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# Thermo-Mechanical Vibration Analysis of Simply Supported Laminated and Flat Plates Under Harmonic Loading

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**Abstract**

*This paper delves into a numerical exploration of the vibration characteristics and frequency response of two types of plates: a simply supported rectangular laminated plate and a simply supported flat plate. These plates are subjected to concentrated harmonic forces within a thermal environment. The determination of the critical buckling temperature guides the assessment of thermal load effects. Analytical formulations, incorporating first-order shear deformation theory (FSDT) to consider shear deformation and rotational inertia effects, are utilized to obtain mode shapes, natural frequencies, and dynamic responses. The application of thermal loads is observed to decrease natural frequencies and shift response peaks towards lower frequencies. It is posited that thermal loads alter the structural stress state, with thermal stresses from uniform temperature changes being determined using thermo-elastic theory. These stresses are then incorporated as pre-stressed factors in subsequent dynamic analyses. The study reveals significant influences of thermal loads on natural frequencies, while the sequence and characteristics of mode shapes remain unchanged. Specific characteristics of laminated plates with different configurations are examined, indicating that increasing young's modulus of the core leads to higher natural frequencies and response peaks shifting towards higher frequencies. The study reveals that natural frequencies get reduced with rising temperature of the plate, with the first natural frequency being especially susceptible to thermal changes. The accuracy of the theoretical formulation is substantiated through a close correlation with numerical results derived from ANSYS simulations.*

**Keywords:** Mode shapes, thermo elastic theory, FSDT, flat plate, laminated plate

## INTRODUCTION

Flat plates and laminated plates found widespread application in mechanical, aerospace, and structural structures [1] on account of high strength-to-weight ratios, design controllable mechanical characteristics [2], and conformity to challenging loading patterns. For numerous actual problems, the structural elements undergo not just mechanical load, but also thermally variable surroundings, which are prone to making notable modifications to the dynamics. An understanding of the vibration behavior of such plates under combined actions of mechanical and thermal loading is required to design safe and efficient future generation engineering systems.

In the current investigation, a thorough thermo-mechanical vibration analysis [3] is provided for two different configurations of plates, i.e., simply supported rectangular laminated plates and flat plates. These plates are exposed to harmonic concentrated loading along with exposure in a uniform temperature field. The thermal environment induces thermal stresses and alters the material stiffness, impacting both mode shapes and

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natural frequencies [4]. Li et al. [5] considered the consequences of both transverse shear deformation and rotary inertia in their study by applying FSDT. Thermo-elastic theory is applied to compute thermal stresses caused by temperature-driven material expansion, which are considered as pre-stressed conditions within the dynamic model. The underlying dynamic equations are studied to find out the natural frequencies and corresponding mode shapes for different thermal conditions [6]. The results showcase the adverse effect of temperature on natural frequency, particularly in the first mode. Additionally, the dynamic response occurring at reducing frequencies with an increase in thermal loads indicates the thermal sensitivity of composite structures. Parametric studies also validate that core stiffening of the laminated plate with increasing stiffness leads to increased natural frequencies [7] and the consequent shift in response peaks. Validation of the analytical calculations against finite element analyses with ANSYS.

This investigation provides significant knowledge on thermal loading effects in the design and behavior of plate structures, with attention drawn to the significance of thermal effects in vibration analysis. Upon exposure to high temperatures, the vibrational response of plate structures varies considerably from that when operating under ambient conditions. Thermal stresses caused by elevated temperatures change the structural resistance and, by extension, the dynamic characteristics of the plate. Major parameters like mode shapes and natural frequencies are influenced by thermal loading and need to be considered in the design process. In tightly constrained plates, excessive thermal stress [8] can cause rupture. Additionally, temperature distribution along the plate surface is generally non-uniform in practical operating conditions, and hence it causes intricate deformation patterns and separate dynamic responses. The research also comprises simulation outcomes derived from the aid of ANSYS, a finite element analysis, to assess the performance of natural frequencies [9] and mode shapes [4] as temperature changes.

## ANALYTICAL FORMULATION

In this section, the frequency response of both flat and laminated plates is mathematically described using FSDT [19]. According to FSDT, the displacement components along the Cartesian coordinates (x,y,z) are characterized by specific kinematic assumptions intrinsic to the theory.

$$u(x,y,z,t) = U(x,y,t) + z\emptyset x(x,y,t)$$
$$v(x,y,z,t) = V(x,y,t) + z\emptyset y(x,y,t)$$
$$w(x,y,z,t) = W(x,y,t) \tag{1}$$

where the mid-surface displacements in the longitudinal (x) direction are denoted by ‘U’ and in the transverse (y) direction by ‘V’, while W denotes the displacement in the out-of-plane direction. The variables $\emptyset \boldsymbol{x}$ and $\emptyset \boldsymbol{y}$ signify the rotations of the normal to the mid-surface about the y-axis and x-axis, respectively.

