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Thermal buckling behaviour of angle-ply laminated composite plate with multi-pole hole

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Abstract

In this present article, we considered thermal buckling analysis of symmetric and antisymmetric cross-ply laminated hybrid composite plates with an inclined multipole hole. Two shear deformable finite element models, one based on first-order shear deformation theory with variational energy method and the other based on higher order shear deformation theory, are employed to obtain thermal buckling solutions. The eight-node Lagrangian finite element technique is employed for obtaining the thermal buckling temperatures of hybrid composite laminates. Results show that the temperature rise are affected by the ratio of h/b for both symmetric and antisymmetric cases, while hole diameter shows, buckling temperature remains almost constant for both symmetric and anti-symmetric plates. The effect of various layers, plate thickness, and the number of holes also affect the bulking temperature for both symmetric and anitsymmetric case. More the size of the geometric defect is less important the buckling load will be. Thus, the amplification factor N⁄ grows with the increase of the ply thickness. The effects of crack size and lay-up sequences on the thermal buckling temperatures for symmetric and antisymmetric plates are also investigated for angular oriented ply laminated composite. The results are also shown in graphical form by considering the various boundary conditions.

Keywords

Thermal Buckling, Laminated Composite Materials, Finite Element Analysis

1. Introduction

Composite materials and laminated structures are being increasingly used in modern engineering applications, because of the advantages offered by their particular mechanical properties coupled with the attractive potential of strength-to-weight ratios. The high specific strength and specific stiffness which is the base of the superior structural performance of composite materials provide composite materials many application choices. For moving parts, weight plays an important role in calculating the structural stability of the system. The fiber reinforced laminated composite plates with holes are widely used especially in weight sensitive structures. Weight reduction is intended by hole opening and fuel, hydraulic and electrical lines can be placed through these holes. The presence of holes in a structural member often complicates the design process. The overall design complexity, owing to the particular constitutive behavior of the material, the complex final shapes of structural members, the external environment, and many other reasons, makes the solution of the design problem extremely difficult.

Buckling is one of the primary modes of failure of these elements when they are subjected to membrane stresses caused by either thermal loads or mechanical loads or a combination of e loads. Thus the buckling strength of skew composite laminates is one of the factors governing their design and its accurate determination is of interest to designers. The thermal buckling analyses of orthotropic plates, including a crack were reported by Avci et al. [1].

Thermal buckling analysis of symmetric and antisymmetric cross-ply laminated hybrid composite plates with a hole subjected to a uniform temperature rise for different boundary conditions was studied by using finite element method by Avci et al. [2]. Then Hoff [3] established a buckling criterion for the cover plates of wing panels subjected to thermal stress. Gowda and Pandalai [4] obtained results for thermally loaded simply supported and clamped isotropic plates by using an energy approach. Huang and Tauchert [5] studied the thermal buckling of clamped symmetric angle-ply laminated plates employing a Fourier series approach and the finite element method. Prabhu and Dhanaraj [9], Chandrashekhara [7], Thangaratnam and Ramaohandran [8], and Chen et al. [9] also studied the thermal buckling of the laminates subjected to uniform temperature rise or non uniform temperature fields using the finite element approach. Finite element analysis was also carried out to obtain the effects of cutout on the buckling behavior of these plates. Jain and Kumar [10] carried out the finite element method for the post buckling response of symmetric square laminates with a central cutout under uniaxial compression. The governing finite element equations were solved using the Newton–Raphson method. Kong et al. [11]analyzed buckling and post buckling behaviors both numerically and experimentally for composite plates with a hole. In the finite element analysis, the updated Lagrangian formulation and the eight-node degenerated shell element were used. The effect of hole sizes and stacking sequences was examined on the compression behavior of the plate. Experiments showed fine agreement with the finite element results in the buckling load and the post buckling strength. Ghannadpour et al. [12] studied the influences of a cutout on the buckling performance of rectangular plates made of polymer matrix composites (PMC). The study was concentrated on the behavior of rectangular symmetric cross-ply laminates. Birman [9] provided buckling analysis of functionally graded hybrid composite plates, and Javaheri and Eslami [10] analyzed the thermal buckling of FGPs based on higher order theory. Liew et al. [11, 12] performed a post buckling analysis of FGPs subjected to thermo-electro-mechanical loading and also considered the thermal post buckling of these plates. Bamberger et al. made a numerical analysis on tension buckling of plates having a hole [4].

