VIBRATION AND SOUND RADIATION ANALYSIS OF COMPOSITE LAMINATED PLATES UNDER THERMAL ENVIRONMENT

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This paper presents a simple closed form solution for vibration and sound radiation of composite laminated plates in thermal environment. The Hamilton’s principle is employed to formulate the governing equation of laminated heated plates. The free and forced vibration of the plate is analytically examined by the separation variables method. The frequency equations and mode functions for the laminated heated plates with any combination of simply supported and clamped boundary conditions can be readily obtained. Based on the Rayleigh integral method, sound radiation from the laminated plate under harmonic excitation is also analysed. To validate the proposed formulation, a considerable number of numerical examples are carried out for the modal, dynamic and acoustic analysis of laminated plates with different combinations of boundary conditions. An excellent agreement can be observed between the obtained results and those from FEM/BEM. The effects of temperature on the free and forced vibration of laminated plates are also presented.

Keywords: Composite laminated plate; Vibration; Sound radiation; Thermal environment; Closed form solution.

1. Introduction

Composite laminated plates have been extensively used in aeronautic, naval, civil and other engineering applications due to their high strength-to-weight ratio. They are usually exposed to the thermal environment during the service, which will significantly affect the dynamic properties of these structures. Therefore, investigations on the vibration and sound radiation characteristics of these structures under thermal environment are of great significance in the optimal design of these structures.

Considerable effort has been devoted to the vibration analysis of heated plates. Jadeja and Loo [¹] studied vibration problems of a rectangular heated plate with one edge clamped and three edges simply supported by using the assumed mode method. Ganesan and Dhotarad [²] applied the finite difference technique and variational approach to analyze the dynamic responses of the plate with thermal stresses. Tauchert et al. Rizzi [³] presented both theoretical and experimental studies for free vibration analysis of a fully clamped rectangular heated plate. Brischetto et al. [⁴] proposed a fully coupled thermo-mechanical model for the free vibration analysis of one-layered and two-layered metallic plates. Mantari et al. [⁵]
employed the Carrera's unified formulation to investigate thermo-elastic problems of simply supported laminated composite and sandwich plates subjected to thermal loads, where various plate theories were adopted in their studies.

In many engineering applications, the sound radiation of the vibrating plates is also a key value for the design of these structures. Jeyaraj et al. [6] investigated the structural and acoustic characteristics of an isotropic plate subjected to thermal loads by using commercial softwares ANSYS and SYNSNOISE. Then they [7] presented numerical simulation studies on the vibration and sound radiation of composite plates and multilayered viscoelastic sandwich plates in thermal environment by using FEM and BEM. Geng et al. [8] investigated the vibration and acoustic characteristics of the simply supported and clamped rectangular heated plates. Liu and Li [9] researched vibration and sound radiation problems of a simply supported sandwich heated plate. Yang et al. [10] presented the investigations for the sound radiation characteristics of simply supported functionally graded materials plates subjected to thermal loads.

From the above survey of researches it can be found that, the analytical solutions only exist for the plates with at least two parallel edges simply supported. Therefore, it’s significant to develop an efficient and convenient method to analyse the structural and acoustic problems of composite laminated heated plates, which is applicable to all combinations of simply supported and clamped boundary conditions. Recently, Xing and Liu [11] presented a closed form method for free vibration of rectangular orthotropic plates, which has a wide application prospect.

The present work aims to develop closed form solutions for predicting structural and acoustic responses of the composite laminated plates with multiple orthotropic and symmetrical layers under thermal environment. The solutions for vibration and acoustic radiation of the laminated heated plates with any combination of simply supported and clamped boundary conditions can be readily achieved by the developed method. The proposed method can give satisfactory predictions for structural and acoustic responses of laminated heated plates.

2. Theoretical formulations

2.1 Kinematic relations and stress resultants

Consider a laminated plate with multiple orthotropic and symmetrical layers in thermal environment shown in Figure 1. The Cartesian coordinate system established on the referenced middle surface is used to describe the geometry dimensions and deformations of the laminated plate. It is assumed that the laminated plate is mounted on an infinite flat rigid baffle and the thermal loads uniformly distribute through the thickness direction.