### Strain-Displacement Relationships for Plates

The linear strain components are given by:

$$\varepsilon_{xx} = \frac{\partial U}{\partial x} + z\frac{\partial \emptyset x}{\partial x}$$
$$\varepsilon_{yy} = \frac{\partial V}{\partial y} + z\frac{\partial \emptyset y}{\partial y}$$
$$\gamma_{xy} = \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} + z\left(\frac{\partial \emptyset x}{\partial x}\frac{\partial \emptyset y}{\partial y}\right)$$
$$\gamma_{xz} = \emptyset x + \frac{\partial W}{\partial x}$$
$$\gamma_{yz} = \emptyset y + \frac{\partial W}{\partial y} \tag{2}$$

## Constitutive Relations

### *Flat Plate*

The constitutive relation of the flat plate is assumed based on the assumption that the plate has uniform material properties, which implies that the material has homogeneous, isotropic, and linearly elastic properties. Based on Hooke's law, the in-plane normal and shear stresses have a direct relationship with their respective strains through the elastic constants. These elastic constants include the young's modulus of elasticity (E) and poisson's ratio (ν). The shear stresses in the transverse direction are calculated using the shear modulus (G), along with the shear correction factor (ks).

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau xy \end{bmatrix} = \frac{E}{1-\nu^2} \begin{bmatrix} 1 & \nu & 0 \\ \nu & 1 & 0 \\ 0 & 0 & \dfrac{1-\nu}{2} \end{bmatrix} \begin{bmatrix} \varepsilon_{\mathrm{xx}} \\ \varepsilon_{\mathrm{yy}} \\ \gamma_{xy} \end{bmatrix}$$

$$\tau_{xz} = k_s G \gamma_{xz}$$

$$\tau_{yz} = k_s G \gamma_{yz} \tag{3}$$

### *Laminated Plate*

In the case of laminated composite plates, the constitutive response is captured using classical laminate stiffness matrices A, B, and D, which capture the extensional, bending-extension coupling, and bending stiffness properties of the laminate, respectively. Stiffness matrices A, B, and D for the laminate are calculated based on the integration of the transformed reduced stiffness matrix of each lamina along the thickness of the laminate. The constitutive formulation is used to establish the relation between the generalized force resultants (membrane forces and bending moments) and the mid-plane strains and curvatures.

$$\begin{bmatrix} N \\ M \end{bmatrix} = \begin{bmatrix} A & B \\ B & D \end{bmatrix} \begin{bmatrix} \varepsilon_0 \\ k \end{bmatrix}, A, B, D = \sum_{k=1}^{n} \int_{z_{k-1}}^{z_k} Q^{(k)} [1, z, z^2]^T \, dz \tag{4}$$

where $Q^{(k)}$ is the rotated stiffness matrix for the kth layer of the composite. $\varepsilon_0$ is the mid-plane strain vector, whereas κ is the curvature vector. 'A' represents extensional stiffness, 'B' is coupling stiffness, and D denotes bending stiffness of the laminate.

## Equations of Motion

According to the study by Li et al. [5], the governing equations for plates based on the FSDT are formulated using either Hamilton's principle or by using governing equations of virtual work. These equations culminate in the generalized governing equations that illustrate the mechanical behavior of the plate.

$$N_{xx,x} + N_{xy,y} = \rho h \ddot{\mathrm{U}}$$

$$N_{yy,y} + N_{xy,x} = \rho h \ddot{\mathrm{V}} \ (5)$$

$$Q_{x,x} + Q_{y,y} + q(x,y,t) = \rho h \ddot{w} \tag{6}$$

$$M_{xx,x} + M_{xy,y} - Q_x = I \ddot{\phi}_x$$

$$M_{yy,y} + M_{xy,x} - Q_y = I \ddot{\phi}_y \tag{7}$$

Where,

q (x, y, t) denotes the time-dependent transverse load and I is the area moment of inertia.

## Shear Correction Factor

To address the uneven distribution of shear stress across the cross-section, a shear correction factor (denoted as $k_s$) is introduced. For rectangular cross-sections, this factor typically takes the value of 5/6.

## Mode Shapes and Natural Frequency

Identification and measurement of a structure's resonance are necessary for comprehending vibration issues. There are two different kinds of vibration: resonant vibration and forced vibration [3]. Forced vibration can result from external loads, internal forces, system imbalances, and environmental disturbances, whereas natural body vibration patterns are the main cause of resonant vibration [10]. At or near natural frequency [11], the mode form of the resonance predominates in the overall vibration shape.

The definition of the operating deflection shape is any forced motion of two or more points on the structure [12]. Mode shapes and inherent frequencies are the essential characteristics of an undamped, freely vibrating system. Do not alter the values of the natural frequencies [13] of an MDOF system at the selected coordinates. This isn't the case with mode forms, though, as they rely on coordinates [14]. Nonetheless, the physical system's motions as specified by the mode shapes will be the same. Accordingly, it is claimed that natural frequencies are essential characteristics independent of coordinate selection. Finding the system's Eigenvalues and Eigenvectors is frequently the first step in determining the mode shapes. The following formula provides the simply supported rectangular plate's [3] mode shape function:

$$\psi_{m,n}(x,y) = sin\left(\frac{m\pi x}{a}\right) sin\left(\frac{n\pi y}{b}\right) \tag{8}$$

Where, $a$ denotes the length and $b$ represents the width of the plate, respectively, whereas $m$ and $n$ represent the corresponding modal integers associated with the mode shapes along the longitudinal and transverse directions [11].