In this articles, my attempt to determine the buckling temperature and buckling mode for angle-ply and cross-ply composite laminated plates with holes using finite element method, based on first-order shear deformation theory in conjunction with the variational energy method. The four-noded finite element approach is used for obtaining the thermal buckling temperature of boron-epoxy/glass-epoxy laminated. The effect of hole diameter, lay-up sequences and boundary conditions with thermal buckling temperatures are investigated. The results are shown in graphical form for various boundary conditions.

2. Problem Formulation Finite Element Formulation

In order to study the buckling of the plate, the eight-node Lagrangian finite element approach is used for obtaining the thermal buckling temperatures of boron–epoxy/glass–epoxy hybrid laminates. The laminated plate of length "a" and width "b" consists of "N" individual layers. Each layer is of thickness t_k, so that $h=\sum_{k=1}^{N} t_k$ is the total thickness of the laminate. The geometry of the laminated ply is shown in Fig.1

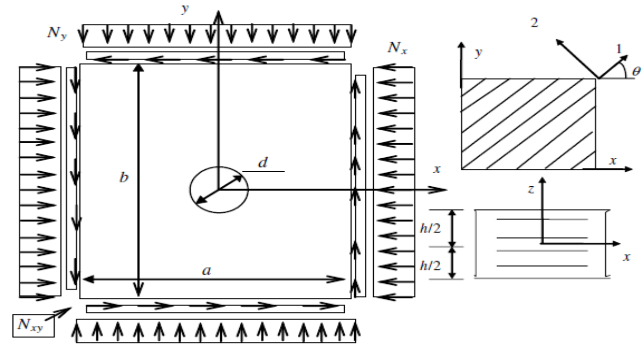


Fig. 1. *Geometry of the problem and coordinate: The difference between the plate and ambient temperatures is ΔT. The displacement fields u, v and w can be written as follows.*

$$\begin{aligned} u(x,y,z) &= u_0(x,y) + z\psi_x(x,y) \\ v(x,y,z) &= v_0(x,y) + z\psi_y(x,y) \\ w(x,y,z) &= w(x,y) \end{aligned} \tag{1}$$

Where u_o, v_o, w are the displacements at any point of the middle surface, and ψ_x and ψ_y are the bending rotations of the normal to the mid plane about the x and y axes respectively. The bending strains $\varepsilon_x, \varepsilon_y$ and shears strains γ_{xy}, γ_{yz} and γ_{xz} at any point of the laminate are

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{vmatrix} \dfrac{\partial u_0}{\partial x} \\ \dfrac{\partial v_0}{\partial y} \\ \dfrac{\partial w}{\partial y} + \dfrac{\partial w}{\partial x} \end{vmatrix} + z \begin{vmatrix} \dfrac{\partial \psi_x}{\partial x} \\ \dfrac{\partial \psi_y}{\partial y} \\ \dfrac{\partial \psi_x}{\partial y} + \dfrac{\partial \psi_y}{\partial x} \end{vmatrix} \tag{2}$$

$$\begin{vmatrix} \gamma_{yz} \\ \gamma_{xz} \end{vmatrix} = \begin{vmatrix} \dfrac{\partial w}{\partial y} - \psi_y \\ \dfrac{\partial w}{\partial x} + \psi_x \end{vmatrix}$$

The linear stress-strain relation for each layer is expressed with x, y $axes$ and has the form

$$\begin{Bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xx}\end{Bmatrix}=\begin{pmatrix}\bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{13}\\ \bar{Q}_{21} & \bar{Q}_{22} & \bar{Q}_{23}\\ \bar{Q}_{31} & \bar{Q}_{32} & \bar{Q}_{33}\end{pmatrix}\begin{Bmatrix}\varepsilon_x-\alpha_x\Delta T\\ \varepsilon_y-\alpha_y\Delta T\\ \gamma_{xy}-\alpha_{xy}\Delta T\end{Bmatrix} \tag{3}$$

$$\begin{Bmatrix}\tau_{yz}\\ \tau_{xz}\end{Bmatrix}=\begin{pmatrix}\bar{Q}_{11} & \bar{Q}_{12}\\ \bar{Q}_{21} & \bar{Q}_{22}\end{pmatrix}\begin{Bmatrix}\gamma_{yz}\\ \gamma_{xz}\end{Bmatrix}$$

