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FREQUENCY OPTIMIZATION AND TRANSIENT ANALYSES OF STIFFENED FOLDED LAMINATE COMPOSITE PLATE USING GENETIC ALGORITHM

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ABSTRACT

In this study, frequency optimization of stiffened folded laminate composite plate is investigated with respect to fiber orientations by using genetic algorithm (GA). The first order shear deformation theory was used for direct frequencies calculations. The Matlab programming using rectangular isoparametric plate element with five degrees of freedom per node was built to solve the problems. The modulus of selection, crossover and mutation were used as standard sub-modulus. The effects of obtained optimal fiber orientation on transient response of the folded plate have been investigated with difference boundary conditions. A good agreement was found between the results of this technique and other published results available in the literature.

Keywords: frequency optimization, folded laminated composite plate, genetic algorithm, transient response, finite element.

1. INTRODUCTION

Now a day, folded laminate composite plates are very useful in engineering. Applications for them have been found almost everywhere in various branches of engineering, such as in roofs, ship hulls, sandwich plate cores and cooling towers, etc. They are lightweight, easy to form, economical, and have much higher load carrying capacities than at plates, which ensures their popularity and has attracted constant research interest since they were introduced.

However, there is very limited information regarding the analysis of composite folded structures. Haldar and Sheikh [1] presented a free vibration analysis of isotropic and composite folded plate by using a sixteen nodes triangular element. Suresh and Malhotra [2] studied the free vibration of damped composite box beams using four node plate elements with five degrees of freedom (DOF) per node. Niyogi et al. [3] carried out a finite element vibration analysis of folded laminates by using first order plate theory and nine nodes elements. In their works, only in axis symmetric cross-ply laminated plates were considered. So that, there is uncoupling between the normal and shear forces, and also between the bending and twisting moments, then besides the above uncoupling, there is no coupling between the forces and moment terms. Peng et al. [4] presented a analysis of folded plates subjected to bending load by the FSDT meshless method. In this, a meshfree Galerkin method based on the first-order shear deformation theory (FSDT) for the elastic bending analysis of stiffened and un-stiffened folded plates is analyzed.

In previous works, Tran Ich Thinh and Bui Van Binh [5 - 8] presented a finite element method to analyze of bending, free vibration and time displacement response of V-shape; W-shape sections and multi-folding laminate plate (which having trapezoidal corrugate plate) and stiffened folded laminate composite plate with in-axis configuration and off-axis configuration. In these studies, the effects of folding angles, fiber orientations, loading conditions, stiffener orientations, and boundary conditions have been investigated.

All of those analyses only investigated for a given fiber configuration. The optimal problem is not readily available.

Recently, Callahan and Weeks [9], Nagendra et al. [10], Le Riche and Haftka [11], Ball et al. [12] are among the first who adopted and used GA for stacking sequence design of laminated composite materials. GA has been used for several objective functions, such as strength [11, 13], buckling loads [11, 14 – 20], dimensional stability [21], strain energy absorption [22], weight (either as a constraint or as an objective to be minimized) [23, 24 - 27], bending/twisting coupling [17], stiffness [22, 28], fundamental frequencies [20, 24, 29], deflection [26] or finding the target lamination parameters [30].

Maximum frequency problems are of practical importance in the design of laminates for against resonance due to external excitation. The frequency of an external excitation can be placed either between zero and the first fundamental frequency or in gap between two consecutive higher-order frequencies depending on its magnitude. Hence, the optimal stacking sequence is to be determined such that the fundamental frequency or the frequency separation is maximized. Add-on, a practical approach is to design a laminate out of plies 0^0, $\pm 45^0$ and 90^0 orientations only that does not cover all possibilities.

On the other hand, only one author [31] has investigated frequency optimization of the laminated folded plate until now. The author presented a frequency optimization of symmetrically *one*- and *two-fold* folded composite plates by using the modified feasible direction method for the optimization routine and a program based on FORTRAN is used. However, the method of feasible directions (MFD) is created to solve optimization problems with inequality constraints. Starting from a feasible initial point, MFD tries to find a move to a better point without violating any of the constraints. Since a composite lay-up design problem usually includes several inequality constraints, MFD has been a good candidate for solving these problems. However, like other gradient-based methods, MFD is not always able to find the global optimum [32].

Therefore, in this study frequency optimization of the stiffened five folds folded laminate composite plate is investigated to fill this gap. The fundamental frequencies of the folded plates are maximized with respect to fiber orientations. The first-order shear deformation theory is used for vibration analysis of the folded plate. The GA method is used for this optimization analysis. The significant effects of obtained optimal fiber orientation on transient response of the folded plate are investigated for different boundary conditions.

2. THEORY AND FORMULATION

2.1. Displacement and strain field

According to the Reissner-Mindlin plate theory, the displacements (u, v, w) are referred to those of the mid-plane (u_0, v_0, w_0) as:

$$\begin{Bmatrix} u \\ v \\ w \end{Bmatrix} = \begin{Bmatrix} u_0 + z\theta_x \\ v_0 + z\theta_y \\ w_0 \end{Bmatrix} \text{ and } \begin{Bmatrix} \theta_x \\ \theta_y \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial w}{\partial x} + \phi_x \\ \dfrac{\partial w}{\partial y} + \phi_y \end{Bmatrix} \tag{1}$$

here, θ_x and θ_y are the total rotations, ϕ_x and ϕ_y are the constant average shear deformations about the y and x-axes, respectively.

The z-axis is normal to the xy-plane that coincides with the mid-plane of the laminate positive downward and clockwise with x and y.

The generalized displacement vector at the mid- plane can thus be defined as

$$\{d\} = \{u_0, v_0, w_0, \theta_x, \theta_y\}^T$$

The strain-displacement relations can be taken as:

$$\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial u_0}{\partial x} + z\dfrac{\partial \theta_x}{\partial x} \\ \dfrac{\partial v_0}{\partial y} + z\dfrac{\partial \theta_y}{\partial y} \\ \left(\dfrac{\partial u_0}{\partial y} + \dfrac{\partial v_0}{\partial x}\right) + z\left(\dfrac{\partial \theta_x}{\partial y} + \dfrac{\partial \theta_y}{\partial x}\right) \end{Bmatrix} \; ; \quad \begin{Bmatrix} \gamma_{yz} \\ \gamma_{xz} \end{Bmatrix} = \begin{Bmatrix} \dfrac{\partial w_0}{\partial y} + \theta_y \\ \dfrac{\partial w_0}{\partial x} + \theta_x \end{Bmatrix} \tag{2}$$