## VALIDATION STUDIES

I'll compare the mode shapes and natural frequencies for the two cases—one with a flat plate and the other with a laminated plate.

- *Case 1:* A comparative study is carried out among the current approach and the effort of Geng et al. [9], where the computed Flat plate's parameters and material characteristics are obtained.
- *Case 2:* A validation is carried out between the current approach and the work of Li et al. [5], where the computed Laminated plate parameters and material properties are obtained.

The behavior of natural vibrations under varying temperature conditions has been thoroughly examined. Tables 1, 2, 3, and 4 illustrate fundamental vibrational modes and their corresponding eigenfrequencies under varying thermal loads, with the corresponding mode shapes shown in Figures 1–4. Notably, although the mode shapes remain consistent, the natural frequencies reduce with the rise in the plate's temperature.

**Table 1.** Natural frequency (Hz) and the sequence mode derived by Geng et al. [9] at a temperature of 273.15K are compared.

| S.N. | Mode sequence | Simulation results | Geng's results | Error % |
|---|---|---|---|---|
| 1 | (1,1) | 414.19 | 420.20 | 1.43% |
| 2 | (2,1) | 859.47 | 874.00 | 1.66% |
| 3 | (1,2) | 1210.00 | 1227.00 | 1.39% |
| 4 | (3,1) | 1600.80 | 1630.40 | 1.82% |
| 5 | (2,2) | 1642.00 | 1680.80 | 2.31% |

**Table 2.** Natural frequency (Hz) and the sequence mode derived by Geng et al. [9] at a temperature of 318.15 K are compared.

| S.N. | Mode sequence | Simulation results | Geng's results | Error % |
|---|---|---|---|---|
| 1 | (1,1) | 73.46 | 100.90 | 27.21% |
| 2 | (2,1) | 627.34 | 646.40 | 2.95% |

 

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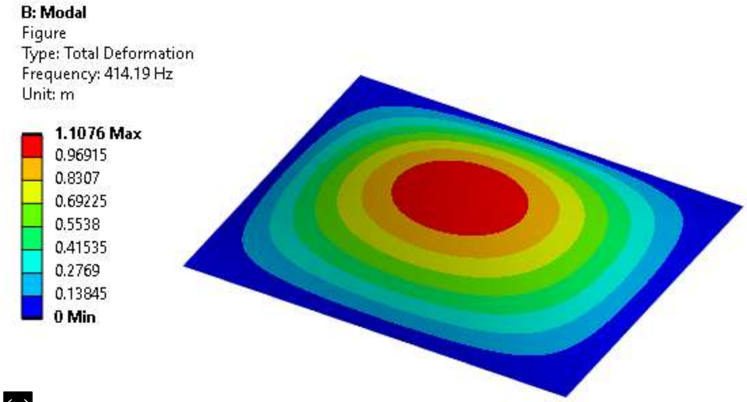
| | | | | |
|---|---|---|---|---|
| 3 | (1,2) | 989.94 | 1009.80 | 1.97% |
| 4 | (3,1) | 1385.70 | 1418.60 | 2.32% |
| 5 | (2,2) | 1426.20 | 1469.50 | 2.95% |

**Table 3.** Natural frequency (Hz) and the sequence mode derived by Li et al. [5] at temperature 273.15K are compared.

| S.N. | Mode sequence | Simulation results | Li's results | Error % |
|---|---|---|---|---|
| 1 | (1,1) | 451.47 | 449.41 | 0.46% |
| 2 | (2,1) | 921.94 | 931.63 | 1.04% |
| 3 | (1,2) | 1292.50 | 1304.98 | 0.96% |
| 4 | (3,1) | 1712.50 | 1728.58 | 0.93% |
| 5 | (2,2) | 1754.80 | 1780.69 | 1.46% |

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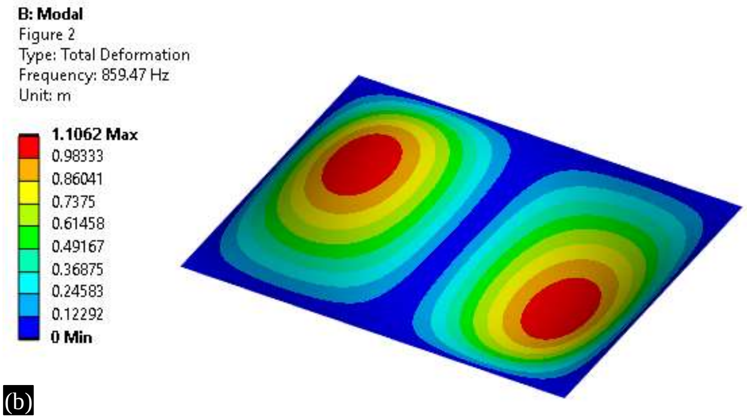