![Lamination, geometry and material coordinate systems of a laminated plate.](image)

The displacement components of the laminated plate can be expressed as

\[ u(x, y, z, t) = u_0(x, y, t) - z \frac{\partial w(x, y, t)}{\partial x} \]
\[ v(x, y, z, t) = v_0(x, y, t) - z \frac{\partial w(x, y, t)}{\partial y} \]
\[ w(x, y, z, t) = w_0(x, y, t) \]

where \( u_0, \ v_0, \ w_0 \) denote the displacement on the middle plane in the \( x, y \) and \( z \) directions, respectively.

The linear strains-displacement relations are defined with elasticity theory.

\[ e_x = \frac{\partial u_0}{\partial x} - z \frac{\partial^2 w}{\partial x^2} \]
\[ e_y = \frac{\partial v_0}{\partial y} - z \frac{\partial^2 w}{\partial y^2} \]
\[ \gamma_{xy} = \left( \frac{\partial^2 u_0}{\partial y^2} + \frac{\partial^2 v_0}{\partial x \partial y} \right) - 2z \frac{\partial^3 w}{\partial x \partial y^2} \]

(2)
For the $k$th orthotropic layer, the relation between Poisson's ratios and elasticity modulus can be expressed by

$$\frac{\nu_{12}^k}{E_1^k} = \frac{\nu_{21}^k}{E_2^k} \hspace{2cm} (3)$$

The corresponding stresses can be obtained by the generalized Hooke’s law

$$\begin{align*}
\sigma_x^k &= \left[ \begin{array}{ccc}
\overline{Q}_{11}^k & \overline{Q}_{12}^k & \overline{Q}_{16}^k \\
\overline{Q}_{21}^k & \overline{Q}_{22}^k & \overline{Q}_{26}^k \\
\overline{Q}_{16}^k & \overline{Q}_{26}^k & \overline{Q}_{66}^k
\end{array} \right] \begin{bmatrix}
\varepsilon_x^k - \alpha_{xx}^k \Delta T \\
\varepsilon_y^k - \alpha_{yy}^k \Delta T \\
\gamma_{xy}^k - \alpha_{xy}^k \Delta T
\end{bmatrix} \\
\end{align*}$$

where $\sigma_x^k$ and $\sigma_y^k$ denote the normal stresses in the $x$ and $y$ directions of the $k$th layer, respectively. $\tau_{xy}^k$ is the corresponding shear stress. $\alpha_{xx}^k$, $\alpha_{yy}^k$ and $\alpha_{xy}^k$ represent the linear thermal expansion coefficients. The temperature variation is defined as $\Delta T = T_1 - T_0$, where $T_0$ is the initial temperature and $T_1$ is the current temperature, respectively. The coefficients $\overline{Q}_{ij}^k$ ($i, j$=1, 2 and 6) and $\alpha_{pq}^k$ ($p, q=x$ and $y$) of the $k$th layer are defined as follows

$$\begin{align*}
\overline{Q}_{ij}^k &= \left[ \begin{array}{ccc}
\overline{Q}_{11}^k & \overline{Q}_{12}^k & \overline{Q}_{16}^k \\
\overline{Q}_{21}^k & \overline{Q}_{22}^k & \overline{Q}_{26}^k \\
\overline{Q}_{16}^k & \overline{Q}_{26}^k & \overline{Q}_{66}^k
\end{array} \right] = T_k \begin{bmatrix}
Q_{11}^k & Q_{12}^k & 0 \\
Q_{21}^k & Q_{22}^k & 0 \\
0 & 0 & Q_{66}^k
\end{bmatrix} \\
\alpha_{xx}^k &= \left( T_k \right)^T \begin{bmatrix}
\alpha_{xx}^k \\
\alpha_{yy}^k \\
\alpha_{xy}^k
\end{bmatrix}
\end{align*}$$

where superscript $T$ denotes the transposition. $Q_{ij}^k$ ($i, j$=1, 2 and 6) and $\alpha_{pq}^k$ ($p, q=x$ and $y$) are the elastic stiffness coefficients and linear thermal expansion coefficients along the principle axes of the $k$th orthotropic layer, where material constants $Q_{ij}$ are given by