2.2. Finite element formulations

The Hamilton variation principle is used here to derive the laminate equations of motion (see [33]). In laminated plate theories, the membrane $\{N\}$, bending moment $\{M\}$ and shear stress $\{Q\}$ resultants can be obtained by integration of stresses over the laminate thickness. The stress resultants-strain relations expressed in the form:

$$\begin{Bmatrix} \{N\} \\ \{M\} \\ \{Q\} \end{Bmatrix} = \begin{bmatrix} [A_{ij}] & [B_{ij}] & [0] \\ [B_{ij}] & [D_{ij}] & [0] \\ [0] & [0] & [F_{ij}] \end{bmatrix} \begin{Bmatrix} \{\varepsilon^0\} \\ \{\kappa\} \\ \{\gamma^0\} \end{Bmatrix} \tag{3}$$

where

$$\left([A_{ij}],[B_{ij}],[D_{ij}]\right) = \sum_{k=1}^{n} \int_{h_{k-1}}^{h_k} \left([Q'_{ij}]_k\right)(1, z, z^2)\,dz \quad i, j = 1, 2, 6 \tag{4}$$

$$[F_{ij}] = \sum_{k=1}^{n} f \int_{h_{k-1}}^{h_k} \left([C'_{ij}]_k\right) dz; \quad f = 5/6;\; i, j = 4, 5 \tag{5}$$

n: number of layers, h_{k-1}, h_k : the position of the top and bottom faces of the k^{th} layer; $[Q'_{ij}]_k$ and $[C'_{ij}]_k$: reduced stiffness matrices of the k^{th} layer [34].

In the present work, eight nodded isoparametric quadrilateral element with five degrees of freedom per nodes is used. The displacement field of any point on the mid-plane given by:

$$u_0 = \sum_{i=1}^{8} N_i(\xi,\eta).u_i \; ; \; v_0 = \sum_{i=1}^{8} N_i(\xi,\eta).v_i \; ; \; \mathrm{w}_0 = \sum_{i=1}^{8} N_i(\xi,\eta).w_i \; ;$$

$$\theta_x = \sum_{i=1}^{8} N_i(\xi,\eta).\theta_{xi} \; ; \; \theta_y = \sum_{i=1}^{8} N_i(\xi,\eta).\theta_{yi} \tag{6}$$

where: $N_i(\xi,\eta)$ are the shape function associated with node i in terms of natural coordinates (ξ,η).

The element stiffness matrix are given by:

$$\left[k\right]_{e(40\times40)} = \int_{A_e} \left(\left[B\right]^{\mathrm{T}}\right)_{40\times8} \left[H\right]_{8\times8} \left[B\right]_{8\times40} t dA_e \tag{7}$$

The element stiffness matrix are given by:

$$\left[k\right]_e = \int_{\mathrm{A}_e} \left(\left[B\right]^{\mathrm{T}}\right)\left[H\right]\left[B\right] t d\mathrm{A}_e \tag{8}$$

The element mass matrix are given by:

$$\left[m\right]_e = \int_{A_e} \rho \left[N_i\right]^{\mathrm{T}} \left[\overline{m}\right]\left[N_i\right] dA_e \tag{9}$$

where $\left[H\right]$ is the material stiffness matrix; ρ is mass density of material; $\left[\overline{m}\right]$ is geometric inertia matrix (see [8]).

Nodal force vector is expressed as:

$$\left\{f\right\}_e = \int_{A_e} \left[N_i\right]^{\mathrm{T}} q dA_e \tag{10}$$

where q is the intensity of the applied load.

For free and forced vibration analysis, the damping effect is neglected; the governing equations are [33]:

$$[M]\{\ddot{u}\} + [K]\{u\} = \{0\} \text{ or } \left\{[M] - \omega^2 [K]\right\} = \{0\} \tag{11}$$

and

$$[M]\{\ddot{u}\} + [K]\{u\} = f(t). \tag{12}$$

In which $\{u\}$, $\{\ddot{u}\}$ are the global vectors of unknown nodal displacement, acceleration, respectively. $\left[M\right], \left[K\right]$, $f(t)$ are the global mass matrix, stiffness matrix, applied load vectors, respectively.

Where

$$[M]=\sum_{1}^{n}[m]_e\,;[K]=\sum_{1}^{n}[k]_e\,;\,\{f(t)\}=\sum_{1}^{n}\{f_e(t)\} \tag{13}$$

with n is the number of element.

When stiffener and folded plates are modeled by eight-nodded isoparametric rectangular plate element, the membrane and bending terms are coupled, as can be clearly seen in Fig.1. Even more since a rotation of the normal appear as unknowns for the Reissner–Mindlin model, it is necessary to introduce a new unknown for the in-plane rotation called drilling degree of freedom, θ_z. The rotation θ_z at a node is not measured and does not contribute to the strain energy stored in the element [8, 35]. The technique is used here: Before applying the transformation, the 40×40 stiffness and mass matrices are expanded to 48×48 sizes, to insert sixth θ_z drilling degrees of freedom at each node of a finite element. The off-diagonal terms corresponding to the θ_z terms are zeroes, while a very small positive number, we taken the θ_z equal to 10^{-4} times smaller than the smallest leading diagonal, is introduced at the corresponding leading diagonal term. The load vector is similarly expanded by using zero elements at corresponding locations. So that, for a folded element, the displacement vector of each node [5-8]:

$$\{u\}=[T]\{u'\} \tag{14}$$

$u=[u,v,w]^T$; $u'=[u',v',w']^T$ is the displacement of any generic point in global and local coordinate system, respectively; $[T]$ is the transformation matrix (see [7, 8]).

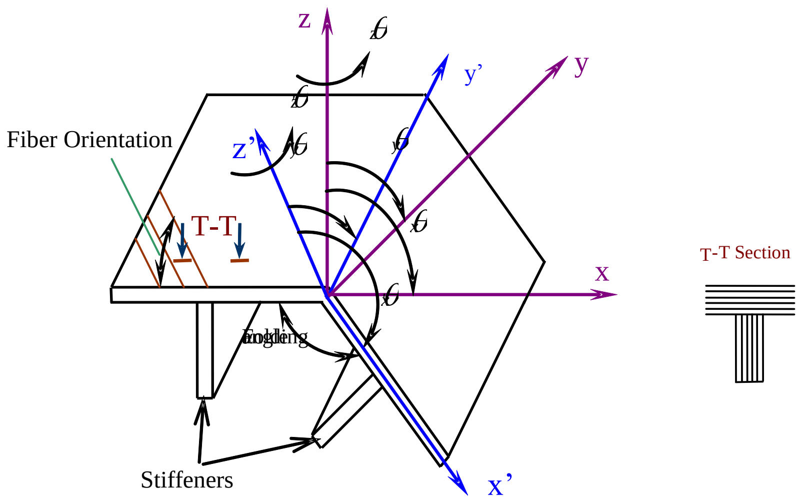


Figure 1. Global (x, y, z) and local (x', y', z') axes system for folded plate element, folding angle α.

2.3. Genetic algorithm

A genetic algorithm (GA) is an evolutionary optimization technique using Darwin's principal of survival of the fittest to improve a population of solutions. If the population size is suitably large, GA is not at the risk of being stuck in a local optimum. However, finding a global solution is not necessarily guaranteed to be successful unless an infinite number of iterations are performed. Despite the high computational cost, GA has been the most popular method for optimizing the stacking sequence of a laminated composite [36]. Its simple coding, which escapes gradient calculations, and its flexibility of being applied to a large variety of problems with different types of variables and objective functions make GA particularly useful for problems with multimodal functions, discrete variables, and functions with costly derivatives. The working principle of a GA is shown in Fig. 2.