Where σ_x , σ_y, τ_{xy}, τ_{yz}, and τ_{xz} are the stress components, $\bar{Q}_{ij}$ are the transformed, reduced stiffness's, which can be expressed in terms of the orientation angle and the engineering constant of the material. ΔT is the temperature increase, α_x and α_y are the coefficients of thermal expansion in directions of x and y axes, respectively. α_{xy} is the apparent coefficient of thermal shear, such as

$$\begin{aligned}\alpha_x &= \alpha_1\cos^2\theta+\alpha_2\sin^2\theta,\\ \alpha_y &= \alpha_2\cos^2\theta+\alpha_1\sin^2\theta,\\ \alpha_{xy} &= 2(\alpha_1-\alpha_2)\sin\theta\cos\theta\end{aligned} \tag{4}$$

Where α_1 and α_2 are the thermal expansion coefficients of the lamina along the longitudinal and transverse directions of fibers, respectively. The resultant forces N_x, N_y, and N_{xy}, the moments M_x, M_y, and M_{xy}, and the shearing forces Q_x and Q_y per unit length of the plate are given as

$$\begin{pmatrix}N_x & M_x\\ N_y & M_y\\ N_{xy} & M_{xy}\end{pmatrix}=\int_{-h/2}^{h/2}\begin{Bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{Bmatrix}(1,z)\,dz$$

$$\begin{Bmatrix}Q_x\\ Q_y\end{Bmatrix}=\int_{-h/2}^{h/2}\begin{Bmatrix}\tau_{xz}\\ \tau_{yz}\end{Bmatrix}dz \tag{5}$$

The total potential energy Π of the laminated plate under the thermal loading is $\prod=U_b+U_s+V$, where U_b is the energy of bending strain, U_s is the energy of shear strain, and V is the potential energy of in-plane strains due to the change in temperature:

$$U_b=\frac{1}{2}\int_{-h/2}^{h/2}\left[\iint\left(\sigma_x\varepsilon_x+\sigma_y\varepsilon_y+\tau_{xy}\gamma_{xy}\right)dxdy\right]dz$$

$$U_s=\frac{1}{2}\int_{-h/2}^{h/2}\left[\iint\left(\tau_{xz}\gamma_{xz}+\tau_{yz}\gamma_{yz}\right)dxdy\right]dz$$

$$V=\frac{1}{2}\iint_R\left[\bar{N}_1\left(\frac{\partial w}{\partial x}\right)^2+\bar{N}_2\left(\frac{\partial w}{\partial y}\right)^2+2\bar{N}_{12}\left(\frac{\partial w}{\partial x}\right)\left(\frac{\partial w}{\partial y}\right)\right]dxdy-\int_{\partial R}\left(\bar{N}_n^b u_n^0+\bar{N}_s^b u_s^0\right)ds \tag{6}$$

Here, R is the region of the plate, excluding the crack. $\bar{N}_n^b$ and $\bar{N}_s^b$ are the in-plane loads applied to the boundary ∂R . For equilibrium, the potential energy Π must be stationary. The equilibrium equations of the cross-ply laminated plate subjected to temperature changes can be derived from the variational principle through the use of stress–strain and strain–displacement relations. The stiffness matrix of the plate is obtained by using minimum potential energy principle. Bending stiffness, shear stiffness and geometric stiffness matrices can be expressed as

$$\begin{aligned}[\kappa_b] &= \int_A [B_b]^T[D_b][B_b]dA\\ [\kappa_s] &= \int_A [B_s]^T[D_s][B_s]dA\end{aligned} \tag{7}$$

And

$$[\kappa_g]=\int_A [B_g]^T[D_g][B_g]dA \tag{8}$$

Where $[D_b]=\begin{bmatrix}A_{ij} & B_{ij}\\ B_{ij} & D_{ij}\end{bmatrix}$, $[D_s]=\begin{bmatrix}\kappa_1^2A_{44} & 0\\ 0 & \kappa_2^2A_{55}\end{bmatrix}$,

$$[D_g]=\begin{bmatrix}\bar{N}_1 & \bar{N}_{12}\\ \bar{N}_{12} & \bar{N}_{22}\end{bmatrix}, \tag{9}$$

$$(A_{ij},B_{ij},D_{ij})=\int_{h/2}^{h/2}\bar{Q}_{ij}(1,z,z^2)dz \text{ Where } (i,j=1,2,6)$$

$$(A_{44},A_{55})=\int_{h/2}^{h/2}(\bar{Q}_{44},\bar{Q}_{55})dz \tag{10}$$

in which the term A_{45} is neglected in comparison with A_{44} and A_{55} ,and shear correction factor for the rectangular cross-section are given by $\kappa_1^2=\kappa_2^2$ =5/6. The total potential energy principle for the plate satisfies the assembly of the element equations. Where

$$[[\kappa_0]-\lambda_b[\kappa_{0g}]]\begin{Bmatrix}u_i\\ v_i\\ w_i\end{Bmatrix}=0 \tag{11}$$

