2007 추계 학술발표회 논문집

Proceeding of the 2007 KSAS Fall Conference

일시 2007년 11월 15일(목)-16일(금)
장소 라마다프라자 제주호텔

주최 한국항공우주학회
후원 유콘시스템(주), 경주전장(주), 한남상사 건국대(BK21 ST·IT, IRH)
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1. Introduction

Piezoelectric materials are widely used as sensors and actuators in the several industrial fields related to noise and vibration control, acoustic speakers, precision position control and health monitoring systems. During the last decades, the application of piezoelectric materials to composite structures has been focused on the development of smart or intelligent structures, which is continuously monitored and controlled using distributed sensors and actuators to achieve the desired performance of the systems. Laminated composite shells are extensively used as structural components in aerospace and other engineering sectors due to their various superior properties. The shells have the capability to carry higher loads due to the coupling between membrane and flexure terms. Piezoelectric laminated structures enhance the use of shells due to the adaptive capabilities.

Marshall et al. [1] presented a theoretical analysis procedure for post buckling behaviour of thin orthotropic curved panels subjected to central point load and uniform pressure loadings. Bushnell [2] described some of the peculiarities in shell buckling: including nonlinear behaviour caused by a combination of large deflection and plasticity, stiffeners and load eccentricity effects and presented some examples of classical bifurcation buckling. Teng and Hong [3] presented the linear and nonlinear analysis of thin shells with a set of nonlinear strain displacement relations, neglecting the transverse shear strains. Kundu and Sinha [4] presented the postbuckling analysis of laminated composite spherical, cylindrical and conoidal shells, considering the strain displacement relations in the curvilinear coordinate system. Varelis and Saravanos [5] presented a coupled nonlinear response of shallow doubly curved adaptive piezocomposite shells. The mechanics was formulated in cylindrical coordinate and developed an eight-noded isoparametric shell element and presented the possibility of snap through of smart shell structures. Crisfield [6] modified Riks approach and made it suitable for use in the finite element. The Arc-length method is applied in conjunction with the Newton-Raphson method in both standard and modified forms. It is a path following technique where both load and displacement are independent parameters. The motivation of present investigation is to study snap through behaviour of laminated composite shells with piezoelectric material. A geometrically nonlinear finite element analysis using nine-noded isoparametric elements is carried out to study the post buckling behaviour of piezolaminated shell panels subjected to transverse load. The
The mathematical formulation is based on the virtual work equations for a continuum with a total Lagrangian approach, and the material behavior is assumed to be linear and elastic. The nonlinear equations are solved by the arc length method, to handle snap through behavior. The piezoelectric bending moments induced by actuators and the bending moments induced by substrate lead to the active snap through buckling. The nonlinear responses of specially piezolaminated cylindrical shell under mechanical and electrical loading are analyzed and discussed.

3. Piezo–Elastic Constitutive Relations

The shell geometry used in the present formulation is derived using an orthogonal curvilinear coordinate system, $a_1, a_2$ along the lines of principal curvatures, and a normal coordinate $\zeta$. The middle surface of the doubly curved laminated shell is assumed to be the reference surface and is shown in Fig. 1. The mid–surface, which defines the shape of a shell, is described by the two Lame parameters, $A_1$ and $A_2$, and the two principal radii of the curvature, $R_1$ and $R_2$. In order to define a valid surface, Gauss–Codazzi conditions [7] are satisfied.

Consider a general thin piezoelectric laminated shell (Fig. 1) of thickness $t$ composed of an arbitrary number of thin unidirectional lamina layers, including the piezoelectric layers. Each lamina may have different thickness as well as arbitrary fiber orientation [8]. In a piezoelectric material, the interaction between the mechanical and electrical fields is defined by

$$\{\sigma\} = [Q]\{\epsilon\} - [E]\{E\}$$
$$\{D\} = [P]^T\{\epsilon\} + \{P\}$$

where $\sigma$, $\epsilon$, $D$, and $E$ are the elastic stress vector, elastic strain vector, electric displacement vector and electric field vector, respectively.

