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Investigation of dynamic properties of sandwich beam using oberst beam method

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Abstract
The Oberst beam method is widely using to measure the mechanical and dynamic properties of materials, such as the Young's modulus, it has been supplying required data for numerical analysis of structures and this method was been used with a composite materials, that has been consist of two skin layers (steel, aluminum) between them the core layer. This paper provides a brief description of the Oberst beam method by applied modal analysis and ASTM E756-05, the impact hammer is used to generate excitation for the beam and obtain a studied response. The boundary condition (Fixed-Free) was applied, it uses three different lengths beams. It was applied this technique in this current work for the two parts. The first part include a composite beam that consists of two skins (Galvanized steel) between them a layer of Polyurethane (PU) Rigid foam core. The second part skin was separated from the core of a composite material beam in (470) mm length beam. It has been obtaining the Young's Modulus for both core and skins over a frequency range of (30-1000) Hz. And compared results between (Experimental and Theoretical) by using finite element method (FEM). Found in low frequencies, the value of the Young's Modulus almost constant, but in high frequencies the value of the Young's Modulus were decreasing with increasing frequency. While found the value Young's Modulus remain almost constant, When the skin layer is separated from the core.

Keywords: Cantilever beam method; Polyurethane (PU) rigid foam; Sandwich panel.

1. Introduction
Weight reduction is one of the main design drivers of modern engineering and transport structures for aerospace and automotive applications. In this context, the sandwich design principle is playing a major role, as it allows for much higher weight-specific bending stiffness compared to a monolithic structure. A sandwich structure typically consists of two thin and stiff skins, separated by a lightweight cellular core. The main purpose of the core is to increase the bending stiffness by separating the skins, to carry transverse shear loads and to withstand compressive loads normal to the sandwich surface. In case of transversal impact loads, the core has to support the skins from local bending and it has to prevent major damage or penetration by a high energy absorption capability Serge Abrate et al [1]. The viscous materials have widely using for damping, and for minimizing resonant vibrations for this materials in a lot of applications. There are many techniques have been using to get the mechanical and dynamic properties such as, Oberst beam method.

Thibault Wassereau et.al [2], addressed the problem of estimating the local viscoelastic parameters of sandwich beams. An original procedure involving an inverse vibratory method (Force Analysis Technique) and the Timoshenko beam theory is detailed and applied experimentally on a sample presenting a honeycomb core. The major philosophy relies in considering multi-layer beams as equivalent homogeneous structures. This simplified approach is thought to be more representative of the global dynamic behaviour, in addition the reduction of degrees of freedom is obviously an improvement for modeling on Finite Element.

H. Abramovich et.al [3], investigated The damping characteristics of composite laminates made of Hexply 8552 AGP 280-5H (fabric), used for structural elements in aeronautical vehicles in depth using the hysteresis loop method and compared to the results for aluminum specimens (2024 T351).It was found that the loss factor, η, obtained by the hysteresis loop method is linearly dependent only on the applied excitation frequency and is independent of the preloading and the stress amplitudes. For the test specimens used in the this tests, it was found that the damping of the aluminum specimens is higher than the composite ones for longitudinal direction damping, while for bending vibrations the laminates exhibited higher damping values.

Gian Luca Ghiringhelli and MauroTerraneo [4], developed a mixed predictive/experimental methodology is developed to determine the frequency behaviour of the complex modulus of such materials. The loss factor of hybrid sandwich specimens, composed of two aluminium layers separated by the damping material, is determined by experimental modal identification. Finite element models and a reversed application of the modal strain energy technique are then used to recover the searched storage modulus and loss factor curves of rubber. The reliability of the results obtained have been verified through an extensive application of a prediction method, based again on the modal strain energy method but also on alternate implementation, to a set of specimens with very different properties. The laminate loss factor was estimated accurately regardless of the amount of rubber in a single viscoelastic inclusion as well as in cases with multiple viscoelastic inclusions.

Previous researchers were shown important work in studying composite material. In particular the study of the sandwich panel consisting of two layers from the skins between them the core material. The surface of the skin has a small thickness compared to the thickness of the core layer. These sandwich panel are lightweight compared to their strength. This makes it is easy to vibrate. Some researchers were addressed the conditional analytical design of the composite material to develop performance and they were used experimental and numerical methods to study the dynamic properties of sandwich panel (for natural frequencies and damping ratio, mode), also they were used the Finite Element Method (FEM) to obtain the mechanical properties. The current study is consider complementary to previous study. Where we are studying the experimental and numerical methods to get the dynamic properties of the sandwich beam, which it is consists of two layers of skin from steel and between them polyurethane foam, to study the (Young's Modulus, natural frequencies).

A sandwich beam were used in this study consist of (Galvanized Steel) thickness 0.5 mm for a skin, and 50 mm thickness for the (Polyurethane foam material) for the core, width the beam was 50 mm and three different free lengths beam (470, 235, 118) mm. The boundary condition (Fixed -Free) are used in this paper and the Oberst beam method. And it was depended in (ASTM E756-05) [5]. It is cannot applied the non-contact transducers to excite the beam in this work, due were not readily available therefor the hammer was used to generate excitation for the beam and obtain a studied response [6] and have been compared the results between (Experimental and Theoretical) by using finite element method (FEM).

2. Theory

2.1 Young's modulus for cantilever beam

Studies showed that the Young Modulus was constant over the low frequency of a sandwich panel consisting of two layers of skin and one layer of core, but with increasing frequency, a reduction in the value of the Young's Modulus was observed, because the Young Modulus for the core less from Young Modulus of skin therefore happened shear in core [7]. According to the Bernoulli-Euler beam theory. Natural frequencies of a homogenous beam in bending vibrations are given by:

$$\omega_n = (\beta_n L)^2 \sqrt{\frac{EI}{mL^4}} \tag{1}$$

where L is the free length of the sandwich beam, E is the Young's Modulus of the materials, I is the mass moment of inertia of the beam, m is the mass per unit length and βn L are the constant values which are given for the first five modes in Table 1. By replacing the appropriate expressions for ω_n, m and I into Equation (1), it is obtain:

$$E = \frac{12\rho L^4 {f_n}^2}{H^2 {C_n}^2} \tag{2}$$

where ρ is the density of the material, H is the thickness of the beam, f_n is the natural frequency [Hz] and C_n are the constants that ability be calculated from:

$$C_n = \frac{(\beta_n L)^{\,2}}{2\pi} \tag{3}$$

Table 1. Constants value for first five mode.

Mode No (n)	$\mathbf{LB_n}$	$\mathbf{C_n}$
1	1.875	0.560
2	4.694	3.507
3	7.855	9.820
4	10.996	19.242
5	14.137	31.809

2.2 Finite element analysis

The structural dynamic properties is estimated by using FEM. The finite element method FEM is used to study the response of the structure under action of impulsive loading and various length for the beams to study the response of the structure. Solid element is suitable for sandwich beam system, it was defined as an object which, for the purpose of frequency analysis. Most sandwich beam system are made of a two skins layers between them one layer for core.

