Free Vibration Analysis of Curved Panels

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

Bachelor of Technology

In

Mechanical Engineering

by

Sandeep Mishra

Roll No.: 110ME0308

Under the supervision of

Prof. Subrata Kumar Panda

Department Of Mechanical Engineering

National Institute Of Technology Rourkela

Rourkela-769008

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C E R T I F I C A T E

This is to certify that the work in this thesis entitled **"*Free Vibration Analysis of Curved Panels*"** by ***Sandeep Mishra (Roll No.: 110ME0308)*** , 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 2013-2014 in the ***Department of Mechanical Engineering, National Institute of Technology, Rourkela*****.**

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

Date:

Place: Rourkela

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Date: 04/05/2014 National Institute of Technology,Rourkela

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Finally, I would also like to appreciate **Mr. Deepro , M. Tech, Department of Mechanical Engineering, NIT Rourkela** for his kind support in helping me to gather all the requisite knowledge related to shells and panels that provided me a thorough insight into this project.

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CONTENTS

Abstract

In the present work, vibration behaviours of cylindrical and/or conical curved laminated /isotropic panel structures have been investigated. The models of cylindrical and conical shell panel have been developed based on first order shear deformation theory (FSDT) using the ANSYS parametric design language (APDL) code in ANSYS 13 environment. The model has been then discretised using a 8 noded isoparametric element (SHELL 281) from ANSYS 13 element library. The model has also been validated by with comparing the its vibration responses with the results of a known literature and the convergence behavior of the present simulation model has also been obtained. Finally, substantial effect of various parameters on the frequency of vibration such as fibre orientation in various layers and number of layers in case of the composite cylindrical shell panel and element size and mode number in case of isotropic/laminated, conical/cylindrical shell panel has been investigated.

Keyword: Cylindrical Panel, Conical Panel, Isotropic/Orthotropic Shell, Vibration, FEM, ANSYS

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Table 2 – Comparison study between present results and that of the reference [8] for conical shell panel

Chapter 1: Introduction

The term ‘curved panel’ basically refers to a geometry bounded by two curved surfaces, where the distance between the curved surfaces is quite less in comparison with other body dimensions as shown in Fig.1 . It has a mid-plane , considered as the reference plane which is the locus of those points that are equidistant from the two curved surfaces . Its thickness ‘h’ is defined as the length of the meridonial axis which is perpendicular to both the curved surfaces. For most panels, 100≤ h/R ≤ 200 where ‘R’ is the radius of curvature.

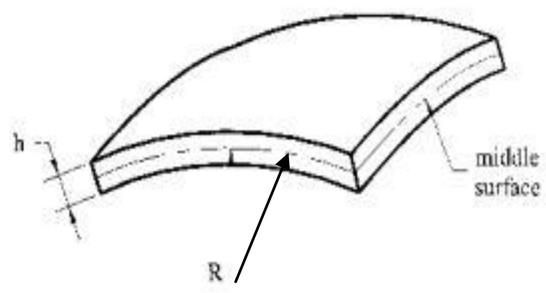


Fig.1 : General geometry of a shell panel [10]

The curved panels possess very unique characteristics like efficiency of load-carrying capacity, high strength to weight ratio, very high stiffness, high degree of reserved strength , structural integrity and containment of space. Owing to these excellent characteristics, today the curved panels (especially the cyindrical and the conical panels) play a vital role in various engineering applications. The well known applications of panel structures in mechanical engineering include piping systems, turbine disks, and pressure vessels. In civil and architectural engineering, they constitute an important part of large-span roofs, liquid-retaining structures and water tanks,. Their utility in aerospace and marine engineering such as in aircrafts, rockets, ships and submarines is simply unmatched.

As the curved panels are frequently subjected to vibration in a wide range of applications, the study of their vibration behaviour is essential **.** So, in the present work, vibration behaviour of cylindrical and conical shell panels have been studied and analysed by developing an APDL code in the commercially available software ANSYS 13. A brief overview on ANSYS software has been discussed in the following paragraph and the methodologies involved have also been discussed.

ANSYS Software – An Overview

Presently, the design and analysis process in every field has become very precise. So the use of finite element method (FEM) has become quite extensive. Thus, ANSYS software which works on FEM has gained widespread reputation as the most innovative ,accurate and fastest tool of design and analysis.

In the current work, a 8-noded isoparametric element viz; SHELL281, as shown in Fig.2, has been used for the study of free vibration behaviour of both the cylindrical as well as conical panel. It is a linear shell element with six degrees of freedom (translations in x, y and z axes and rotations about x, y and z axes) at each node. It is used to analyse shell structures having a wide range of thickness and even whose thickness changes during non-linear analysis. Owing to its plasticity, this element can also be used in the structural analysis of laminated composite and sandwiched structures. In all these analysis processes, this element is formulated using logarithmic strain and true stress measurements based on first order shear deformation theory(FSDT).

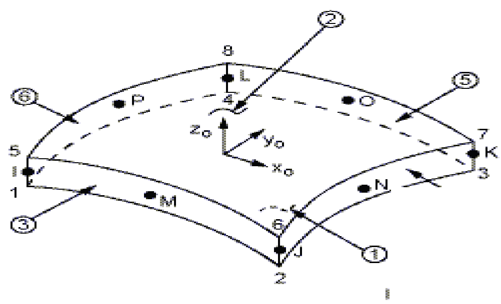


Fig. 2 : Geometry of an isoparametric Serendipity eight noded element [11]

x_0 = x-axis of the element if its orientation is not provided

x = x-axis of the element if its orientation is provided

Literature Review

In view of the widespread utilities and unique characteristic properties of curved panels, many shell theories have been developed till today, most of which are based on linear elasticity concepts just because these theories help to predict accurately the stresses and extremely small deformations that occur during vibration of the curved panels. In most of these theories, the curved panel is treated as a 2 D body in lieu of the cumbersome calculations involved in the analysis of a 3 D curved panel.

