

**Static and Buckling analysis of Laminated
Sandwich Plates with Orthotropic Core using FEM**

*A Thesis Submitted In Partial Fulfilment
of the Requirements for the degree of*

**Bachelor of Technology
in
Mechanical Engineering**

by

Santanu Kumar Sahoo

Roll No: 109ME0413

Under the supervision of

Dr. S. K. Panda



**DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA**

May 2013

Static and Buckling analysis of Laminated Sandwich Plates with Orthotropic Core using FEM

*A Thesis Submitted In Partial Fulfilment
of the Requirements for the degree of*

**Bachelor of Technology
in
Mechanical Engineering**

by

Santanu Kumar Sahoo

Roll No: 109ME0413

Under the supervision of

Dr. S. K. Panda



**DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY, ROURKELA**

May 2013



NATIONAL INSTITUTE OF TECHNOLOGY

ROURKELA

C E R T I F I C A T E

This is to certify that the work in this thesis entitled “Static and Buckling analysis of Laminated Sandwich Plates with Orthotropic Core using FEM” by *Santanu Kumar Sahoo*, has been carried out under my supervision in partial fulfillment of the requirements for the degree of *Bachelor of Technology in Mechanical Engineering* during session 2012-2013 in the *Department of Mechanical Engineering, National Institute of Technology, Rourkela*.

To the best of my knowledge, this work has not been submitted to any other University/Institute for the award of any degree or diploma.

Dr. Subrata Kumar Panda

(Supervisor)

Assistant Professor

Department of Mechanical Engineering

Date: 08/05/2013

National Institute of Technology, Rourkela

ACKNOWLEDGEMENT

I take this opportunity as a privilege to thank all individuals without whose support and guidance I could not have completed our project in this stipulated period of time. First and foremost I would like to express my gratitude to **Project Supervisor Prof. S K Panda**, Department of Mechanical Engineering, National Institute of Technology, Rourkela for his precious guidance, support and encouragement during the tenure of this work. His insights, comments and undaunted cooperation in every aspect of the project work have led to the successful completion of the project.

I would like to thank **Mr. Girish Kumar Sahu, M.Tech** and **Mr. Pankaj Katariya, M.Tech(Res)** , Department of Mechanical Engineering, National Institute of Technology, Rourkela for their constant help in understanding of the technical aspects of the project. I will also be grateful to **Ph.D scholar Mr. Vishesh Ranjan Kar**, for his constant help in the successfully carrying out the new results.

And finally I also extend my heartfelt thanks to my families, friends and the Almighty.

Santanu Kumar Sahoo (109ME0413)

Department of Mechanical Engineering

National Institute of Technology

Rourkela

CONTENTS

	Page No.
Abstract	I
List of tables	II
List of Figures	III
1. Introduction	1
2. Literature Review	2-4
3. ANSYS and its application	5-6
4. Mathematical Formulation	7-10
5. Result and Discussion	11-16
6. Conclusion	17
References	18-20

Abstract

In this study, static and buckling behavior of sandwich plate with orthotropic core has been presented. The present model is developed based on first order shear deformation theory and discretised using a finite element method in ANSYS environment to obtain the responses. The convergence test has been carried out and the results are compared with those available open literature. We note substantial effect of different parameters such as, support conditions, number of layers and thickness ratio on static and stability behavior of laminated structures.

List of Tables

1. Material properties
2. Non-dimensional center deflection of clamped sandwich square plate with $a/h= 10$ under uniformly distributed load for SHELL181
3. Non-dimensional center deflection of clamped sandwich square plate with $a/h= 10$ under uniformly distributed load for SHELL281
4. Convergence of Non-dimensional buckling load with simply supported condition

List of Figures

1. SHELL181 geometry
2. SHELL281 Geometry
3. Non-dimensional deflection V/S a/h for $0/C/0$
4. Non-dimensional deflection V/S a/h for $0^0/90^0/C/90^0/0^0$
5. Non-dimensional deflection v/s a/h for $(0^0/90^0)_5/C/(90^0/0^0)_5$
6. Non-dimensional buckling load for thickness ratio ($a/h=10$) under uniaxial loading (a) SHELL181 (b) SHELL281
7. Non-dimensional buckling load for aspect ratio ($a/b=1$) under uniaxial loading (a) SHELL181 (b) SHELL281
8. Non-dimensional buckling load for aspect ratio ($a/b=1$) under biaxial loading (a) SHELL181 (b) SHELL281

1 Introduction

Sandwich structures are such, when two hard materials are joined using a low molecular weight structural core then it is called a sandwich plate. It consists of face plate, core and an adhesive attachment which is capable of transmitting shear and axial loads to and from the core. Face sheets are the external strong layers and in between material is called the core. Core material separates both face sheets, transfers the load from one plate to the other and resists deformations perpendicular to the face plane. They can have different shapes and provide resistance to shear stress along the planes perpendicular to the face sheets. In general core has less stiffness and less strength. It enhances the plate performance of crushing and impulsive loading. The main contribution of the sandwich plate is to provide lateral support to core members and thereby enhancing the buckling strength of these members. Polymeric foam gives unique structural and thermal property to the structure. They transfer shear between each plate and eliminate the need for stiffeners. It is a good alternative to stiffened steel and reinforced concrete. It has high stiffness to weight ratio.

It is used because of its simplicity in design and less time consuming as compared to other similar equivalents. It is corrosion and fatigue resistant. The elastomer core dissipates the strain energy generated over a large area, avoiding stress concentration at a particular point and thereby avoiding permanent deformations and cracks. This makes the sandwich plates more robust and increases the working life. The soft core also provides good vibration and noise bearing ability.

There is a need of composite that has a high transverse shear modulus to high in-plane tensile modulus. So the transverse shear modulus plays a significant role in sandwich plates. Most important property of sandwich plates is its high strength to weight ratio and high stiffness to weight ratio. To account for the correct structural behavior it has to be modeled properly and analyzed to exploit their design strength. Because of its wide application in various sectors there is a need for examination of its behavior. Parametric study gives the idea of how to use and where to use with the consequences of loading and limiting conditions. The property and orientation can be appropriately judged from this and hence helps in correct operation.