The product of λ_b and the initial guest value ΔT is the critical buckling temperature T_{cr} , that is $T_{cr}=\lambda_b\Delta T$.

3. Result and Discussion

There are many techniques to solve Eigen value problems. In this study the finite element method is applied to obtain critical buckling temperature for thermal buckling due to temperature change in the plate, the uniaxial or biaxial in-plane loads are developed along the rectangular edges, while the circular hole edge is free. The angle-ply laminated composite plates used here have several thicknesses and bonded symmetrically and anti-symmetrically. Stacking sequence of hybrid composite plates have been taken both symmetric and antisymmetric. For the computations thermo-elastic properties considered for the E-glass/epoxy, and boron/epoxy composites are given in Table1. Here E_1 and E_2 are elastic moduli in 1 and 2 directions, respectively, ϑ_{12} is the Poisson's ratio and α_1 and α_2 are thermal expansion coefficients of the material used in the solution. The effect of α_{12} is neglected.

Table 1. *Material properties of finite element solutions.*

Material	$E_1(GPa)$	$E_2(GPa)$	$G_{12}(GPa)$	ϑ_{12}	$\alpha_1(1/℃)$	$\alpha_2(1/℃)$
E-glass/epoxy	15	6	3	.3	7.0×10^{-6}	2.3×10^{-5}
Boron/epoxy	207	19	4.8	.21	4.14×10^{-6}	1.9×10^{-5}

Each layer has 0.25 mm thickness and the length of one edge of plate is 100 mm, h/b ratio, represents the total thickness of composite plate to the length of one side of composite plate and d/b ratio, represents the hole size to the length of one side of composite plate. A wide range of boundary condition can be accommodated, but only four kinds of boundary conditions are taken as.

1. Four edges simply supported (SSSS)

At $x=-a/2,a/2;u=w=\psi_y=0$

At $y=-b/2,b/2;v=w=\psi_x=0$

2. Two edges simply supported (SS)

At $x=-a/2,a/2;u=w=\psi_y=0$

At $y=-b/2,b/2;w=0$

3. Four edges clamped (CCCC)

At $x=-a/2,a/2;u=w=\psi_y=\psi_x=0$

At $y=-b/2,b/2;v=w=\psi_x=\psi_y=0$

4. Two edges clamped (CC):

At $x=-a/2,a/2;u=w=\psi_y=\psi_x=0$

At $y=-b/2,b/2;w=0$

Fig.1 shows the meshed plate. Four edges of plate have divided into 20 parts disregarding the hole size.

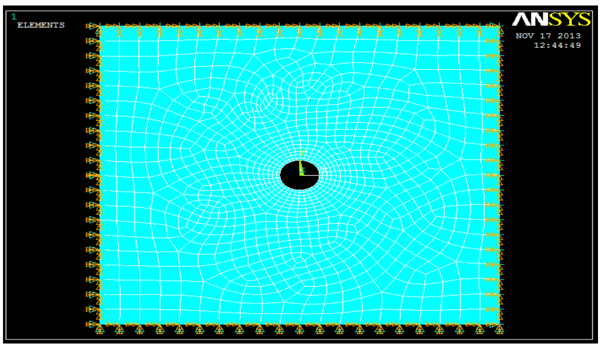


Fig 2. *Typical meshed for four edge clamped (CCCC) plate.*

(A) Effect of plate thickness

The variation of critical buckling temperature of plate without hole for $(15/-15)_2$ configuration is shown in Fig.2. As expected, temperature increases as the h/b ratio increases for both symmetric and anti-symmetric cases. But critical buckling temperatures of anti-symmetric laminates are higher than that of symmetric plates. The highest buckling temperatures are obtained for four edge clamped plate and the lowest temperatures obtained for four edges simply supported plate. Above result shows that present result has good agreement with paper result. The finite element result follows the same trend as Newton-Raphson method.

