.*[17] discussed a procedure which can be used to evaluate the modal damping at panel level starting from the experimental characterization of the damping properties at ply level. Studied theory of beams or plates and the strain energy method to aircraft sub-component structures.The analysis procedure was developed for structures characterized by the regions with different stacking sequences, as well as lay-ups with plies of different materials and thicknesses.Illustrated the evaluation of the specific damping capacities of the structure and direct transient analyses to investigate the effect of damping on the panel response to pulse loadings.

P. Nagashankar*et al.*[18]studied the prediction of the dynamic characteristics of PPHC sandwich composites experimentally and theoretically results which have been compared. The paper has studied the effect of the different orientations of FRP skins and the different thicknesses of the skin and core on the transverse shear damping of the sandwich with the help of appropriate equations. The effect of different orientations of fiber and different thicknesses of the skins and polypropylene honeycomb core (PPHC) on the transverse shear damping of the sandwich using experimental and theoretical studies has been investigated.

P.Malekzadeh.Y.Heydarpour *et al.*[19]presented the three-dimensional static and free vibration analysis of multi-layered plates with FGCNTRC layers. The extended rule of mixture as a simple and convenient micromechanical model for prediction of the overall properties of the CNTRC materials was employed to estimate the effective material properties of the CNTRC layers. Types of CNTs distribution and volume fractions and also lamination scheme on the natural frequencies, stress components and displacement of the FG-CNTRClayered plates were investigated.

Cheng Shen*et al.*[20]presented the layerwise shear deformable theory which was applied to model the vibration of the laminate composite base plate and utilized this theory to model the vibration of arbitrary thin-walled composite beam stiffeners. On the basis of the theoretical model, numerical investigations were conducted specially focusing on the influence of stiffeners spacing and stacking geometry on the vibro-acoustic property of the structure.

M. Jaber*et al.*[21]studied the modal loss factor of sandwich structures comprised of varying laminas with respect to their mass as well as to the improvement of their noise, vibration, and harshness (NVH) performance. Multiple optimization algorithms were used to determine the most efficient laminate make-up for damping purposes, as well as the modal strain energy method to calculate strain energies stored and dissipated in the composite layers, and finally optimization algorithms to calculate the modal loss factor for the complete structure, which utilizes a dynamic response in terms of the undamped natural frequencies. A comparison between the effectiveness of the different optimization algorithms was also carried out.

DeveshPratap Singh Yadav*et al.*[22]showed free vibration analyses of plates withvarrying stiffeners location which were evaluated by finite element method. Using the first order shear deformation theory ,the effects of transverse shear deformation and rotary inertia were studied. The main advantage of stiffeners was to increase the bending strength of plate with minimum material requirement which made the better performance of plate. A numerical tool Finite element method which was used to solve the structural and solid mechanics problems which can handle the complex restraints and loading conditions. This paper had utilized the finite element method to find the natural frequency of stiffened plates for different boundary conditions and to study the effect of skew angle, stiffener to thickness ratio of plate on the natural frequency. The result showed that the frequency of stiffener 1 was higher than the other three cases of stiffenersconsidered.Even the observation was made that the non-dimensional frequencies increased by increasing the stiffener to thickness ratio of plate and skew angle.

The metal plates are used in services today in more and more extent in automobile structures and panels and aeronautical industries. Due to high load acting the structures vibrate with higher frequency leading to resonance condition. The plates of aluminium and mild steels fails in service due to vibration it ultimately results in generation of cracks and occurrence in residual stresses these effects to be eliminated by using composite plates and honeycomb structure.Thus,honeycomb plates exhibit good damping properties as compared to other composite plates.

The main objective of this present research work is to develop an organized and better composite structurefor car body panel and structure application and comparison to be done for better composite plate instead of metals plates in order that structures must safely work during its service life. The maximum amplitude of vibration must be in limit for the safety of structure. From the frequency response natural frequency and mode shapes were obtained. The first mode is called as fundamental mode in bending, second mode as twisting and rest of the modes as combination of both bending and twisting. The phases of the process plan for the present investigation are as follows: First Finite Element Analysis was performed to obtain the relative values of first, second and third modal natural frequencies. After FEA, Experimental Analysis was performed to obtain the relative values of first, second and third modal natural frequencies considering two boundary conditions. Finally a comparative study was made between Finite Element Method and Experimental method for the composite plates.The advances in damping evaluation of composite and conventional materials using FFT technique for frequency response carried out for different plate structures using suitable boundary conditions.

