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EFFECT OF CORE AND FACE SHEET THICKNESS ON DYNAMIC PARAMETER OF BEAMS USING FEM AND ABAQUS

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ABSTRACT

In this paper, finite element analysis of isotropic and sandwich beams are presented by using ABAQUS software keeping time constant. ***.*** *Various combinations have been chosen by varying beam thickness in case of isotropic beam and core thickness and face sheet thickness in case of sandwich beam. The results are generated on ABAQUS to compare the fundamental frequency of vibration for selected beams against the first order shear deformation theory (FSDT) model. It is perceived from isotropic and sandwich beam results that an increase in the face sheet thickness produces an increase in modal frequency. The material chosen for this purpose is Aluminum 6061 T-6. Different modes of vibration frequency are alsodiscussed in the present scenario.*

Keywords- ABAQUS,Beams, FEM

1. INTRODUCTION

In the competitive aerospace industry which is based on light weight and high strength structures, the usage of composite sandwich materials are essential in meeting the severe performance requirements. Foam core sandwich structures are utilized in a number of aerospace applications such as fabrication of wing flaps, aircraft flooring, and interior secondary structures. The benefits of using a foam core are reduced parts count, seamless construction, light weight, low cost and offers higher reliability of product quality [1]. Knowing these numerous applications, vibration of sandwich structures is of extreme interest to engineers and designers in the aerospace industry as these structures experience a wide range of dynamic loads during flight profile [2].The core material is usually low strength material, but its higher thickness provides the sandwich composite with high bendingstiffness with overall low density.The core is bonded to the skins with an adhesive or with metal components by brazing together [2]. The advantages of sandwich material are high bending stiffness combined with low mass, smooth surfaces, good fatigue behavior, good thermal insulation, good dampening capacity and high energy absorption [3, 4].Application of sandwich materials in damped are structures for effective vibration damping, aerospace field, building construction, naval ships, rail industry and automotive industry[2].

1.1 FEM and ABAQUS

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Finite element method (FEM) is the dominant discretization technique in structural mechanics. The basic concept in the physical interpretation of FEM is the subdivision of the mathematical model into disjoint components of simple geometry called finite elements. The response of each element is expressed in terms of a finite number of degrees of freedom characterized as the value of an unknown function, at a set of nodal points. The response of the mathematical model is then considered to be approximated by that of the discrete model obtained by connecting or assembling the collection of all elements. Finite elements do not overlap in space.A typical finite element analysis on a software system requires the nodal point spatial locations, elements connecting the nodal points, mass properties, boundary conditions, loading function details and analysis options. Abaqus is a powerful engineering simulation software based on the FEM. The unique features of Abaqus include an extensive library of elements that can model virtually any geometry. A complete Abaqus analysis usually consists of three distinct stages pre-processing, simulation and post –processing. The Abaqus/CAE is the complete Abaqus environment that provides a simple, consistent interface for creating Abaqus models, interactively submitting and monitoring Abaqus jobs, and evaluating results from Abaqus simulations. Abaqus/CAE is divided into modules, where each module defines a logical aspect of the modelling process [5-6].

II MODELLING OF BEAMS

In the present work, the geometric parameters that define the isometric and sandwich beams are illustrated in Fig.2 (a & b). Material properties of A6061-T6 [1] is tabulated in Table 1. Procedure of modeling of isotropic beam is shown in the Fig. 1 while Fig. 2(a & b) shows the geometry of the isotropic beam and sandwich beam. Modelling of sandwich beam is same as that of isotropic beam except partition of the beam. The sandwich is partitioned in upper and lower face sheet by central core.

Table 1 Material Properties of Aluminum 6061-T6

Material Property	Young’s Modulus	Poisson’s Ratio	Density
Aluminium 6061-T6	68.9GPa	0.33	2700kg/m^3