2.2.1 Geometrical and material properties of sandwich beam

In the current work, three types of sandwiches beams were modeled depending on the length of sandwich beam, it was composed of two layers of the skins and one layer from core. The skins layer (Preprinted Galvanized steel sheet), they have same parameters physical and engineering properties. Each layer for the skin have thickness (0.5) mm, density 2800 kg/m^3, Young modulus of elasticity 200 GPa, Poisson ratio (0.3). The core material for a sandwich beam are made from Polyurethane PU foam and for all the models. Thickness of core layer (50 mm and 100 mm), density ± 40 kg/m^3, Young Modulus of elasticity (0.31) MPa, Poisson ratio (0.3).

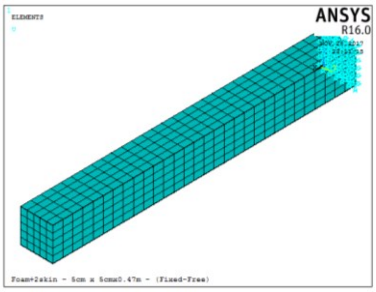
In the current study, three types of sandwiches beams were testing depending on the length of sandwich beam, different free lengths the beam (470, 235, 118) mm, it was composed of two layers of the skins and one layer from core. The material properties and dimension of the composite material beams used it shown in Table 2.

Table 2. Material properties wall panel composite sandwich material.

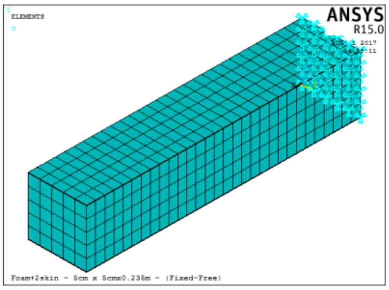
Material	Thickness (mm)	Width (mm)	Density (kg/m^3)	Constant Young's modulus
Skin (1&2) Preprinted galvanized steel sheet.	0.5	50	2800	200 GPa
Polyurethane (PU) rigid foam.	50	50	40(∓2)	0.31 MPa

2.2.2 Boundary condition

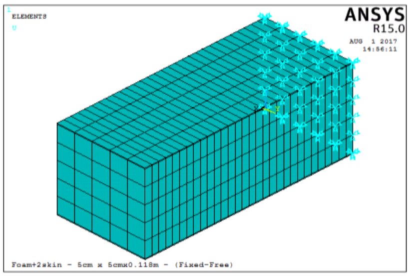
Initial conditions (the condition at time = 0) default to zero, boundary conditions of all models (Fixed – Free) for all three beams (DOF sets to zero), and free for the all other sides. The boundary conditions are shown in Figures 1 and 2.



Boundary Condition's Sandwich Beam L= (470) mm

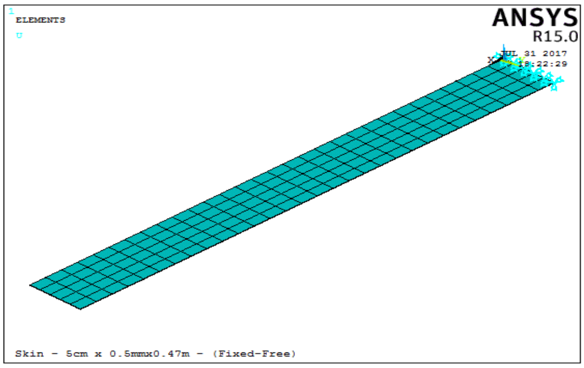


Boundary Condition's Sandwich Beam L= (235) mm

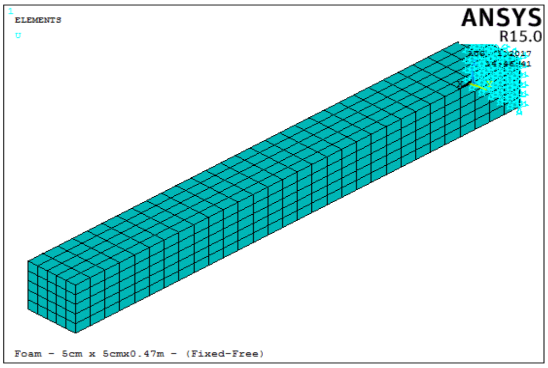


Boundary Condition's Sandwich Beam L= (118) mm

Figure 1. Boundary conditions (fixed-free) of all models cantilever sandwich beam of current work.



Boundary Condition Skin Beam L= (470) mm



Boundary Condition Foam Beam L= (470) mm

Figure 2. Boundary conditions (fixed-free) of all models cantilever (skin and core) beam of current work.

2.2.3 Element type

The element that used to build the sandwich beam models was SOLID185 Element. Figure 3 shows the geometry, node locations, and the element coordinate system for this element ANSYS, Release (15.0).

2.2.4 Mesh convergence

The sandwich plate models are solved using modal analysis to find the first five natural frequencies and mode shapes. Figures 4 and 5, shows the mesh (FEM) models of cantilever beams.

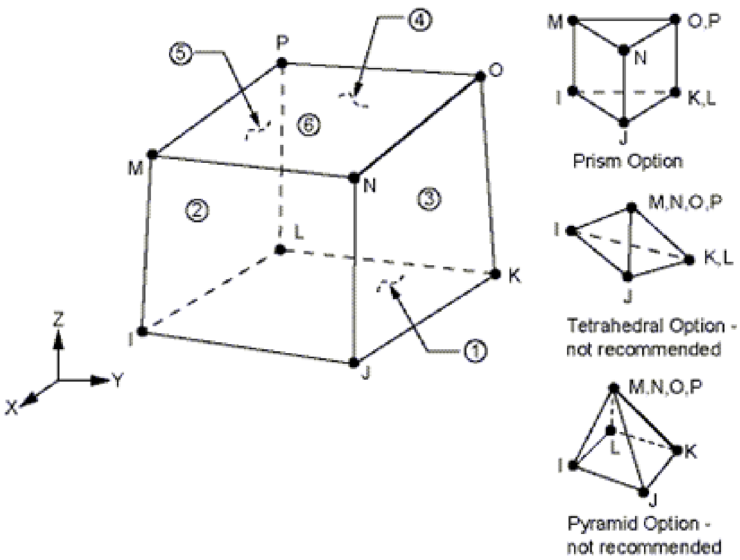
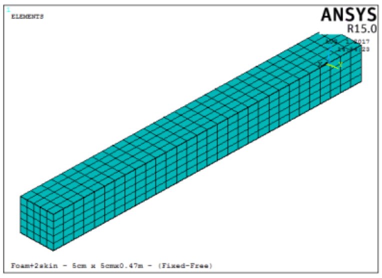
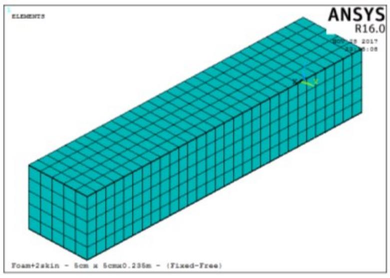


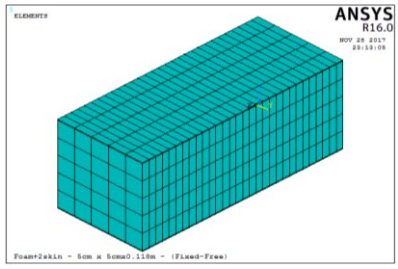
Figure 3. Solid 185.



