

Modal Damping Ratio of Symmetric Laminate Composite Under the Effect of Attached Mass Using Experimental Design

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Nowadays, the use of composite materials has taken a large place in civilian industries as well as in military and aerospace industries. Therefore, significant investigations about their mechanical and physical properties are needed. The present study addresses the effect of attached mass on damping ratio of symmetric angle ply laminate composite. Furthermore, factor influencing the effect of attached mass on damping ratio of laminate composite are studied using Taguchi method. The considered factors parameters are: attached mass locations from the clamped edge, stacking sequences and boundary conditions. The results of this study indicate that the damping ratio of the laminate composite plates is sensitive to the attached mass, where the damping ratio is found to be proportional to the locations of the attached mass. The findings of this study indicate that the attached mass decreases frequency parameter and increase the damping ratio of the composite plate, if it is inserted at a point other than a nodal line. In addition, the paper presents a good correlation between the numerical results of the fundamental frequency obtained by the ANSYS software and those obtained experimentally.

Keywords : Laminate composite, Fundamental frequency, Damping ratio, Attached mass, vibration..

1. INTRODUCTION

The composite materials are fabricated to have better engineering properties than the conventional ones, such as metals. Some of the properties that can be improved by forming composite materials are: stiffness-to-weight ratio, strength, corrosion resistance, thermal properties, fatigue life and wear resistance. With regard to the vibration behaviour of the composite structures, many studies were carried out to control and determine the dynamic characteristics of those materials. So, several authors have performed vibration analysis of laminated composite under various conditions of reinforcement and configuration [1-7].

In many engineering applications such as satellites systems and aviations, substructures are added to the structures. These attached masses have an effect on the response of the main structures, principally, the vibration behaviour [8-10]. Therefore, it is important to understand properties of vibration of the composite plates under diversity of loading conditions with and without attached mass. Several studies have presented vibration analysis of composite materials with concentrated attached masses at separate locations under various arrangements and conditions of reinforcement. The free vibration of a simply supported laminated composite plate with distributed attached mass was presented by Alibeigloo et

al. [11]. The problem is solved using Hamilton's Principle. The obtained results showed that the fundamental frequencies depend on the amount of the patch mass and its location. The numerical simulations and experiments of free vibration behaviours of carbon fibre reinforced composite lattice-core sandwich cylinder (LSC) were studied under different boundary conditions, including end-free constraints by Yongshuai Han et al. [12] the results revealed that the attached masses affect the vibration modes and decrease the fundamental frequency. S.M.R. Khalili et al. [13] investigated the free vibrations of a cross-ply composite shell with and without regularly distributed attached mass, under a diversity of conditions such as the thickness of the shell and the thickness of the distributed attached mass using higher order shell theory. The thick laminated circular plates with attached rigid core were investigated on free vibration by Hosseini-Hashemi et al. [14], the results confirm that the present study is in excellent agreement with those of 3-D FEM for thick laminated composites structures. K. Malekzadeh and A. Sayyidmousavi [15] used the finite element method (FEM) under ANSYS, for analysis the free vibration of rectangular sandwich plates with a flexible core, with and without distributed attached mass on the top face sheet. The results show that the attached mass reduces the frequencies of the sandwich plate. A. Rahmane et al. [16] presented a numerical and experimental method for evaluation the effect of attached mass on dynamic properties of composite laminate plates. The study examines the effect of various factors such as number of layers, position of attached mass and fiber orientation) on frequencies.

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With regard to the experiments plan using Taguchi method, different studies were used to investigate the mechanical behaviour of the composite structures. In the study of H. Omidvar et al. [17], the experimental method of Taguchi was used to find the relationship between different factors that control the optimal ballistic behaviour in Kevlar-epoxy composite materials. Tom Sunny et al. [18] used the Taguchi method to study delamination in drilling GFRP composites. The results indicated that feed rate is the more influential factor on delamination than spindle speed. Manjeet Singh et al. [19], studied geometric parameters for double-spindle joint configurations made of nanoclay glass / epoxy laminates by the use of the Taguchi method. The results show that the most significant factor to increase the bearing strength is the E/D (the diameter of the first hole) ratio in serial and parallel pin joint configurations. Ch.-Sh. Chen et al. [20] developed an efficient method based on Taguchi method and finite elements to determine the optimum conditions providing the greatest thickness of the shear layer in composites and to quantify the distributions of the fibre orientation in short-fibre-reinforced composites.