2 Literature review

The most attractive properties of composite materials are the high strength-to-weight and high stiffness-to-weight ratios and because of many other superior physical properties of unidirectional fiber reinforced polymer matrix composites, they are excellent material for high-performance structures. The core material in between the face sheets, increase the moment of inertia with little increase of weight producing an efficient structure for resisting bending and buckling load. The sandwich plates will have to meet the stiffness and strength criteria. In addition to that, they need to meet typical modes of failure of face yielding, face wrinkling, intra-cell dimpling, core shear or local indentation. The critical failure mode and the corresponding failure load depend on the properties of the face and the core material, on the geometry of the structure and on the loading arrangement. The skin can be modeled in a very simple manner while the modeling of the core is quite sophisticated.

Many studies related to sandwich structure are reported in the open literature time to time to fill the gap. Mahendran *et al.* [1] investigated and designed sandwich panels subjected to local buckling effects using finite element method and carried out an experiment of the model. Carrea *et al.* [2] explained a refined multilayered finite-element model applied to linear and non-linear analysis of sandwich plates. Zhen *et al.* [3] studied on accurate higher-order theory and C^0 finite element for free vibration analysis of laminated composite and sandwich plates using the Hamilton's principle and Navier's technique. Mihir *et al.* [4] proposed an improved higher order zigzag theory for the static analysis of laminated sandwich plate with soft core. They assumed that the variation of in-plane displacements is cubic for both the face sheets and the core and transverse displacement is vary quadratic ally within the core while it remains constant through the faces. Kant *et al.* [5] reported on analytical solutions for the static analysis of laminated composite and sandwich plates based on a higher order refined theory. Ramtekkar *et al.* [6] used the layer wise (three-dimensional), mixed, 18-node finite element (FE) for the accurate evaluation of transverse stresses in sandwich laminates. Pokharel *et al.* [7] analysed the sandwich panels which was subjected to local buckling effects by using finite element method. Palazotto *et al.* [8] studied the sandwich panels using finite element analysis and designed it which was subjected to local buckling effects by a displacement-based, plate bending, finite element algorithm. Yeh *et al.* [9] used the finite element method and Hamilton's principle to derive the equations of motion for the orthotropic sandwich plates. Malekzadeh *et al.* [10] worked on higher-order dynamic response of composite sandwich panels by using First shear deformation theory (FSDT).

Ivanez *et al.* [11] studied on numerical modeling of foam-cored sandwich plates under high-velocity impact using finite element method. Skvortsov *et al.* [12] studied the overall behavior of singly curved shallow sandwich panels by using mathematical model developed by using Reissner Mindlin plate theory. Pandit *et al.* [13] studied stochastic perturbation-based finite element for deflection statistics of soft core sandwich plate. Skvortsov *et al.* [14] studied two-dimensional analysis of shallow sandwich panels. Pandya *et al.* [15] studied higher order shear deformable theories for flexure of sandwich plates-finite element evaluations. Kant *et al.* [16] studied the finite element transient analysis of composite and sandwich plates based on a refined theory and a mode superposition method. Parton *et al.* [17] presented the finite element analysis for cement composite sandwich plates. Kant *et al.* [18] described finite element transient dynamic analysis of isotropic and fiber reinforced composite plates using higher-order theory. Pandya *et al.* [19] studied finite element analysis of laminated composite plates using a higher order displacement model. Marur *et al.* [20] studied the free vibration analysis of fiber reinforced composite beams using higher order theories and finite element modeling. Flores-Jhonson *et al.* [21] studied the structural behavior of composite sandwich panels with plain and fiber-reinforced foamed concrete cores and corrugated steel faces. Chakrabarti *et al.* [22] explained the behavior of laminated sandwich plates based on refined higher order plate theory. Chakrabarti *et al.* [23] proposed an efficient C^0 FE model for the analysis of composites and sandwich laminates by using higher order zigzag theory. SChakrabarti *et al.* [24] studied on the buckling behavior of laminated sandwich beam with soft core by developing FE model and using higher order zigzag theory. Zhen *et al.* [25] reported the vibration and stability analysis of laminated composite and sandwich beams by using displacement-based theories. Khdeir *et al.* [26] worked on the analysis of symmetric cross-ply laminated elastic plates using a higher order theory. Kant *et al.* [27] used the analytical solutions for free vibration of laminated composites and sandwich plates based on a higher order refined theory. C'etkovic' *et al.* [28] worked on bending, free vibrations and buckling of laminated composite and sandwich plates using a layerwise displacement model. Kheirikhah *et al.* [29] studied the buckling analysis of soft-core composite sandwich plates using improved high-order theory. Khare *et al.* [30] studied on free vibration of composite and sandwich laminates with a higher-order face shell element.

Based on the above literature it is true that many work has been done on the laminated composites and sandwich structures with and without orthotropic core to find the responses due to different loading condition. It is also true that studies based on the available finite

element software are less in number. In this proposed work a general finite element model of laminated composite sandwich plate with orthotropic core has to be develop using commercial finite element package ANSYS. The model has been verified and a set of parametric study using developed model is obtained by ANSYS parametric design language (APDL) code.

3 ANSYS and its application

In modern world design process has been too close to precision so the use of finite element method is extensive. It is being used as the most trustworthy tool for designing. It helps to predict the behavior of various products, parts, subassemblies and assemblies. Analyzing the results helps to prevent the time of prototyping and reduces the expense due to physical test. It also increases the innovation at a faster and more accurate way. Analysts and designers work together to find the most appropriate answer using the most optimized tool. ANSYS is now being used in a number of different engineering fields such as power generation, transportation, medical components, electronic devices, and household appliances.

The first ANSYS seminar was held in 1976. The designing was improved from 2D modeling to 3D modeling. Beam models to shell and then to volume elements were used for modeling. Graphics were introduced for better modeling and analysis. The substructure technique was introduced to divide the structure and analyze it element wise. The first task was to discretize the structure into nodes and elements. ANSYS gradually entered to a number of fields making it handy for fatigue analysis, nuclear power plant, medical applications, to find the eigenvalues of magnet, etc. Thermal analysis of various structures based on the thermal and mechanical loading was also done.

ANSYS is also very useful in electro thermal analysis of switching elements of a super conductor, ion projection lithography, detuning of an HF oscillator by the mechanical vibration of an acoustic sounder. It is used to analyze the vehicle simulation and in aerospace industries as well.