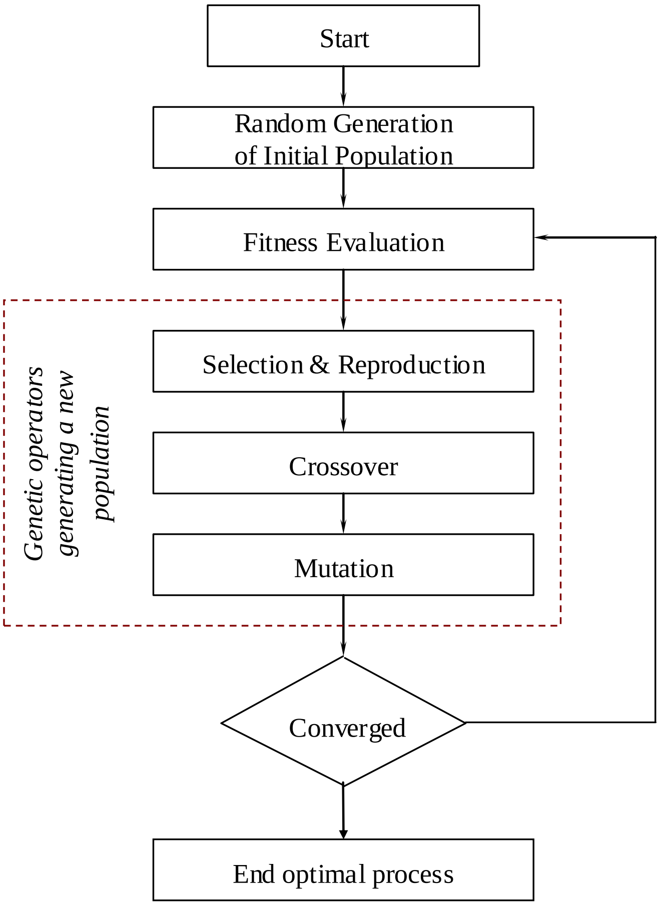


Figure 2. Flow chart of genetic algorithm.

When GA is applied to solve a practical problem, the parameter set of the problem first needs to be coded as a finite-length string (an individual). These to fall strings is known as a population. Each string represents one possible solution to the problem. GA begins by randomly generating an initial population of strings. Then, the fitness of individuals is evaluated using a cost function and each solution is ranked based on its fitness value. Selection, crossover and mutation are known as genetic operators through which new solutions to the problem are reproduced. A selection criterion is imposed to determine which solution should be kept and which should be discarded. According to the fitness of chromosomes the selection process chooses solutions to be advanced to the next generation. A crossover operator is applied to a pair of solution strings (parents) by exchanging a part of one string with another part of the other string to create two new solution strings (children). A mutation operator operates on only one string, thus creating a new string (a child). These three steps are repeated until a termination criterion is satisfied. The stopping criterion might be a maximum number of generations or it might be that the score of a potential solution must lie within a certain boundary [37].

In optimal problem of this paper, the GA is applied to determine optimal fiber orientation that the first frequency of the problem maximized. So that, the objective of optimization problem is formulated as:

Maximization: $\omega_1 = \omega_1(\theta_i)$: i=1,2,3,4

Subjected to $0^0 \le \theta_1, \theta_2, \theta_3, \theta_4 \le 90^0$

The natural frequency ω_1 for a given fiber orientation is determined from the finite element program given by Eq. (13).

The parameters of the GA are given in Table 1 for all cases.

Table 1. Parameter of genetic algorithm.

Parameter	Value
Number of individuals	10
Number of generations	100
Generation gap	0.9
Precision of variables	5
Number of variables	4 (for θ_i: i = 1 - 4)

3. NUMERICAL RESULTS

3.1. Validation examples

Firstly, to observe the accuracy of the present Matlab code and applied GA, the optimal fiber orientations of one fold and two folds folded laminate plate which plotted in Fig.3 are recalculated, which is a previously reported by Topal et al [31], 2008. Dimension parameters of the plate are illustrated in Fig 3 with L = 1 m, thickness of h = 0.01 L; for one-fold folded laminate plate: $b_1 = b_2 = L/2$; for two-folds folded laminate plate: $b_1 = b_2 = b_3 = L/3$. Material properties (T300/5208 graphite/epoxy): E_1 = 181 GPa, E_2 = 10.3 GPa, G_{12} = 7.17 GPa, υ_{12} = 0.28, density ρ = 1600kg/m^3. The symmetrically laminates folded plate is constructed of four layers with $\theta_1 = -\theta_2 = -\theta_3 = \theta_4 = \theta$. The thickness of each lamina is the same and not varied during the optimization. The results are compared with numerical results given by [31].

The first natural frequencies obtained from the present code and the results obtained by [31] are present in table2 for comparison. It is observed that the optimal fiber orientations and non-dimensional frequencies are in good agreement with other authors results.

Table 2. First five natural frequencies of isotropic stiffened flat plate.

Boundary conditions	One-fold				Two-folds			
	Topal et al [31]		Present (GA)		Topal et al [31]		Present (GA)	
	θ_{opt}(deg)	$\bar{\omega}$	θ_{opt}(deg)	$\bar{\omega}$	θ_{opt}(deg)	$\bar{\omega}$	θ_{opt}(deg)	$\bar{\omega}$
CCFF	0	27.738	0	27.468	0	29.159	0	28.912
CCCC	90	76.245	90	74.614	74.3	129.852	74.557	127.782

The non-dimensional frequency is defined as $\overline{\omega} = \omega L^2 \sqrt{\rho / E_2} / h$.

The boundary conditions are:

(CCFF): Two edges clamped and two edges free: At y = 0 and y = L: clamped; At straight lines: free.

(CCCC): Four edges clamped: At y = 0 and y = L: clamped; At straight lines: clamped.

The geometry parameters are taken as: L = 1 m; total thickness t = 0.01 L; folding angle $\alpha = 150^0$.

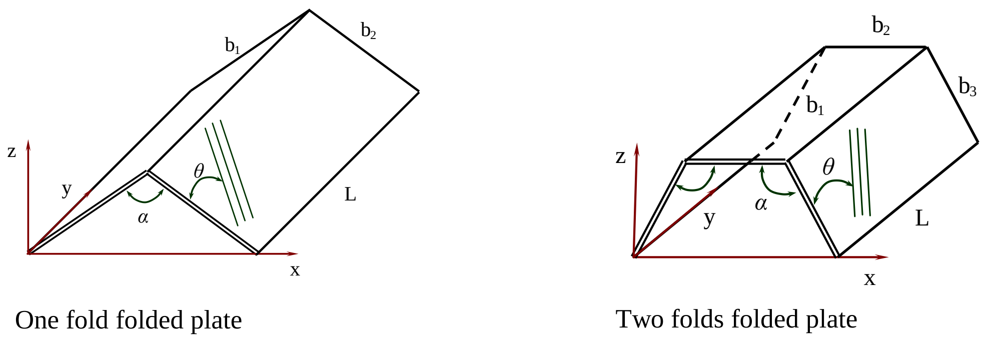


Figure 3. Geometry of one- and two folds folded laminated plates.