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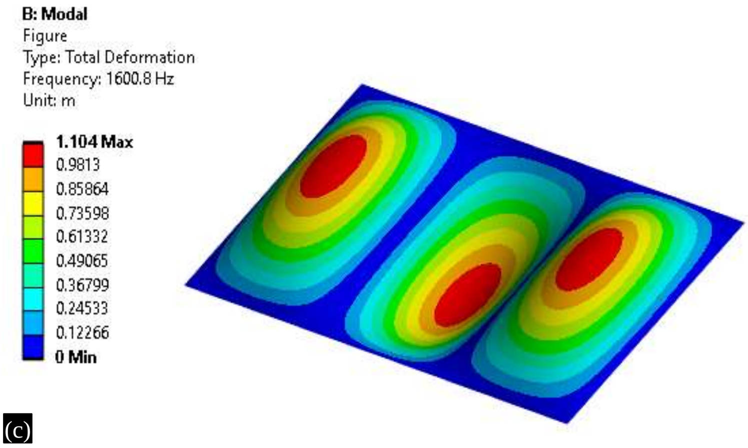


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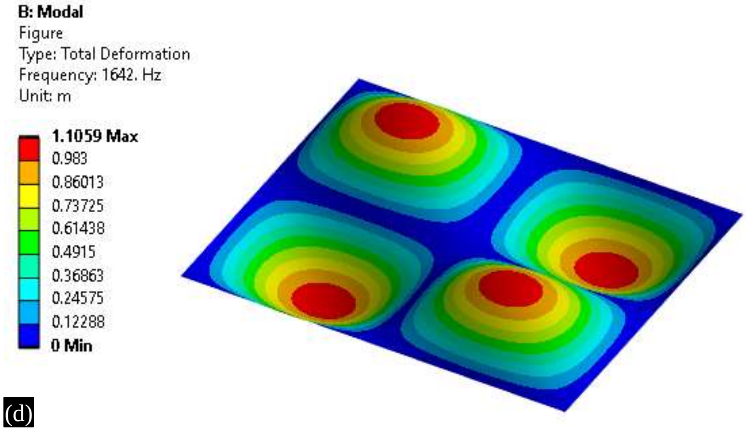


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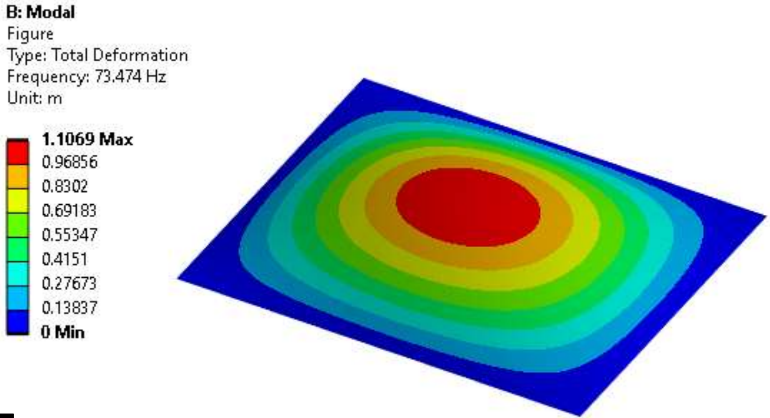
**Figure 1.** (a–d) Mode shapes for the flat plate at temperature 273.15 k.

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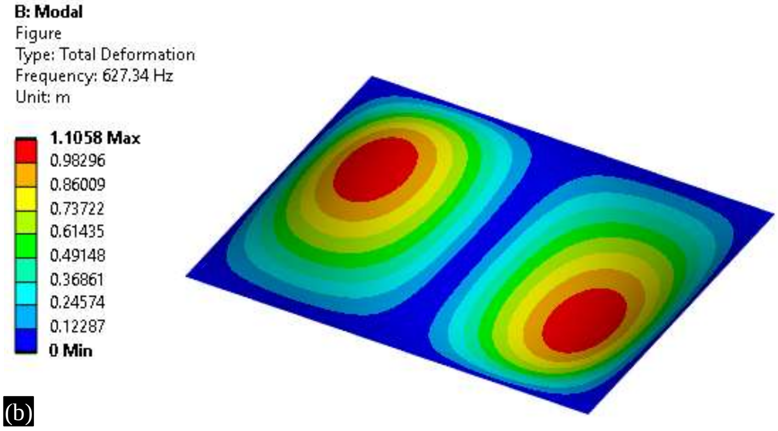