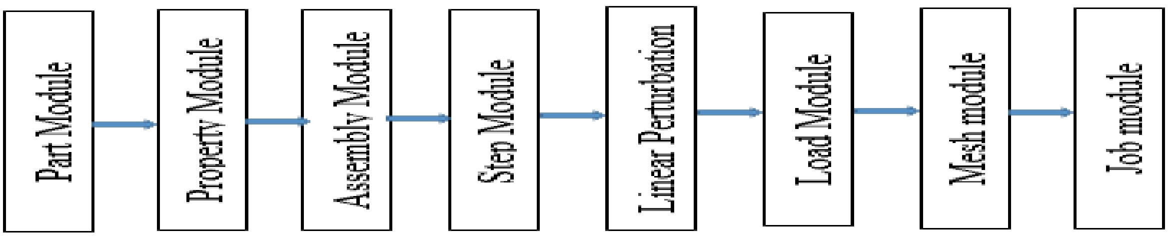

Figure1. Procedure of isotropic beam modeling

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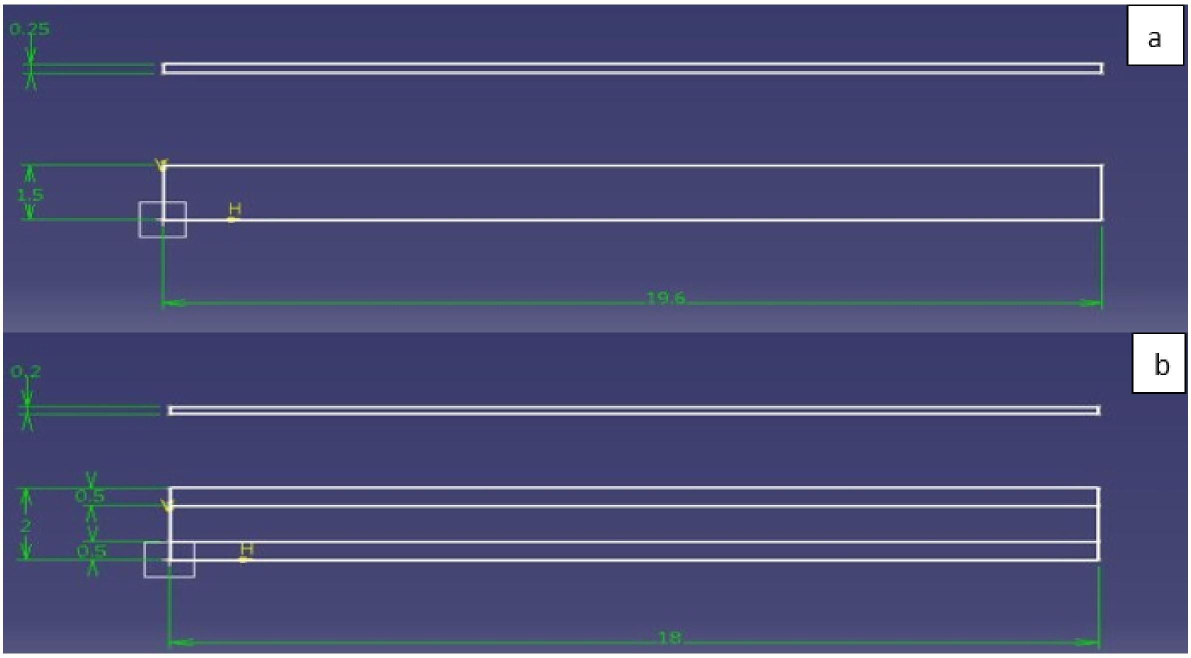

Figure2. Geometry of (a) Isotropic beam and (b) Sandwich beam

III RESULTS AND DISCUSSION

The result section demonstrates the validity of the analytical model by first verifying an aluminum isotropic case.

This is followed by sandwich beam trends associated with varying face sheet thickness and core thickness. The present analysis shows the effect of thickness variation of isotropic and sandwich beams consist of face sheets and core of the same material (A6061-T6) on the fundamental frequency of vibration. The results are generated on Abaqus and are compared against FSDT model [1].

3.1 Isotropic Beam Thickness Variation

Comparison of the fundamental frequency of vibration for the isotropic beam model of varying beam thickness [1] against the result fromFSDT model has shown in Table 2.Table 3 illustrates the five model frequency using the Abaqus and evaluated alongside FSDT values. These results are calculated for a beam with constant thickness of 0.02 inch. Fig.3shows the result of isotropic validation graphically. The first three mode variations of isotropic beam of different thickness is shown in the Fig.4. Isotropic beam bending images of different modes are shown in the Fig.5(a, b & c). It is seen that as the beam thickness increases the frequencies of FSDT and ABAQUS model also increases as shown in the Table 2 & 3. The difference in frequencies are evaluated in the form of % error (see Table 2 & 3). The FSDT and ABAQUS frequencies are validated with respect to the variation of the beam thickness as shown in the Fig.3. It revealed from the results that all frequencies increases with the thickness of the beam for all three modes. But the slope of the frequency of mode 3 is more as compared to the mode 1 and mode 2. Similarly, mode 2 has more slope of frequency than mode 1. It means that

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vibration in the beam for mode 3 will be large and hence the deflection will be higher as compared to the mode 1 and 2. This can be clearly seen in Fig. 4 and Fig. 5 for deflection.

Table 2 Isotropic Beam Thickness Variation

Fundamental Mode			
Beam Thickness	Natural Frequency FSDT (Hz)	Natural Frequency ABAQUS (Hz)	% Error
0.02	5.527	4.73	-15.16
0.03125	8.636	7.32	-15.23
0.04	11.055	11.193	1.24
0.0625	17.217	17.472	1.14
0.08	22.109	20.334	8.04
0.091	25.148	24.269	-3.53
0.125	34.543	30.626	0.24

Table 3 Isotropic Beam Model Frequency

Constant Beam Thickness=0.02 inch			
Mode	Natural Frequency FSDT (Hz)	Natural Frequency ABAQUS (Hz)	% Error
1	5.527	4.73	-15.16
2	22.109	18.96	-14.24
3	49.745	42.58	-14.40