Figure 4, figure 5 plotted the variation of the population distribution as generations proceed in order to maximize the first fundamental frequency for different boundary conditions of one- and two folds folded laminated plates.

From Figs. 4, 5 we can see that the optimal value converges around the generation of 40.

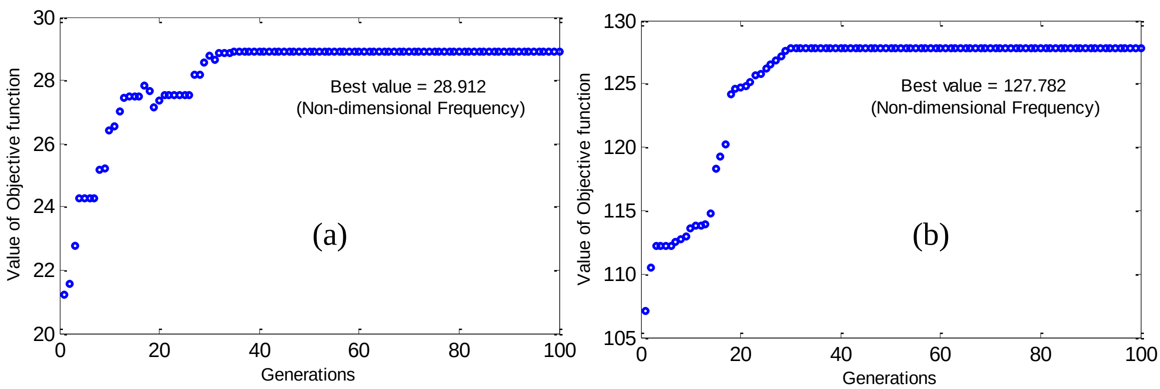


Figure 5. The variation of the objective function value with generation using GA of two-folded plate. (a) and (b): Two-folds with CCFF and CCCC boundary condition, respectively.

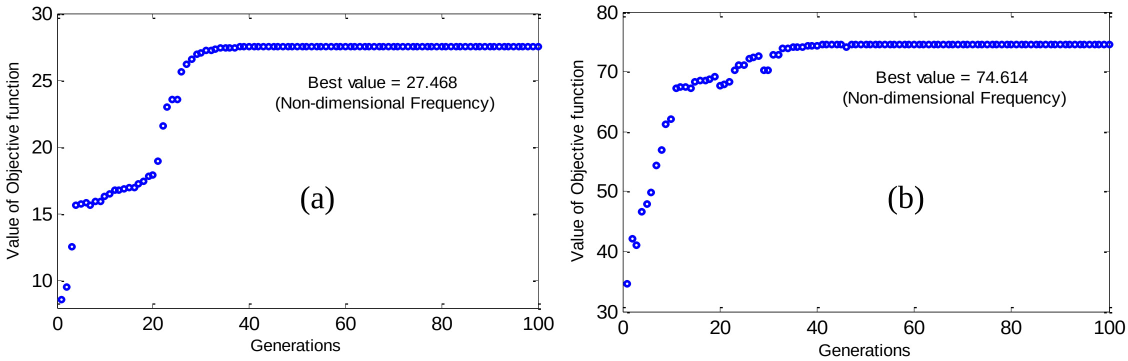


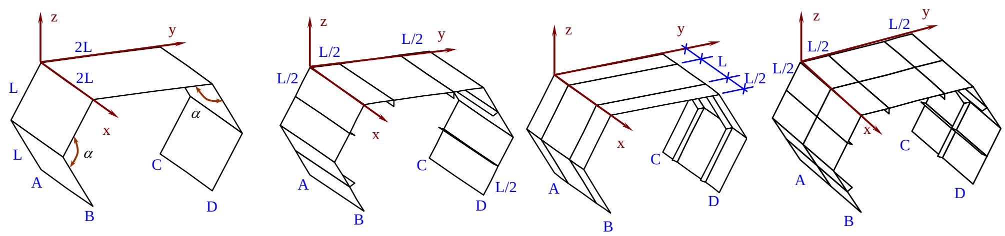
Figure 4. The variation of the objective function value with generation using GA of one-folded plate (a) and (b): One-fold with CCFF and CCCC boundary condition, respectively.

In the following subsections, several new numerical examples have been analyzed.

3.2 Study case: Folded laminated plate

3.2.1. Free vibration analysis and frequency optimization

Consider a five folds folded composite plate is shown in Fig. 6, the material properties shown in table 3, the dimension L = 1 m, total thickness t = 0.02 m, folding angle $\alpha = 120^0$.



a - Case 1 b - Case 2 c - Case 3 d - Case 4

Figure 6. Geometry of stiffened five-folds folded composite plate.

Four following cases for different stiffener orientations are studied:

Case 1: Unstiffened folded composite plate (Fig. 6a).

Case 2: Six x-stiffeners are attached below the folded plate running along the length of the clamped edges; width of stiffening plate taken equal to 5 cm and thickness remaining same as the original folded plate (Fig. 6b).

Case 3: Two y-stiffeners are attached below the folded plate along transverse direction; width of stiffening plate taken equal to 5cm and thickness remaining same as the original folded composite plate (Fig. 6c).

Case 4: Six x-stiffeners and one y-stiffener are attached below the folded plate (Fig. 6d).

The boundary conditions are:

- Clamped: at edges AB and CD: $u = v = w = \theta_x = \theta_y = \theta_z = 0$.

- Cantilever plate: clamped all edges at x = 0; ($u = v = w = \theta_x = \theta_y = \theta_z = 0$).

Table 3. Graphite-Epoxy (AS4/3501) material properties.

E_1 (GPa)	E_2 (GPa)	G_{12} (GPa)	G_{23} (GPa)	G_{13} (GPa)	υ_{12}	ρ (kg/m^3)
144.8	9.67	4.14	3.45	4.14	0.3	1500

Before frequency optimization, the folded plate with two basically lamination schemes: symmetric and anti-symmetric off-axis configurations [45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]; [45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0] and [60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]; [60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0] are taken to free vibration analysis. The reasons that we take the configurations to investigate in this section are [34]: For symmetric laminates, from the definition of $[B_{ij}]$ (see Eq. 4) matrix, it can be proved $[B_{ij}] = 0$. Hence, there is uncoupling between the bending deformation and shear strain. For anti-symmetric laminate, the matrix $[B_{ij}]$ do not vanish. However, these two types of laminate configurations have uncoupling bending-twisting. For this reason, the method used for calculating natural frequency should take into account this effect in order to avoid the associated error and the resulting false optimum designs.

Then, the optimization procedure involves the stages of evaluating the natural frequencies and improving the fiber orientation θ_i to maximize the first frequency ω_1 using genetic algorithm.