Sandwich Beam L= (470) mm

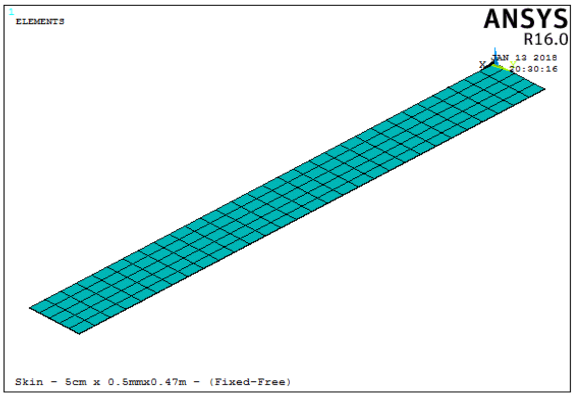


Sandwich Beam L= (235) mm

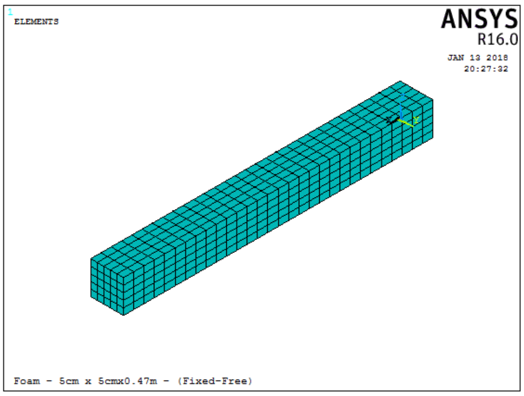


Sandwich Beam L= (118) mm

Figure 4. Finite element models (FEM) of sandwiches beams.



Skin Beam L= (470) mm



Foam Beam L= (470) mm

Figure 5. Finite Element Models (FEM) of (Skin and Foam) Beams.

3. Experimental work

The signal obtained is displayed by oscilloscope advise then it is convert into FFT function by (Sig-View) program. The purpose of this conversion is to obtain the natural frequency of the intact beam (Fixed-Free) boundary condition, for calculated Young's Modulus. The signal obtained in a moving disk is saved and data is then opened through the (Excel) program. Through the (Excel**)** program, the response to the signal is determined by sending the response file to the (Sig-View) program. Through the latter program, the signal is converted by the use of spectral analysis FFT from t domain to ω domain in order to obtain the natural frequencies of the samples used in the vibration test as shown in Figure 6. The accelerometer used in this test was placed along the middle path of the beam and was traveling at five different locations along the path and for three different free lengths the beam (470, 235, 118) mm. The excitation point was at the end of the free edge of the beam. Selection of the first five mode to calculate the Young's Modulus according to the Equations (2) as shown in Figures 7 and 8.

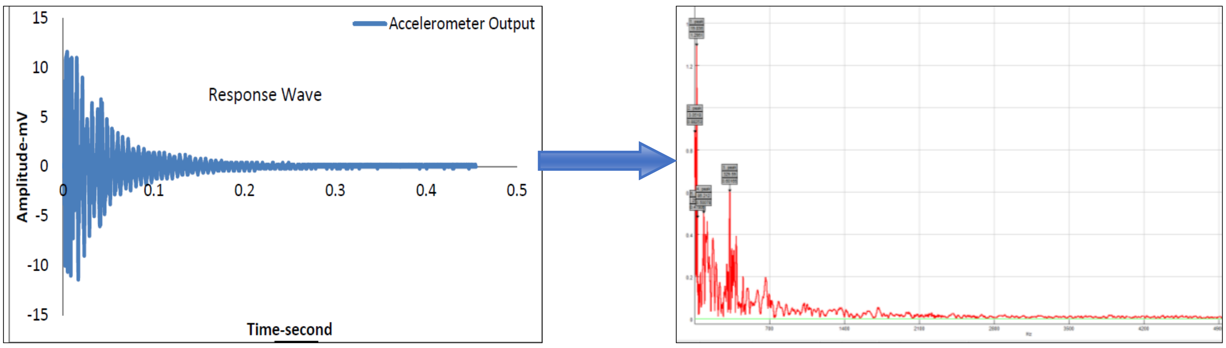
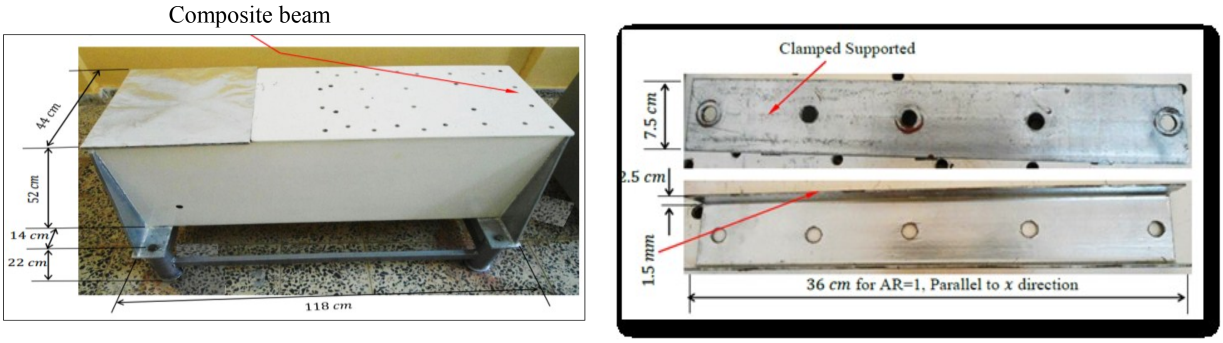


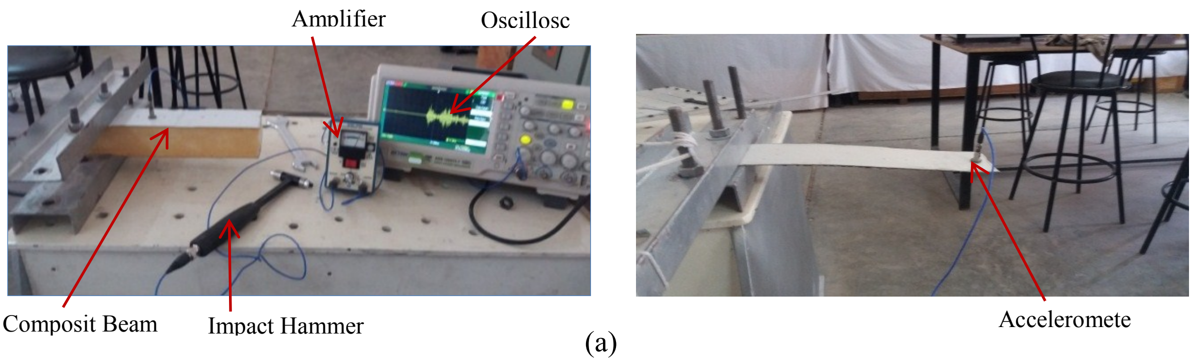
Figure 6. (Sig-View) Program and FFT Analysis.

The rig structure used in the first part of the vibration test consists of a (10 mm) thick steel plats while the dimensions other are shown in Figure 7. It was used as a clamped support on which the beam sample was fixed, it appear location of accelerometer in test vibration and design of the vibration test as shown in Figure 8. The hammer was used to generate excitation for the beam and obtain a studied response. The mass hammer consists of (0.1 kg) of head with a diameter of (1.57 cm), the tip of diameter is (0.63 cm), the length of it was (21.6 cm).