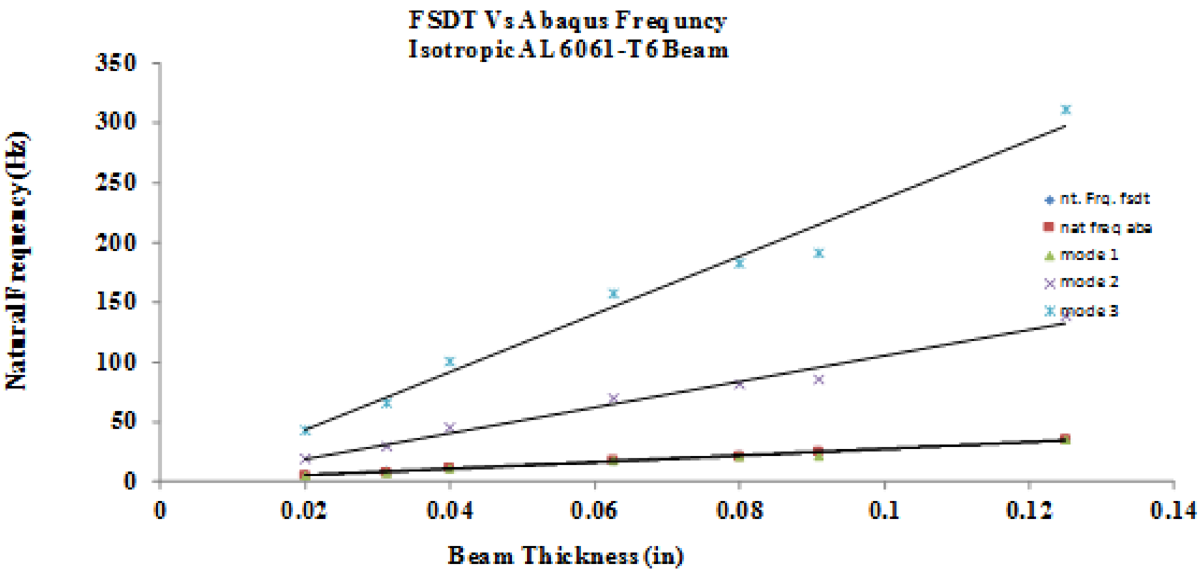

Figure 3. Isotropic validation

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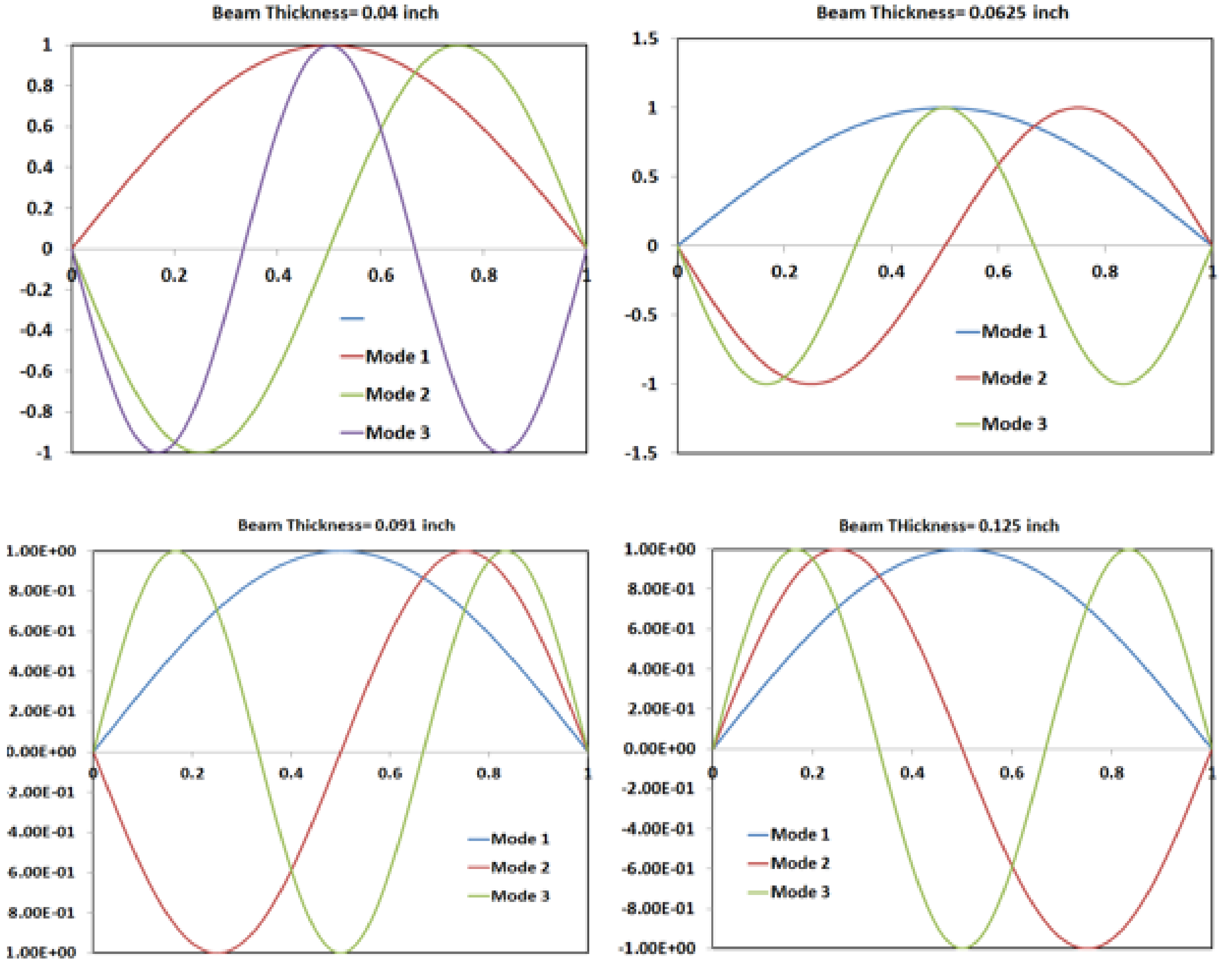