For present work the analysis is done by choosing two different shell elements from ANSYS library. In the present work an element SHELL181 is used for the static and buckling analysis of sandwich plates and for more refinement an element SHELL281 is used in the buckling analysis of sandwich structures.

SHELL181

This is a four-node linear shell element with six degrees of freedom at each node. Those are translation in x, y, z direction and rotation about x, y, z axis. It has plasticity and hence can be used in laminated composites and sandwich structures. Thin to moderately thick structures can be analyzed using this shell element. It can also be used for analyzing also for shells whose thickness changes under nonlinear analysis of plates. This shell element can also be used for analyzing layered shell structures. All the analysis is governed by First order Shear Deformation Theory (FSDT).

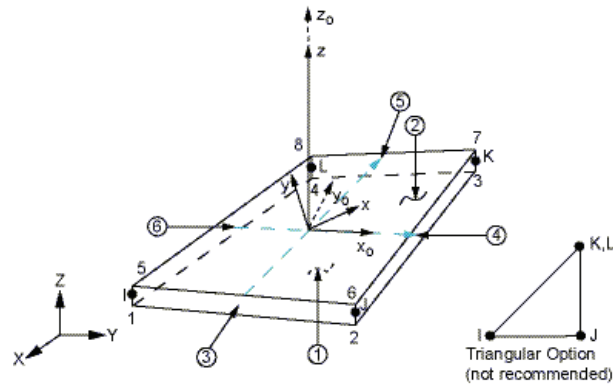


Fig. 1 SHELL181 geometry [31]

x_o = Element x-axis if ESYS is not provided.
 x = Element x-axis if ESYS is provided.

SHELL281

This is an eight-node linear shell element with six degrees of freedom at each node. Those are translation in x , y , z direction and rotation about x , y , z axis. It is well-suited for linear, large rotation, and/or large strain nonlinear applications. It uses the same theory and analyses all the elements that uses Shell181. The element formulation is based on logarithmic strain and true stress measures.

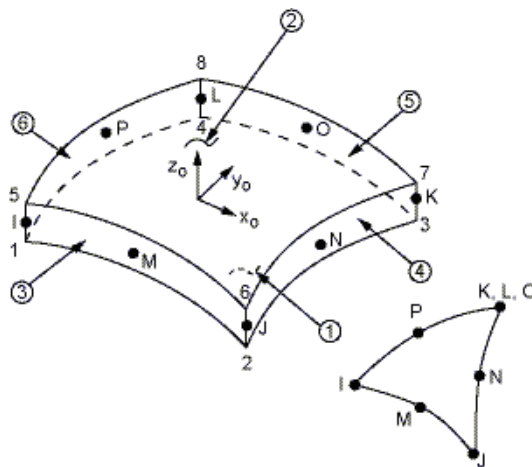


Fig. 2 SHELL281 Geometry [31]

x_o = Element x-axis if element orientation is not provided.

x = Element x-axis if element orientation is provided.

4 Mathematical Formulation

For the modeling purpose, a SHELL181 element is being selected among the available elements in ANSYS 12.0 element library. Fig. 1 shows solid geometry, node locations and the element coordinate system of the SHELL181 element. The element is defined by four nodes (I, J, K and L). This is a 2D four noded element with six degree of freedom per node and the degree of freedoms of each node are the translations and rotations in the respective axis. It is suitable for thin to moderately thick structures and well suited for linear, large rotation and large strain nonlinear applications.

It is well known that the mathematical model in ANSYS is based on the FSDT as follows:

$$\begin{aligned} u(x, y, z) &= u_0(x, y) + z\theta_x(x, y) \\ v(x, y, z) &= v_0(x, y) + z\theta_y(x, y) \\ w(x, y, z) &= w_0(x, y) + z\theta_z(x, y) \end{aligned} \quad (1)$$

The displacements are expressed in terms of shape functions (N_i)

$$\delta = \sum_{i=1}^j N_i \delta_i \quad (2)$$

where, $\delta_i = [u_0, v_0, w_0, \phi_x, \phi_y, \phi_z]^T$. The shape functions for four noded shell element ($j=4$) and eight noded shell element ($j=8$) are represented in Eqn. (3) and (4), respectively in natural (ξ - η) coordinates, and details of the element are given as

$$\left. \begin{aligned} N_1 &= \frac{1}{4}(1-\xi)(1-\eta) \\ N_2 &= \frac{1}{4}(1+\xi)(1-\eta) \\ N_3 &= \frac{1}{4}(1+\xi)(1+\eta) \\ N_4 &= \frac{1}{4}(1-\xi)(1+\eta) \end{aligned} \right\} \quad (3)$$

$$\begin{aligned} N_1 &= \frac{1}{4}(1-\xi)(1-\eta)(-\xi-\eta-1); & N_2 &= \frac{1}{4}(1+\xi)(1-\eta)(\xi-\eta-1) \\ N_3 &= \frac{1}{4}(1+\xi)(1+\eta)(\xi+\eta-1); & N_4 &= \frac{1}{4}(1-\xi)(1+\eta)(-\xi+\eta-1) \\ N_5 &= \frac{1}{2}(1-\xi^2)(1-\eta); & N_6 &= \frac{1}{2}(1+\xi)(1-\eta^2) \\ N_7 &= \frac{1}{2}(1-\xi^2)(1+\eta); & N_8 &= \frac{1}{2}(1-\xi)(1-\eta^2) \end{aligned} \quad (4)$$

where, u , v and w represents the displacements of any point along the (x, y, z) coordinates. u_0 , v_0 are the in-plane and w_0 is the transverse displacements of the mid-plane and θ_x , θ_y are the

rotations of the normal to the mid plane about y and x axes respectively and θ_z is the higher order terms in Taylor's series expansion.

Strains are obtained by derivation of displacements as:

$$\{\boldsymbol{\varepsilon}\} = \left\{ u_{,x} \quad v_{,y} \quad w_{,z} \quad u_{,y} + v_{,x} \quad v_{,z} + w_{,y} \quad w_{,x} + u_{,z} \right\}^T \quad (5)$$

where, $\{\boldsymbol{\varepsilon}\} = \{\varepsilon_x \quad \varepsilon_y \quad \varepsilon_z \quad \gamma_{xy} \quad \gamma_{yz} \quad \gamma_{xz}\}^T$, is the normal and shear strain components of in plane and out of plane direction.