III. MATERIALS AND PROPERTIES

I) Aluminium Alloy (A5052 - H34/ISO AlMg2.5):

It has good workability, high fatigue strength, weldability, exhibits moderate strength and resistant to corrosion. As it has low density is probably used in many applications e.g. car body panel structures, aircraft and composite plate structures as it exhibits good stiffness and strength properties.(Ref. from 4).

Young's Modulus(MPa)=70000, Poisson's Ratio = 0.33, Density (kg/m^3) = 2700

II) Mild Steel (CR 1010): It is also known as plain carbon steel i.e low carbon steel which contains iron and carbon approx.-0.05-0.15%.It is the most commonly used mild and cold rolled steel. It has excellent welding properties and is suitable for grinding, punching, tapping, drilling and machining processes also used in car panels (Ref. from 22).Young's Modulus (MPa) = 210000, Poisson's Ratio = 0.3, Density (kg/m^3) = 7850

III)Fiber Reinforced Plastic (Glass epoxy fiber [1] and polypropylene (PP) [2]):

The glass fibre are fabricated in rectangular shape fabric mats which are oriented at 0^0 non-woven type and cured at room temperature of 24^0C. E-Glass fiber plates made by glass fibers 60% and resin 40% (Ref. from 7). PP used for variety of parts including panels and door trims was compounded with materials to form copolymer such EP copolymer was formed which ascts as impact resistance and has good rigidity.Young's Modulus (MPa) E_1 = 74000, Poisson's Ratio υ_1 = 0.25, Density (kg/m^3) ϱ_1 = 1566 .Young's Modulus (MPa) E_2 = 1500, Poisson's Ratio υ_2 = 0.26, Density (kg/m^3) ϱ_2 = 90

IV) Honeycomb Core:

It is a hexwebi.ehexagonal cell type structure a core made of PP thermoplastic with facing sheets on both sides of core made of aluminium alloy. It has a wide range of applications in automobile panel structures, aerospace, ship building, industrial lightweight construction etc. Therefore, can be used in cars for weight saving. Thus, sandwich panels made of these cores and laminated skins are light weight, high specific strength, impact strength, flexural damping and rigidity and acoustic damping properties .They were bonded together with adhesive epoxy resin which has good mechanical and damping properties as compared to others. Material nomenclature: PP1-5-N1-8, Cell size = 8mm, Thickness of core = 8 mm, Cell wall thickness = 1mm,Young's Modulus (MPa) = 1500, Poisson's Ratio = 0.39, Density (kg/m^3) = 80, Tensile Strength= 0.89 MPa (ASTM C297), Compression Strength = 1.89 MPa, Compression modulus = 79.2 MPa (ASTM C365), Shear Strength = 0.58 MPa, Shear Modulus = 15.2 MPa (ASTM C273)(Ref from 16)

V) Epoxy Resin LY556 and Hardner HY951:

The resin and hardener was mixed with weight ratio of 10:1 and used for bonding of composite plates(Ref. from 22). The mixer was applied on plates for proper bonding and with thickness of around 0.2mm. These have good fatigue resistance, tensile strength, stiffness and damping properties.Young's Modulus (MPa) = 4000, Poisson's Ratio = 0.4, Density (kg/m^3) = 1200. The constants were obtained by performing tensile test on specimens according to standards ASTM: D 638-08 and D 3039. Tension dominated structural components and a small saving in weight leads to fuel saving in aerospace and automobile structures.(Ref. from 5).Parameter which control impact properties of hybrid composite system are thickness of layers, bonding material used, impact velocity and energy.