Figure 4. Mode shapes of isotropic beam of different thickness

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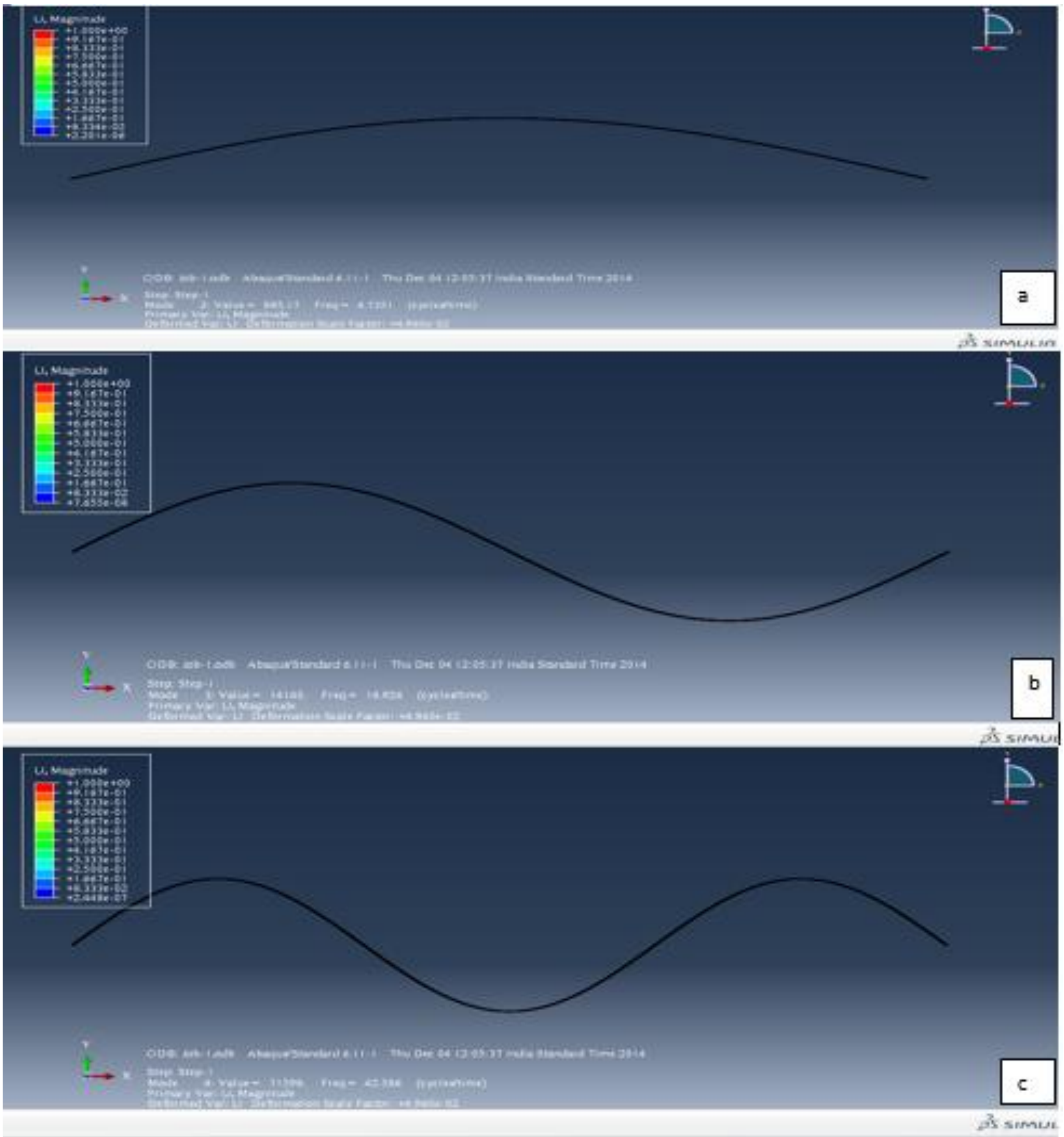

Figure 5. Images of isotropic beam of thickness 0.02 inch for (a) first mode, (b) second mode and (c) third mode

3.2 Sandwich Beam Thickness Variation

3.2.1 Constant Core Thickness 0.25 inch

The first analysis is performed by varying face sheet thickness and keeping core thickness constant. This analysis is performed using an aluminum 6061-T6 metal sandwich beam. This signifies that the core and the face sheet are all manufactured from same material [1]. The results are presented in Table 4 and Fig 6. The variation of the frequencies with face sheet thickness for constant core thickness as resulted in Table 4

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demonstrate that frequencies increases very fast as compared and hence the deflection too as shown in the Fig.8. Also, increment of frequency in case of sandwich beam is large as compared to the isotropic beam.The first three mode variations of sandwich beam of different face sheet thickness and constant core thickness 0.25 inch is shown in the Fig.7. Sandwich beam bending images of different modes are shown in the Fig.8(a, b & c).