A well known theory was developed by Love [1] which is a first-order approximation shell theory of thin elastic shells based on classical linear elasticity. In this theory, Kirchhoff's plate theory hypotheses was applied to simplify the strain–displacement relationships and the constitutive relations. Sanders [2] developed another first-order-approximation shell theory based on the principle of virtual work by incorporating the Kirchhoff–Love assumptions. Timoshenko [3] put forth another theory of thin elastic shells where the general relations and equations were derived by applying the Kirchhoff–Love hypotheses. Lurye [4], Flugge [5], and Byrne [6] developed a second-order approximation theory of shells by applying Kirchhoff hypotheses and small-deflection assumption to the corresponding equations of the three-dimensional theory of elasticity. Novozhilov [7] developed a new second-order approximation theory of shells wherein the strain–displacement relations were derived by applying Kirchhoff's hypotheses to the three-dimensional theory of elasticity. Zhao and Liew [8] presented the vibration behaviour of functionally graded conical panels using element-free *kp*- Ritz method. The mathematical model was developed based on first order shear deformation shell theory in order to account for the transverse shear strains and rotary inertia and mesh-free kernel particle functions were employed to approximate the two-dimensional displacement fields. Nanda and Bandyopadhyay [9] investigated linear free vibration behaviour of laminated composite cylindrical panel with and without cutouts.

The objective of the present work is to study the vibration responses of laminated cylindrical and conical shell panels. A computer code has been developed in APDL for isotropic/ orthotropic shell panels. The model has been discretised using an eight noded isoparametric Serendipity element (SHELL281), available in ANSYS library. The free vibration response of the present model is obtained using Block Lanczos method. Further, the effects of different structural parameters namely the lamination schemes and the mode numbers on the vibration behaviour of cylindrical and conical shell panels have to be examined so that these parameters can be modified in future works to yield desired vibration responses.

Chapter 2 : Mathematical Formulation

The modelling of shell panels (both cylindrical and conical) is done using SHELL 281 element which is selected from ANSYS 13 element library . This is a 8 noded isoparametric element having six degrees of freedom (three translation about x,y and z axis and three rotations about x,y and z axis) at each node.The dispacement field based on FSDT is assumed as per [12] as follows :

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

where u,v and w denote the displacements of any point on the panel along (x,y,z) coordinate axes. In equation 1 , u_0 and v_0 are the in-plane and w_0 is the transverse displacements of the mid-plane , θ_x and 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.

The displacements u,v and w can be expressed in terms of shape functions (N_i) as [12] :

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

where $\delta_i = [\ u_{0i}\ \ v_{0i}\ \ w_{0i}\ \ \theta_{xi}\ \ \theta_{yi}\ \ \theta_{zi}\ ]^T$

The shape functions for the 8 noded(j=8) shell element (SHELL 281) are given by [12]:

$$\left.\begin{aligned} N_1 &= 1/4\times(1-\zeta)(1-\eta)(-\zeta-\eta-1); \quad & N_2 &= 1/4\times(1+\zeta)(1-\eta)(\zeta-\eta-1) \\ N_3 &= 1/4\times(1+\zeta)(1+\eta)(\zeta+\eta-1); \quad & N_4 &= 1/4\times(1-\zeta)(1+\eta)(-\zeta+\eta-1) \\ N_5 &= 1/2\times(1-\zeta^2)(1-\eta); \quad & N_6 &= 1/2\times(1+\zeta)(1-\eta^2) \\ N_7 &= 1/2\times(1-\zeta^2)(1+\eta); \quad & N_8 &= 1/2\times(1-\zeta)(1-\eta^2) \end{aligned}\right\} (3)$$

Strains are obtained by derivation of displacements as [12] :

$$\{\ \varepsilon\ \} = \{u_{,x}\ \ v_{,y}\ \ w_{,z}\ \ u_{,y} + v_{,x}\ \ v_{,z} + w_{,y}\ \ w_{,x}+u_{,z}\}^T \ldots\ldots(4)$$

where $\{\varepsilon\} = \{\ \varepsilon_x\ ,\ \varepsilon_y\ ,\ \varepsilon_z\ ,\ \gamma_{xy}\ ,\ \gamma_{yz}\ ,\ \gamma_{zx}\ \}^T$ is the strain matrix containing normal and shear strain components of the mid-plane in in-plane and out of plane direction.

The strain components are now rearranged in the following steps.