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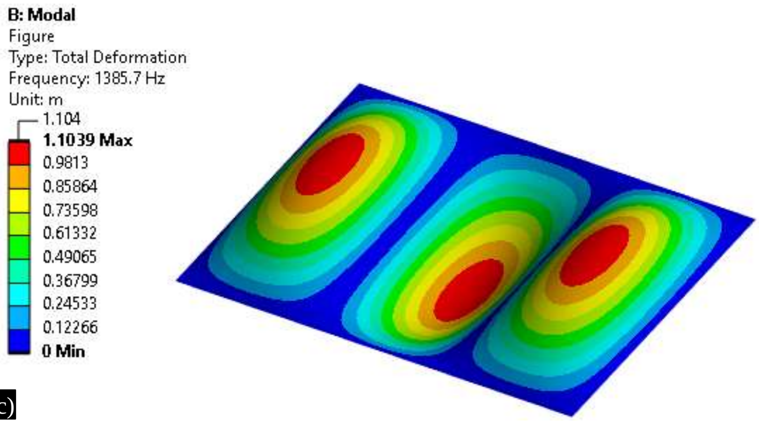


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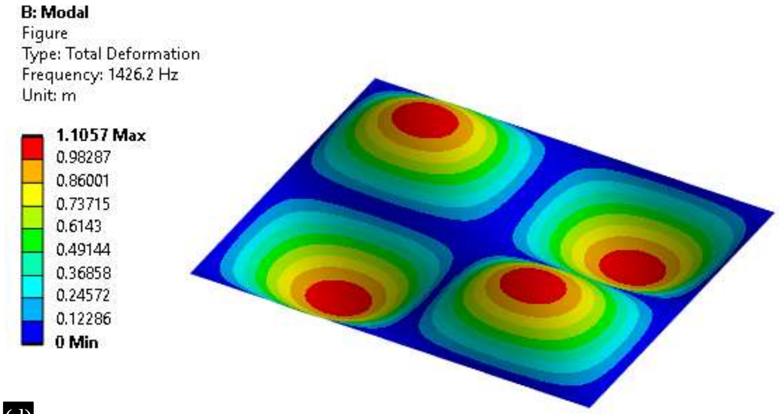


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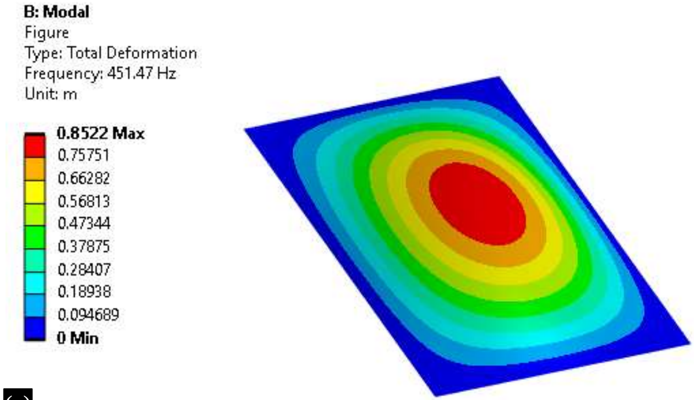
**Figure 2.** (a–d) Mode shapes for the flat plate at temperature 318.15 K.

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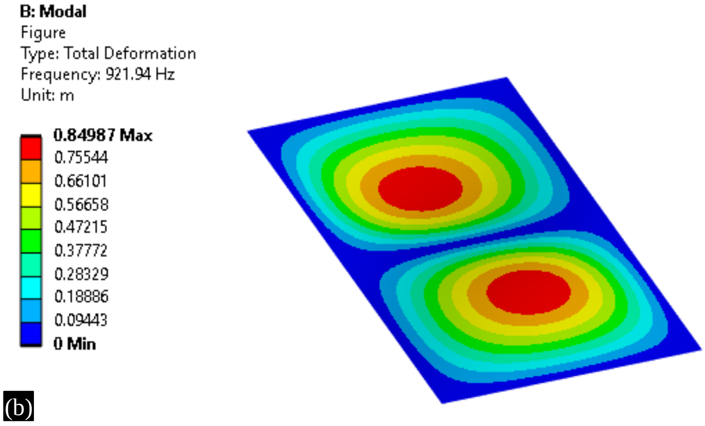


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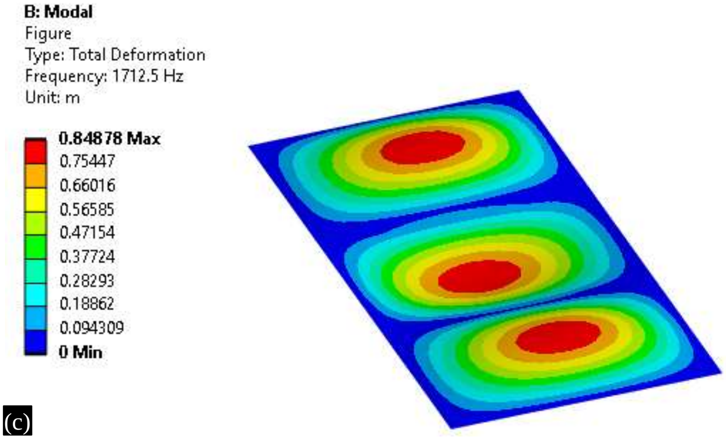