IV. METHODOLOGY

Considering the ratio of number of aluminium to FRP to be (n+1)/n(Ref. from 22).the composite plates of aluminium ,mild steel,fiber reinforced plastic i.e Glass Epoxy Fiber and polypropylene, polypropylene honeycomb core as mentioned above in materials were formed. The standard specimen plates considered were of dimension (cover plates of Al and MS of 300x 150x 1mm thickness and middle layered plates of FRP's 8mm thickness) .In order to form three layered laminated sandwich structure plates of 10mm thickness using the ratio as mentioned above.Each plate was cleaned first and was applied with epoxy resin as given in materials and according to the ratio they were compiled to form composites. In order to obtain proper bonding of the composite materials they were applied with pressure of 17 MPa under hydraulic press and kept for 24 hrs (Ref. from 11).For the stiffened plate stiffeners of the required dimension were formed and placed along the longitudinal direction between the two plates. The stiffen plate was made of aluminium having good strength properties according to application(Ref. from 22).Thus,the honeycomb core of the required specification as mentioned in materials above were obtained and sandwiched between the two aluminium alloy plates as shown in figure no.1(a) because the middle core resists the transverse shear deformation and the outer layer acts as high strength facing sheets. As all the cores act as viscoelastic material even the honeycomb exhibits the same with good damping properties than other materials. Composite plate are formed in combination as shown in figure no.(1)and (1)a: Plate no.1 Al-FRP-Al, Plate no.2 MS-FRP-MS, Plate no.3 Al-PP-Al, Plate no.4 MS-PP-MS these are composite layered plates ,while Plate no.5 Stiffen plate Al-Al-Al and Plate no.6 honeycomb core sandwich plate Al-PP-Al.

The present study composite plates were formed of the given materials and thus effects of layers on vibrational behavior were examined. Tests were conducted experimentally to determine the influence of the above parameters and the obtained results were validated using finite element software ANSYS. In this work there was a consideration of different structures such as laminated and Viscoelastic honeycomb cored anisotropic plate structure and the vibrational (Modal) behavior were seen for the same. The methodology was divided in following stages as Stage 1: The composite structures consisting of aluminum, mild steel and FRP materials are to be constructed. Stage 2: The natural frequencies for the different structures were found out by using ANSYS software and FFT analyzer.The damping ratios were evaluated using half power band width method.

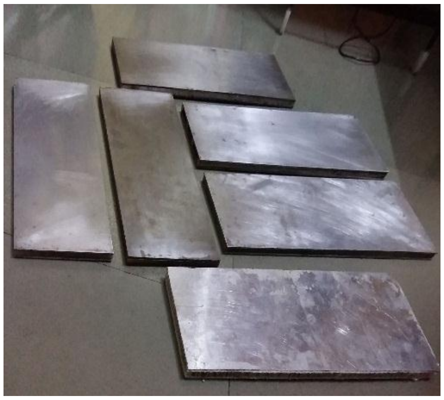

Figure No.1 All Composite Plates

FigureNo.1(a) Honeycomb composite plate

V. MODELLING AND SIMULATION

Plate models were obtained of dimensions i.ecover plates of 300 x 150 x 1 mm and middle layer plates of 8mm thickness in CATIA using part modelling and using assembly modelling 12mm thickness composite plates were formed and converted to the .igs format as shown in figureno.3and imported to ANSYS Workbench14 for the further modal analysis.Similalry for honeycomb core the heaxagon was made of cell size as mentioned in materials using sketch , part modelling and assembly modelling as shown in figure no.2 ,4 (a) and 5

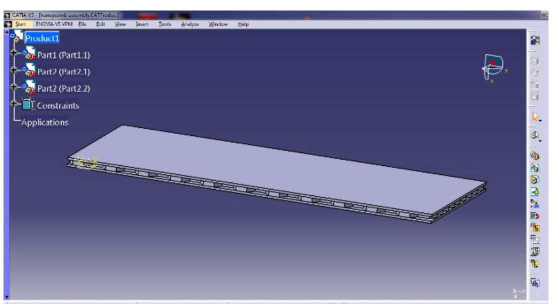

Figure No.2Honeycomb composite plate

There are three stages of activity:

1) Preprocessing:

Engineering data was given as input such as material properties Young’s Modulus,Poisson’s ratio and density as mentioned in section VI above for the given materials.Thencomposite plate geometry was imported and generated as shown in figure no.3

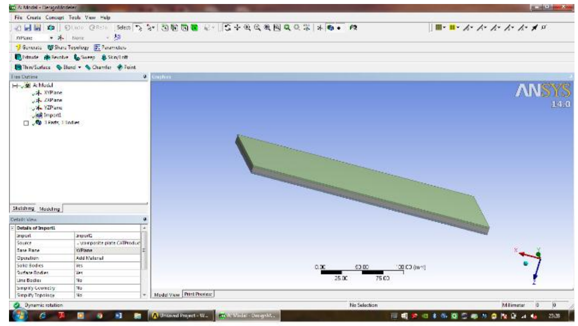

Figure No.3Model of Composite plate

2) Processing :

In this process ,the model was meshed with the elements 162 and nodes 1272. Meshing was done to obtain proper results at the required nodes thus, tetrahedral and Hex 20 mesh was used for accurate results as shown in figure no.4.Then Modal analysis was done by applying the two boundary conditons cantilever i.e one short end fix thus constraining all degrees of freedom x=0 and other end free and simply supported condition i.e remote displacement at one end of the plate and other end fix.Thus, the total deformation were obtained.