The strain components are now rearranged in the following steps.

The in-plane strain vector:

$$\begin{Bmatrix} \boldsymbol{\varepsilon}_x \\ \boldsymbol{\varepsilon}_y \\ \boldsymbol{\gamma}_{xy} \end{Bmatrix} = \begin{Bmatrix} \boldsymbol{\varepsilon}_{x_0} \\ \boldsymbol{\varepsilon}_{y_0} \\ \boldsymbol{\gamma}_{xy_0} \end{Bmatrix} + z \begin{Bmatrix} \boldsymbol{\kappa}_x \\ \boldsymbol{\kappa}_y \\ \boldsymbol{\kappa}_{xy} \end{Bmatrix} \quad (6)$$

The transverse strain vector:

$$\begin{Bmatrix} \boldsymbol{\varepsilon}_z \\ \boldsymbol{\gamma}_{yz} \\ \boldsymbol{\gamma}_{xz} \end{Bmatrix} = \begin{Bmatrix} \boldsymbol{\varepsilon}_{z_0} \\ \boldsymbol{\gamma}_{yz_0} \\ \boldsymbol{\gamma}_{xz_0} \end{Bmatrix} + z \begin{Bmatrix} \boldsymbol{\kappa}_z \\ \boldsymbol{\kappa}_{yz} \\ \boldsymbol{\kappa}_{xz} \end{Bmatrix} \quad (7)$$

where, the deformation components are described as:

$$\begin{Bmatrix} \boldsymbol{\varepsilon}_{x_0} \\ \boldsymbol{\varepsilon}_{y_0} \\ \boldsymbol{\gamma}_{xy_0} \end{Bmatrix} = \begin{Bmatrix} \frac{\partial u_0}{\partial x} \\ \frac{\partial v_0}{\partial y} \\ \frac{\partial u_0}{\partial y} + \frac{\partial v_0}{\partial x} \end{Bmatrix}; \quad \begin{Bmatrix} \boldsymbol{\kappa}_x \\ \boldsymbol{\kappa}_y \\ \boldsymbol{\kappa}_{xy} \end{Bmatrix} = \begin{Bmatrix} \frac{\partial \theta_x}{\partial x} \\ \frac{\partial \theta_y}{\partial y} \\ \frac{\partial \theta_x}{\partial y} + \frac{\partial \theta_y}{\partial x} \end{Bmatrix} \quad (8)$$

$$\begin{Bmatrix} \varepsilon_{z_0} \\ \gamma_{yz_0} \\ \gamma_{xz_0} \end{Bmatrix} = \begin{Bmatrix} \theta_z \\ \frac{\partial w_0}{\partial y} + \theta_y \\ \frac{\partial w_0}{\partial x} + \theta_x \end{Bmatrix}; \begin{Bmatrix} \kappa_z \\ \kappa_{yz} \\ \kappa_{xz} \end{Bmatrix} = \begin{Bmatrix} 0 \\ \frac{\partial \theta_z}{\partial y} \\ \frac{\partial \theta_z}{\partial x} \end{Bmatrix} \quad (9)$$

The strain vector expressed in terms of nodal displacement vector:

$$\{\varepsilon\} = [B]\{\delta\} \quad (10)$$

where, $[B]$ is the strain displacement matrix containing interpolation functions and their derivatives and $\{\delta\}$ is the nodal displacement vector.

For a laminate, the generalized stress strain relationship with respect to its reference plane may be expressed as:

$$\{\sigma\} = [D]\{\varepsilon\} \quad (11)$$

where, $\{\sigma\}$ and $\{\varepsilon\}$ is the stress and strain vector respectively and $[D]$ is the rigidity matrix.

The element stiffness matrix $[K]$ can be easily derived with the help of virtual work method which may be expressed as:

$$[K] = \int_{-1}^{+1} \int_{-1}^{+1} [B]^T [D] [B] |J| d\xi d\eta \quad (12)$$

where, $|J|$ is the determinant of the Jacobian matrix, $[N]$ is the shape function matrix and $[m]$ is the inertia matrix. The integration has been carried out using the Gaussian quadrature method.

The static analysis determines the deflections as:

$$[K]\{\delta\} = \{P\} \quad (13)$$

where, $\{P\}$ is the static load vector acting at the nodes.

For determining the buckling load in FEM, it is necessary to discretise the governing differential equations. In this analysis an eight noded shell element (SHELL 281) is used from the ANSYS library. The element geometry can be seen in the Figure 2. This element is suitable for analysing thin to moderately thick shell/plate structures. This element has six degrees of freedom at each node: translations in the x , y , and z axes, and rotations about the x , y , and z axes. It includes the effects of transverse shear deformation. The nodal displacements can be presented in the form of their shape functions and their corresponding nodal values as follows:

$$\begin{aligned}
u^0 &= \sum_{i=1}^8 N_i u_i^0 \quad v^0 = \sum_{i=1}^8 N_i v_i^0 \\
w^0 &= \sum_{i=1}^8 N_i w_i^0, \quad \theta_x = \sum_{i=1}^8 N_i \theta_{xi}, \\
\theta_y &= \sum_{i=1}^8 N_i \theta_{yi} \quad \text{and} \quad \theta_z = \sum_{i=1}^8 N_i \theta_{zi}
\end{aligned} \tag{14}$$

The above equation can be rewritten in i^{th} nodal displacement as follows:

$$\{\delta_i^*\} = \{u_i^0 \quad v_i^0 \quad w_i^0 \quad \theta_{xi}^0 \quad \theta_{yi}^0 \quad \theta_{zi}^0\} \tag{15}$$

$$\{\delta_i^*\} = [N_i]\{\delta_i\} \tag{16}$$

where, $[N_i] = \begin{bmatrix} N_i & 0 & 0 & 0 & 0 & 0 \\ 0 & N_i & 0 & 0 & 0 & 0 \\ 0 & 0 & N_i & 0 & 0 & 0 \\ 0 & 0 & 0 & N_i & 0 & 0 \\ 0 & 0 & 0 & 0 & N_i & 0 \\ 0 & 0 & 0 & 0 & 0 & N_i \end{bmatrix}_{i=1 \text{ to } 8}$ nodal shape function of a 8 noded element and

the values can be seen in Reddy (2005).