Damping is very important to characterize the dynamic behaviour of fibre-reinforced composites regarding to minimization of resonant vibrations. In addition, its measurements are essential for performance control and structural safety. Several authors have studied the effect of damping on laminated composite under different conditions of reinforcement and orientations of layers. Among that studies, the damping behaviour of carbon fibre and flax fibre reinforced polymer (CFRP and FFRP) composites are studied by M. Rueppel et al. [21] the authors used the logarithmic decrement measurements, dynamic analysis and vibration beam measurements, this work demonstrates that the FFRP may efficient at damping compared to CFRP. Wang and Guild [22] used the vibration technique to measure the dynamic modulus of elasticity and damping ratio of wood-based on composites. The results revealed that a good linear agreement exists between dynamic modulus of elasticity and static modulus of elasticity. Aytac Arikoglu and Ahmet Gokay Ozturk [23] presented a novel approach for evaluation of the damping and vibration analysis of arbitrarily curved laminated composite and sandwich beams. The results obtained were indicated that the present study is in very good agreement with the ones that exist in the literature. The paper of Yi He et al [24] presents an efficient finite element method (FEM) for computing the modal damping of carbon/epoxy laminated composites using ABAQUS software. The obtained results from this model are in good agreement with the test data. Thus, the proposed by using finite element method is capable of providing an effective method to determine the modal damping of composite materials. D.A. Pereira et al. [25] presented a numerical and experimental method for evaluation of the influence of fibre orientations on the modal characteristics of composite laminates, emphasizing values of modal damping. The results confirm the efficiency of design strategies intended to regulate and augment the damping levels of composite laminates by exploring the different steering of fibre. El Mahi et al.

[26] modelled the damping ratio of the laminate composites. The obtained results showed that the analysis by using Ritz method is able to evaluate fairly well the damping proprieties of unidirectional materials and laminates.

For all of the literature mentioned above the effects of the attached mass on the modal damping ratio of the composite plate were not considered. Therefore, the aim of this work is to fill this gap by studying the effects of the attached mass on the modal damping ratio as a function of three factors (mass locations, boundary conditions: Clamped-Free-Free-Free (CFFF) and Clamped-Free-Clamped-Free (CFCF) and stacking sequences) on damping ratio for fundamental frequency (1st Mode) of laminate composite under flexural vibration. Furthermore, the paper presents a finite element model (FEM) carried out by using ANSYS software and validated experimentally. In the study, factors influencing the effect of attached mass on damping ratio of fundamental frequency are studied using experiments plan by Taguchi method.

2. EXPERIMENTAL SET-UP

Experimental tests have been carried out with the aim to understand the dynamic behaviour of laminate composite. The dimensions of the specimens are: a = 270 mm (along x direction), b = 300 mm (along y direction). They consist of 8 layers of unidirectional fibre carbon / epoxy (IM7/8552). The thickness of each layer is 0.125 mm.

IM7 carbon fibre is a continuous, high performance, intermediate modulus. The unique properties of IM7 fibre, such as modulus, higher tensile strength and good shear strength, allow structural designers to achieve higher safety margins for the stiffness and strength critical applications. 8552 epoxy matrix is a high performance tough for use in primary aerospace structures. It exhibits good impact resistance and damage tolerance for a wide range of applications. The volume fraction of fibres is calculated and found to be about 0.577. The mechanical properties for the carbon fibre and the matrix used in the analysis are given in Table 1 [27].

Table1. Mechanical properties of carbon fibre and matrix

Properties	E ₁ (GPa)	E ₂ (GPa)	G ₁₂ (GPa)	ν ₁₂
Fibre	276	19.5	70	0.28
Matrix	4.76	4.76	1.74	0.37

The mechanical properties of the laminates are given in Table 2.

Table 2. Mechanical properties of IM7/8552 carbon/epoxy [28]

Young's modulus (GPa)		Shear modulus (GPa)		Poisson ratio	
E ₁₁	E ₂₂ , E ₃₃	G ₁₂ , G ₁₃	G ₂₃	ν ₁₂ , ν ₁₃	ν ₃₂
164.87	9.81	4.8	3.2	0.31	0.52

Three laminated composite plates with the symmetrical stacking sequences of $[\pm 20]_{2s}$, $[\pm 25]_{2s}$ and $[\pm 30]_{2s}$ are excited in free vibration (These staking sequences were chosen on the basis of the great variation in the damping ratios between them, compared to the remaining sequences). The specimens are tested for

clamped-free-free-free (CFFF) and clamped-free-clamped-free (CFCF) boundary conditions. The experiments are carried out to identify the effect of attached mass on natural frequencies and damping ratio as a function of mass locations from the clamped edge. Figure 1 represents the locations of the attached mass to the plates. The masses are located in the middle of the direction (y) and spaced 67.5 mm in the (x) direction. The attached mass weight is $M/4$, Where M is about 120 g and represent the weight of the plates contain 8 layers.

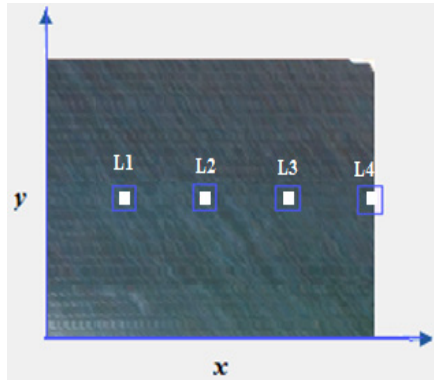


Figure 1. Location of the attached mass

The experimental tests are carried out by using a vibration analyzer of the type B&K. It is connected to an impact hammer used to induce excitation signal. A low mass accelerometer is used to detect the vibration response. During the measurements, the mass including with the mass of accelerometer was moved on all the mesh measurement points presented in Figure. 1. The experimental equipments are shown in Figure. 2.