$$\begin{align*}
Q_{11}^k &= \frac{E_1^k}{1-\nu_{12}^k \nu_{21}^k}, & Q_{12}^k &= \frac{\nu_{12}^k E_2^k}{1-\nu_{12}^k \nu_{21}^k}, & Q_{21}^k &= \frac{\nu_{21}^k E_1^k}{1-\nu_{12}^k \nu_{21}^k}, & Q_{22}^k &= \frac{E_2^k}{1-\nu_{12}^k \nu_{21}^k}, & Q_{66}^k &= G_{12}^k \\
T_k &= \begin{bmatrix}
\cos^2 \theta_k & \sin^2 \theta_k & -2 \sin \theta_k \cos \theta_k \\
\sin^2 \theta_k & \cos^2 \theta_k & 2 \sin \theta_k \cos \theta_k \\
\sin \theta_k \cos \theta_k & -\sin \theta_k \cos \theta_k & \cos^2 \theta_k - \sin^2 \theta_k
\end{bmatrix}
\end{align*}$$

where $\theta_k$ denotes the angle between the principal direction and the $x$-axis for the $k$th orthotropic layer. The lamina stiffness coefficients can be obtained by substituting Eq. (7) to Eqs. (5) and (6)
\[ \alpha_n = \alpha_n^k \cos^2 \theta_k + \alpha_n^s \sin^2 \theta_k - \alpha_n^{ks} \sin \theta_k \cos \theta_k \]
\[ \alpha_{ns} = \alpha_n^s \sin^2 \theta_k + \alpha_n^k \cos^2 \theta_k + \alpha_n^{ks} \sin \theta_k \cos \theta_k \]
\[ \alpha_{ns} = 2(\alpha_n^s - \alpha_n^k) \sin \theta_k \cos \theta_k + \alpha_n^k (\cos^2 \theta_k - \sin^2 \theta_k) \]

Due to the symmetry of the laminated plate, the thermal stresses can be expressed by

\[
\begin{bmatrix}
\sigma_x \\
\sigma_y \\
\tau_{xy}
\end{bmatrix} = \begin{bmatrix}
Q_{11}^T & Q_{12}^T & Q_{16}^T \\
Q_{21}^T & Q_{22}^T & Q_{26}^T \\
Q_{61}^T & Q_{62}^T & Q_{66}^T
\end{bmatrix} \begin{bmatrix}
\alpha_x \Delta T \\
\alpha_y \Delta T \\
\alpha_{xy} \Delta T
\end{bmatrix}
\]

The strains induced by temperature gradation can be written as

\[
d_k^1 = z^2 \left( \frac{\partial^2 W}{\partial x^2} \right)^2 + z^2 \left( \frac{\partial^2 W}{\partial x \partial y} \right)^2 + \left( \frac{\partial W}{\partial x} \right)^2
\]
\[
d_k^2 = z^2 \left( \frac{\partial^2 W}{\partial y^2} \right)^2 + z^2 \left( \frac{\partial^2 W}{\partial x \partial y} \right)^2 + \left( \frac{\partial W}{\partial y} \right)^2
\]
\[
d_k^3 = z^2 \left( \frac{\partial^2 W}{\partial x^2} \right)^2 + \left( \frac{\partial^2 W}{\partial x \partial y} \right)^2 + \left( \frac{\partial W}{\partial x} \right)^2
\]

2.2 Energy expressions

The strain energy expression of the laminated plate can be expressed as

\[
U = \frac{1}{2} \int_{\Omega} \int \sum_{k=1}^{n} \sigma_k^k \left( \varepsilon_k^k - \alpha_k^k \Delta T \right) + \sigma_k^s \left( \varepsilon_k^s - \alpha_k^s \Delta T \right) + \tau_{xy}^k \left( \gamma_{xy}^k - \alpha_{xy}^k \Delta T \right) \, dx \, dy
\]

The strain energy due to the thermal stresses can be given as

\[
U' = \frac{1}{2} \int_{\Omega} \int \sum_{k=1}^{n} \sigma_k^k d_k^1 + \sigma_k^s d_k^2 + 2 \tau_{xy}^k d_k^3 \, dx \, dy
\]

The kinetic energy of the laminated plate can be written as

\[
T = \frac{1}{2} \int_{\Omega} \int \sum_{k=1}^{n} \rho_k (z_{k+1}^i - z_k^i) \, dx \, dy
\]

The work done by external load can be given as

\[
W = \int_{\Omega} F \, dx \, dy
\]

where \( F \) denotes the external load.