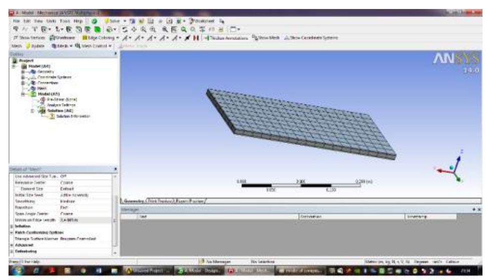

Figure No.4 Meshing of Composite plate

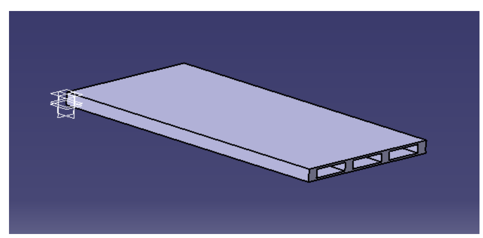

Figure No.4(a) Model of Stiffen plate

3) Post-processing:

In this process the frequencies were obtained for the given three modes as input and mode shapes of deformation.

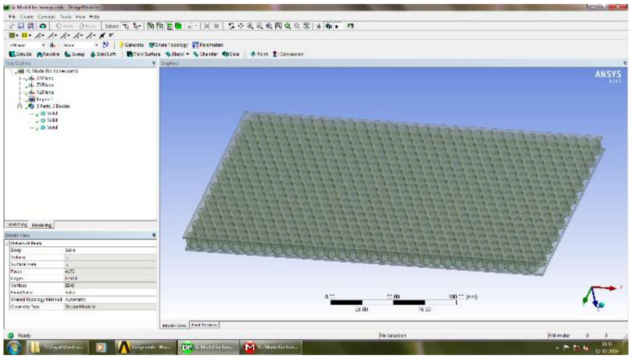

Figure No.5 Honeycomb Core

Ansys results for the three mode shapes as shown in table no.1 were obtained for the given cantilever conditions and we observed that the modal frequency for plate no.6 i.e honeycomb plate as shown in figure no.6 were less as compared to other plates.Similarly results were obtained for simply supported condition where frequencies amplitude values were less as compared to other composite plates.The variation in frequency and damping ratio was observed for 0^0 orientation of composite plates and cell size with given wall thickness of honeycomb .

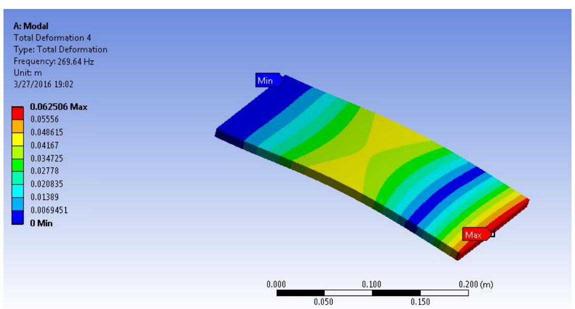

Plate no.1 (269 Hz)

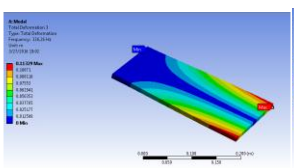
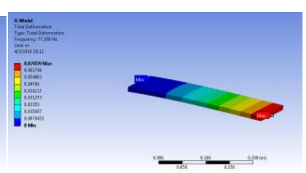

Plate no.1 (326 Hz)Plate no.2 (77.6 Hz)

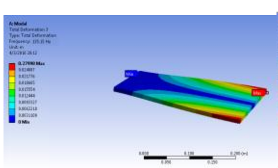
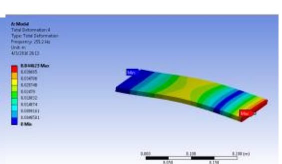

Plate no.2 (135 Hz and 255 Hz)