3. Nonlinear Finite Element Formulation

3.1 Displacement Models

The displacements at any point can be described as

$$u(\alpha_1, \alpha_2, \zeta) = u^0(\alpha_1, \alpha_2) + \zeta \theta_1$$
$$v(\alpha_1, \alpha_2, \zeta) = v^0(\alpha_1, \alpha_2) + \zeta \theta_2$$
$$w(\alpha_1, \alpha_2, \zeta) = w^0(\alpha_1, \alpha_2)$$

where $u^0$, $v^0$, and $w^0$ are the displacements on the reference surface along $a_1$, $a_2$ and $\zeta$ directions, respectively; and $\theta_1$ and $\theta_2$ are the rotations. The isoparametric formulation [9] has been used to implement the nine–noded shell element, where each node has five degrees of freedom, $u_1^0$, $u_2^0$, $w^0$, $\Theta_1$ and $\Theta_2$.

3.2 Strain–Displacement Relations

The strains at any point are expressed as follows [4]:

Fig.1. A doubly curved piezoelectric laminated shell panel
\[
\begin{align*}
\varepsilon_{11} &= \varepsilon_1^0 + \frac{1}{2} \left[ \varepsilon_1^0 \right]^2 + \left[ \varepsilon_2^0 \right]^2 + \left[ \varepsilon_3^0 \right]^2 + \xi \kappa_1 \\
\varepsilon_{22} &= \varepsilon_2^0 + \frac{1}{2} \left[ \varepsilon_2^0 \right]^2 + \left[ \varepsilon_3^0 \right]^2 + \left[ \varepsilon_6^0 \right]^2 + \xi \kappa_2 \\
\gamma_{12} &= \left( \varepsilon_1^0 \varepsilon_2^0 \varepsilon_4^0 \varepsilon_5^0 \varepsilon_6^0 \varepsilon_0^0 \right) + \xi \kappa_1 \kappa_2 \\
\gamma_{13} &= \xi \kappa_1 \kappa_3 \\
\gamma_{23} &= \xi \kappa_2 \kappa_3
\end{align*}
\]

(3)

where \( \varepsilon_1^0, \varepsilon_2^0, \varepsilon_3^0, \varepsilon_4^0, \varepsilon_5^0, \varepsilon_6^0 \) correspond to the mid-surface strains, and \( K_1, K_2, K_3, K_4 \) correspond to the curvatures. The strain and curvature terms of Eqn. (3) are expressed in terms of the field variables as

\[
\begin{align*}
\varepsilon_1^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\varepsilon_2^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\varepsilon_3^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\varepsilon_4^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\varepsilon_5^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\varepsilon_6^0 &= \frac{1}{\varepsilon_1^0} \left[ \frac{u_1^0}{\varepsilon_1^0} + \frac{u_2^0}{\varepsilon_2^0} + \frac{u_3^0}{\varepsilon_3^0} + \frac{u_4^0}{\varepsilon_4^0} + \frac{u_5^0}{\varepsilon_5^0} + \frac{u_6^0}{\varepsilon_6^0} \right] \\
\kappa_1 &= \frac{1}{A_1} \left\{ \theta_1 \right\} \frac{A_1}{A_1} + \frac{\theta_2}{A_1} \frac{A_1}{A_1} + \frac{\theta_3}{A_1} \frac{A_1}{A_1} + \frac{\theta_4}{A_1} \frac{A_1}{A_1} + \frac{\theta_5}{A_1} \frac{A_1}{A_1} + \frac{\theta_6}{A_1} \frac{A_1}{A_1} \\
\kappa_2 &= \frac{1}{A_1} \left\{ \theta_1 \right\} \frac{A_1}{A_1} + \frac{\theta_2}{A_1} \frac{A_1}{A_1} + \frac{\theta_3}{A_1} \frac{A_1}{A_1} + \frac{\theta_4}{A_1} \frac{A_1}{A_1} + \frac{\theta_5}{A_1} \frac{A_1}{A_1} + \frac{\theta_6}{A_1} \frac{A_1}{A_1} \\
\kappa_3 &= \frac{1}{A_1} \left\{ \theta_1 \right\} \frac{A_1}{A_1} + \frac{\theta_2}{A_1} \frac{A_1}{A_1} + \frac{\theta_3}{A_1} \frac{A_1}{A_1} + \frac{\theta_4}{A_1} \frac{A_1}{A_1} + \frac{\theta_5}{A_1} \frac{A_1}{A_1} + \frac{\theta_6}{A_1} \frac{A_1}{A_1}
\end{align*}
\]

(4)