The in-plane strain vector is given by [13] :

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} \varepsilon_{x0} \\ \varepsilon_{y0} \\ \gamma_{xy0} \end{Bmatrix} + z \begin{Bmatrix} \kappa_x \\ \kappa_y \\ \kappa_{xy} \end{Bmatrix} \quad \text{.....(5)}$$

The transverse strain vector is given by [13] :

$$\begin{Bmatrix} \varepsilon_z \\ \gamma_{yz} \\ \gamma_{xz} \end{Bmatrix} = \begin{Bmatrix} \varepsilon_{z0} \\ \gamma_{yz0} \\ \gamma_{xz0} \end{Bmatrix} + z \begin{Bmatrix} \kappa_z \\ \kappa_{yz} \\ \kappa_{xz} \end{Bmatrix} \quad \text{.....(6)}$$

where the deformation components are given by [13] :

$$\begin{Bmatrix} \varepsilon_{xo} \\ \varepsilon_{y0} \\ \gamma_{xy0} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial u_0}{\partial x} \\ \dfrac{\partial v_0}{\partial y} \\ \dfrac{\partial u_0}{\partial y} + \dfrac{\partial v_0}{\partial x} \end{Bmatrix}; \begin{Bmatrix} \kappa_z \\ \kappa_{yz} \\ \kappa_{xz} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial \theta_x}{\partial x} \\ \dfrac{\partial \theta_y}{\partial y} \\ \dfrac{\partial \theta_x}{\partial y} + \dfrac{\partial \theta_y}{\partial x} \end{Bmatrix} \quad \text{.....(7)}$$

$$\begin{Bmatrix} \varepsilon_{zo} \\ \gamma_{yz0} \\ \gamma_{xz0} \end{Bmatrix} = \begin{Bmatrix} \theta_z \\ \dfrac{\partial w_0}{\partial y} + \theta_y \\ \dfrac{\partial w_0}{\partial x} + \theta_x \end{Bmatrix}; \begin{Bmatrix} \kappa_z \\ \kappa_{yz} \\ \kappa_{xz} \end{Bmatrix} = \begin{Bmatrix} 0 \\ \dfrac{\partial \theta_z}{\partial y} \\ \dfrac{\partial \theta_z}{\partial x} \end{Bmatrix} \quad \text{.....(8)}$$

The strain vector in terms of the nodal displacement vector is given by :

$$\{\varepsilon\}=[B]\{\delta\} \quad \text{.....(9)}$$

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

The generalised stress-strain relation with respect to the reference plane is given by :

$$\{\sigma\}= [D]\{\varepsilon\} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

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

The element stiffness matrix [K] can be easily derived with the help of virtual work method which is expressed as [13] :

$$[K] = \int_{-1}^{+1}\int_{-1}^{+1} [B]^T [D] [B]|J|d\zeta d\eta \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(11)$$

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 total strain energy of the curved panel is given by [14] :

$$U= ½ \int_v\{\varepsilon\}\{\sigma\}dV \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(13)$$

Substituting the values of $\{\varepsilon\}$and $\{\sigma\}$ from Eq.(9) and Eq.(10) respectively, the total strain energy of the panel becomes

$$U = ½ \int_v [B]^T\{\delta\}^T[D][B]\{\delta\}dV \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(14)$$

The kinetic energy of the shell panel is given by [14] :

$$T = ½ \int_v \rho\{\delta\}^T\{\delta\}dV \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(15)$$

where ‘ρ’ is the density and δ is the first order differential with respect to time.

The final form of governing equation for the free vibrated shell panel is obtained by using Hamilton’s principle given by :

$$\delta \int_{t_1}^{t_2} (T – U)\, dt = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(16)$$

The final expression for free vibration of shell panel is obtainedusing Eqns. (14)-(16) and conceded as

$$([K] – \omega_n^2 [M]) \{\delta\} = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(17)$$

where [K] and [M] are computed from Eqn.(11) and Eqn.(12) respectively.

Methodology adopted

In this section, the methodology involved in the development of simulation models of the cylindrical shell panel and the conical shell panel have been discussed. As the first step, the geometry of the cylindrical / conical panel with the desired material properties is created in the ANSYS 13.0 environment using APDL code. The model is then discredited into the required mesh configuration using the eight noded isoparametric Serendipity element (SHELL281) from the ANSYS element library. Further, the model is subjected to desired boundary conditions in order to constrain its degrees of freedom at all the nodal position of the edges. Finally, free vibration analysis of the present model is carried out by Block Lanczos method in ANSYS 13.0 environment based on inbuilt FSDT. The present model is examined for convergence and then the present results are compared with the available literatures for validation purpose.

Numerical results and validation

Cylindrical shell panel

A general composite cylindrical shell panel model is shown in Fig. 3. The same panel (Fig. 4) is modelled using the APDL code of ANSYS software with the dimensions : A = B = 1mm, h = 0.01mm , R_X = 3mm,$R_Y = \infty$. Its material properties are defined as [9] : $E_{XX} = 40\times10^3$ N/mm^2 , $E_{YY} = E_{ZZ} = 10^3$ N/mm^2, $\rho = 10^{-9}$ kg/mm^3 , $G_{XY} = G_{YZ} = G_{XZ} = 0.5\times10^3$ N/mm^2, $\nu_{XY} = \nu_{YZ} = \nu_{ZX}$ = 0.25.