(B) Effect of hole diameter

The variation of critical temperature of plate with hole for $(60/-60)_2$ configuration is shown in Fig.3. For this layer configuration, for two and four edge simply supported plates, as hole diameter increases, buckling temperature remains almost constant for both symmetric and anti-symmetric plates. As hole diameter increase, the buckling temperature of four edges clamped increases.

(C) Effect of various layers

The variation of critical buckling temperature with hole diameters for various layer configuration is shown in Fig.4. As the number of layers, namely, thickness of plate increases, the critical temperature increases. This behaviour is same for all layer configurations used in this solution.

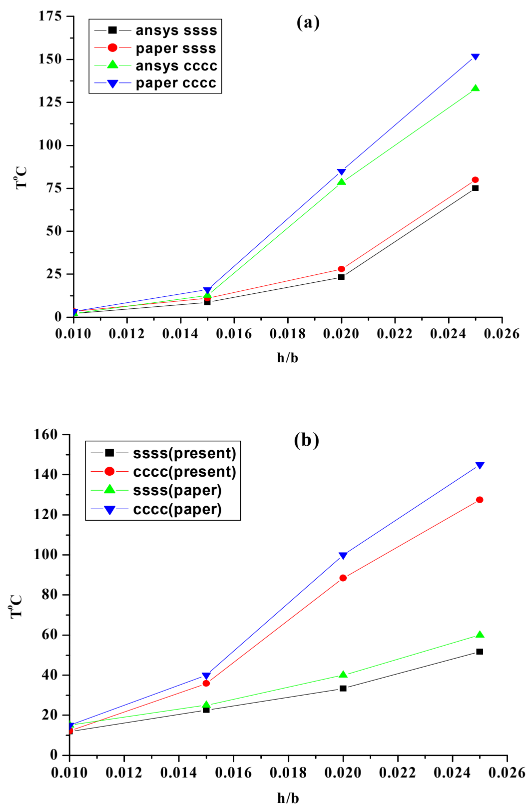


Fig. 3. *Variation of buckling temperature with thickness of (a) anti-symmetric, (b) symmetric laminates without hole for various boundary conditions for* $(15/-15)_2$ *configuration.*

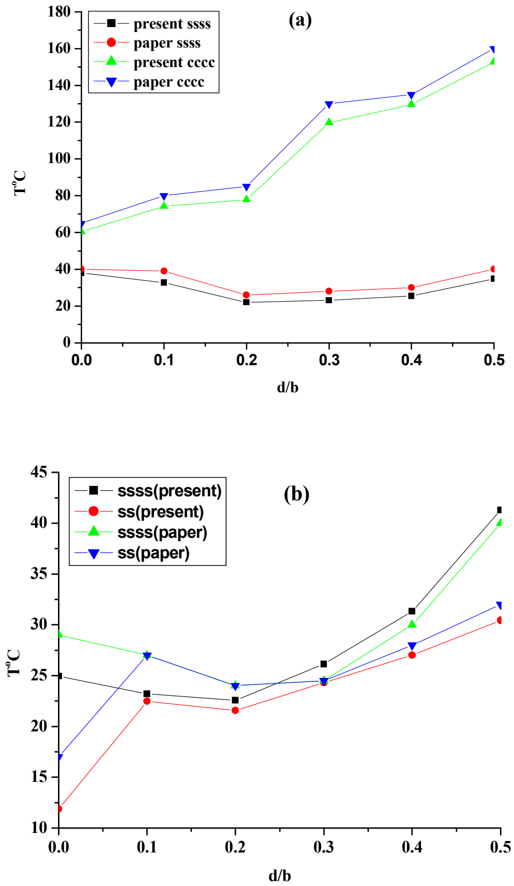


Fig. 4. Variation of buckling temperature with hole size for six layered (a) anti-symmetric (b) symmetric plate for various boundary conditions for $(60/-60)_2$ *configuration. As seen buckling temperature for anti-symmetric plate are higher than that of symmetric plate.*