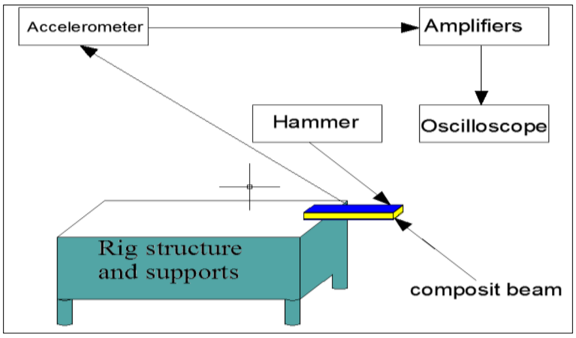


(a) (b)

Figure 7. (a) Rig structure, (b) Clamped supported.



(a)



(b)

Figure 8. (a) Location of accelerometer in test vibration, (b) Design of the vibration test.

4. Results and discussions

4.1 Young's modulus for sandwich beam depended frequency

Through experimental work was observed that some mechanical properties cannot depend on the physical properties, but in fact that these properties may depend on the frequency. Among these characteristics is the Young's Modulus and the literary research that proved this phenomenon A.C. Nilsson [8]. Fifteen measurements for test were obtained using three different lengths for beams of composite materials instead of one beam as mentioned above over the range frequency of (100-3000) Hz. The following table demonstrates (Experimental and Theoretical) results for the test condition (Fixed-Free) and for three different lengths. It shows in Table 3. The comparison between (Experimental and Theoretical) Young's Modulus result, natural frequency and error percentage. For (Fixed-Free) boundary condition. Figures 9-11 shown the compared between (Theoretical and Experimental) results for defriends lengths sandwiches beams.

In the range of frequencies that are (<300 Hz), it was observed that the lateral movement of the composite material depends on the pure bending, as shown in Figures 9-11. While in the mid-high frequency that are more than (>300Hz) it is clearly stated that the Young's Modulus changes with frequency, where it decreases by increasing frequency. These results are consistent with A.C. Nilsson's study of the sandwich [8]. The FEM model as shown in Figures 12-14. The comparison of the results (Theoretical and Experimental) showed a good.

Table 3. Comparison between (experimental and theoretical) Young's modulus result, natural frequency and error percentage for (fixed-free) boundary condition.

L= 470 mm						
Mode	Frequency (Hz)			Young's Modulus(MPa)		
	Experimental	Theoretical	Error%	Experimental	Theoretical	Error%
1	112	106	5.357	2.27E+11	2.45E+11	7.930
2	195	166.6	14.564	1.60E+11	1.86E+11	16.25
3	348	368.8	5.977	7.50E+10	8.50E+10	13.333
4	770	764	0.779	1.48E+10	1.68E+10	13.513
5	915	957	4.590	1.20E+10	1.18E+10	1.667
L = 235 mm						
Mode	Frequency (Hz)			Young's Modulus(MPa)		
	Experimental	Theoretical	Error%	Experimental	Theoretical	Error%
1	246.6	222.17	9.907	3.12E+11	2.85E+11	8.653
2	647	635	1.855	6.80E+10	5.96E+10	12.353
3	845	750	11.2 42	3.22E+10	3.78E+10	17.391
4	1401.4	1525	8.843	8.52E+09	7.18E+09	15.727
5	2240	1876	16.25	3.13E+09	2.83E+09	9.584
L = 118 mm						
Mode	Frequency(Hz)			Young's Modulus(MPa)		
	Experimental	Theoretical	Error%	Experimental	Theoretical	Error%
1	440.6	445.3	1.067	6.11E+10	7.10E+10	16.203
2	1435.2	1479.5	3.087	5.90E+09	5.20E+09	11.864
3	1744.4	1654.4	5.159	3193719995	3273615696	2.502
4	2099	2188	4.240	1171750682	1090816777	6.907
5	2316.1	2315.9	8.635e-3	791270248	911112001.3	15.146

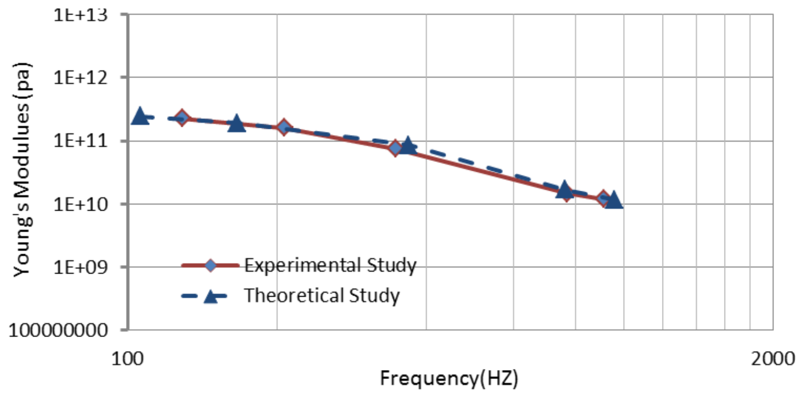


Figure 9. Comparison Young's modulus of study the (theoretical and experimental) of the sandwich beam, l = (470) mm.

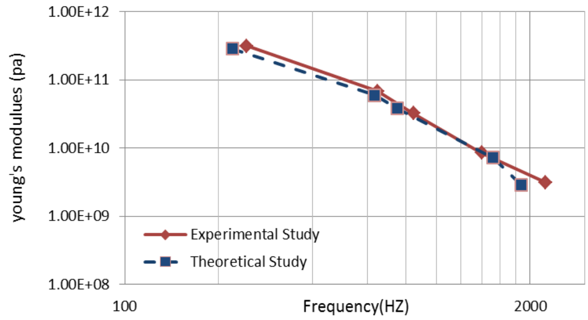


Figure 10. Comparison Young's modulus of study the (theoretical and experimental) of the sandwich beam L = (235) mm.

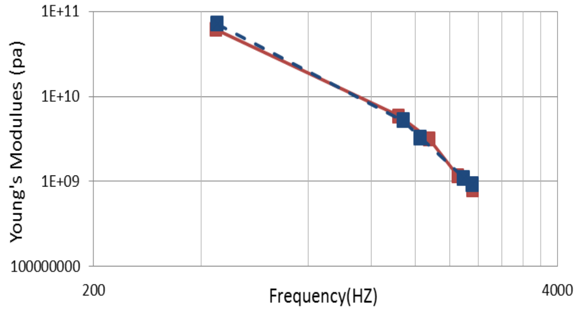
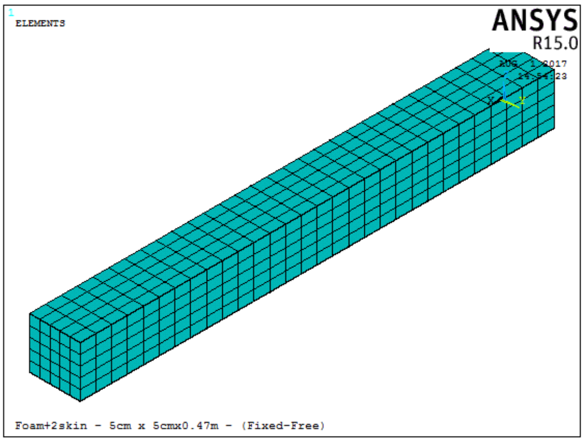
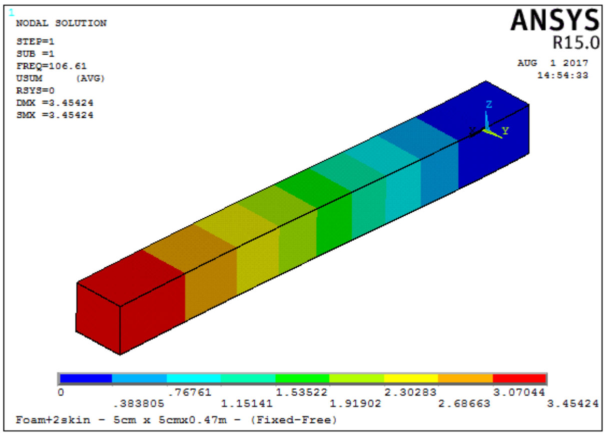