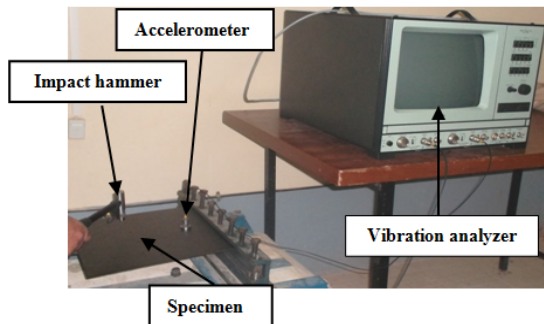


Figure 2. Experimental equipment

The method used for determining the damping at a resonance, is called the Half-Power Bandwidth Method. The approach is based on finding the bandwidth for each mode. The bandwidth is the normalized Δf across the resonant response between the -3 dB points on the transfer magnitude curve (Figure 4). For a particular mode, the damping ratio (ζ) can be found from the following equation (1):

$$\zeta = \frac{\Delta f}{2f_r} \quad (1)$$

where Δf : is the frequency bandwidth between the two half power points. f_r : is the resonance frequency.

A model of the response on frequency with attached mass (L3) of the CFFF specimen used in the tests are presented in Figure. 3.

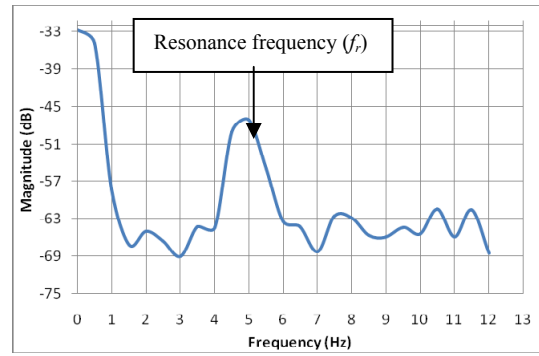


Figure 3. Response of frequency with attached mass of the CFFF specimen $[\pm 20]_{2s}$

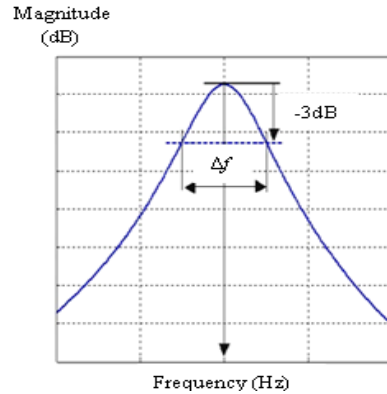


Figure 4. The half-power bandwidth method

3. ANALYSIS

The general governing equation of motion of free vibration of the plate system is given by (2):

$$[M]\{\ddot{y}\} + [K]\{y\} = \{0\} \quad (2)$$

where $[M]$ is the structural mass matrix, $[K]$ is the structural stiffness matrix, $\{\ddot{y}\}$ is the nodal acceleration vector and $\{y\}$ is the nodal displacement vector. Equation (2) is used to determine the mode shapes and fundamental frequencies of the plates.

In this work, the plate with and without attached mass is considered for the analysis of modes and fundamental frequencies. The composite plate having thickness h and the x - y plane is the mid-plane of the plate and the z -axis is normal to the plate (Figure. 1). Ordinarily, the resultant forces and moments are written in:

$$\begin{Bmatrix} N \\ M \end{Bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix} \begin{Bmatrix} \varepsilon^0 \\ K \end{Bmatrix} \quad (3)$$

$$A_{ij} = \sum_{K=1}^n (Q'_{ij})_K e_k \quad (4)$$

$$B_{ij} = \sum_{K=1}^n (Q'_{ij})_K e_k z_K \quad (5)$$

$$D_{ij} = \sum_{K=1}^n (Q'_{ij})_K (e_k z_K^2 + \frac{e_k^3}{12}) \quad (6)$$

where the A_{ij} , B_{ij} , and D_{ij} are conventional laminate stiffness coefficients and Q'_{ij} are the components of transformed lamina stiffness matrix.

The attached mass to the composite plate is assumed to be a point mass. It is assumed that the attached mass does not prevent any bending of the plate. The strain energy of deformation for a thin plate is:

$$U_d = \frac{1}{2} \iiint (\sigma_{xx}\varepsilon_{xx} + \sigma_{yy}\varepsilon_{yy} + \sigma_{zz}\varepsilon_{zz} + \sigma_{xz}\gamma_{xz} + \sigma_{yz}\gamma_{yz} + \sigma_{xy}\gamma_{xy}) dx dy dz \quad (7)$$

where ε_x and ε_y are the strains along the x and y directions, respectively, and γ_{xy} is the shear strain

Taking into account the assumptions of laminate theory: $\sigma_{zz} = \gamma_{xz} = \gamma_{yz} = 0$ and in the case of symmetrical laminates, the membrane-bending coupling terms B_{ij} are zero. The strain energy is written:

$$U_d = \frac{1}{2} \iiint (+Q_{11}^k \varepsilon_{xx}^2 + 2Q_{12}^k \varepsilon_{xx} \varepsilon_{yy} + 2Q_{16}^k \varepsilon_{xx} \gamma_{xy} + 2Q_{26}^k \varepsilon_{yy} \gamma_{xy} + Q_{22}^k \varepsilon_{yy}^2 + Q_{66}^k \gamma_{xy}^2) dx dy dz \quad (8)$$