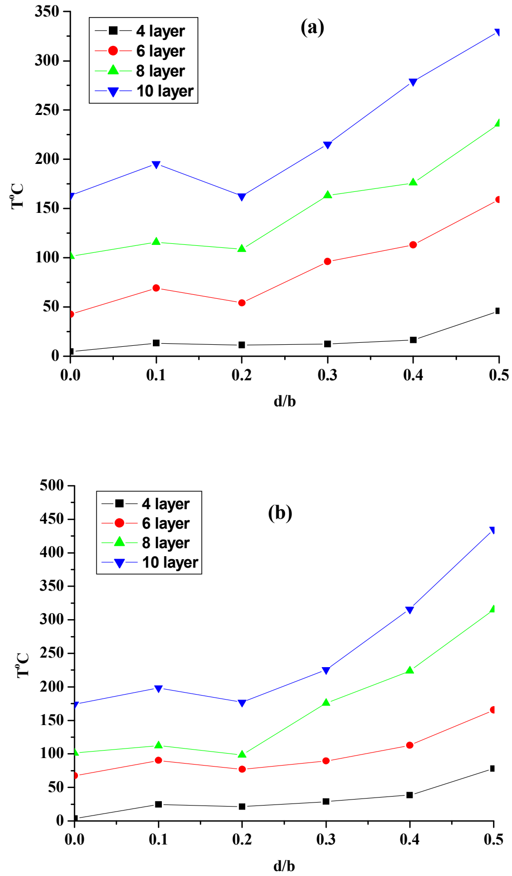


Fig. 5. Variation of buckling temperature with hole size for anti-symmetric four edges clamped plate for various layer number for (a) $(30/-30)_2$ *(b)* $(15/\text{-}15)_2$ *configuration.*

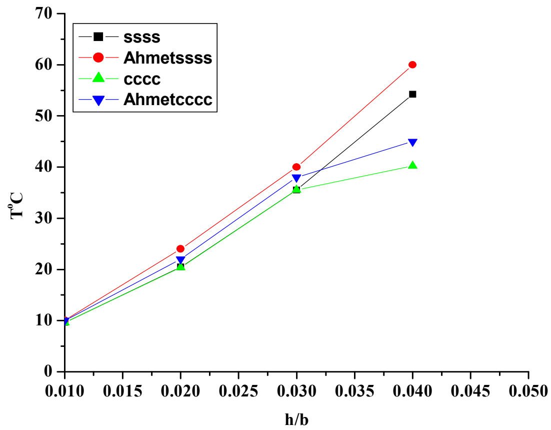


Fig. 6. Variation of buckling temperature with thickness for anti-symmetric, laminates without hole for various boundary conditions for $(0/90)_2$ *configuration.*

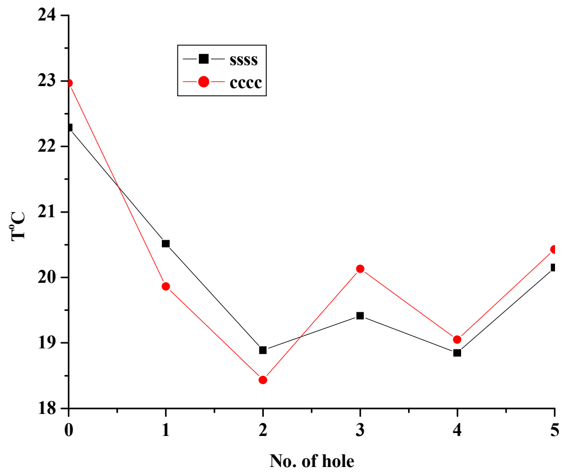
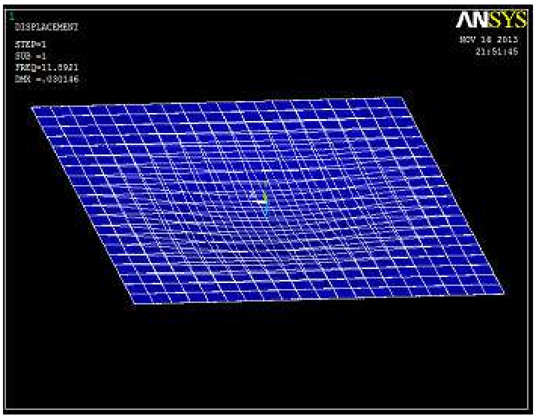
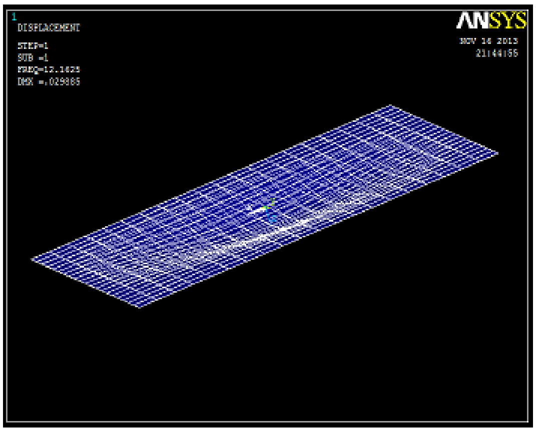