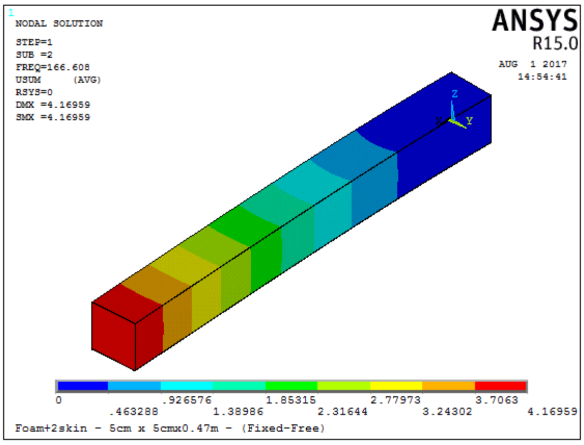
Figure 11. Comparison Young's modulus of study the (theoretical and experimental) of the sandwich beam L= (118) mm.



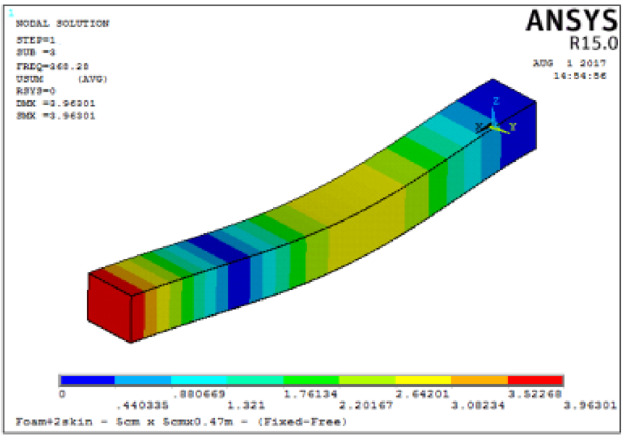
ELEMENT



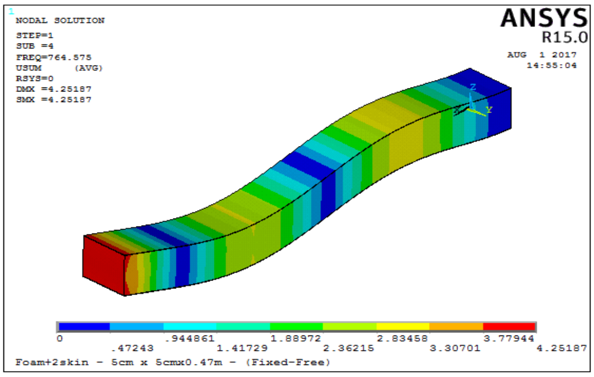
Fn=106 Hz



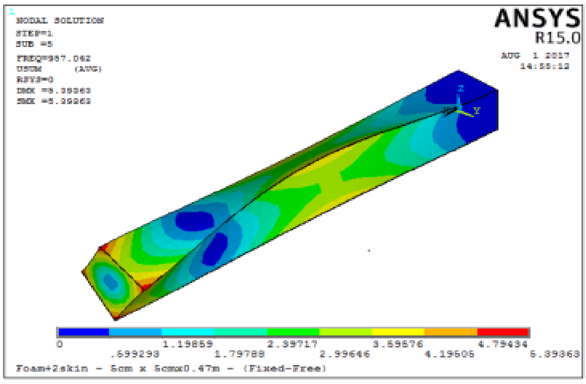
Fn =166.6 Hz



Fn=368.8 Hz

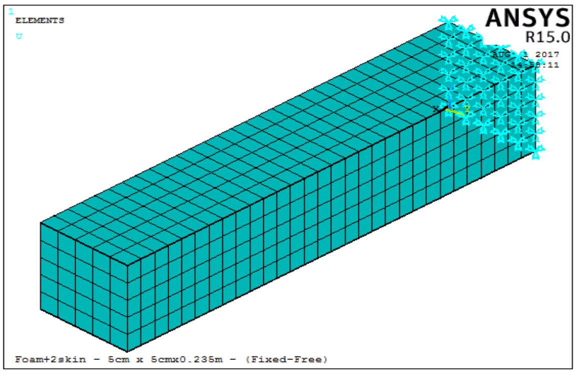


Fn=764 Hz

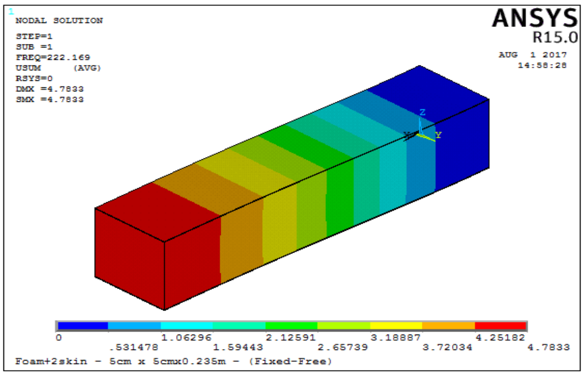


Fn=957 Hz

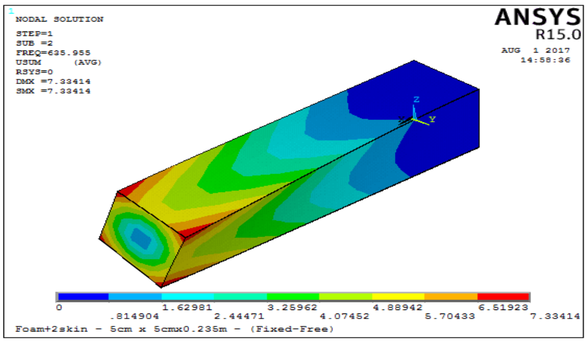
Figure 12. FEM model (fixed-free) condition for beam composite material, L= (470) mm.