The configurations of the folded laminate plates for optimal design are:

Symmetric configurations: [θ_1^0/$-\theta_2^0$/θ_3^0/$-\theta_4^0$/$-\theta_4^0$/θ_3^0//$-\theta_2^0$/θ_1^0]: and we denoted as OPT_SC1; OPT_SC2; OPT_SC3 and OPT_SC4 for cases (1 to 4), respectively.

Anti-symmetric configurations: [θ_1^0/$-\theta_2^0$/θ_3^0/$-\theta_4^0$/θ_4^0/$-\theta_3^0$//θ_2^0/$-\theta_1^0$]: similarly, we denoted as OPT_Anti_SC1, OPT_Anti_SC2, OPT_Anti_SC3 and OPT_Anti_SC4 for cases (1 to 4), respectively.

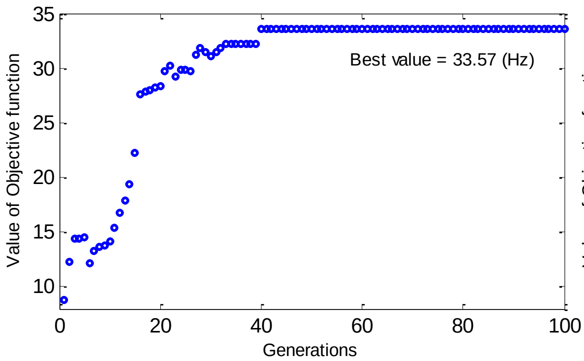


Figure 7. The variation of the objective function value for Case 1 with anti-symmetric configurations and clamped at edges AB, CD: **OPT_Anti_SC1.**

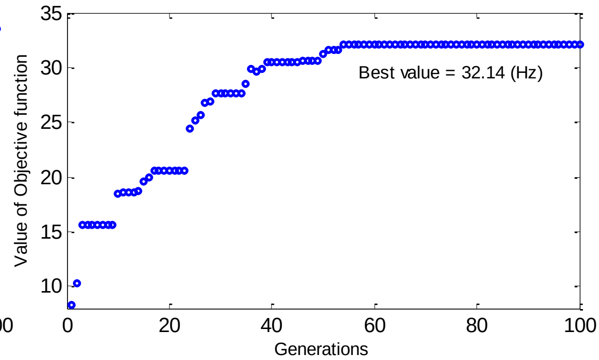


Figure 8. The variation of the objective function value for Case 2 with anti-symmetric configurationsand clamped at edges AB, CD: **OPT_Anti_SC2.**

The first three natural frequencies of the given symmetric and anti-symmetric off-axis configurations and obtained optimal frequencies of the plates are compared in Table 4 for clamped at edges AB,CD and in Table 5 for folded cantilever plates, respectively.

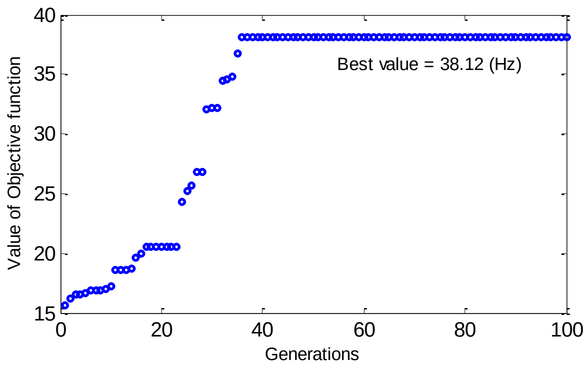


Figure 9. The variation of the objective function value for Case 3 with anti-symmetric configurations and clamped at edges AB, CD: **OPT_Anti_SC3.**

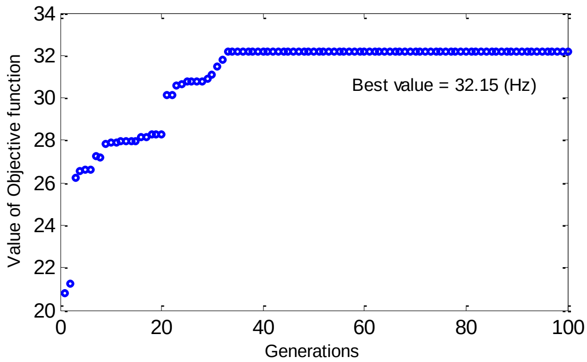


Figure 10. The variation of the objective function value for Case 4 with anti-symmetric configurations and clamped at edges AB, CD: **OPT_Anti_SC4**.

Figure 7 to figure 10 are shown for the variation of the population distribution as generations proceed in order to maximize the first fundamental frequency for cases (1 to 4) with clamped at edges AB and CD.

From Table 4 and Table 5, it is observed that the optimal natural frequencies of anti-symmetric and symmetric configurations are similar for all studied cases. All the obtained optimal solutions exhibit higher natural frequencies than those given routine configurations. Case 3 gives the highest optimal frequency among the four cases, although mass added is least in the stiffened folded plates.

When the plates clamped at edges (AB, CD), case 1 and case 2 give the same optimal stacking sequences but the first optimal frequency f_1 of case 2 is lower than optimal frequency of case 1. The phenomenon should be explained that the flexural rigidity of the plate decrease as the effect of inertial momentum of *x*-stiffeners. The optimal frequency of case 4 do not make any significant change over the unstiffened folded plates, although mass of structure is increase.

Figures 11 to figure 14 are shown for the variation of the population distribution as generations proceeds in order to maximize the first fundamental frequency for cases (1 to 4) with cantilever boundary condition. The optimal values were determined around the generation of 45.

Table 4. Frequency optimization of five folds folded plate with respect to fiber orientations.

Boundary condition: Clamped at edges AB and CD					
		Fiber orientations	Natural frequencies		
			f_1	f_2	f_3
Case 1	SC1_1	$[45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]$	17.38	36.27	49.15
	SC1_2	$[60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]$	11.46	23.88	39.56
	Anti_SC1_1	$[45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0]$	17.54	36.57	49.12
	Anti_SC1_2	$[60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0]$	11.54	24.15	40.05
	OPT_SC1	$\mathbf{[0^0/0^0/0^0/0^0/0^0/0^0/0^0/0^0]}$	**33.57**	56.47	71.68