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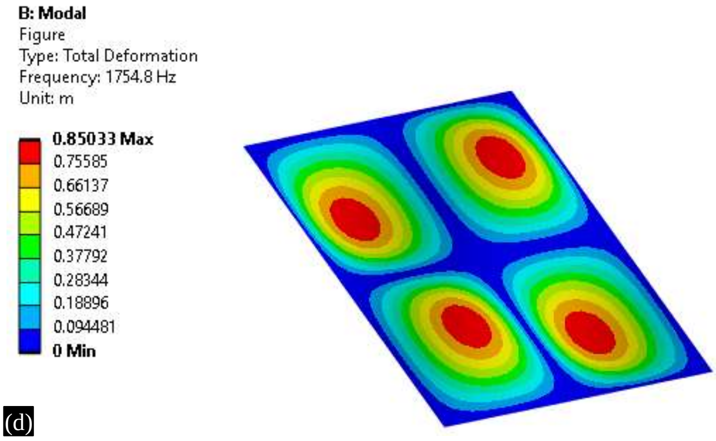


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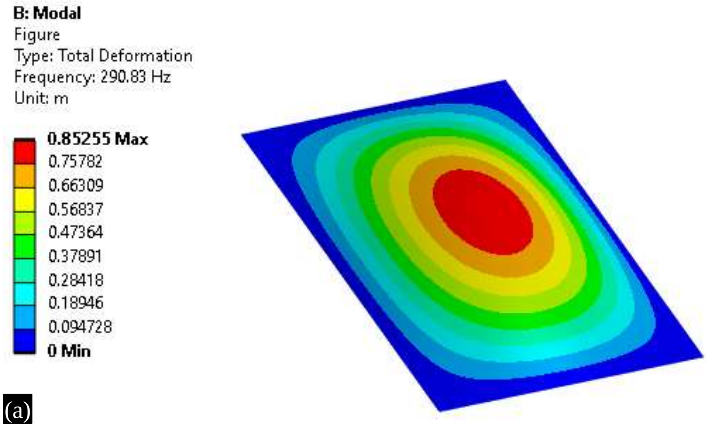
**Figure 3.** (a–d) Mode shapes for the laminated plate at temperature 273.15 K.

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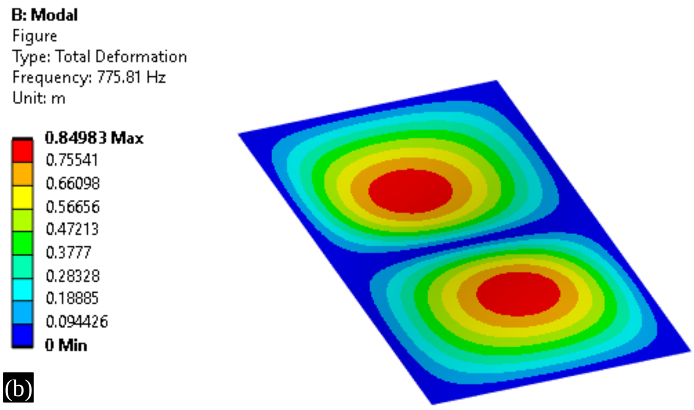


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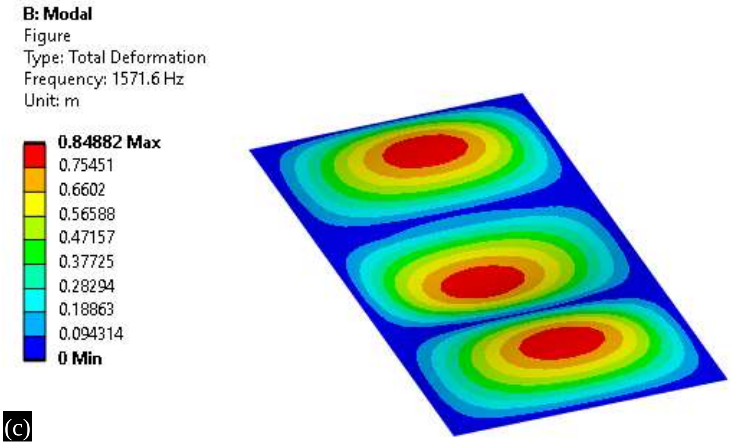


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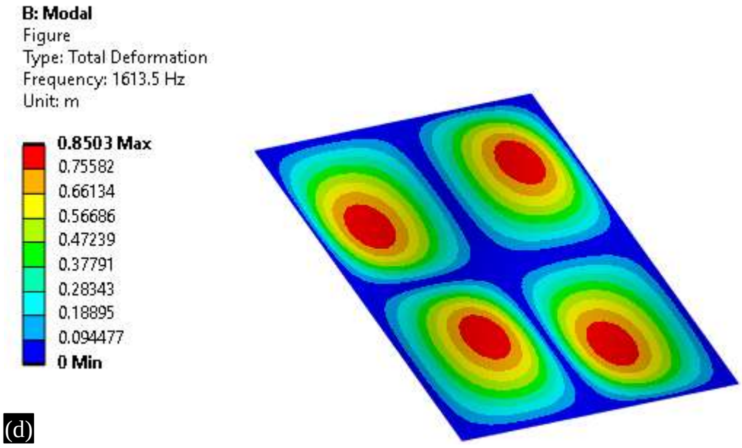