Fig. 7. Variation of buckling temperature with no. hole for six layer anti-symmetric four edges clamped and four edges simply supported plate for various boundary conditions for $(60/-60)_2$ *configuration*

The first buckled mode shapes generated glass-epoxy angle-ply six layered plate with four boundary conditions are shown in figs.7-9.It is found that the critical temperature for $(15/-15)_2$ configuration plate without hole for four edge clamped plate is 78.4364 ℃, for four edges simply supported plate is 23.3067 ℃. For $(60/-60)_2$ configuration six layered plates for four edge simply supported is 37.957 ℃ and for four edge clamped is 60.416℃. For $(60/-60)_2$ configuration plate with hole size of d/b=0.2 for six layered for four edge simply supported and four edges clamped are 21.958 ℃ and 77.901 ℃ respectively. The first buckled mode shapes generated glass-epoxy angle-ply six layered plates with four boundary conditions are shown in figs.8-9.It is found that critical temperature for $(15/-15)_2$ configuration plate without hole for four edge clamped plate is 78.4364 ℃, for four edges simply supported plate is 23.3067 ℃. For $(60/-60)_2$ configuration six layered plate for four edge simply supported is 37.957 ℃ and for four edge clamped is 60.416℃. For $(60/-60)_2$ configuration plate with hole size of d/b=0.2 for six layered for four edge simply supported and four edges clamped are 21.958 ℃ and 77.901 ℃ respectively.

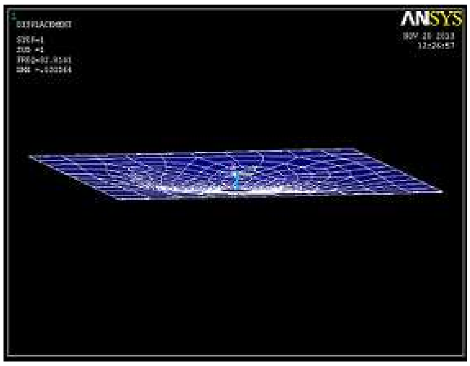


(a)

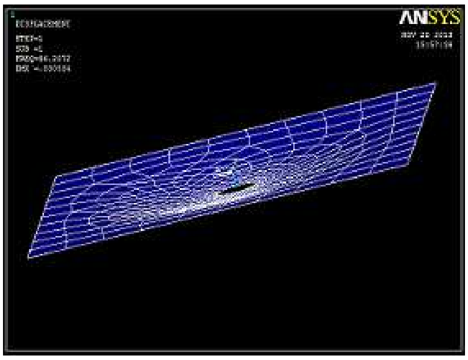


(b)

Fig. 8. *First buckling mode shapes for* $(15/-15)_2$ *configuration four layered laminate without hole for boundary condition (a) four edge clamped (CCCC), (b) four edge simply supported (SSSS) plates.*

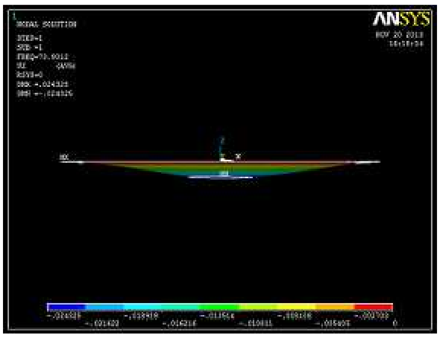


(a)

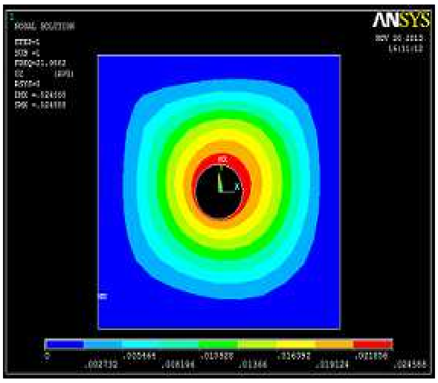


(b)