	OPT_Anti_SC1	**[$0^0/0^0/0^0/-1^0/1^0/0^0/0^0/0^0$]**	**33.57**	56.34	71.78
Case 2	SC2_1	[$45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0$]	16.83	35.42	47.71
	SC2_2	[$60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0$]	11.14	23.18	39.42
	Anti_SC2_1	[$45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0$]	16.97	35.28	48.15
	Anti_SC2_2	[$60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0$]	11.25	23.48	39.52
	OPT_SC2	**[$0^0/0^0/0^0/0^0/0^0/0^0/0^0/0^0$]**	**32.13**	54.42	68.26
	OPT_Anti_SC2	**[$0^0/0^0/0^0/0^0/0^0/0^0/0^0/0^0$]**	**32.14**	54.17	68.34
Case 3	SC3_1	[$45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0$]	23.15	51.41	55.23
	SC3_2	[$60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0$]	20.15	49.36	50.97
	Anti_SC3_1	[$45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0$]	23.13	51.27	54.93
	Anti_SC3_2	[$60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0$]	19.46	49.05	51.23
	OPT_SC3	**[$0^0/-6^0/27^0/-85^0/-85^0/27^0/-6^0/0^0$]**	**38.15**	64.86	84.73
	OPT_Anti_SC3	**[$0^0/0^0/1^0/-68^0/68^0/-1^0/0^0/0^0$]**	**38.12**	63.96	84.72
Case 4	SC4_1	[$45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0$]	19.62	43.58	48.61
	SC4_2	[$60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0$]	21.46	39.15	50.48
	Anti_SC4_1	[$45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0$]	19.92	43.46	48.21
	Anti_SC4_2	[$60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0$]	21.43	38.97	51.24
	OPT_SC4	**[$88^0/0^0/9^0/-24^0/-24^0/9^0/0^0/88^0$]**	**32.05**	41.32	67.24
	OPT_Anti_SC4	**[$1^0/-90^0/1^0/-30^0/30^0/-1^0/90^0/-1^0$]**	**32.15**	40.18	67.93

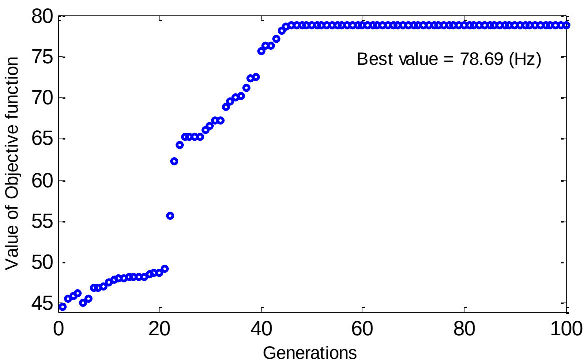


Figure 11. The variation of the objective function value for cantilever plate of case 1 with anti-symmetric configurations.

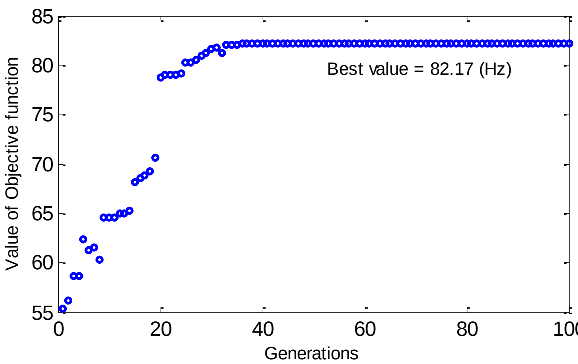


Figure 12. The variation of the objective function value for cantilever plate of case 2 with anti-symmetric configurations.

Table 5. Frequency optimization of five folds folded plate with respect to fiber orientations.

Boundary condition: Cantilever plate: clamped at x = 0					
		Fiber orientations	Natural frequencies		
			f_1	f_2	f_3
Case 1	SC1_1	$[45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]$	65.93	73.76	87.82
	SC1_2	$[60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]$	60.54	67.17	72.48
	Anti_SC1_1	$[45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0]$	69.93	70.77	87.97
	Anti_SC1_2	$[60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0]$	63.48	65.12	72.99
	OPT_SC1	$\mathbf{[8^0/-25^0/56^0/-77^0/-77^0/56^0/-25^0/8^0]}$	**78.61**	79.23	121.58
	OPT_Anti_SC1	$\mathbf{[7^0/-28^0/45^0/-85^0/85^0/-45^0/28^0/-7^0]}$	**78.69**	79.32	121.19
Case 2	SC2_1	$[45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]$	67.48	75.34	86.37
	SC2_2	$[60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]$	65.15	71.83	76.15
	Anti_SC2_1	$[45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0]$	71.24	73.48	86.37
	Anti_SC2_2	$[60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0]$	67.83	69.91	74.92
	OPT_SC2	$\mathbf{[11^0/-35^0/90^0/-86^0/-86^0/90^0/-35^0/11^0]}$	**81.52**	82.06	116.17
	OPT_Anti_SC2	$\mathbf{[16^0/-33^0/74^0/-87^0/87^0/-74^0/33^0/-16^0]}$	**82.17**	82.93	115.78
Case 3	SC3_1	$[45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]$	77.73	85.15	103.46
	SC3_2	$[60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]$	77.87	89.14	113.24
	Anti_SC3_1	$[45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0]$	81.56	82.94	103.48
	Anti_SC3_2	$[60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0]$	75.23	92.46	111.98
	OPT_SC3	$\mathbf{[15^0/-35^0/66^0/-84^0/-84^0/66^0/-35^0/15^0]}$	**98.67**	100.58	147.99
	OPT_Anti_SC3	$\mathbf{[12^0/-32^0/60^0/-83^0/83^0/-60^0/32^0/-12^0]}$	**98.39**	100.77	147.79
Case 4	SC4_1	$[45^0/-45^0/45^0/-45^0/-45^0/45^0/-45^0/45^0]$	75.18	77.47	93.14
	SC4_2	$[60^0/-60^0/60^0/-60^0/-60^0/60^0/-60^0/60^0]$	83.42	89.15	112.76
	Anti_SC4_1	$[45^0/-45^0/45^0/-45^0/45^0/-45^0/45^0/-45^0]$	76.37	78.43	91.19
	Anti_SC4_2	$[60^0/-60^0/60^0/-60^0/60^0/-60^0/60^0/-60^0]$	85.46	87.94	113.52
	OPT_SC4	$\mathbf{[86^0/-63^0/58^0/-86^0/-86^0/58^0/-63^0/86^0]}$	**90.42**	91.07	127.73
	OPT_Anti_SC4	$\mathbf{[86^0/-64^0/59^0/0^0/0^0/-59^0/-64^0/-86^0]}$	**92.16**	93.12	131.11

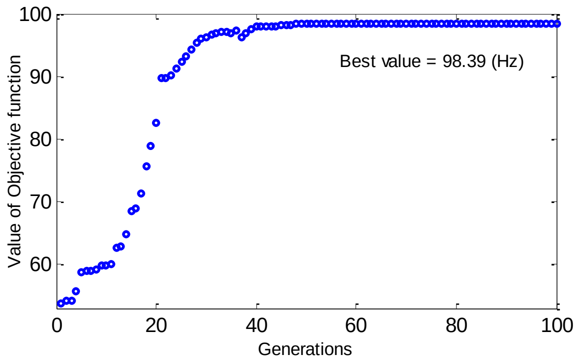


Figure 13. The variation of the objective function value for cantilever plate of case 3 with anti-symmetric configurations.