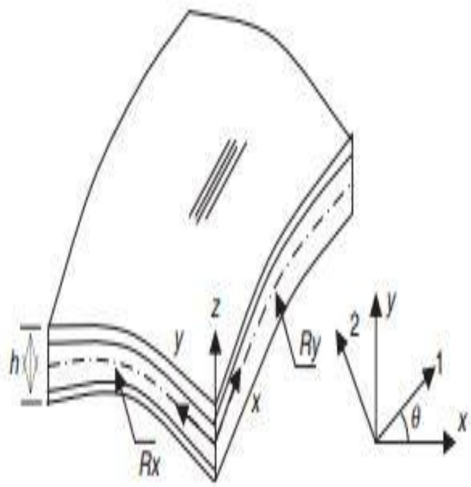


Fig. 3: Geometry of the composite cylindrical shell panel [9]

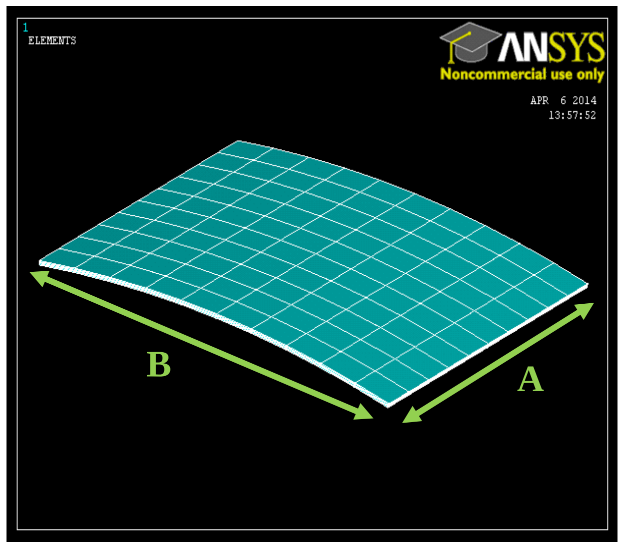


Fig. 4: ANSYS model of the cylindrical shell panel

The panel model is developed into four different (symmetric as well as anti-symmetric) lamination schemes : ($[0^0/90^0]$, $[0^0/90^0/0^0]$, $[0^0/90^0]_s$ and $[0^0/90^0]_2$) each of which is used for computation of vibration response. Fig. 5 presents one such lamination scheme ($[0^0/90^0]_s$) for reference.

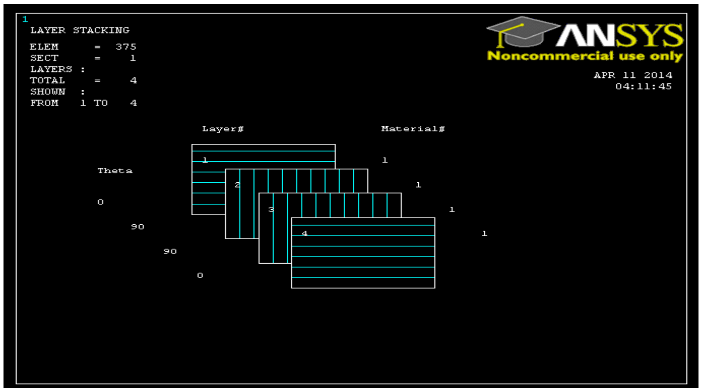


Fig.5 : Lamination scheme of $[0^0/90^0/90^0/0^0]$ configuration

The model of the panel is dicretised into four mesh configurations (10×10,14×14, 18×18 and 22×22) using the eight noded isoparametric Serendipity element SHELL281. Simply supported boundary conditions [9] given by the following constraints are applied at all the four edges of the panel.

$$v = w = \theta_Y = 0 \text{ at } x = 0,\ x = a$$

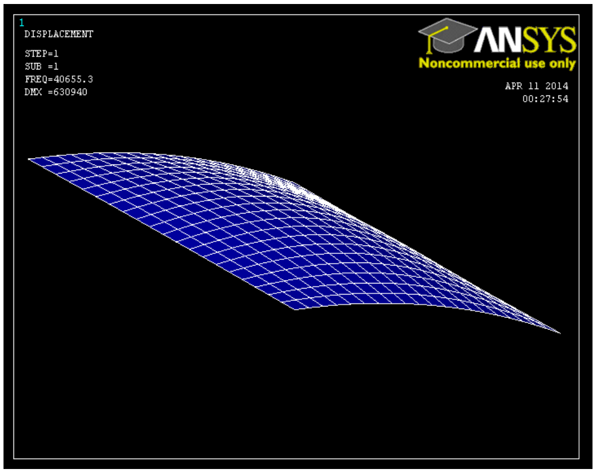
$$u = w = \theta_X = 0 \text{ at } y = 0,\ y = b$$

Free vibration analysis of the panel is done by Block Lanczos method using the APDL code of ANSYS 13.0 software and the results (frequency 'f' values) are used to compute the non-dimensional frequency parameters 'ω_l' given by $\omega_l = \omega A^2\ (\rho/E_{YY}\ h^2)^{0.5}$ where $\omega = 2\pi \times f$. The values of 'ω_l' are presented in table 1 and compared with reference [9]. It is clear that the present results are showing good agreement with that of the referred literature.