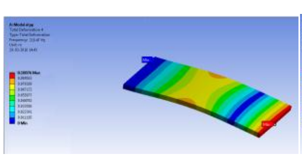
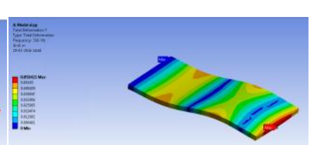

Plate no.3 (218 and 318 Hz)

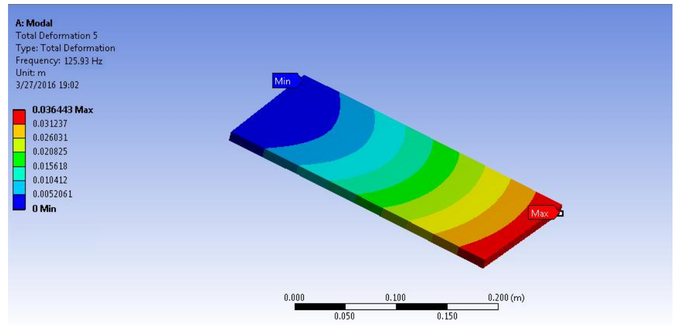

Plate no.1 (125 Hz)

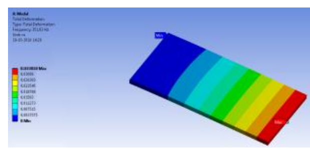
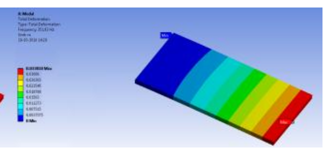

Plate no.3 (351 Hz)Plate no.4 (65 Hz)

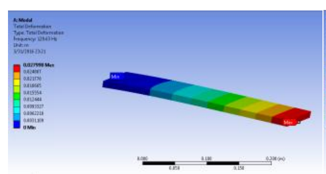
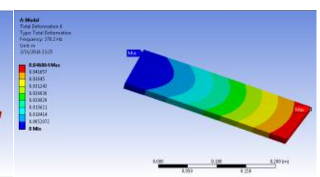

Plate no.4 (129 and 270 Hz)

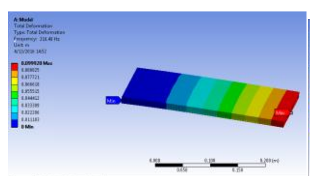
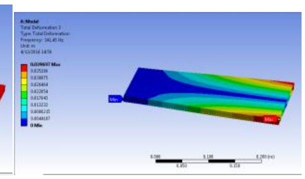

Plate no.5 (216 and 341 Hz)

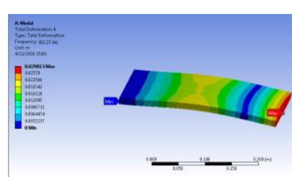
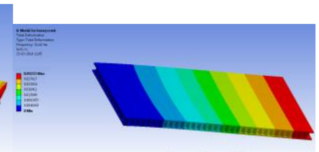

Plate no.5 (432 Hz)Plate no.6 (52.6 Hz)

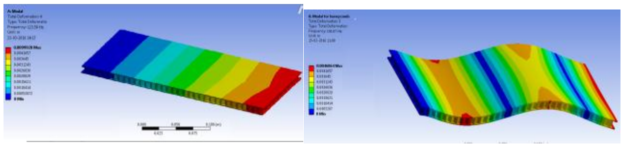

Plate no.6 (123.5 Hz and 141 Hz)

Figure No.6 Results of ANSYS Mode Shapes with frequency values

VI. EXPERIMENTAL TEST PROCEDURE

In order to carry out the experimentation the fixture was made for holding of plate for both boundary conditions with fix lever at one end and roller atthe other as shown in figure no.7.

Figure No.7Fixture for holding of plates

Different specimens of composite plates of dimension (300 x 150 x 10 mm) were marked with grid points of 25 x 25 mm array in order to obtain results at various points on plates.Firstly for cantilever condition and measurement purpose the plate was fix at one end as shown in figure no.8and was applied with constant force at free end of plate with piezoelectric impact hammer and for simply supported condtionforce was applied at the middle as it was observed that maximum deflection occurs at these pointsas shownin figure no.9.