This relation can be expressed according to the displacements u_0 , v_0 and w_0 :

$$U_d = \frac{1}{2} \iint \left\{ A_{11} \left(\frac{\partial u_0}{\partial x} \right)^2 + 2A_{12} \frac{\partial u_0}{\partial x} \frac{\partial v_0}{\partial y} + A_{22} \left(\frac{\partial v_0}{\partial y} \right)^2 + 2 \left(A_{16} \frac{\partial u_0}{\partial x} + A_{26} \frac{\partial v_0}{\partial y} \right) \left(\frac{\partial u_0}{\partial y} + \frac{\partial v_0}{\partial x} \right) + A_{66} \left(\frac{\partial u_0}{\partial y} + \frac{\partial v_0}{\partial x} \right)^2 + D_{11} \left(\frac{\partial^2 w_0}{\partial x^2} \right)^2 + 2D_{12} \frac{\partial^2 w_0}{\partial x^2} \frac{\partial^2 w_0}{\partial y^2} + D_{22} \left(\frac{\partial^2 w_0}{\partial y^2} \right)^2 + 4 \left(D_{16} \frac{\partial^2 w_0}{\partial x^2} + D_{26} \frac{\partial^2 w_0}{\partial y^2} \right) \frac{\partial^2 w_0}{\partial x \partial y} + 4D_{66} \left(\frac{\partial^2 w_0}{\partial x \partial y} \right)^2 \right\} dx dy \quad (9)$$

The kinetic energy of a laminate is written:

$$E_c = \frac{1}{2} \iiint \rho \left[\left(\frac{\partial u}{\partial t} \right)^2 + \left(\frac{\partial v}{\partial t} \right)^2 + \left(\frac{\partial w}{\partial t} \right)^2 \right] dx dy dz + \frac{1}{2} \sum_{i=1}^{N_i} M_i w^2(x_p, y_p) \quad (10)$$

where ρ is the mass per unit area of the plate, M_i is the mass of the attached particle,

(x_p, y_p) is the position, and N_i is the number of particles attached to the composite plate[29].

$$u = u_0 - z \frac{\partial w_0}{\partial x}; v = v_0 - z \frac{\partial w_0}{\partial y}; w = w_0(x, y) \quad (11)$$

$$E_c = \frac{1}{2} \iiint \left[\left(\frac{\partial u_0}{\partial t} - z \frac{\partial^2 w_0}{\partial x \partial t} \right)^2 + \left(\frac{\partial v_0}{\partial t} - z \frac{\partial^2 w_0}{\partial y \partial t} \right)^2 + \left(\frac{\partial w_0}{\partial t} \right)^2 \right] dx dy dz + \frac{1}{2} \sum_{i=1}^N M_i w^2(x_p, y_p) \quad (12)$$

The equation of the free transversal vibration is written:

$$-\rho_s \omega^2 w_0 + D_{11} \frac{\partial^4 w_0}{\partial x^4} + 4D_{16} \frac{\partial^4 w_0}{\partial x^3 \partial y} + 2(D_{12} + 2D_{66}) \frac{\partial^4 w_0}{\partial x^2 \partial y^2} + 4D_{26} \frac{\partial^4 w_0}{\partial x \partial y^3} + D_{22} \frac{\partial^4 w_0}{\partial y^4} = 0 \quad (13)$$

The solution of this equation is sought, by the method of separation of the variables of spaces and time, in the form:

$$w_0(x, y, t) = w_0(x, y) e^{j\omega t} \quad (14)$$

$w_0(x, y)$: is the spatial or shape solution.

3.1 Finite Element simulation

The FEM simulation is realized by using ANSYS software. For the composite laminate material, the element type used is SHELL281.

The element is suitable for analyzing thin to moderately-thick shell structures. The element has eight nodes with six degrees of freedom at each node: translations in the x , y , and z axes, and rotations about the x , y , and z -axes.

The point mass element type which has been used is 3D mass 21 in ANSYS. Figure. 5 shows the finite element model of the attached mass on the composite laminate plates.