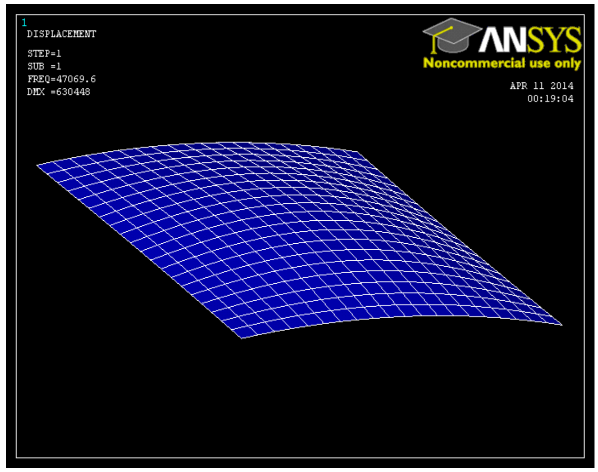
Table 1 : Comparison study between current results and that of the reference [9]

Lay-up	Present (ω_l)	Reference (ω_l) [9]	Percentage of difference
$[0^0/90^0]$	25.5445	25.4511	0.36
$[0^0/90^0/0^0]$	29.5747	29.3565	0.74
$[0^0/90^0/0^0/90^0]$	28.6644	28.5362	0.45
$[0^0/90^0/90^0/0^0]$	29.6284	29.4214	0.70

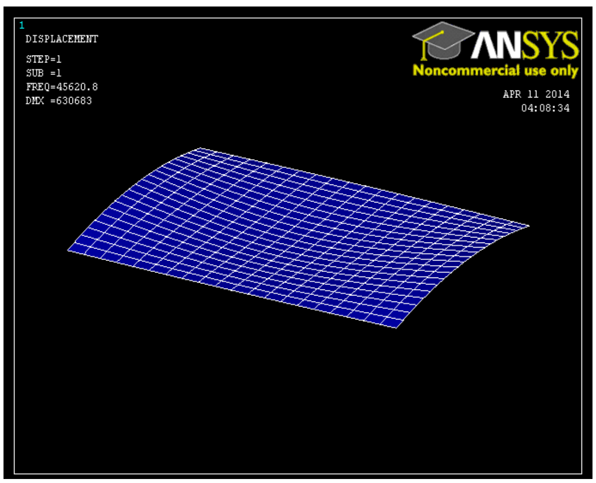
The deformed shapes for all the four lamination schemes are shown in Fig.6.



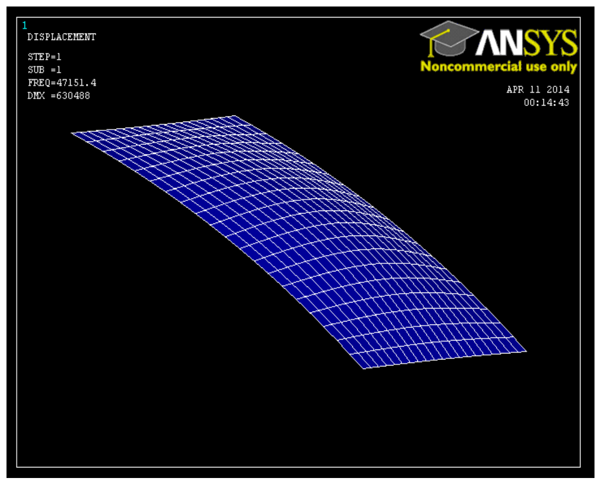
(a)



(b)



(c)



(d)

Fig.6 (a), (b) , (c) and (d) : Deformed shapes for first mode of vibration for four different lamination schemes ($[0^0/90^0]$, $[0^0/90^0/0^0]$, $[0^0/90^0]_s$ and $[0^0/90^0]_2$).

Conical shell panel

A general model of the isotropic conical shell panel is shown in Fig. 7. The same panel (Fig. 8) is modelled using the APDL code of ANSYS software with the dimensions : L = 300mm , s = 500mm , h = 3mm , R_1 = 100mm , R_2 =250mm , $\alpha = 30^0$, $\theta_0 = 60^0$. Its material properties are defined as [1] : $E = 70\times10^3$ N/mm^2 , $\nu = 0.3$, $\rho = 2707\times10^{-9}$ kg/mm^3.

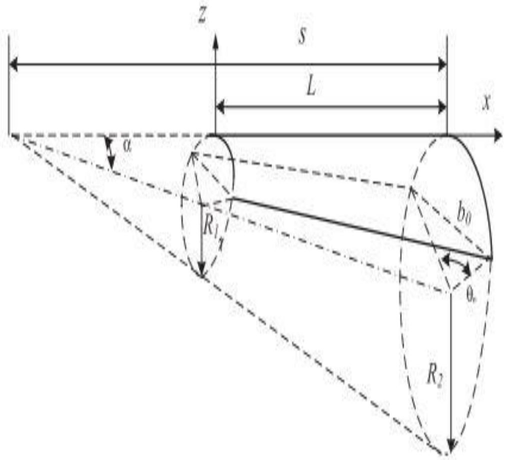


Fig. 7: Geometry of the isotropic conical shell panel [8]