Fig. 9. *First buckling mode shapes for* $(60/-60)_2$ *configuration six layered laminate with hole for boundary conditions (a) simply supported (SSSS), (b) four edge clamped (CCCC) plates.*

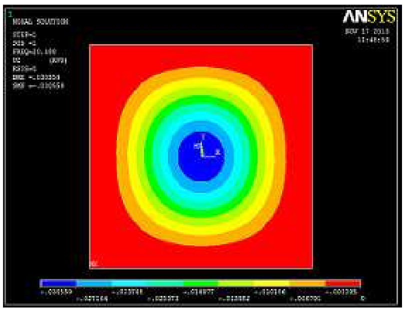


(a)

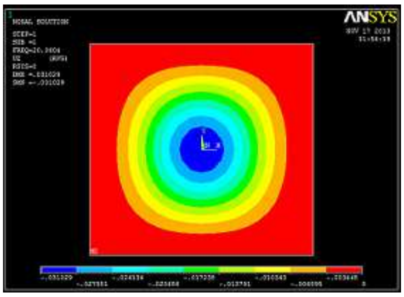


(b)

Fig. 10. *First nodal solution for* $(60/-60)_2$ *configuration six layered laminate with hole size of d/b=0.2 for boundary conditions (a) four edge clamped, (CCCC) (b) four simply supported (SSSS) plates.*

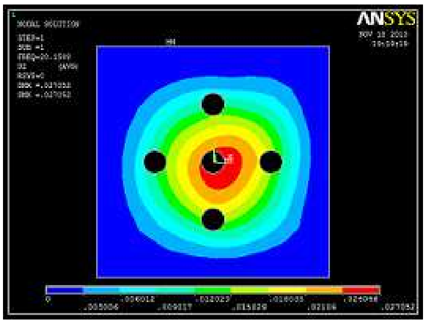


(a)

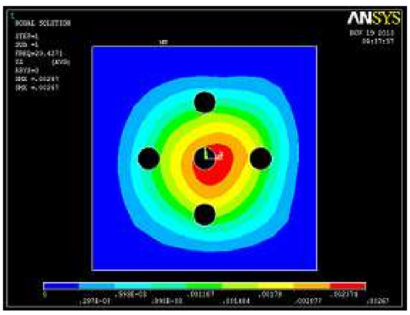


(b)

Fig. 11. *First nodal solution for* $(0/90)_2$ *configuration four layered laminate without hole for boundary conditions (a) foue edge simply supported (SSSS), (b) four edge clamped (CCCC) plates.*



(a)



(b)

Fig. 12. *First nodal solution for* $(60/-60)_2$ *configuration six layered laminate for five no. of hole with hole sized d/b=0.1 for boundary conditions (a) four edge simply supported (SSSS), (b) four edge clamped (CCCC) plates.*

The first buckled mode shapes generated glass-epoxy cross-ply four and six layered plates with four boundary conditions are shown in figs.10-11.For $(0/90)_2$ configuration plate without hole for four edges simply supported and four edges clamped are 54.241 ℃ and 40.242 ℃ respectively. For $(0/90)_2$ configuration with multi hole plates with holes size d/b=0.1 for six layered for four edges simply supported and four edge clamped are shown respectively.

4. Conclusions

Thermal buckling behaviors of angle-ply laminated composite plates with holes have been examined by employing the first-order shear deformation theory and finite element technique. Both symmetric and anti-symmetric lay-up sequence are considered and various boundary conditions are taken into account. Four edge clamped plate with hole have the high buckling temperature and four edges simply supported plate have the lowest buckling temperature for all layer configurations. The critical buckling temperature strongly depends on hole size for two edges and four edges clamped plates. For four edge clamped plates, as the hole size increases, the buckling temperature shows an increase. But, as the hole size increases, the buckling temperatures for two edges clamped plate decreases. The critical buckling temperature of four edges simply supported and simply supported plates do not affect so much by hole size and plates have this boundary conditions result the lowest buckling temperature. In this case, there is a little difference between symmetric and anti-symmetric lay-up. As the number of layers, namely the plate thickness increases, more temperature difference is needed for buckling. This behaviour is same for both anti-symmetric and symmetric lay-up. But, the higher temperature difference is needed for buckling of anti-symmetrical stacked plate than the symmetric one.

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