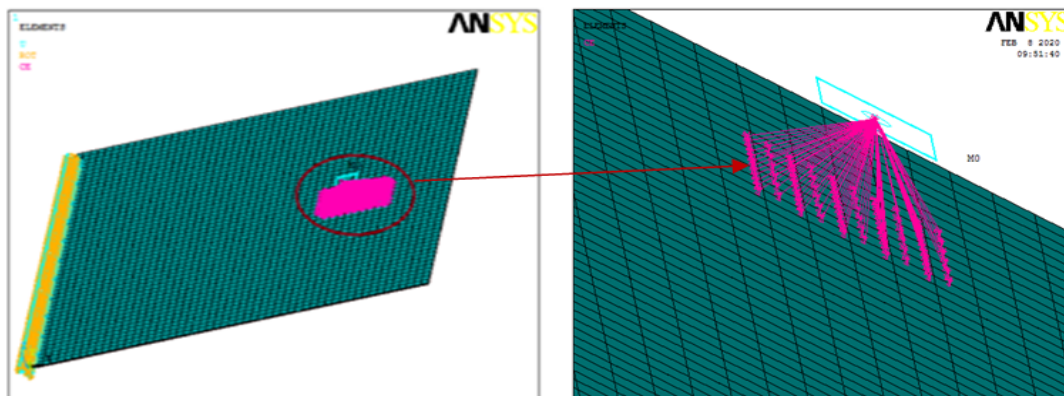


Figure 5. Finite element model of composite laminate plates with mass loading

4. THE TAGUCHI METHOD AND DESIGN OF EXPERIMENTS

The Taguchi methods are statistical methods, developed to improve the quality of manufactured goods, and more recently also applied to biotechnology and engineering. As it can improve the processing quality, and reduce the number of experiments. In addition, the advantage of this method is to save experimental time and locate important factors fast, and save effort during testing.

There are many standard orthogonal arrays available, each of the arrays is meant for a specific number of independent design factors and levels. So, the selection of orthogonal arrays is based on the number of factors and the levels for each experimental application [30]. In this work, we need a large number of experimental tests to achieve the objective of the studies. The Taguchi method uses a small number of tests for research, by the use of a special design of orthogonal arrays. The most suitable mixed orthogonal array L18 ($2^1 \times 3^2$) has been used to determine the optimal factors influencing the effect of attached mass on damping ratio for composite laminate plates (Table 3). These factors include attached mass locations and the staking sequences for three levels (L1, L2, and L3), $[\pm 20]_{2s}$, $[\pm 25]_{2s}$ and $[\pm 30]_{2s}$, and two levels in the case of the boundary conditions (CFFF, and CFCF) (Table 4).

The Taguchi method uses a loss function (signal-noise (S/N) ratio) to calculate the deviation between the desired values and the experimental values.

Better quality characteristic was used as shown in Eq. (15), the large-the-best:

$$\mu = \frac{S}{N} = -10 \log \left[\frac{1}{n} \sum_{i=1}^n \frac{1}{y_i^2} \right] \quad (15)$$

where y_i is the observed data at the i^{th} experiment and n is the number of observations of the experiment.

Table 3. Taguchi L18 array test design

Tests	Factors		
	A	B	C
1	1	1	1
2	1	1	2
3	1	1	3
4	1	2	1
5	1	2	2
6	1	2	3
7	1	3	1
8	1	3	2
9	1	3	3
10	2	1	1
11	2	1	2
12	2	1	3
13	2	2	1
14	2	2	2
15	2	2	3
16	2	3	1
17	2	3	2
18	2	3	3

The percentage differences between FEM results and the experimental results for $[\pm 20]_{2s}$, $[\pm 25]_{2s}$, $[\pm 30]_{2s}$ laminated composite plates are less than 8.7%.

Table 4. Factors and their levels

Parameters	Symbol	Level 1	Level 2	Level 3
Boundary conditions	A	CFFF	CFCF	—
Staking sequences	B	$[\pm 20]_{2s}$	$[\pm 25]_{2s}$	$[\pm 30]_{2s}$
Attached mass locations	C	L1	L2	L3

5. RESULTS AND DISCUSSION

5.1 Effect of the attached mass on modal damping and natural frequencies

In Table 5 the experimental results of the fundamental frequencies of plates composite without attached mass as a function of lamination angle are given. These results are compared with the results of the finite element method (FEM) and good agreement is observed between the two results.

Table 5. Fundamental natural frequencies and modal damping ratio of the CFFF specimens, without attached mass

Staking sequences	Fundamental frequency f_r (Hz)			Modal damping (%)
	Exp	FEM	Error (%)	
$[\pm 20]_{2s}$	6.0	5.7913	3.3	5.480
$[\pm 25]_{2s}$	5.7	6.2831	8.7	4.990
$[\pm 30]_{2s}$	6.6	7.0480	6.6	4.190

Table 6. Fundamental natural frequencies and modal damping ratio with attached mass of the CFFF specimen $[\pm 20]_{2s}$

Locations of attached masses	Fundamental frequency f_r (Hz)			Modal damping (%)
	Exp	FEM	Error (%)	
L1	5.6	6.084	8.5	06.01
L2	5.2	5.683	7.6	08.1
L3	4.8	4.993	4.04	08.5
L4	4.4	4.393	0.16	9.09

Table 7. Fundamental natural frequencies and modal damping ratio with attached mass of the CFFF specimen $[\pm 25]_{2s}$

Locations of attached masses	Fundamental frequency f_r (Hz)			Modal damping (%)
	Exp	FEM	Error (%)	
L1	5.8	6.595	12.06	5.5
L2	5.4	6.038	11.1	7.75
L3	5.1	5.1985	1.9	8.0
L4	4.8	4.6506	3.1	8.33

Table 8. Fundamental natural frequencies and modal damping ratio with attached mass of the CFFF specimen $[\pm 30]_{2s}$