3.3 Electric Field Equation

The active layers can act as an actuator or a sensor and they can be placed anywhere in the thickness of the laminate. It is assumed that the electric potential of every point on the surface of the piezoelectric layer has the same value and the electric potential variation is linear across the thickness because the piezoelectric layer is very thin. Piezoelectric layers are attached to the top (convex) and concave (bottom) surface of the shell surface. The variation of electric potential in the piezoelectric layer attached to the surface is given by,

\[
\phi \left( \alpha_1, \alpha_2, \varsigma \right) = \frac{\varsigma - \varsigma_{i-1}}{\varsigma_{i-1}} \phi \left( \alpha_1, \alpha_2 \right)
\]

(6)

The arc-length method solution technique is used to solve the nonlinear problem. The tangent stiffness matrix as defined is given by,

\[
[K'] = [K]_L + [K]_N + [K]_G
\]

(6)

where \([K]_L, [K]_N\) and \([K]_G\) are the linear, nonlinear and geometric stiffness matrices. An object oriented finite element code in C++ language is written to implement the above finite element formulation.

4. Numerical Results and Discussions

4.1 Example 1

The post-buckling response of an isotropic cylindrical panel \((a = b = 508 \text{ m}, R_1 = 2544.53 \text{ mm}, h = 12.7 \text{ mm})\) is considered having additionally two piezoelectric continuous layers attached on the upper and lower surfaces of the substrate. The thickness of each piezoelectric layer is 0.1 mm. The material properties of the piezoelectric laminated cylindrical shell panel are as follows

\[
E = 3100 \text{ N/mm}^2, \quad v = 0.3
\]

PZT 5: \(E_{11} = 62 \text{ GPa}, G_{12} = 23.6 \text{ GPa}, G_{23} = 18 \text{ GPa}, v_{12} = 0.31, d_{31} = d_{32} = -200 \times 10^{-12} \text{ m/V}, d_{34} = 670 \times 10^{-12} \text{ m/V}, p_{11} = p_{22} = 2992 \times 10^{-12} \text{ farad/m}
\]

The mechanical point load is applied on the center of the cylindrical panel in combination with bipolar electric fields imposed on the actuators (+ve Volts on the outer electrode and the inner electrode is connected to earth). The nonlinear load-deflection curves, for
Various applied electric volt are shown in Fig. 2. When positive voltage is applied, the shell buckle under higher load value compared to zero electric voltage. Conversely, for negative voltage the shell buckle at a lower load compared to that of zero voltage. It is noted here that the negative voltage indicates an early buckling. The present results and the published results (Varellis and Saravanos (5)) are in very good agreement.

It is observed that the curve corresponding to $\pm 300$ Volts under-goes snap through at a higher load values compared to that of 0.0 Volt.

Fig. 3. Post buckling responses of isotropic cylindrical shells

4.3 Example 3

The post buckling behaviour of $[\pm 45/\mp 45]$ piezolaminated cylindrical shell panels are considered. The panels ($a = b = 500$ mm) are subjected to a central transverse point load. The total thickness of the each panel is considered as 6.25 mm, and the thickness of the upper and lower piezoelectric layer is assumed as 0.1 mm. The load-deflection curves are shown in Fig. (4). For 0.0 Volt the limit point is observed at 7.854 kN. When $+300.0$ Volts is applied to each piezoelectric layer, the limit point is observed at 8.095 kN and for $-300.0$ Volts, the limit point is observed at 7.617 kN.
5. Conclusions

The geometrically nonlinear analysis of doubly curved piezoelectric laminated shells is carried out using finite element method. The coupled nonlinear responses of smart cylindrical shells undergoing large displacement (snap through and snap back) are predicted. The buckling load increases for positive electric potential applied. Thin smart shells having integrated piezoelectric actuators undergo large deformation resulting in snap-through and snap-back phenomenon from one equilibrium state to another. The limit point load can be increased by introducing piezoelectric layers in the laminated shells to control the deflections.

Acknowledgements

This work was supported by Defense Acquisition Program Administration and Agency for the Defense Development under the contract UD060009AD. The first author thanks the support of the second stage of the Brain Korea 21 project in 2007.

References

2007 추계 학술발표회 논문집

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주최 한국항공우주학회
후원 한국학술진흥재단
유콘시스템(주)
경주전장(주)
한남상사건국대(BK21 ST-IT, IRH)

일시 2007년 11월 15일(목)-16일(금)
장소 라마다프라자 제주호텔