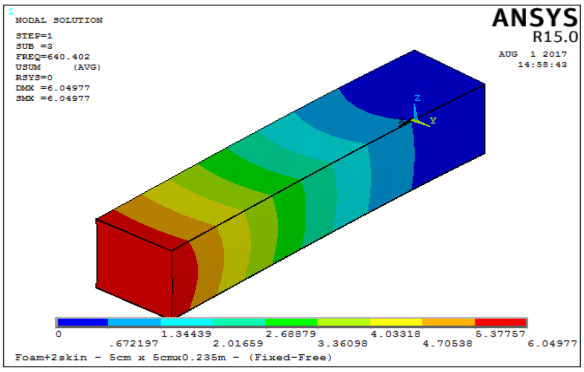
ELEMENT



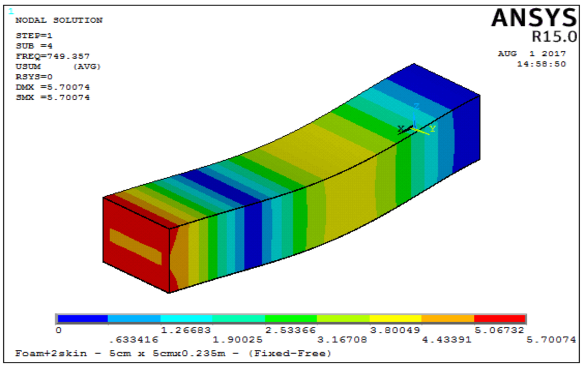
Fn=222.17 Hz



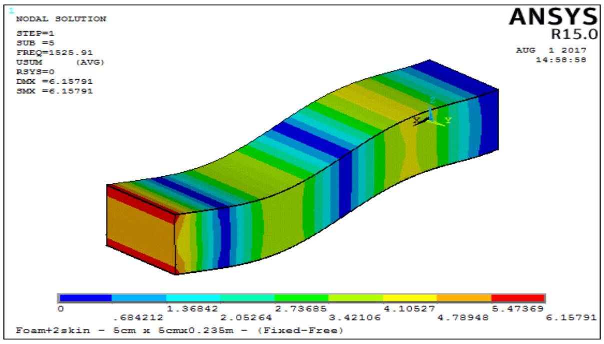
Fn =635Hz



Fn=750 Hz



Fn=1525Hz



Fn=1876Hz

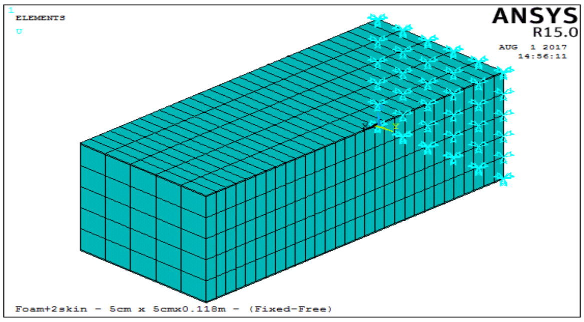
Figure 13. FEM model (fixed-free) condition for beam composite material, L= (235) mm.

4.2 Young's modulus for (skin and core) beam depended frequency

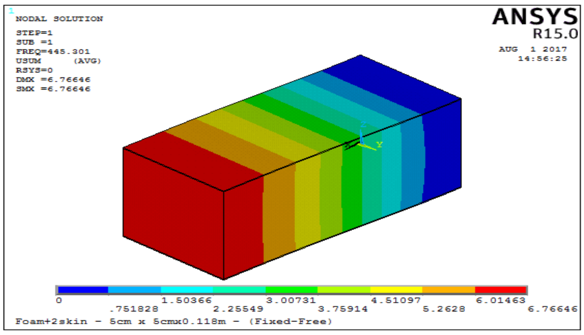
The skin was separated from the core of a composite material beam for (470) mm length. The Young's Modulus for both core and skins over a frequency range of (30-1000) Hz, and (Fixed -Free) boundary condition. It has been using to obtain five mods shapes and five natural frequency. Table 4 provides the experimental result for Young's Modulus.

Figure 15 showed that the Young's Modulus of skin and the core was affected by pure bending, because the Young's Modulus is constant at frequencies from (30 to 605) Hz. When applied the modal analysis in each beams that observed the Young's Modulus for skin is the highest value on the core.

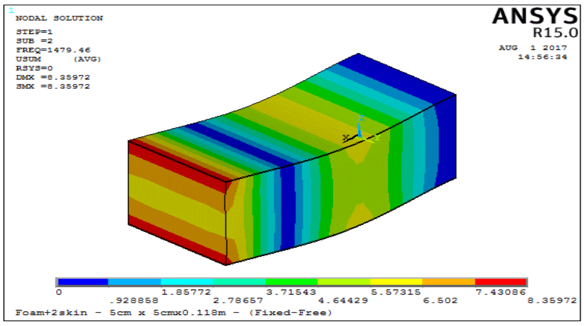
Table 5 provides theoretical Young's Modulus result, at (Fixed-Free) boundary condition and for (Core-Skin) beam. Table 6 represented the (Theoretical & Experimental) result for Young's Modulus and also the error Percentage at (Fixed-Free) boundary condition for Core. The comparison between the (Theoretical and Experimental) results it were showed a good. Table 7 represented the (theoretical and experimental) result for Young's Modulus and also, the error percentage at (Fixed-Free) boundary condition for (skin). Figures 16 and 17 depicts compared between (Theoretical and Experimental) result for Young's Modulus. Figures 18 and 19 were showed FEM models for the (core and skin).



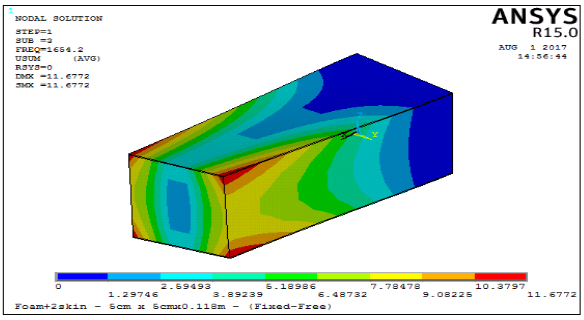
ELEMENT



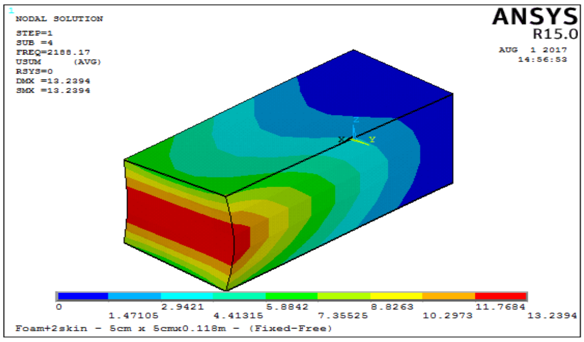
Fn=445.3Hz



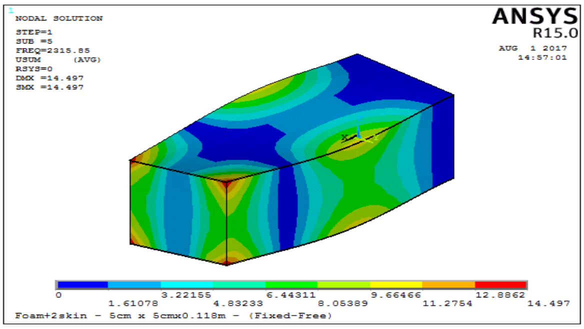
Fn =1479.5Hz



Fn=1654.4hz



Fn=2188Hz



Fn = 2315.9Hz

Figure 14. FEM model (fixed-free) condition for beam composite material, L= (118) mm.