Locations of attached masses	Fundamental frequency f_r (Hz)			Modal damping (%)
	Exp	FEM	Error (%)	
L1	6.8	7.2254	5.8	4.6
L2	6.4	6.7913	4.7	7.4
L3	5.9	5.8178	0.2	7.6
L4	5.6	5.2503	6.25	7.76

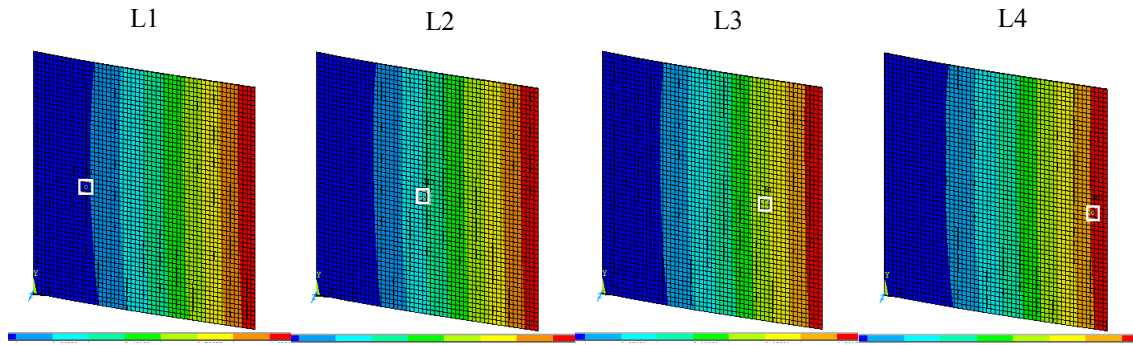


Figure 6. Fundamental numerical mode shapes of $[\pm 20]_{2s}$ plate with 30g (M/4) attached mass

The Tables 6-8 the experimental results of the fundamental natural frequencies of plates composite with attached mass (as a function of lamination angle for different locations) are compared with the finite element method (FEM). The percentage differences between FEM results and the experimental results are less than 8.5 %, 12.06 % and 6.25 % for $[\pm 20]_{2s}$, $[\pm 25]_{2s}$, $[\pm 30]_{2s}$ laminated composites plates respectively. Generally, it is seen that a good agreement exists between numerical and experimental results.

The differences are probably due to the geometric imperfections in the structure of plate's composite, where the difficult to define a perfect clamped edges in experimental tests (due to possible flexibility of the clamping apparatus, neglecting shear deformation), uniform variations in ply thickness and the effect of misalignments in ply orientation, that have not been considered in the numerical model. These errors parameters affect the stiffness, consequently on the frequencies and damping ratio of the plates.

The present result in the Tables 6-8 are compared with that in the Table 5 and shows variation of fundamental frequencies between plates, with and without attached mass. Generally, the fundamental frequency decreases as the location of the attached mass (L1, L2, L3, and L4) is closer to the clamped edge (Figure 6). With regard to the damping ratio of fundamental mode of CFFF composite, if the location of the mass is approached to the clamped edge, the change of the damping ratio is not more pronounced. So, the change of damping compared with damping obtained in plate without attached mass is larger when the distance between mass and clamped edge is greater.

All changes in the frequency and damping ratio of the composite plate are related with position of the attached mass with respect to nodal lines in the corresponding mode shapes. If the mass is attached near to a nodal line, attached mass has small amplitudes that lead to small kinetic energy. Therefore, a small change is observed for the frequency and damping ratio of the composite plates.

5.2 Evaluation and analysis of experimental results using taguchi design

The values of damping ratio of fundamental frequencies between plates with and without attached mass are very important for the improvement of the quality

of dynamic measurements due to the loading of added masses to the composite structures, that they are frequent in the fields of mechanical industry and civil engineering.

Table 9. The results of experiments

Tests	Factors			Damping ratio (%)	S/N Ratio for Damping ratio (%)
	A	B	C		
1	CFFF	$[\pm 20]_{2s}$	L1	6.01	15.5775
2	CFFF	$[\pm 20]_{2s}$	L2	8.10	18.1697
3	CFFF	$[\pm 20]_{2s}$	L3	8.50	18.5884
4	CFFF	$[\pm 25]_{2s}$	L1	5.50	14.8073
5	CFFF	$[\pm 25]_{2s}$	L2	7.75	17.7860
6	CFFF	$[\pm 25]_{2s}$	L3	8.00	18.0618
7	CFFF	$[\pm 30]_{2s}$	L1	4.60	13.2552
8	CFFF	$[\pm 30]_{2s}$	L2	7.40	17.3846
9	CFFF	$[\pm 30]_{2s}$	L3	7.60	17.6163
10	CFCF	$[\pm 20]_{2s}$	L1	1.20	1.5836
11	CFCF	$[\pm 20]_{2s}$	L2	1.6	4.0824
12	CFCF	$[\pm 20]_{2s}$	L3	1.53	3.6938
13	CFCF	$[\pm 25]_{2s}$	L1	1.04	0.3407
14	CFCF	$[\pm 25]_{2s}$	L2	1.32	2.4115
15	CFCF	$[\pm 25]_{2s}$	L3	1.21	1.6557
16	CFCF	$[\pm 30]_{2s}$	L1	0.80	-1.9382
17	CFCF	$[\pm 30]_{2s}$	L2	0.93	-0.6303
18	CFCF	$[\pm 30]_{2s}$	L3	1.1	0.8279

The changes in damping ratio were obtained as the result of experimental study are seen in Figures. 7 and 8.