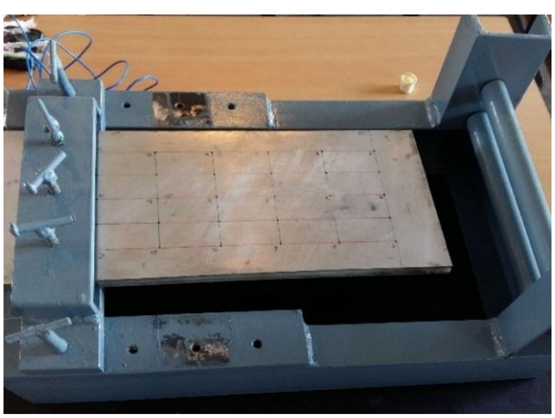

Figure No.8Cantilever condition of plate

A two channel SKF Microlog analyzer GX series FFT analyzer was usedwith Bump testprocedure to compute frequency domain response function (FRF's)i.e the measured data was transformed from the time domain to the frequency domain which was displayed in terms of graphs as acceleration vs frequency and the whole arrangement as shown in figure no.10.An impact modal hammer with a load cell attached to its head to measure the input force of the structure and the test plate was tapped to induce the required vibration and it acts as input channel(i.e channel 1 CH1)as shown in figure no.12 and the response was measured due to excitation with an accelerometer (coiled cable CMAC 5209)which was attached to one corner of plate and acts as output channel 2 (CH2) as shown in figure no.11

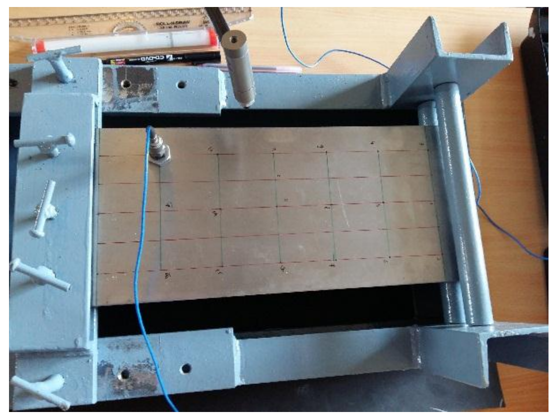

Figure No.9Simply Supported Condition of plate

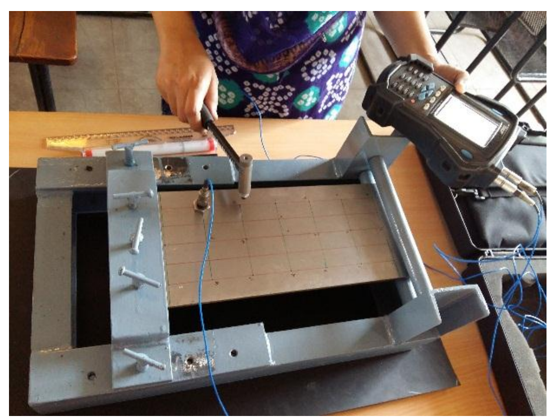

Figure No.10Experimental setup using FFT analyser

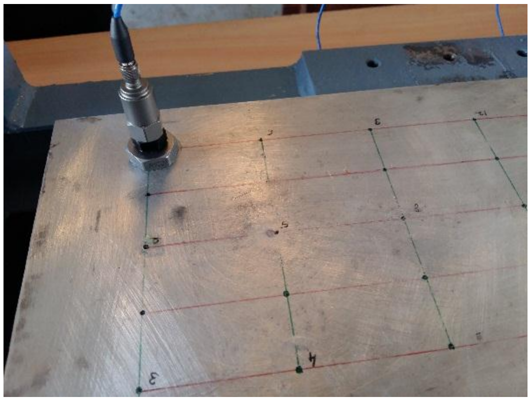

Figure No.11Accelerometer

Figure No.12 Impact modal hammer

Accelerometer was mountedwith SKF adhesive atvarious points on plate with wax for aluminium composite plates and for others with magnetic action attached as shown in figureno.11.Specifications of FFT:Testtype=Bump test,Type= linear,resolution lines=400,Frequency range for amplitude(g) and impact hammer=1000Hz,averaging type=Peak hold, filter = 2Hz.Using a 2-channel FFT analyzer, FRF's were computed one at a time, between each impact DOF and the fixed response DOF. FRF's were used to extract such modal parameters as natural frequency and mode shape. For good experimental results and to avoid vibration during measuremantsthe fixture was placed on Neoprene rubber sheet of 10 mm thickness which has good fatigue and damping properties. The force input sensitivity of impact hammer is 2.25 mV/N and the response has been measured through the accelerometer with sensitivity of 100mV/g.From the response natural frequency and mode shapes were obtained with required maximum amplitude values.The first three modes were consideredi.efirst mode is called as fundamental mode in bending, second mode as twisting and rest of the modes as combination of both bending and twisting degree of movement(Ref. from 12).The simulation carried out using ANSYS Software and the FRF results were compared.