Table 4. Experimental Young's modulus result by (fixed-free) boundary condition for (core and skin) beam.

L= (470mm)				
	Core		Skin	
mode	Frequency(Hz)	Young's Modulus(MPa)	Frequency(Hz)	Young's Modulus(MPa)
1	33.755	83585252.7	60.12	7.25E+12
2	128.94	57815823.7	194.7	2.52E+12
3	134.28	64603592.3	220.03	1.72E+12
4	230.41	69900554.1	267.33	1.74E+12
5	373	82253675.96	605	4.01E+12

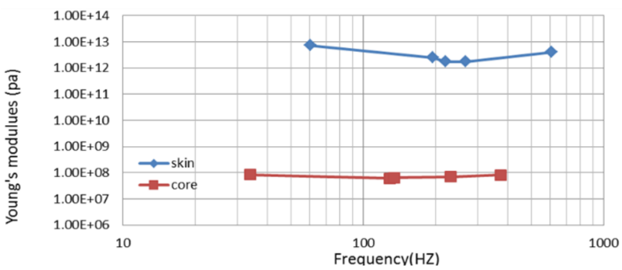


Figure 15. Experimental Young's modulus (core & skin) cantilever beams by using (fixed-free) boundary condition, and length (L= 470 mm).

Table 5. Theoretical Young's modulus result at (fixed-free) boundary condition for (core-skin) beam.

L= (470mm)				
	Core		Skin	
Mode	Frequency(HZ)	Young's Modulus (MPa)	Frequency (HZ)	Young's Modulus (MPa)
1	24	77704054.7	56.865	6.49E+12
2	141	60928476.8	186.39	2.30E+12
3	145.79	75645496.1	226.78	1.51E+12
4	221.44	64563953.5	265.96	1.61E+12
5	341.59	68983932.6	636.89	4.50E+12

Table 6. Theoretical and experimental young's modulus and error percentage by (fixed-free) boundary condition for (core).

L= (470mm)			
Mode	Experimental Young's Modulus (MPa)	Theoretical Young's Modulus (MPa)	Error %
1	83585252.7	77704054.7	7.04
2	57815823.7	60928476.8	5.38
3	64603592.3	75645496.1	17.09
4	69900554.1	64563953.5	7.63
5	82253675.96	68983932.6	16.13

Table 7. Experimental and theoretical Young's modulus and error percentage at (fixed-free) boundary condition for (skin).

L= (470mm)			
Mode	Experimental Young's Modulus (MPa)	Theoretical Young's Modulus (MPa)	Error %
1	7.25E+12	6.49E+12	10.48
2	2.52E+12	2.30E+12	8.73
3	1.72E+12	1.51E+12	12.2
4	1.74E+12	1.61E+12	7.47
5	4.01E+12	4.50E+12	12.22

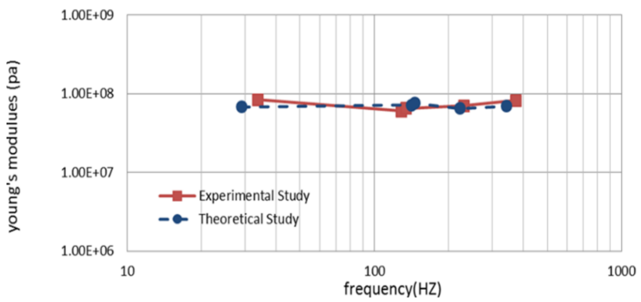


Figure 16. Comparison Young's modulus of the study (theoretical & experimental) of the core cantilever beam, L= (470) mm.

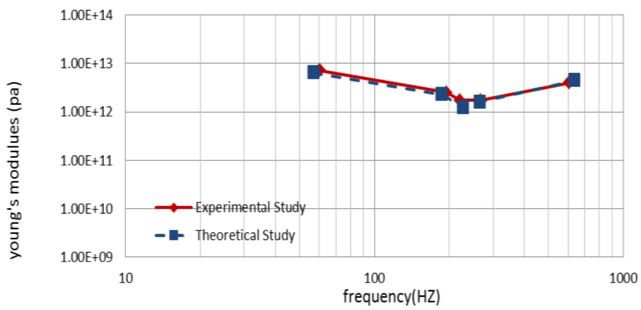
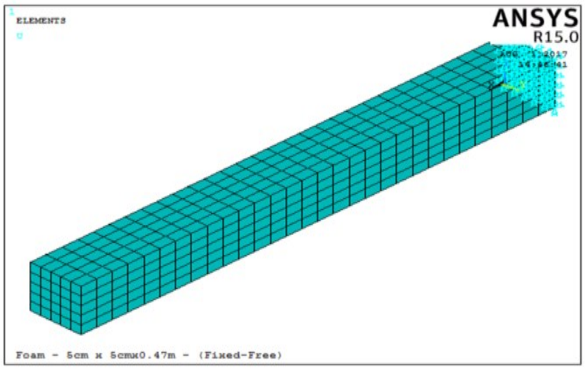
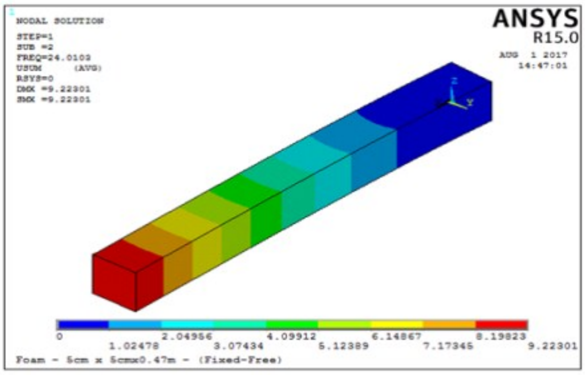


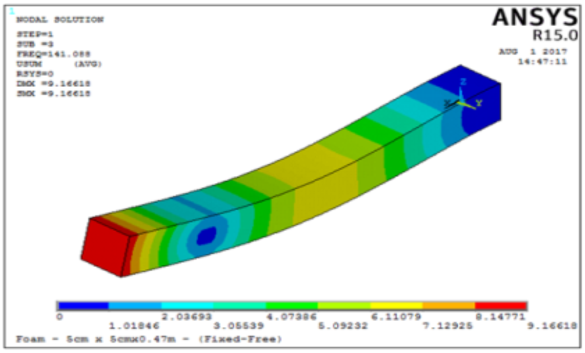
Figure 17. Comparison Young's modulus of the study (theoretical & experimental) of the skin cantilever beam L= (470) mm.