Table 4 Sandwich Beam Face Sheet Thickness Variation

Constant core thickness= 0.25 inch			
Fundamental Mode			
Face Sheet	Natural Frequency	Natural Frequency	% Error
0.0200	81.936	80.353	-1.70
0.0312	96.215	97.257	1.080
0.0400	105.011	106.326	1.150
0.0625	123.369	125.589	1.799
0.0800	135.547	131.378	-3.070
0.0910	142.710	146.384	2.570
0.1250	163.535	173.259	5.940

Table 5 Sandwich Beam Modal Variation

Constant Core Thickness= 0.25 inch			
Constant Face Sheet Thickness=0.02 inch			
Mode	Natural Frequency	Natural Frequency	% Error
1	81.936	80.352	-1.93
2	326.909	320.976	-1.81
3	732.444	720.594	-1.61

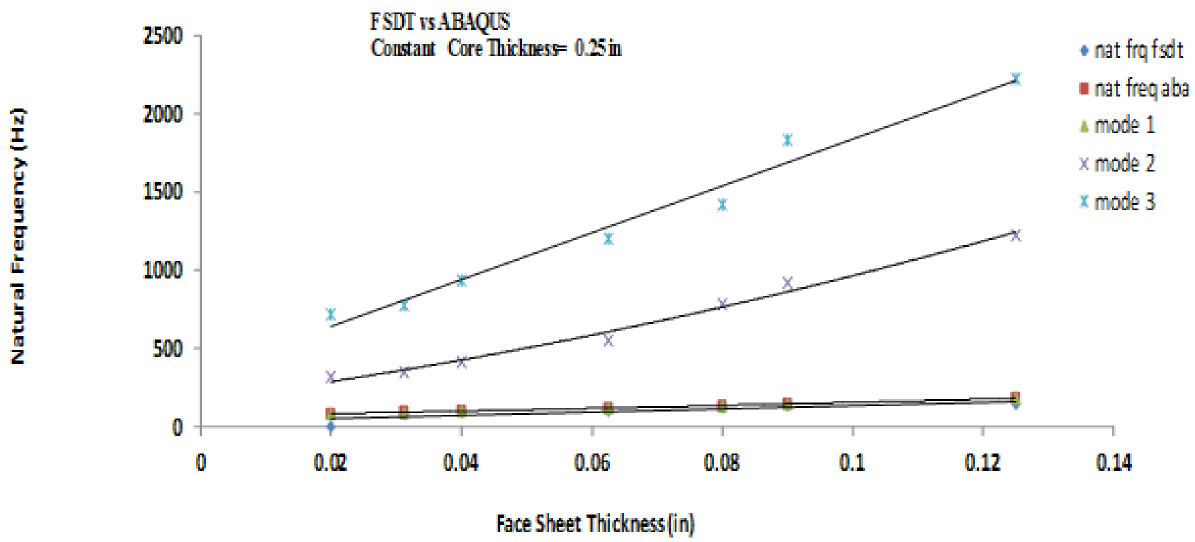

Figure 6. Model frequency response with increasing face sheet thickness

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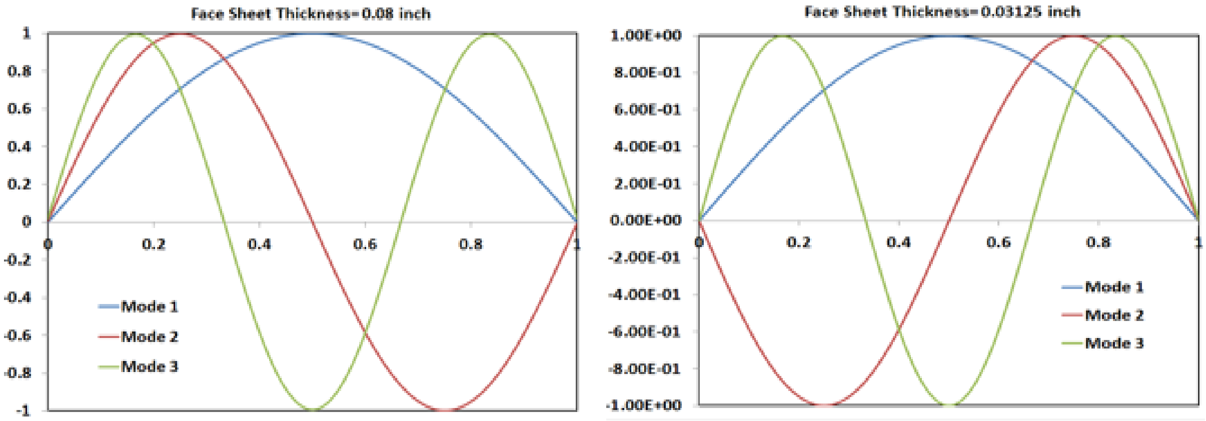