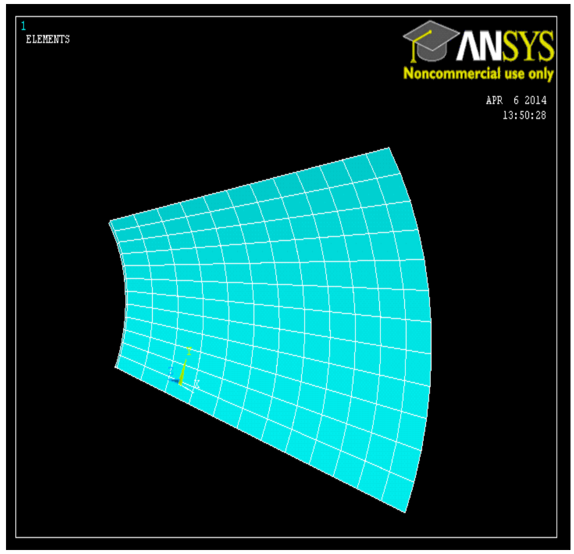


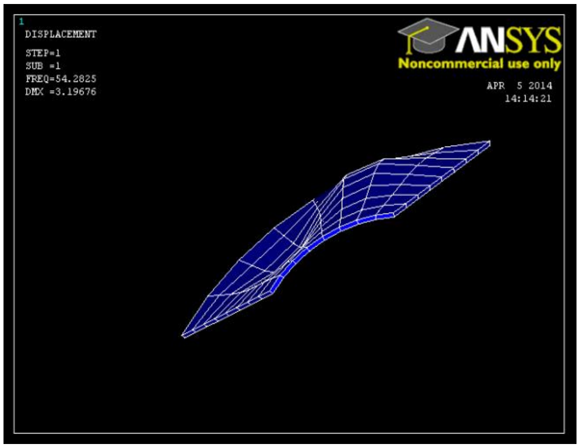
Fig. 8 : ANSYS model of the conical shell panel

The model of the panel is dicretised into four mesh configurations (8×8,10×10, 14×14 and 18×18) using the eight noded isoparametric Serendipity element SHELL281. Then, clamped boundary conditions [8] are applied at four edges of the panel. Free vibration analysis of the panel is done by Block Lanczoz method using the APDL code of ANSYS 13.0 software and results(frequency 'f' values) are used to compute the non-dimensional frequency parameters '$\tilde{\omega}$' given by $\tilde{\omega} = \omega L^2 \times (\rho h/D)^{0.5}$ where $D = Eh^3/12(1\text{-}\nu^2)$ and $\omega = 2\pi \times f$. The values of '$\tilde{\omega}$' are presented in table 2 and and compared with reference [8]. It is clear that the present results are showing good agreement with that of the referred literature.

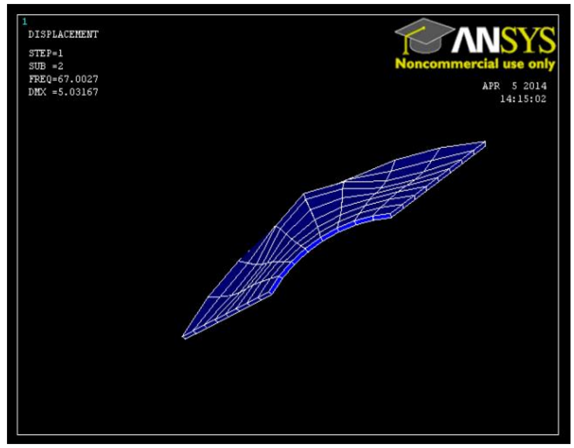
Table 2 : Comparison study between current results and that of the reference [8]

Mode number	Present ($\tilde{\omega}$)	Reference ($\tilde{\omega}$) [8]	Percentage of difference
1	210.263	206.75	1.7
2	259.538	251.51	3.1
3	310.173	300.70	3.1
4	356.199	344.84	3.2

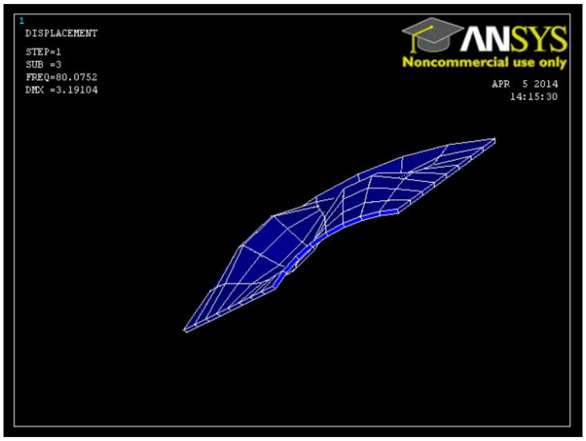
The deformed shapes for all the four modes of vibration are shown in Fig.9.



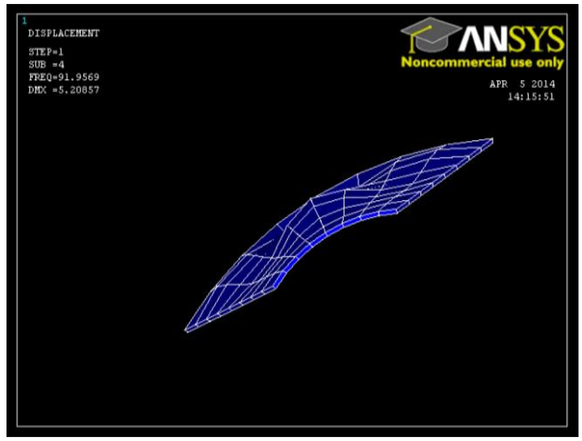
(a)



(b)



(c)



(d)

Fig. 9 (a),(b),(c) and (d) : Deformed shapes for first four modes of vibration of the conical shell panel

Convergence studies

Cylindrical shell panel

The current results 'ω_l' obtained for the lamination scheme [$0^0/90^0/90^0/0^0$] are tested for convergence. The convergence results are shown in the plot in Fig. 10