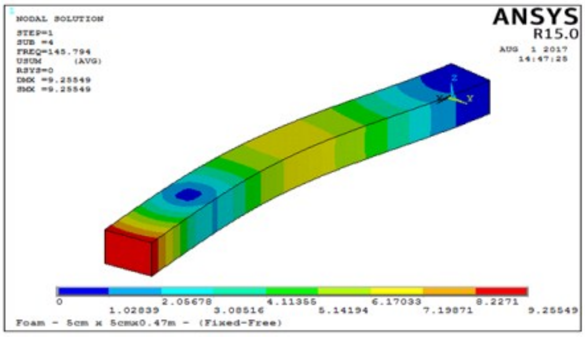
ELEMENT



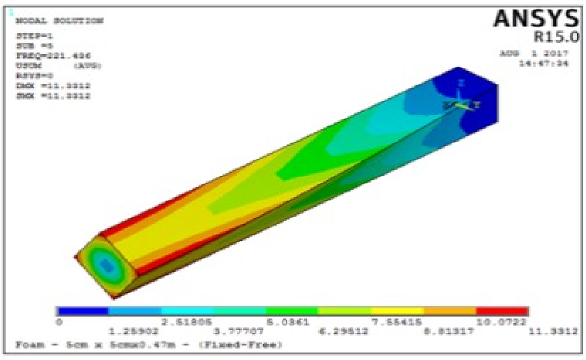
Fn=24 Hz



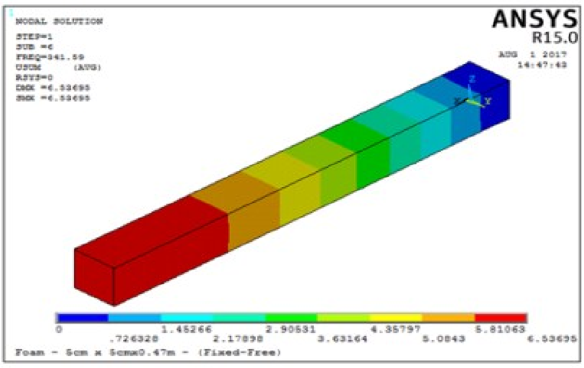
Fn=141Hz



Fn=145.79Hz



Fn=221.44Hz



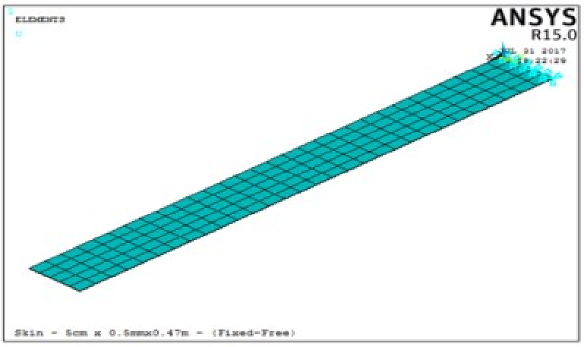
Fn=341.59 Hz

Figure 18. FEM model (fixed-free) boundary condition for core, L= (470) mm.

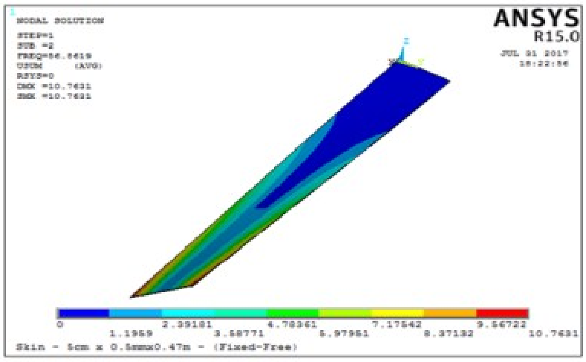
5. Conclusions

The conclusions have been obtain from the results as the following:

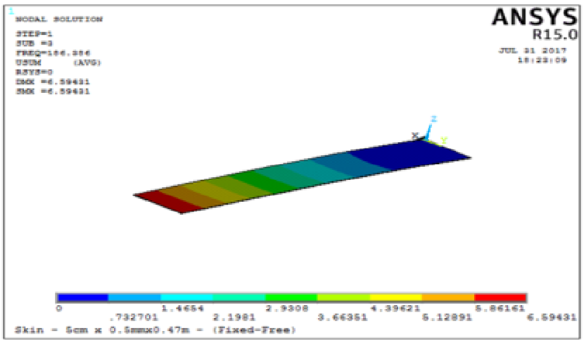
1. At low frequencies, the sandwich beam are subjected to pure bending, which makes the value of the young's modulus almost constant.
2. At high frequencies the sandwich beam is affected by shear bending within the material core, causing the particles to move towards the plane. This displacement occurs because of the difference in the value of the Young's Modulus between the skin and the core, which causes a decrease in the Young's Modulus with increasing frequency.
3. When the skin layer is separated from the core, the Young's Modulus remains almost constant, so that all the plates remain subject to pure bending.
4. A comparison between Experimental & Theoretical results by applied FEM for (Fixed-Free) boundary condition a good requirement, the larger error percentage in composite beam (235) mm nearly (17.391%).



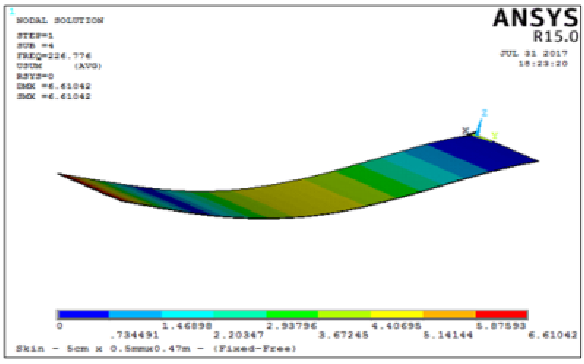
ELEMENT



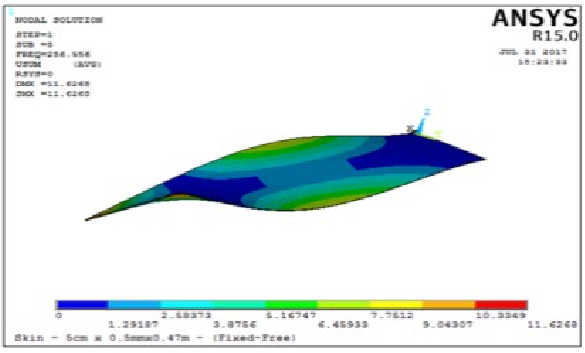
Fn=56.865Hz



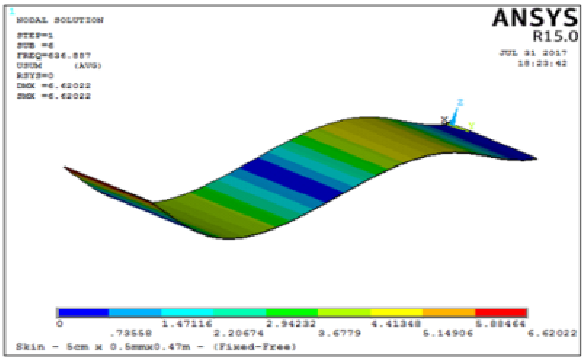
Fn =186.39Hz



Fn= 226.78 Hz



Fn= 265.96Hz



Fn= 636.89Hz

Figure 19. FEM model (fixed-free) boundary condition for skin, L= (470) mm.

Nomenclature

Symbol	*Definition*	*Units*
E	Young's Modulus	N/m^2
L	Free Length	m
I	Mass Moment of Inertia	m^4
m	Mass Per Unit Length	Kg/m
βn L	Constant Values	-
ρ	Density the Material	kg
H	Thickness the Beam	m
C_n	Constants	-
ω_n	Natural Frequency	rad/s
ω_d	Damped Natural Frequency	rad/s

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