Substituting the value of nodal displacement in strain, strain energy we will get

$$\{\varepsilon\} = [B_i]\{\delta_i\} \tag{17}$$

$$U = \frac{1}{2} \int [B_i]^T \{\delta_i\}^T [D] [B_i] \{\delta_i\} dA - \{F\}_{mi} \tag{18}$$

where, $[B_i]$, and $\{F\}_m$ are the strain displacement relation matrix, mechanical force, respectively.

The final form of the equation can be obtained by minimizing the total potential energy (TPE) as follows:

$$\partial \Pi = 0 \tag{19}$$

where, Π is the total potential energy.

$$[K]\{\delta\} = \{F\}_m \tag{20}$$

where, $[M]$ and $[K]$ are the global mass and stiffness matrix.

The buckling equation can be obtained for the laminated shell/plate and conceded as

$$[K]\{\delta\} = \{F\}_m \tag{21}$$

In this study eigenvalue type of buckling has studied and the eigenvalue equation will be obtained by dropping the force vectors and considering their effects in geometry the new form of the Eq. (21) will be expressed as:

$$\{[K] + \lambda_{cr}[K_G]\}\{\delta\} = 0 \tag{22}$$

where, $[K_G]$ is the geometric stiffness matrix and λ_{cr} is the critical mechanical load at which the structure buckles.

5 Result and Discussion

Numerical examples are taken with various face sheet and core properties and are subjected to uniformly distributed load for static analysis and axial and bi-axial loading condition for buckling analysis. The proposed model is solved in the ANSYS 12.0 environment to find the static behavior and buckling behavior of sandwich plates. Some new results are obtained with different thickness ratios (a/h), lay up scheme, support conditions, aspect ratios (a/b) and loading type.

Validation and Convergence for Static Analysis

For the validation purpose a square based ($a/b=1$) symmetric cross-ply sandwich plate (0/90/C/90/0) is taken, where two-ply laminated stiff sheets at the two faces under uniformly distributed load of intensity q_0 . The square sandwich plate is analyzed with the thickness ratio ($a/h=10$) with all edges clamped condition. The thickness distribution is taken for core as $0.8h$ and thickness of face sheet is taken as $0.05h$ and the material properties are shown in Table 1.

Table 1: Material properties

Material	E_1	E_2	E_3	G_{12}	G_{13}	G_{23}	γ_{12}	γ_{13}	γ_{23}
Core	0.04E	0.04E	0.5E	0.016E	0.06E	0.06E	0.25	0.25	0.25
Face Sheet	25E	E	E	0.5E	0.5E	0.2E	0.25	0.25	0.25

The non-dimensional values of deflection $\hat{w} = 100wEh^3 / (q_0a^4)$ at the plate center obtained by using the present developed model and presented in Table 2. The results are compared with refined higher order shear deformation theory (RHSDT) by Chakrabarti and Sheikh (2005). For the computation purpose, a four noded shell element (SHELL 181) and an eight noded serendipity shell element (SHELL 281) is utilized from ANSYS element library. It can be easily understood that the present results are showing good convergence rate with refined mesh size and negligible difference. A mesh (18×18) is sufficient to obtain responses and it used for further computation.

Table 2. Non-dimensional center deflection of clamped sandwich square plate with $a/h= 10$ under uniformly distributed load for SHELL181

Mesh Size	SHELL181
12×12	1.5
14×14	1.499
16×16	1.498
Chakrabarti and Sheikh	1.474

Table 3. Non-dimensional center deflection of clamped sandwich square plate with $a/h= 10$ under uniformly distributed load for SHELL281

Mesh Size	SHELL281
2×2	1.492
4×4	1.494
6×6	1.496
Chakrabarti and Sheikh	1.474

It is observed from Table 2 that the present result obtained through ANSYS 12.0 shows a good agreement with the published literature. The element taken here is 4-noded SHELL181. However, it also showed compliance when the element is 8-noded SHELL281 as well. It suggests that there is no discontinuity in between the elements. New results are found by taking shell element 181. Fig. 3 shows the variation of deflection at the midpoint with thickness ratio (a/h) and this is compared for various sandwich constructions. From the Fig. 3 it can be observed that for $0^0/C/0^0$ sandwich plate the non-dimensional deflection value decreases with increasing thickness ratio. It is well known that, as the thickness ratio increases the structure becomes thinner. The aspect ratio is changed and observed to find that it gradually decreases for all but the value decreases as aspect ratio increases for a particular thickness ratio.

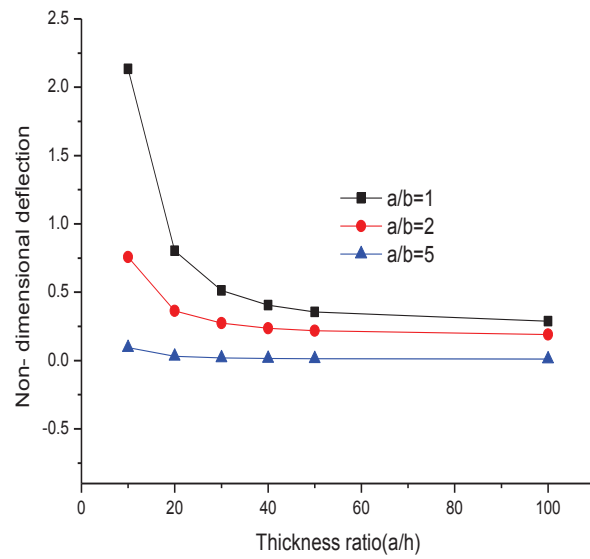


Fig. 3. Non-dimensional deflection V/S a/h for $0/C/0$

Fig. 4 shows the non-dimensional deflection values with thickness ratio (a/h) and similar trend can be seen in this case also as the thickness ratio increases the non-dimensional deflection decreases which is showing good result. The aspect ratio is changed and observed to find the same consequence for all. Also the value decreases as aspect ratio increases for a particular thickness ratio. Fig. 5 shows the sandwich plate having configuration as $(0^0/90^0)_5/C/(90^0/0^0)_5$ and in this condition also same trend can be seen. Hence it can be concluded that for any sandwich plate with any configuration of face sheets the deflection at the centre decreases as the thickness ratio is increased. Fig. 5 shows the variation of deflection at the midpoint with a/b and this is compared for various sandwich constructions.