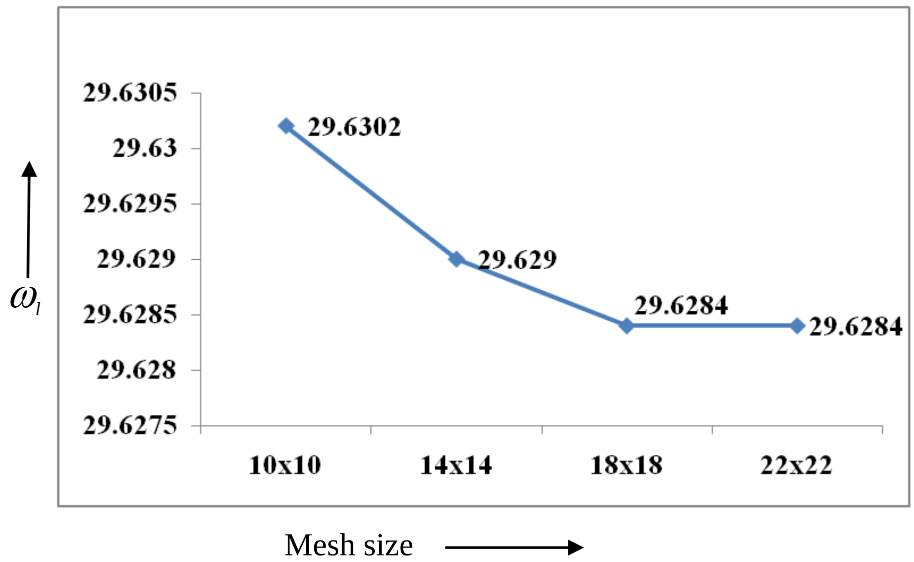
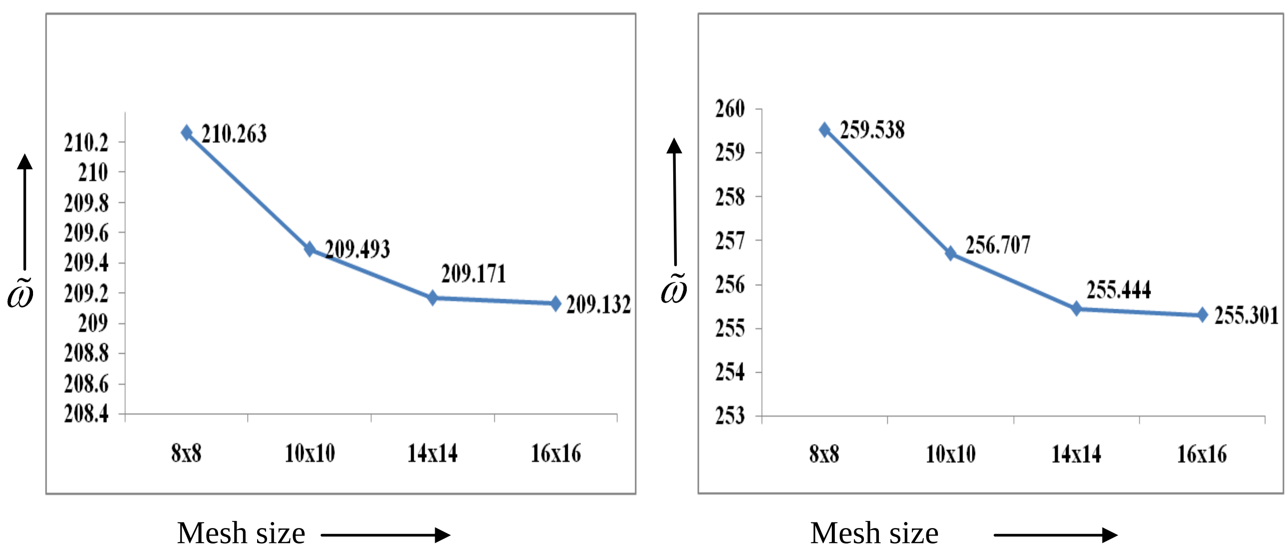


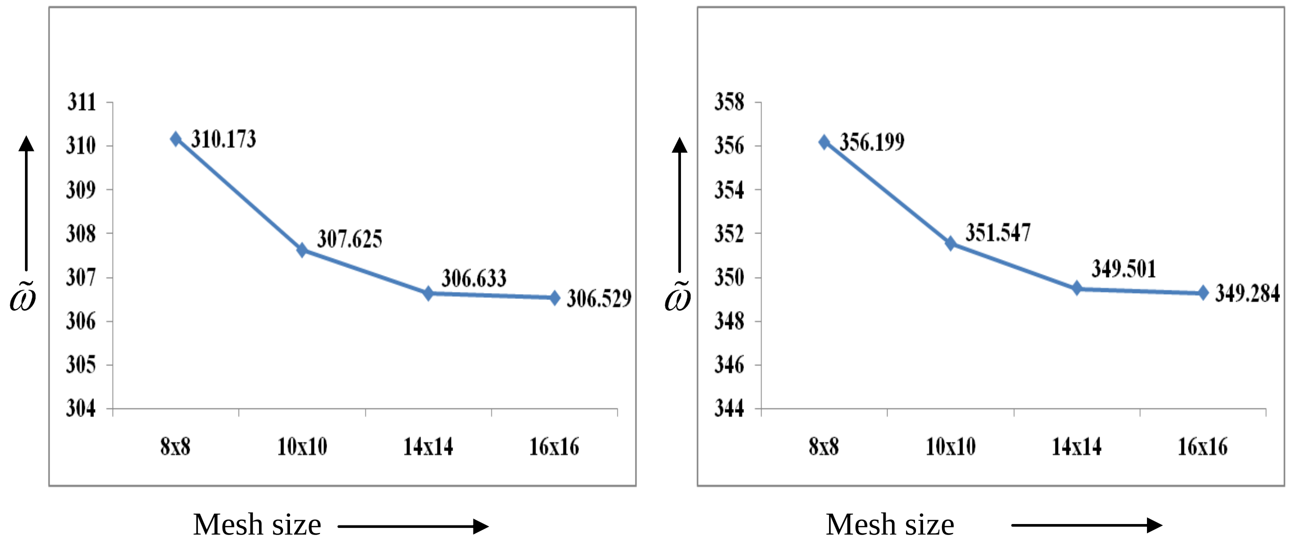
Fig. 10 : Convergence study plot for the cylindrical shell panel