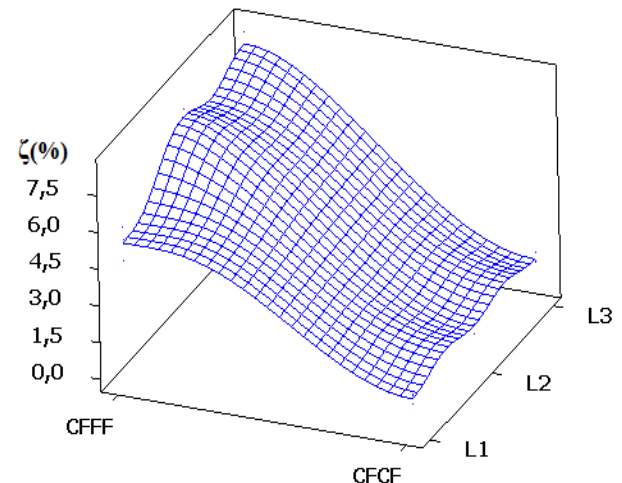


Figure 7. Effect of boundary conditions and attached mass locations on damping ratio

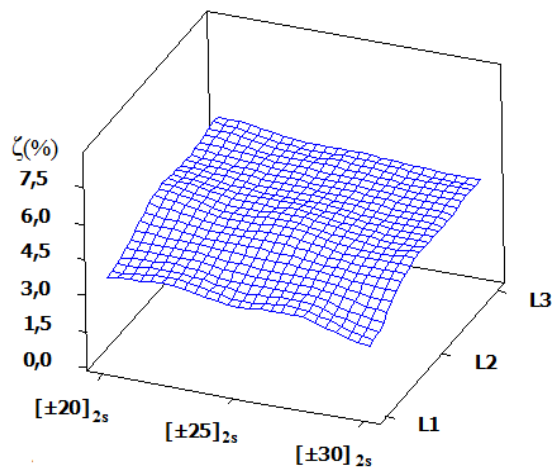


Figure 8. Effect of staking sequences and attached mass locations on damping ratio

Depending on the difference of the attached locations mass (L1, L2 and L3), there was not much change in the damping ratio values for CFCF boundary conditions. However, in CFFF boundary conditions, the values of damping ratio had an important value compared with the values obtained for CFCF boundary conditions (Figure. 7).

The values of damping ratio compared with damping obtained in plate without attached mass is larger when the distance between location of the mass and clamped edge is greater. All changes in the damping ratio of the composite plate are related with position of the attached mass with respect to nodal lines in the corresponding mode shapes (For the first mode the clamped edge is only a nodal line) (Figure 6).

The data is plotted in Figure 8. It indicates that the damping ratio increases with the increase in staking sequences ($[\pm 20]_{2s}$, $[\pm 25]_{2s}$ and $[\pm 30]_{2s}$) for all attached mass locations (L1, L2 and L3) and for both CFCF and CFFF boundary conditions. This is due to the facts that, the number of lay fiber and staking sequences between lateral and parallel to the bending axis for each modes. Consequently, the bending stiffness of the $[\pm 20]_{2s}$ becomes lower than that of the $[\pm 25]_{2s}$ and $[\pm 30]_{2s}$, and its damping becomes higher.

The “larger-the better” equation was used for the calculation of the S/N ratio. Table 9 shows the values of the S/N ratios for observations of damping ratio of the fundamental frequencies. In addition, the effect of each control factor was presented with S/N response table, and is shown in Table 10. The bold values shows the optimal levels of control factors.

Table 10. S/N response table for the damping ratio of the fundamental mode

Levels	Control factor (Damping ratio (%))		
	A (Boundary conditions)	B (Staking sequences)	C (Attached mass locations)
1	16.805	10.283	7.429
2	1.442	9.335	9.867
3		7.753	10.074
Delta	15.363	2.530	2.645
Rank	1	3	2

The level values of control factors are shown in paragraph forms in Figure 9.

The best level was found according to the highest S/N ratio in the levels of that control factors. According to this, the levels and S / N ratios for the factors giving the best value of damping ratio were specified as A (Level 1, S/N = 16,805), factor B (Level 1, S/N = 10,283), and factor C (Level 3, S/N = 10,074). In other words, the optimum value of damping ratio was obtained with a CFFF boundary condition at the point L3 and has a symmetrical staking sequence $[\pm 20]_{2s}$.

6. CONCLUSION

This paper presents the findings of an experimental and numerical investigation the dynamic behavior of the composite laminate plates. The focus is made on the effects of attached mass on the damping ratio. In the research, the three factors (attached mass locations, the staking sequences and the boundary conditions) influencing the effect of the attached mass on the damping ratio are studied using the Taguchi method. The paper presents a finite element modeling carried out by using ANSYS software and validated experimentally. Through the results obtained above, the following conclusions could be drawn:

- Attached masses greatly increase damping ratio and reduce the fundamental frequency.
- By changing staking sequences, the attached mass changes the position and shape of nodal lines.
- The attached mass increases the damping ratio of the composite plate if it is inserted at a point other than a nodal line. This change is more pronounced if the mass is attached farther from the nodal lines of the mode shapes.