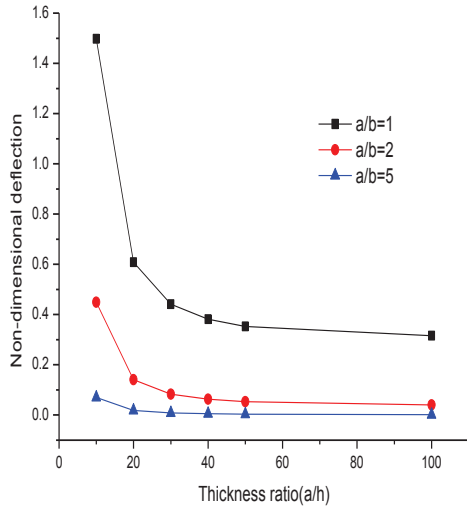


Fig. 4. Non-dimensional deflection V/S a/h for $0^0/90^0/C/90^0/0^0$

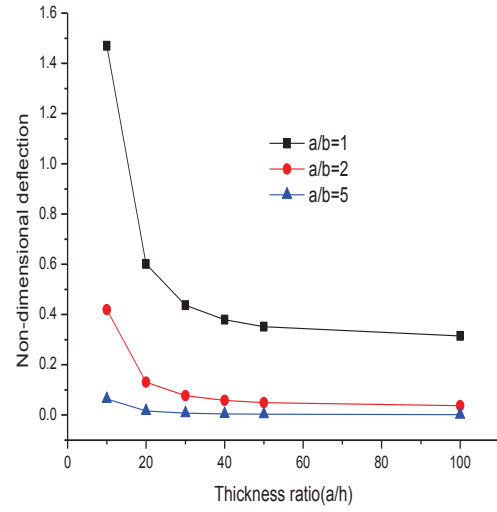


Fig. 5. Non-dimensional deflection v/s a/h for $(0^0/90^0)_5/C/(90^0/0^0)_5$

Validation and Convergence for Buckling Analysis

A simply supported square sandwich plate with face sheet thickness h and the stacking formation of $(0^0/90^0)_5/C/(90^0/0^0)_5$ is taken for the analysis. The face sheet contains 10 layers of equal thickness cross-ply layers on either side of the soft orthotropic core. The buckling analysis is performed under uniaxial loading for different thickness ratio ($a/h=10, 20, 30, 40, 50$ and 100).

The material properties of the face sheet are taken as:

$$E_1 = 19E, E_2 = E_3 = E, G_{12} = G_{13} = 0.52E, G_{23} = 0.338E, \nu_{12} = \nu_{13} = 0.32, \nu_{23} = 0.49$$

The material properties for core are taken as:

$$E_x = 3.2 \times 10^{-5}E, E_y = 2.9 \times 10^{-5}E, E_z = 0.4E, G_{xy} = 2.4 \times 10^{-3}E, G_{yz} = 6.6 \times 10^{-2}E, G_{xz} = 7.9 \times 10^{-2}E, \nu_{xy} = 0.99, \nu_{xz} = 3 \times 10^{-5}, \nu_{yz} = 3 \times 10^{-5}E$$

It can be seen that the present results for uniaxial buckling loads are in good agreement with improved higher order theory. The same sandwich arrangement $(0^0/90^0)_5/C/(90^0/0^0)_5$ with simply supported condition and under uniaxial loading with the thickness ratio ($a/h=10$) and $h_t/h=0.1$ is taken as the reference (Kheirikhah *et al.* [29]) and the maximum discrepancy is less than 2 percent which is negligible which is shown in Table 4.

Table 4. Convergence of Non-dimensional buckling load with simply supported condition

Mesh size	Buckling load
16x16	5.8043
17x17	5.7821
18x18	5.7616
19x19	5.7441
20x20	5.728
Kheirikhah	5.6215

Some new results are obtained by shell element 181 and 281 with various thickness ratio (a/h), aspect ratio (a/b) and face sheet layup scheme. The non-dimensional buckling load is found out by $\bar{N} = (N_0 a / E_2)$ if it is not stated elsewhere. The new results have been obtained for all the arrangement of face sheet and core by two different loading conditions as uniaxial and biaxial. Fig. 6 shows the non-dimensional buckling loads characteristics with the aspect ratios (a/b) with two different shell elements under uniaxial load with simply supported condition. Fig 7 shows the non-dimensional buckling loads characteristics with different thickness ratios (a/h) under simply supported condition and uniaxial loading. It can be seen from figure that as the aspect ratio increases the non-dimensional buckling loads also decreases because the geometry of the structure changes. From all the figures it is clear that as the thickness ratios (a/h) increases the non-dimensional buckling loads also increases in both the cases of uniaxial loading and biaxial loading. It is well known that as the thickness ratio increases the buckling loads has to decrease but it is not true for small strain and large deformation regime.