(d)

**Figure 4.** (a–d) Mode shapes for the laminated plate at temperature 333.15 K.

**Table 4.** Natural frequency (Hz) and the sequence mode derived by Li et al. [5] at a temperature of 333.15 K are compared.

| S.N. | Mode sequence | Simulation results | Li's results | Error % |
|---|---|---|---|---|
| 1 | (1,1) | 290.83 | 289.00 | 0.634% |
| 2 | (2,1) | 775.81 | 772.00 | 0.494% |
| 3 | (1,2) | 1150.30 | 1140.00 | 0.904% |
| 4 | (3,1) | 1571.70 | 1560.00 | 0.750% |
| 5 | (2,2) | 1613.50 | 1610.00 | 0.217% |

This clearly demonstrates the impact of thermal conditions on the vibrational response of the structure. The reduction in fundamental frequency, as also discussed in prior studies [15], highlights the significant softening effect caused by thermal stresses, an observation strongly supported by the findings of Geng et al. [9] and Li et al [5]. The observed reduction is consistent with the general principle that thermal expansion causes compressive stresses, which reduce the structural stiffness, a trend also noted by the research on high-temperature composite structures by Kim et al [17].

This study conducts a theoretical analysis to support and validate the findings obtained from numerical simulations, focusing on structural dynamics and acoustic radiation behavior under thermal loading conditions [9]. It is examined how a change in the thermal environment causes an isotropic rectangular flat and laminated plate to vibrate when it is pre-stressed. It is examined how heat loads-induced membrane forces affect dynamic responsiveness [16]. Subsequently, the thermally stressed plate's radiation characteristics are examined. Temperature has no effect on the material's characteristics.

## RESULTS AND DISCUSSION

Figure 5 compares the natural frequencies of a simply supported flat plate under two thermal conditions, viz. 273.15 K and 318.15 K. Simulation results in this study are compared with reference values reported by Geng et al. For all mode pairs (1, 1), (2, 1), (1, 2), (3, 1), (2, 2), the agreement is found to be very close, with the differences being less than 3%. At the lower temperature (273.15 K), natural frequencies are higher, in accordance with the higher stiffness of the plate at ambient conditions. With the temperature rise to 318.15 K, the frequencies drop significantly, most notably for the first mode, as a result of the thermal stresses' softening effect. The almost parallel shapes of the curves mean that although temperature significantly affects frequency values, it has no effect on the order of mode shapes. This finding is both a testament to the validity of the suggested analytical formulation and to the thermal sensitivity of flat plates. This result is both evidence of the correctness of the proposed analytical formulation and of the temperature sensitivity of flat plates, a widely known phenomenon in structural dynamics in which the vibration behavior of a plate is extremely sensitive to temperature changes caused by the ensuing changes in material properties and internal stress fields, a relation well defined by Reddy et al [18].

***Case 1***

For the flat plate case, natural frequencies were calculated for two thermal conditions of 273.15 K and 318.15 K, and are compared with the reference data given by Geng et al.[9] Figure 5 shows the comparison, which proves a consistently close agreement between the current simulation and the reference study. With the lower temperature of 273.15 K, the error percentage is still less than 2.5%, while with the higher temperature of 318.15 K, the difference is still less than 3%. Such minimal error levels verify the accuracy and precision of the formulation established. The findings also reflect the anticipated thermal effect on frequency response. Natural frequencies reduce as temperature increases from 273.15 K to 318.15 K, with the greatest reduction in the fundamental (1, 1) mode. Such a reduction is due to the thermal stressing causing the plate's stiffness to soften, thereby reducing its capacity for vibrational deformation. The good agreement with Geng's findings not only confirms the validity of the current method but also shows its ability to predict thermal sensitivity for flat plates. As a whole, the results build confidence in the analytical model for thermomechanical vibration analysis.

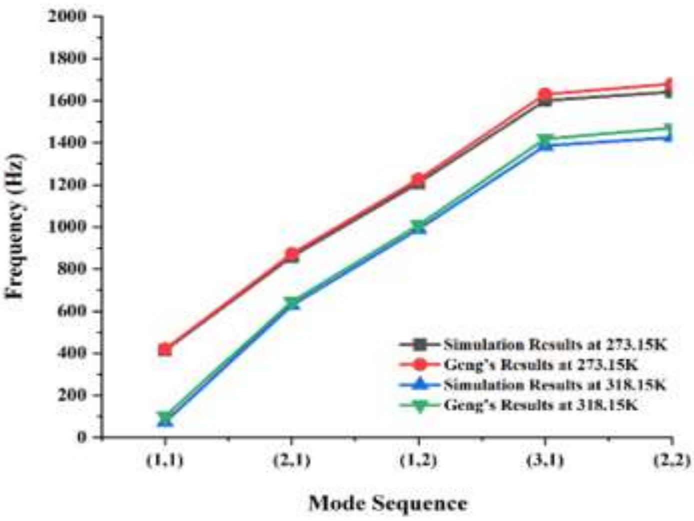


**Figure 5.** Comparison of simulation results with Geng et al. [9] for fat plate at a temperature of 273.15 K and 318.15 K.