Figure 7 Mode shapes of sandwich beam of constant core thickness (0.25inch) and different face sheet thickness

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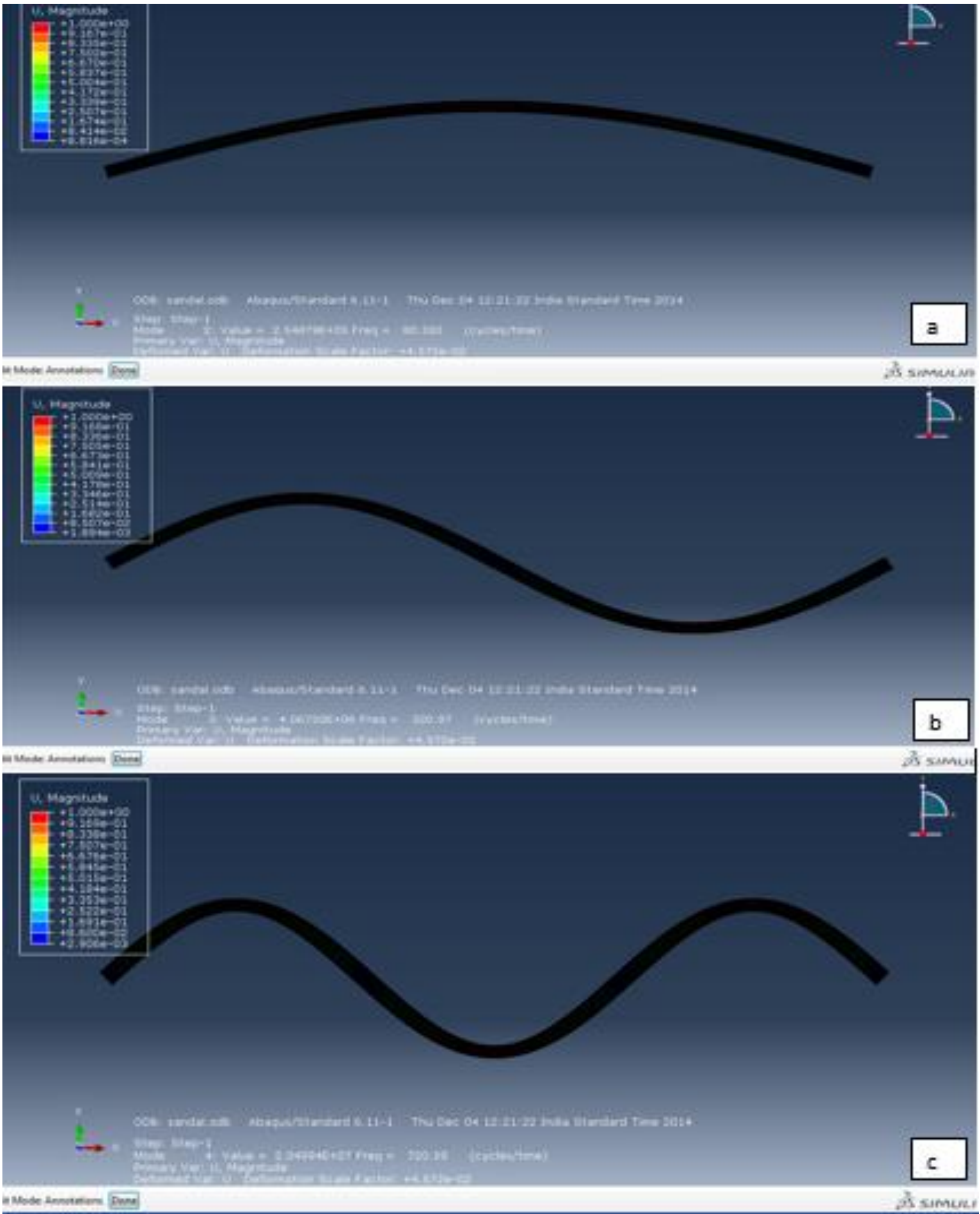

Figure8. Images of sandwich beam for (a) first mode, (b) second mode and (c) third mode

3.2.2Constant Face Sheet Thickness

Figure 9 shows the mode variation of sandwich beam for constant face sheet thickness 0.03125 inch. Images of sandwich beam bending of different modes are shown in the Fig.10 (a, b & c).The variation of the core

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thickness of sandwich beam with the frequencies are shown in the Table 6 and predicted that frequency increases with the core thickness. Although, for constant core and face sheet thickness, frequency variation is dependent on mode variation and there is very high increment is depicted with higher mode (see Table 7 and Fig.9). It is clearly evident from the results that % error decreases at higher mode as shown in the Table 7. The deflection in the sandwich beam for three modes are shown in the Fig.11. The higher frequency shows the large deflection in the beam and vice-versa. Due to the high frequency in case of mode 3 as compared to the mode 1 and mode 2, the greater deflection are predicted.