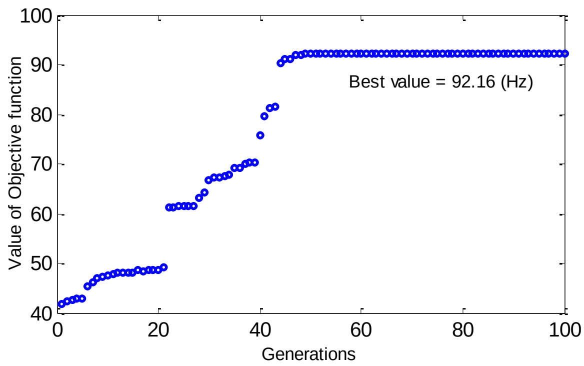


Figure 14. The variation of the objective function value for cantilever plate of case 4 with anti-symmetric configurations.

The optimal frequencies of cantilever folded plates are extremely higher the others which having clamped at edges (AB, CD). The obtained stacking sequences of cantilever plates are more different comparing with clamped at edges (AB, CD) cases. The phenomenon makes sense to us because the difference of mode shapes between two kinds of boundary conditions. We can conclude that the optimal result is significant that depend on geometry and boundary condition of the plates.

3.2.2 Transient displacement response

In order to investigate the effect of optimal fiber orientation on transient response of the plates: the plates are subjected to a uniformly exploded load of intensity q = 10 kN/m^2 on the top individual plate, towards the negative direction of the z-axis (plotted in Fig. 16).

The exploded loading condition scheme (with t_1 = 1 ms, t_2 = 2 ms, t_3 = 50 ms) is illustrated in Fig. 15.

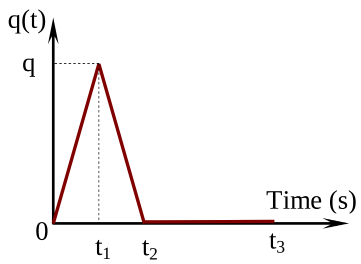


Figure 15. Exploded loading condition scheme

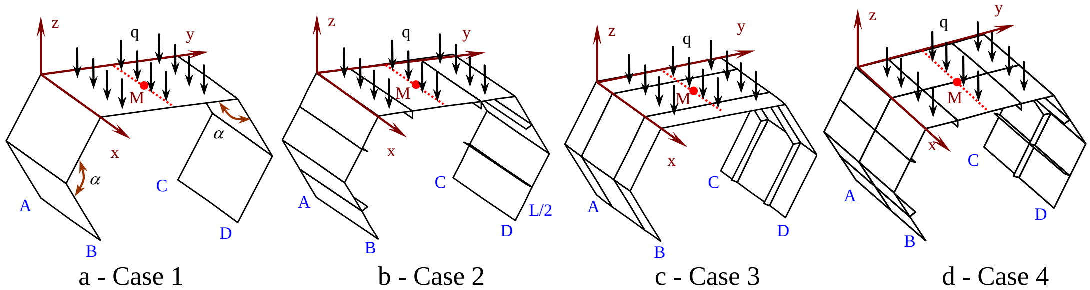
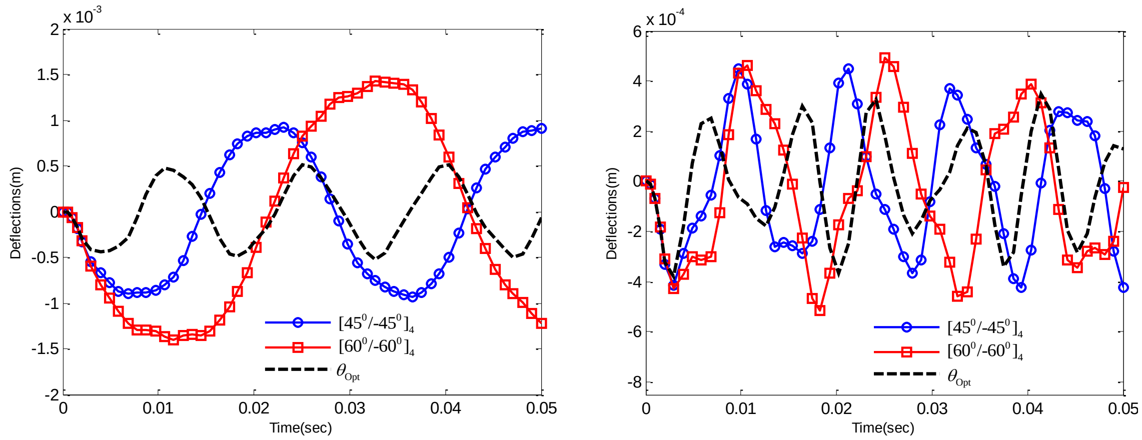


Figure 16. Unstiffened and stiffened five folds folded composite plate subjected to uniformly distributed load of density q.

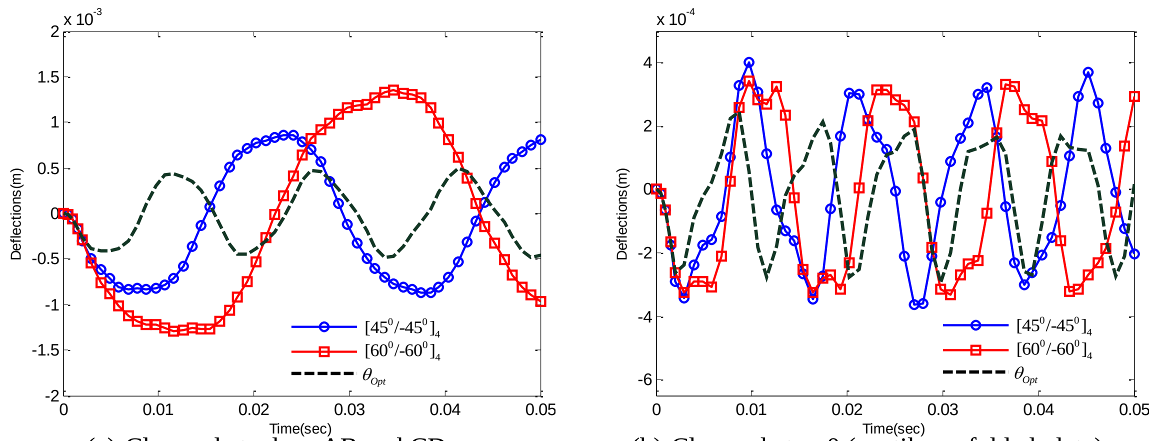
Two lamination schemes: symmetric and anti-symmetric off-axis configurations: $[45^0/-45^0]_4$; $[60^0/-60^0]_4$ are re-taken as routine configurations. The comparisons of displacement responses at the center point M of the plates are plotted in Fig. 17 - Fig. 20 for different boundary condition.