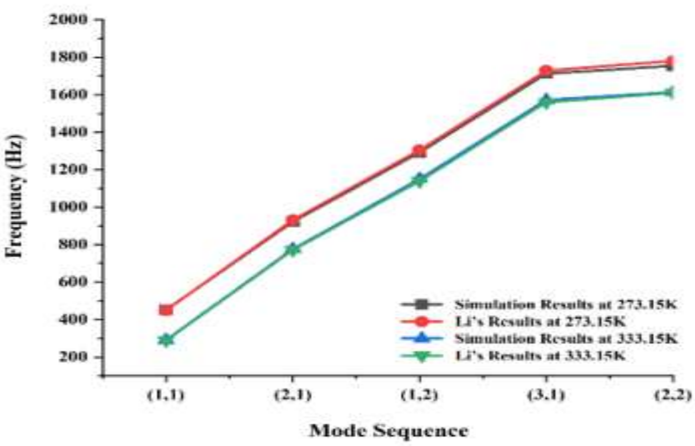


**Figure 6.** Comparison of simulation results with Li et al. [5] for laminated plate at a temperature of 273.15 K and 333.15 K

 

Figure 6 illustrates the natural frequencies of a simply supported laminated plate under two thermal conditions, 273.15 K and 333.15 K, with results compared against the reference study by Li et al. [5] The simulation values align very closely with Li's reported data, with error percentages remaining below 1.5%, confirming the robustness of the present formulation. As with the flat plate, natural frequency decreases with increasing temperature, again most significant in the fundamental mode (1, 1). Nevertheless, the general decrease in frequency with temperature is less extreme than that seen in flat plates, which verifies the higher thermal resistance of laminated composites. The results also further affirm that while thermal effects lower resonance to frequencies, mode order and mode shape are unchanged. This confirms the theoretical model and emphasizes the benefit of laminated plates for structural use at higher thermal conditions.

***Case 2***

For the laminated composite plate, natural frequencies were computed at two thermal states, i.e., 273.15 K and 333.15 K, and compared with the benchmark values of Li et al. [5]. The results, as shown in Figure 6, confirm an outstanding degree of agreement with all error percentages being less than 1.5%. The maximum observed deviation is only 0.904%, which is negligible in the context of practical structural vibration analysis and well within the acceptable range for engineering applications.

The close match between the present simulation results and Li's published data confirms the robustness and accuracy of the FSDT-based formulation employed in this study. Additionally, the results clearly show the influence of thermal environments on laminated plates. Like flat plates, thermally induced temperature increase results in a decrease in natural frequencies but with a smaller magnitude due to increased stiffness and anisotropic tailoring in the laminated composites. This shows the higher resistance of laminated plates against thermally induced softening effects. The very high accuracy of the validation results not only validates the analytical framework but also enhances the applicability of laminated composites towards high-temperature applications where dynamic stability is one of the critical design specifications.

## CONCLUSION

This research has conducted a thorough exploration of the thermo-mechanical vibration response of simply supported laminated composite plates and rectangular flat plates subjected to uniform thermal conditions. The analysis is rigorously limited to basic simply supported boundary conditions and linear thermal loads, excluding mixed boundary conditions or non-uniform temperature distributions. Additionally, the research only addresses simple rectangular geometries, excluding the response of plates with curvatures. The material model is in the linear elastic regime and neglects advanced material phenomena such as creep, degradation, and post-buckling effects that heavily affect structural response under extreme thermal stress.

The research started by constructing a theoretical framework using the First-Order Shear Deformation Theory (FSDT), which successfully incorporated transverse shear deformation and rotary inertia terms, thus allowing for a realistic representation of moderately thick and laminated structures. Thermal stresses based on thermo-elastic theory were integrated as pre-stress conditions, enabling the effect of uniform temperature increase on structural dynamics to be accurately modeled. The predictions made analytically were tested firmly through ANSYS simulations, and the results showed very good correlation with benchmarked data available in the literature. For flat plates, the error remained less than 3% at all times, and for laminated plates, the error did not exceed 1.5% at any time. These findings confirm the validity and applicability of the proposed formulation. Among the prominent findings is that natural frequencies reduce with increasing temperature, most notably for the fundamental mode, due to thermal softening reducing the effective stiffness. Although this frequency drop was noticed for both plate types, laminated composites showed greater capacity to resist thermal effects with the help of their engineered anisotropy and enhanced stiffness. The implications of the results are important for engineering design, especially for aerospace, naval, and power plant applications where plates are often subjected to combined thermal and mechanical loads. Through the determination of vibration characteristic sensitivity to temperature, this research yields useful information for more efficient and safer structural designs operating under thermally fluctuating conditions.

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