Table 6 Sandwich Beam Core Thickness Variation

Constant Face Sheet Thickness= 0.03125 inch			
Fundamental Mode			
Core	Natural Frequency	Natural Frequency	% Error
0.1	53.82	54.2	0.772
0.2	83.337	85.25	2.29
0.3	108.147	104.55	-.3.37
0.4	129.79	128.49	-1.00
0.5	149.165	156.78	4.71
0.6	166.823	183.96	10.1

Table 7 Sandwich Beam Model Variation

Constant Core Thickness = 0.2 inch			
Constant Face Sheet Thickness = 0.03125			
Mode	Natural Frequency	Natural Frequency	% Error
1	83.337	80.352	-3.88
2	332.687	288.800	-13.18
3	746.087	648.813	-13.03

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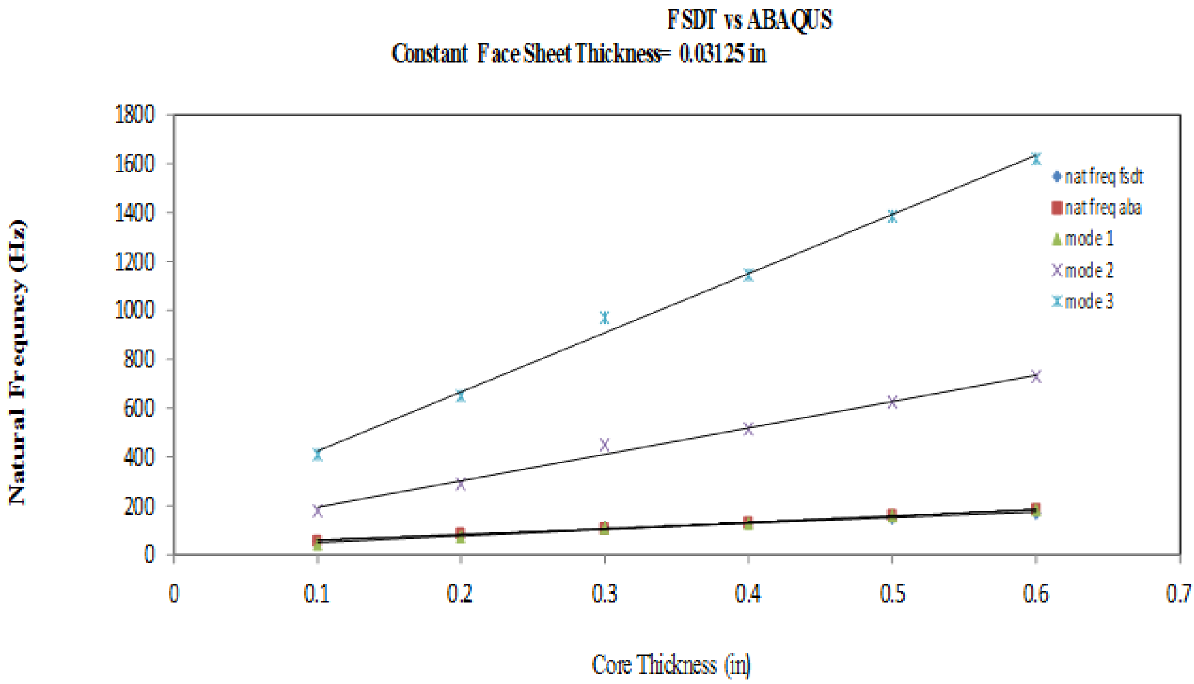