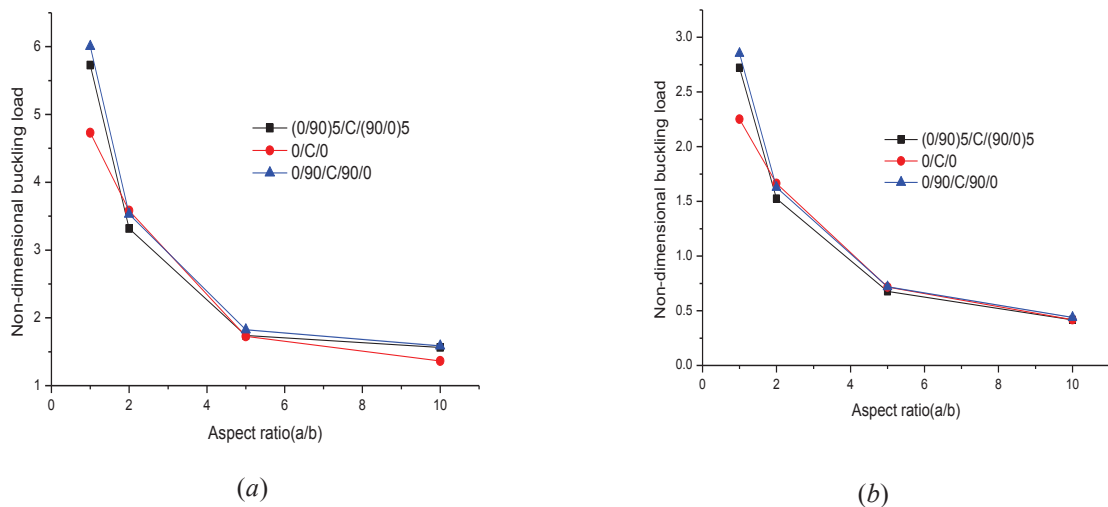
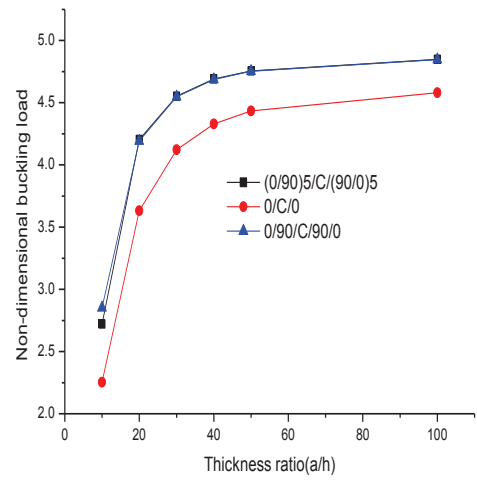
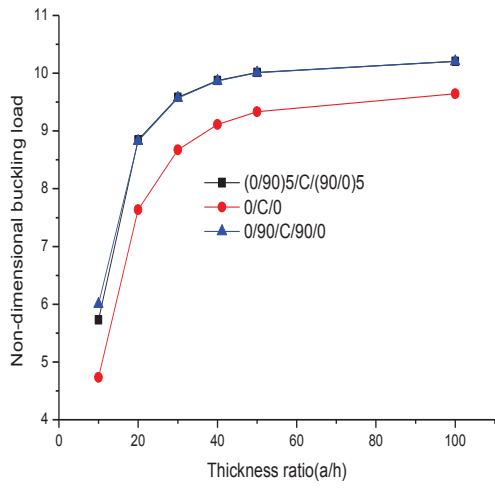


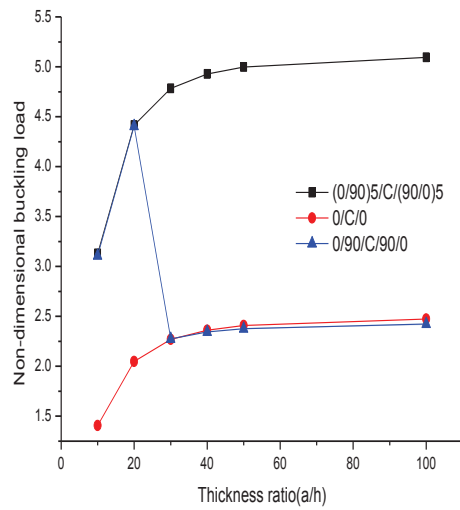
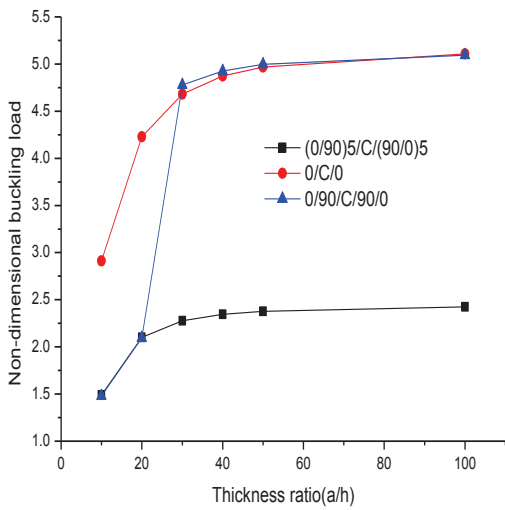
Fig. 6. Non-dimensional buckling load for thickness ratio ($a/h=10$) under uniaxial loading (a) SHELL181 (b) SHELL281



(a)

(b)

Fig. 7. Non-dimensional buckling load for aspect ratio ($a/b=1$) under uniaxial loading (a) SHELL181 (b) SHELL281



(a)

(b)

Fig. 8. Non-dimensional buckling load for aspect ratio ($a/b=1$) under biaxial loading (a) SHELL181 (b) SHELL281

6 Conclusions

In this present work, a finite element model is being developed in ANSYS using APDL code and its comprehensive testing has been done. In this work ANSYS was used to analyze static deflection and buckling load for sandwich plates with different aspect ratios and thickness ratios. The convergence study is clearly showing that the present developed model is capable to solve different static and buckling problems with good accuracy with less computational cost. For the static analysis a uniformly distributed load is taken and center deflection is obtained with clamped condition and for buckling, simply supported conditions under uniaxial and biaxial loadings are computed. The static deflection behavior of sandwich plate has been obtained for various thickness ratios (a/h), layup of face sheet, under distributed loading. It is observed that as the thickness ratio (a/h) increases the center deflection decreases. The buckling analysis of sandwich plates has been carried out for different parameters to check the efficacy of the developed model. The effect of different thickness ratios (a/h), stacking sequences of face sheets and loading condition on the buckling behavior of the sandwich plates has been obtained. It is seen that, the values of the non-dimensional critical buckling load value increases as thickness ratio increases. It is well known that as the thickness ratio increases the buckling loads has to decrease but it is not true for small strain and large deformation regime.