(a)-Clamped at edges AB and CD (b)-Clamped at x = 0 (cantilever folded plate)

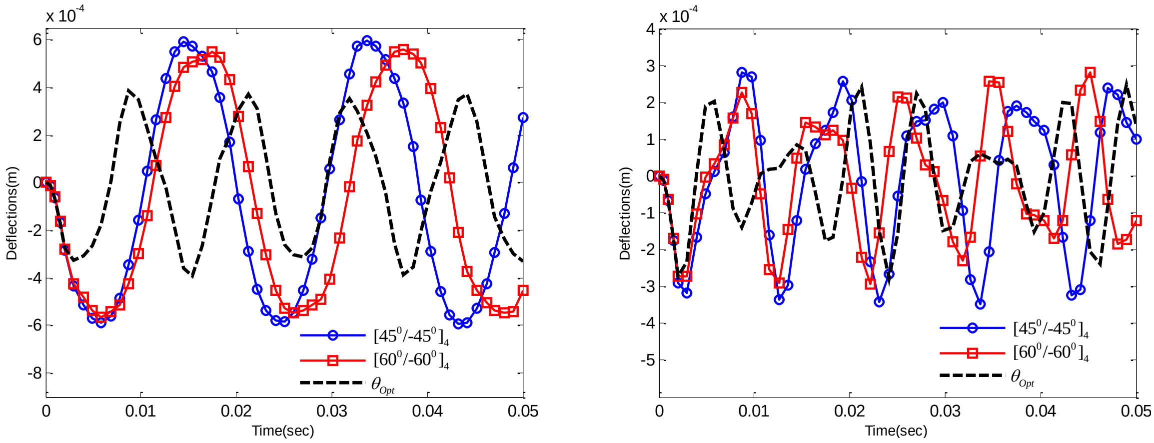
Figure 17. Center displacement response of the unstiffened folded plate (Case 1) with and without optimal design subjected to uniformly distributed load.

Figures 17a, Fig. 18a, Fig. 19a, Fig. 20a and Fig. 17b, Fig. 18b, Fig. 19b, Fig. 20b plotted the comparison of center displacement response of case (1 to 4) for clamped at edges (AB, CD) and cantilever boundary condition, respectively.



(a)-Clamped at edges AB and CD (b)-Clamped at x=0 (cantilever folded plate)

Figure 18. Center displacement response of the stiffened folded plate (Case2) with and without optimal design subjected to uniformly distributed load.

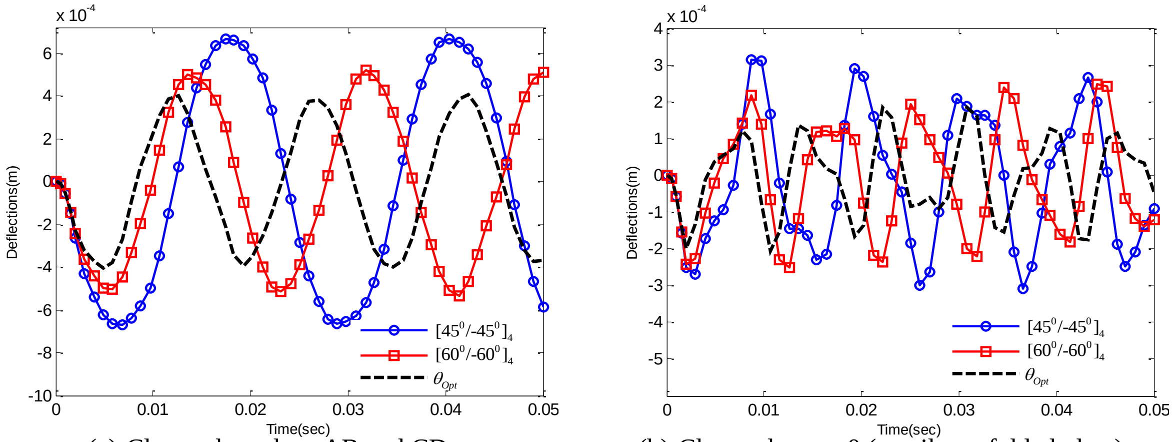


(a)-Clamped at edges AB and CD (b)-Clamped at *x*=0 (cantilever folded plate)

Figure 19. Center displacement response of the stiffened folded plate (Case3) with and without optimal design subjected to uniformly distributed load.

From Figs. 17 to 20, it is observed that the displacement amplitude and wave of optimal configurations change more dramatic. It is a significant increase of frequencies and significant decrease of amplitudes. The effects become more rapidly for cantilever plates. This observation provides us a clue that the cantilever plate could be vibrative extinction more quickly than other case.

For the same boundary condition and loading condition, the displacement response of case 3 is least.



(a)-Clamped at edges AB and CD (b)-Clamped at x = 0 (cantilever folded plate)

Figure 20. Center displacement response of the stiffened folded plate (Case4) with and without optimal design subjected to uniformly distributed load.

4. CONCLUSION

In this study, a computer code has been developed for optimization of the unstiffened and stiffened folded laminate composite plate using genetic algorithm. The code has two distinct modules. First, a finite element code using an eight-nodded isoparametric plate elements, based on the first order shear deformation theory for calculating natural frequencies of the folded laminate composite plate. In this module, the transverse shear deformation, the rotary inertia of plate and stiffeners are considered to see that the more advanced in presented model. Second one is the GA module for solving optimization problem.

We conclude that GA can be successfully employed for optimal design the unstiffened and stiffened folded laminate composite plate with any number of design variables. The GA is guided random and exhaustive search process, hence the probability of finding the global optimum is high and the variables could be real.

By using more than one variables approach, the optimal results of stacking sequence can be modified to suit the designers' requirement.

Some set of new results are presented to see the effects of optimal fiber orientations on dynamic responses of unstiffened and stiffened folded laminate composite plates for different boundary conditions.

The applicability of the present approach covers a wide range of forced vibration problems, with varying material combinations, geometric features, and boundary conditions.

The results of this study will serve as a benchmark for future research for designing folded composite structures and sandwich structures made of composite materials, as it was extremely quick and reliable in producing design results.

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TÓM TẮT

TỐI ƯU TẦN SỐ VÀ PHÂN TÍCH ĐÁP ỨNG TỨC THỜI CỦA TẤM COMPOSITE GẤP NẾP CÓ GÂN GIA CƯỜNG

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Bài báo trình bày bài toán tối ưu tần số của tấm composite lớp có gân gia cường bằng thuật toán di truyền (GA). Chương trình Matlab bằng phương pháp PTHH dựa trên lí thuyết biến dạng cắt bậc nhất với việc sử dụng phần tử đẳng tham số 8 nút, mỗi nút có 5 bậc tự do được thiết lập cho bài toán phân tích dao động riêng và phân tích đáp ứng tức thời của chuyển vị tấm theo thời gian. Mô đun thuật toán GA được tác giả áp dụng cho bài toán tối ưu tần số với việc điều khiển các biến số thiết kế của bài toán (một biến và nhiều biến thiết kế). Ảnh hưởng của cấu hình tối ưu đến đáp ứng tức thời của tấm được xem xét với các điều kiện biên khác nhau. Sự đúng đắn của thuật toán và chương trình được khẳng định khi so sánh các kết quả với kết quả giải bằng các phương pháp khác đã công bố trên các tạp chí thế giới có uy tín.

Từ khoá: bài toán tối ưu, tấm compozit cố gân, thuật toán di truyền, điều kiện biên.