Figure 9.Modal frequency response with increasing core height

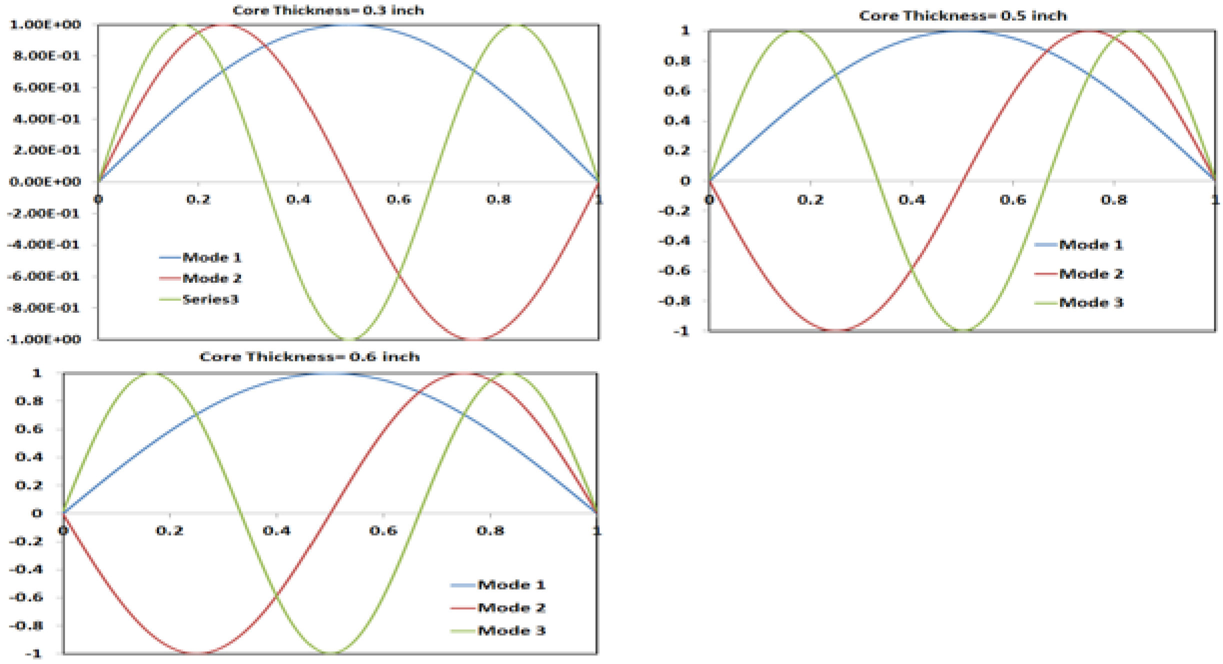

Figure 10. Mode shapes of sandwich beam at different core thickness

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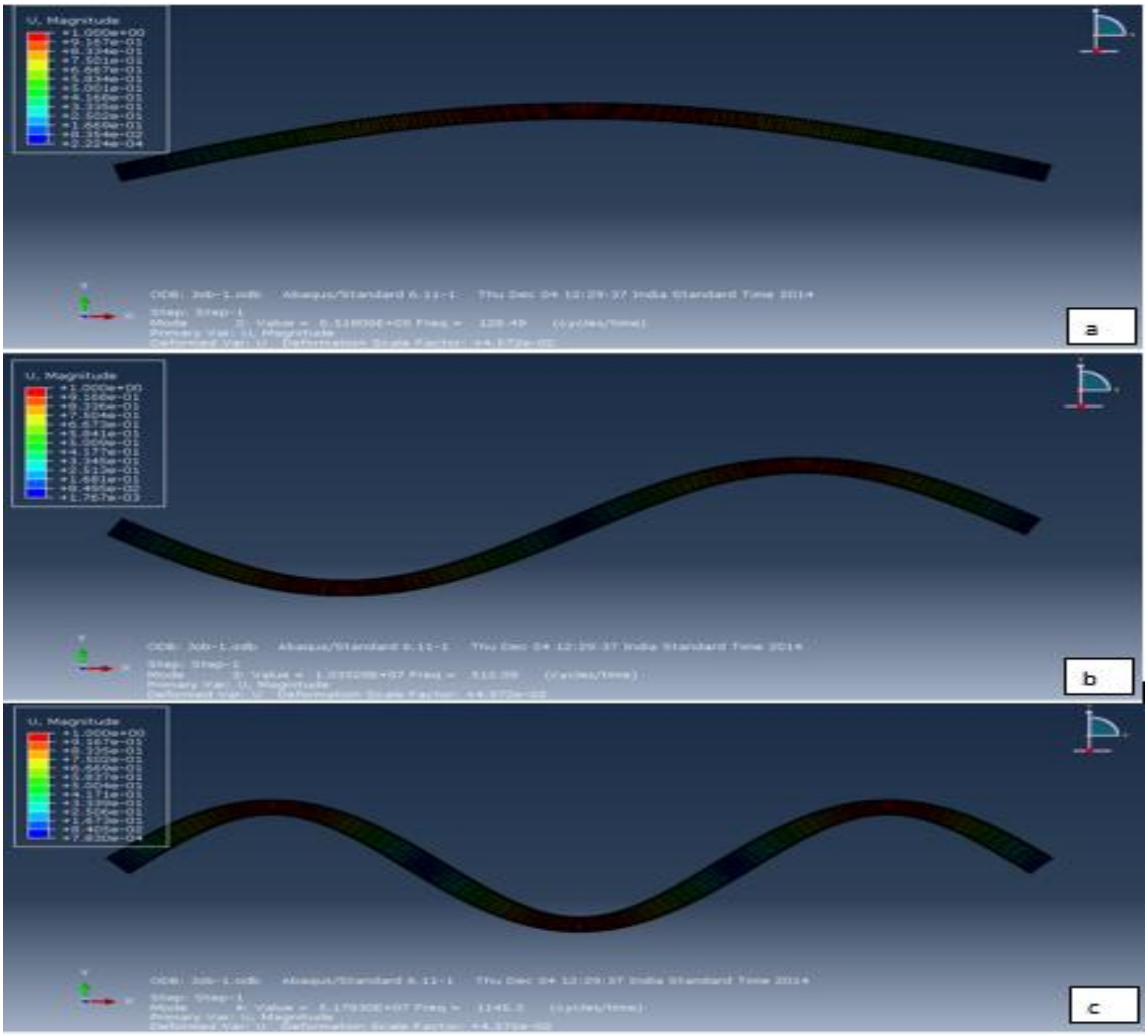

Figure 11. Images of sandwich beam for (a) first mode, (b) second mode and (c) third mode

IV CONCLUSION

The analytical model matches exceptionally well with the results obtained from ABAQUS output for the isotropic beam validation. Additionally, it is observed from isotropic and sandwich beam results that an increase in the face sheet thickness produces an increase in modal frequency. Also, the effect of FSDT is not critical for the fundamental mode of vibration. However, at higher modes ABAQUS greatly under predicts the natural frequency response of the sandwich beam. Similar trends are observed if the face sheet thickness is kept constant and the core thickness is increased. Additionally, an increase in core thickness can be construed as an increase in beam inertia that augments the overall structural stiffness.

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