Energy dissipation depends upon viscoelastic nature of materials so damping ratio of composite plates were evaluated using half power band width method through FRF curves obtained from FFT analyser.So the values were calculated based on equation (1):

$\zeta = \Delta\omega/2\omega_n$(1)

whereζ –damping ratio, $\Delta\omega$ -bandwidth and ω_n -natural frequency

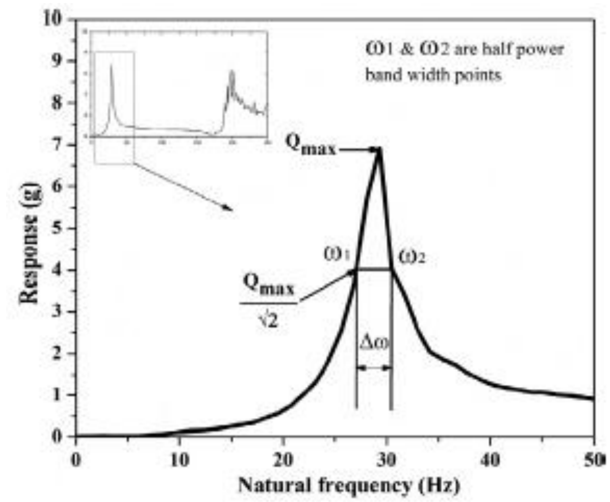

Figure no.13 Frequency response by half power band width method

From the above figure no.13 the half power points i.e ω_1 and $\omega_2$were obtained by the intersection lines from the Frequency response function (FRF) curves.The maximum amplitude obtained from resonant peak of the curve was Q_{max},$\Delta\omega$ was difference between frequency valueswhich were cut by horizontal line taking $Q_{max}/\sqrt{2}$ below the amplitude values.These half power points were used to determine the damping ratios $\Delta\omega = \omega_1 - \omega_2$.(Ref. from 11).

VII. Results and Discussions

Frequency response domain function(FRF)i.e.amplitude(acceleration(g)) versus frequency(Hz) graphs were plotted using SKF analyst software. FRF response curves were obtained when FFT was connected to PC and the details were extracted according to the required frequency range with the maximum amplitude indication for the given frequency range of 1000Hz as shown in figure no.14 and 14(a)and for resonance frequency of 10kHz. Thus, the maximum amplitude i.e acceleration and natural frequency of honeycomb sandwich for two boundary conditions and three modes were observed to be less as compared to other composite plates has greater damping and high stiffness as shown in figure no.15.Weobserved that deflection of composite plates was maximum at free end for

Plat-es	No. of Modes	ANSYS Frequen-cies (Hz)	Experime-ntalFrequenci-es (Hz)	Amplitu-de By FFT (g)	Amplitu-de By ANSYS (g)
1	Mode 1	125.93	128	0.0201	0.03
	Mode 2	269.64	265	0.0407	0.06
	Mode 3	326.26	330	0.123	0.113
2	Mode 1	85	77.5	0.093	0.07
	Mode 2	135	130	0.152	0.27
	Mode 3	255.2	258	0.0244	0.0446
3	Mode 1	218	225	0.178	0.1
	Mode 2	318	315	0.0356	0.058
	Mode 3	351	348	0.026	0.03
4	Mode 1	65	72	0.04	0.05
	Mode 2	129	128	0.0201	0.0279
	Mode 3	270	275	0.0507	0.046
5	Mode 1	216	220	0.102	0.09
	Mode 2	341	335	0.022	0.039
	Mode 3	432	420	0.19	0.29
6	Mode 1	52	50	0.011	0.032
	Mode 2	123	120	0.009	0.009
	Mode 3	141	145	0.00363	0.004

the cantilever condition and at the middle of plates for the simply supported condition.