<u>Conical shell panel</u>

The present results '$\tilde{\omega}$' for all the four modes are tested for convergence. The convergence results for all the four modes are shown by four plots in Fig. 11.



(a) (b)



(c) (d)

Fig.11 (a),(b),(c) and (d) : Convergence study plot for first four modes frequency of the conical shell panel

Discussions :

- Effect of lamination scheme on vibrationa behaviour of the cylindrical shell panel

As the lamination scheme changes in case of the cylindrical shell panel , the non-dimensional frequency parameter 'ω_l' and hence the frequency of vibration 'f' changes as shown in Fig. 12 This is due to modifications in fibre orientations in successive layers that changes the overall effective properties of the composite cylindrical panel.

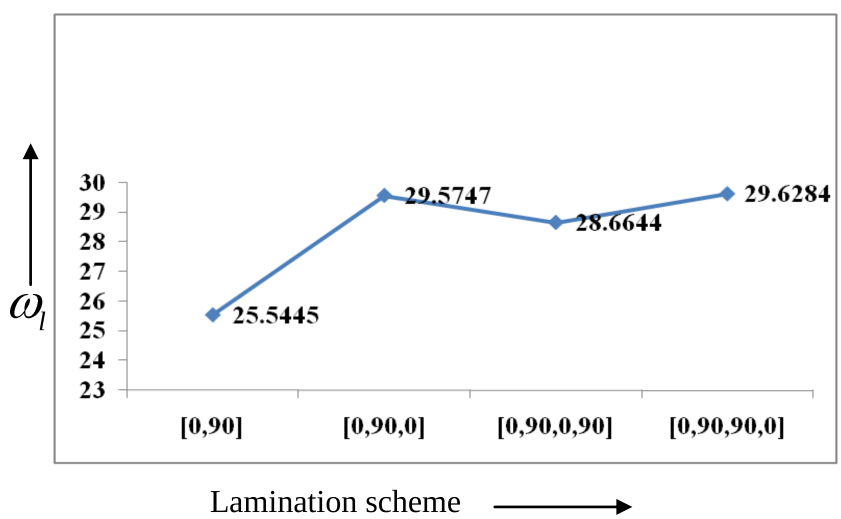


Fig.12 : Effect of lamination scheme on vibration characteristics of the cylindrical shell panel

- <u>Effect of mode number on vibration behaviour of the conical shell panel</u>

As the mode number increases in case of the conical shell panel, the non-dimensional frequency parameter '$\tilde{\omega}$' and hence the frequency of vibration 'f' of the conical panel becomes more as shown in Fig. 13.

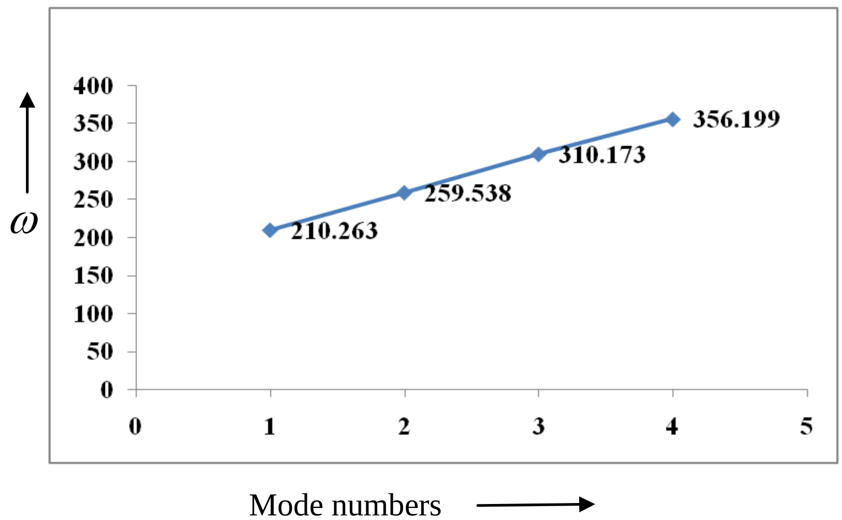


Fig.13 : Effect of mode numbers on vibration characteristics of the conical shell panel

Chapter 4: Conclusion

The vibration behaviours of the cylindrical and the conical shell panel have been thoroughly studied by using a simulation (developed in APDL code) model ANSYS 13 environment. Numerical results have been computed for vibration responses of laminated panel using the steps of Block Lanczos method as in ANSYS. It is well known that ANSYS mathematical model is developed based on the FSDT and the responses are compared with availabe literature. It is believed that the present model is showing good agreement. It is also worthy to mention that the present responses are showing higher value as compared to the reference and it is within the expected line. Finally, effects of different parameters like lamination scheme and mode number on vibration behaviour of cylindrical and conical panel respectively have been discussed.

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