References

- [1] Pokharel N. and Mahendran M., 2003, "Experimental investigation and design of sandwich panels subject to local buckling effects," *Journal of Constructional Steel Research*, 59, pp. 1533-1552.
- [2] Carrea E., 1998, "A refined multilayered finite-element model applied to linear and non-linear analysis of sandwich plates," *Composite Science and Technology*, 58, pp. 1553- 1569.
- [3] Zhen Wu, Wanji Chen and Xiaohui Ren, 2010, "An accurate high order theory and C finite element for free vibration analysis of laminated composite and sandwich plates," *Composite Structures*, 92, pp. 1299-1307.
- [4] Pandit M. K., Sheikh A. H. and Singh B. N., 2008, "An improved high order zig-zag theory for the statistical analysis of laminated sandwich plate with soft core," *Finite Elements in Analysis and Design*, 44, pp. 602-610.
- [5] Kant T. and Swaminathan K., 2002, "Analytical solutions for the static analysis of laminated composite and sandwich plates based on higher order refined theory," *Composite Structures*, 56, pp. 329-344.
- [6] Ramtekkar G.S., Desai Y.M. and Shah A.H, 2003, "Application of three dimensional mixed finite element model to the flexure of sandwich plate," *Computers and Structures*, 81, pp. 2183-2198.
- [7] Pokharel N. and Mahendran M., 2004, "Finite element analysis and design of sandwich panels subject to local buckling effects," *Thin Walled Structures*, 42, pp. 589-611.
- [8] Palazotto A.N., Herup E.J. and Gummadi L.N.B, 2000, "Finite element of low velocity impact on composite sandwich plates," *Composite Structures*, 49, pp.209-227.
- [9] Yeh Jia-Yi and Chen Lien-Wen, 2007, "Finite element dynamic analysis of orthotropic sandwich plates with an electrorheological fluid core layer," *Composite Structures*, 78, pp. 368-376.
- [10] Malekzadeh K., Khalili M.R., Olsson R. and Jafari A., 2006, "Higher order dynamic response of composite sandwich panels with flexible core under simultaneous low velocity impacts of multiple small masses," *International Journal of Solids and Structures*, 43, pp. 6667-6687.

- [11] Ivanez I., Santiuste C., Barbero E. and Sanchez-Saez S., 2011, "Numerical modeling of foam cored sandwich plates under high-velocity impact," *Composite Structures*, 93, pp. 2392-2399.
- [12] Skvortsov V., Bozhevolnaya E. and Kildegaard A., 2000, "Overall behavior of singly curved shallow sandwich panels: the case of general boundary conditions," *Composite Structures*, 49, pp. 95-109.
- [13] Pandit M.K., Singh B.N. and Sheikh A.H., 2009, "Stochastic perturbation-based finite element for deflection statistics of soft core sandwich plate with random material properties," *International Journal of Mechanical Sciences*, 51, pp. 363-371.
- [14] Skvortsov V. and Bozhevolnaya E., 2001, "Two-dimensional analysis of shallow sandwich panels," *Composite Structures*, 53, pp. 43-53.
- [15] Pandya B. N. and Kant T., 1988, "Higher order shear deformable theories for flexure of sandwich plates-finite element evaluations," *Solid Structures*, 24, pp. 1267-1286.
- [16] Kant T., Arora C. P. and Varaiya J. H., 1992, "Finite element transient analysis of composite and sandwich plates based on a refined theory and a mode superposition method," *Composite Structures*, 22, pp. 109-120.
- [17] Parton G. M. and Shendy-EI-Barbary M. E., 1983, "A finite element analysis for cement composite sandwich plates," *International Journal of Cement Composites and Light Weight Concrete*, 5, pp. 181-191.
- [18] Kant T., Ravichandran R. V., Pandya B. N. and Mallikarjuna, 1988, "Finite element transient dynamic analysis of isotropic and fiber reinforced composite plates using higher-order theory," *Composite Structures*, 9, pp. 319-342.
- [19] Pandya B. N. and Kant T., 1988, "Finite element analysis of laminated composite plates using a higher order displacement model," *Composite Science and Technology*, 32, pp. 137-155.
- [20] Marur S. R. and Kant T., Free vibration analysis of fibre reinforced composite beams using higher order theories and finite element modeling, *Journal of Sound and Vibration* 194(1996), 337-351.
- [21] Flores-Johnson E. A. and Li Q. M., 2012, "Structural behaviour of composite sandwich panels with plain and fibre-reinforced foamed concrete cores and corrugated steel faces," *Composite Structures*, 94, pp. 1555-1563.
- [22] Chakrabarty A. and Sheikh A. H., 2005, "Analysis of Laminated Sandwich Plates Based on Interlaminar Shear Stress Continuous Plate Theory", *Journal of Engineering Mechanics*, 131, pp. 377-384.

- [23] Singh S. K., Chakrabarti A., Bera P. and Sony J. S. D., 2011, "An efficient C0 FE model for the analysis of composites and sandwich laminates with general lay-up," *Latin American Journal of Solids and Structures*, 8, pp. 197-212.
- [24] Chakrabarti A. , Chalak H. D. , Iqbal Md. A. and Sheikh A. H. , 2012, "Buckling analysis of laminated sandwich beam with soft core," *Latin American Journal of Solids and Structures*, 1, pp. 1-15.
- [25] Zhen W. and Wanji C., 2008, "An assessment of several displacement-based theories for the vibration and stability analysis of laminated composite and sandwich beams," *Composite Structures*, 84, pp. 337-349.
- [26] Khdeir A. A. and Librescu L., 1988, "Analysis of symmetric cross-ply laminated elastic plates using a higher order theory- part II-buckling and free vibration," *Composite Structures*, 9, pp. 259-277.
- [27] Kant T. and Swaminathan K., 2001, "Analytical solutions for free vibration of laminated composites and sandwich plates based on a higher order refined theory," *Composite Structures*, 53, pp. 73-85.
- [28] C'etkovic' C. and Vuksanovi Dj., 2009, "Bending, free vibrations and buckling of laminated composite and sandwich plates using a layerwise displacement model," *Composite structures*, 88, pp. 219-227.
- [29] Kheirikhah M M, Khalili S.R.M and Malekzadeh Fard K., 2012, "Biaxial buckling analysis of soft-core composite sandwich plates using improved high-order theory," *European Journal of Mechanics and A/solids* 31, pp. 54-66.
- [30] Khare R. K., Kant T. and Garg A. K., 2004, "Free vibration of composite and sandwich laminates with a higher-order face shell element," *Composite Structures*, 65, pp. 405-418.
- [31] Reddy J. N., 2005. "An Introduction to the Finite Element Method", McGraw-Hill Companies, Inc.
- [32] ANSYS Inc., "ANSYS 12.0 reference manual", 2009.