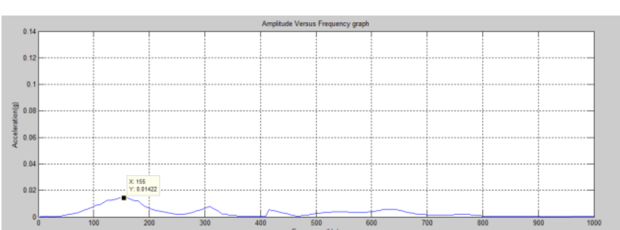

Figure No.14FRF graph of honeycomb core composite plate for damping ratio

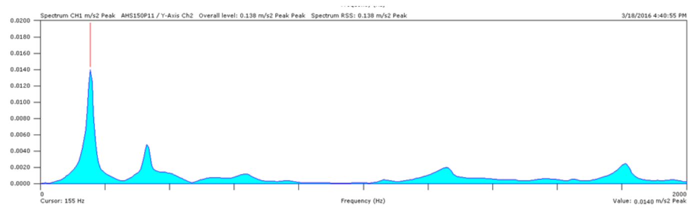

Figure No.14 (a) FRF graph of honeycomb core composite plate

The natural frequency obtained were used to determine damping ratio using half power band width method. Thus, from the half power band width method the damping ratio was observed to be increased in honeycomb plate as compared to other composite plates. Damping ratios calculated for cantilever condition were obtained to be 0.019, 0.015, 0.01, 0.024, 0.015, 0.03.

Table no.2 Results of cantilever condition of plates

Table no.3 Results of simply supported condition of plates

Plat -es	No. of Modes	ANSYS Frequenci -es (Hz)	Experimen - talFrequen cie-s (Hz)	Amplit- ude Acceler -ation (g)	Amplitu- de By ANSYS (g)
1	Mode 1	138.5	135	0.152	0.25
	Mode 2	304	320	0.075	0.09
	Mode 3	432	400	0.018	0.021
2	Mode 1	86	80	0.04	0.06
	Mode 2	195	185	0.06	0.075
	Mode 3	284	280	0.275	0.23
3	Mode 1	269	265	0.0179	0.015
	Mode 2	356	355	0.0108	0.03
	Mode 3	438	433	0.0335	0.06
4	Mode 1	88	85	0.031	0.021
	Mode 2	255	260	0.030	0.05
	Mode 3	304	305	0.0338	0.045
5	Mode 1	395	390	0.0186	0.014
	Mode 2	456	450	0.0363	0.051
	Mode 3	827	823	0.0157	0.012
6	Mode 1	70	76	0.007	0.004
	Mode 2	123.5	125	0.0014	0.0016
	Mode 3	152.1	155	0.014	0.012

From the result table no.2 and 3 we observed that the frequency and amplitude values of honeycomb plate which were obtained from FFT graphs were increased in simply supported condition than comparison with cantilever condition but in plate no.6 values had decreased than other plates.

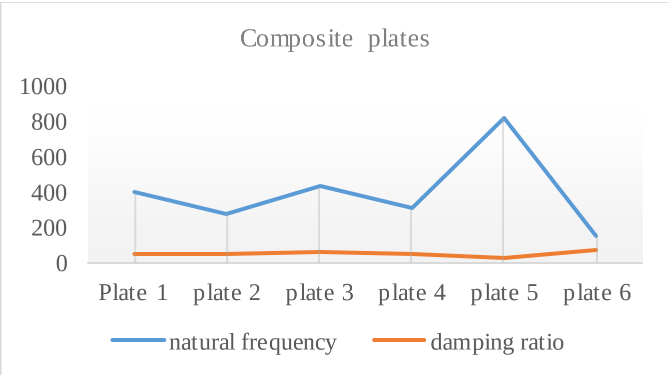

Figure No.15 Comparison of results

Thus,for different materials middle layer which resists transverse shear deformation and outer layer for load bearing strength we observed that honeycomb structure has greater resistance to transverse shear deformation which leads in good damping effect.

VIII. Conclusions

We observed that with the increase in constrains the frequency values also increased and the cantilever boundary condition has less frequency value as compared to simply supported. Even the honeycomb core of greater cell size acts as a viscoelastic layer having less stiffness has low amplitude and low natural frequency as compared to other composite structured plate. Honeycomb structure had good impact resistance i.e good damping as compared to other composite structure load bearing capacity.Thestrength, reliability and energy absorption capacity has increased by the use of honeycomb core in the applications of automotive, aerospace panels and structures.Using the half power band width method the damping ratio of honeycomb was increased as compared to other composites as shown in figure no.15.The results such as frequency and damping ratio wereinfluenced due to 0^0 orientation of composite plates and even for cell size with the given wall thickness of honeycomb.

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