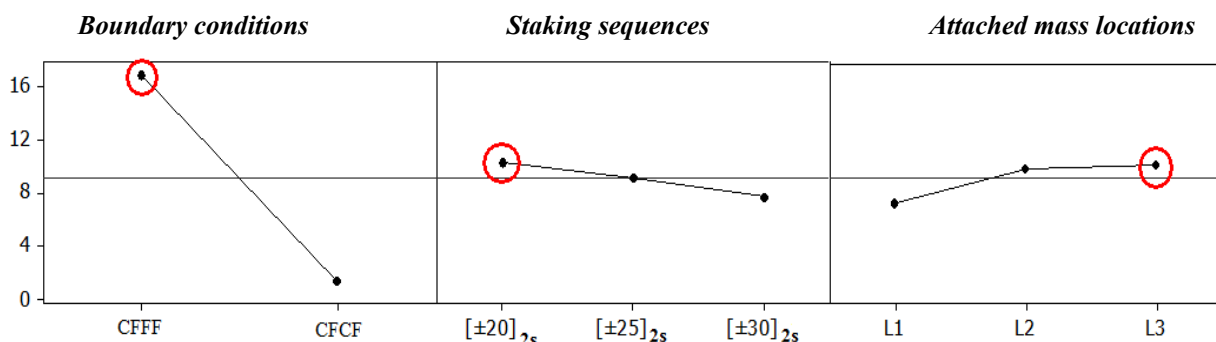


Figure 9. Plot of main effects based on Means and S/N ratios for damping ratio

- The optimum levels of the control factors for indicate the effect of attached mass on damping ratio using S/N ratio were determined. The optimal conditions for fundamental vibration mode were observed at A1B1C3.

This result is important from the point of view of the location of the sensor on the plate and the improvement of vibratory measurements by the optimization of emplacement. In addition, can be used to eliminate the mass of accelerometers used in the experimental tests for detection of defaults in the machines by vibration analysis.

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NOMENCLATURE

f_r	resonance frequency
ζ	damping ratio
$\{y\}$	nodal displacement vector
γ_{xz}, γ_{yz}	shear strain
u, v, w	displacements of a point in the x, y and z directions
$\{j\}$	the nodal acceleration vector
$\sigma_{xx}, \sigma_{xy}, \sigma_{yy}$	stress components in Cartesian coordinates
$\epsilon_{xx}, \epsilon_{yy}, \epsilon_{xy}$	strains along the x and y directions
Q_{ij}^k	components of transformed lamina stiffness matrix
e_k	layer thickness
A_{ij}, B_{ij}, D_{ij}	extensional, coupling and bending stiffness coefficients
G_{12}, G_{13}, G_{23}	shear modulus

$\nu_{12}, \nu_{13}, \nu_{23}$	Poisson's ratio
N	stress resultant
$\gamma_{xx}, \gamma_{yy}, \gamma_{xy}$	shear strain
M	components of bending moment
E_c	kinetic energy of a laminate
U_d	strain energy of a laminate
u_0, v_0, w_0	displacements of a point in the midplane
ρ	mass per unit area of the plate
ω	frequency of free vibrations
M_i	mass of the attached particle
N_i	number of particles attached to the composite plate
$[M]$	structural mass matrix
$[K]$	structural stiffness matrix
Δf	frequency bandwidth between the two half power points
(x_p, y_p)	position of particles attached to the composite plate
s	standard deviation of measured values
n	number of observations of the experiment
u	signal-noise ratio (S/N)
y_i	observed data

ОДНОС МОДАЛНОГ ПРИГУШЕЊА КОД СИМЕТРИЧНОГ ЛАМИНАТНОГ КОМПОЗИТА ПОД УТИЦАЈЕМ ПРИЧВРШЋЕНЕ МАСЕ ПРИМЕНОМ ЕКСПЕРИМЕНТАЛНОГ ДИЗАЈНА

Р. Абделхафид

Ламинатни материјали имају широку примену у грађевинарству као и у војној и авио индустрији. Потребно је истражити њихова механичка и физичка својства. Рад се бави проблемом утицаја причвршћене масе на однос пригушења угла симетричног слоја ламинатног композита коришћењем Тагучи методе. Разматрају се следећи параметри фактора: локација причвршћене масе од притегнуте ивице, редослед слагања и гранични услови. Резултати показују да је однос пригушења ламинатних плоча осетљив на причвршћену масу, при чему је пропорционалан локацијама причвршћене масе. Утврђено је да причвршћена маса смањује фреквенцију а да повећава однос пригушења, ако је уметнута у тачки изван нодалне линије. Постоји добра корелација између нумеричких резултата добијених за основну фреквенцију применом софтвера ANSYS у поређењу са резултатима добијеним експерименталним путем.