2.3 Equation of motion

The Hamilton’s principle is written as follows

\[
\delta \int_{t_0}^{t_f} \left( T - U - U' + W \right) dt = 0
\]

For a constrained laminated plate, \( w_0 \) and \( v_0 \) are equal to zero. Substituting the Eqs. (13)-(16) into the Eq. (17), the governing equation can be obtained

\[
D_1 \frac{\partial^4 W}{\partial x^4} + 2D_3 \frac{\partial^3 W}{\partial x^2 \partial y^2} + D_2 \frac{\partial^2 W}{\partial y^2} - N_x \frac{\partial^2 W}{\partial x^2} - N_y \frac{\partial^2 W}{\partial y^2} + \rho \frac{\partial^2 W}{\partial t^2} = F
\]

in which

\[
D_1 = D_{12} + 2D_6; \quad D_3 = \frac{1}{3} \sum_{k=1}^{n} (Q_{11}^k - Q_{16}^k \alpha_n^k \Delta T - Q_{12}^k \alpha_{ns} \Delta T - Q_{16}^k \alpha_{ns} \Delta T)(z_{k+1}^i - z_k^i);
\]
\[ D_2 = \frac{1}{3} \sum_{k=1}^{n} (Q_{22}^i - Q_{21}^i \alpha_x^i \Delta T - Q_{22}^i \alpha_y^i \Delta T - Q_{26}^i \alpha_{xy}^i \Delta T)(z_{k+1}^i - z_k^i); \]

\[ D_6 = \frac{1}{3} \sum_{k=1}^{n} (Q_{26}^i - Q_{21}^i \alpha_x^i \Delta T + Q_{22}^i \alpha_y^i \Delta T + Q_{26}^i \alpha_{xy}^i \Delta T + Q_{22}^i \alpha_{xy}^i \Delta T + Q_{26}^i \alpha_{xy}^i \Delta T) / 4(z_{k+1}^i - z_k^i); \]

\[ N_x = -\sum_{k=1}^{n} (Q_{21}^i \alpha_x^i \Delta T + Q_{22}^i \alpha_y^i \Delta T + Q_{26}^i \alpha_{xy}^i \Delta T)(z_{k+1}^i - z_k^i); \]

\[ N_y = -\sum_{k=1}^{n} (Q_{21}^i \alpha_x^i \Delta T + Q_{22}^i \alpha_y^i \Delta T + Q_{26}^i \alpha_{xy}^i \Delta T)(z_{k+1}^i - z_k^i); \]

\[ D_{12} = \frac{1}{3} \sum_{k=1}^{n} Q_{12}^i (z_{k+1}^i - z_k^i); \]

\[ \bar{\rho} = \sum_{k=1}^{n} \rho_k(z_{k+1}^i - z_k^i); \]

3. **Solutions for the structural and acoustic response**

3.1 **Free vibration**

The solution of the homogeneous equation can be assumed as

\[ w = W(x, y) e^{i\omega t} \]  

Substituting Eq. (19) into Eq. (18) leads to the following differential equation

\[ D_x \frac{\partial^2 W}{\partial x^2} + 2D_x \frac{\partial^2 W}{\partial x \partial y} + D_y \frac{\partial^2 W}{\partial y^2} + N_x \frac{\partial W}{\partial x} + N_y \frac{\partial W}{\partial y} - \bar{\rho} \omega^2 W = 0 \]  

The mode function \( W(x, y) \) can be expressed by \( W(x, y) = Ae^{\mu_x x} \cdot Be^{\mu_y y} \)

Substituting into Eq. (20), the characteristic equation of Eq. (18) can be given by

\[ D_\mu^4 + 2D_\mu^2 \lambda^2 + D_\mu^4 - N_\mu^2 - N_\mu^2 - R^4 = 0 \]  

where the frequency parameter is defined as \( R^4 = \bar{\rho} \omega^2 \).

By solving the Eq. (22), the corresponding characteristic roots can be obtained as

\[ \mu_{1,2} = \pm i \alpha_1 = \pm i \sqrt{\frac{N_x + D_x D_y \beta_1^2}{2D_x D_y} \beta_1^4 + \frac{N_x D_y - N_y D_x}{D_x D_y} \beta_1^2 + \frac{N_y}{2D_y} \beta_1^2 + \frac{R^4}{D_y}} \]  

\[ \mu_{3,4} = \pm \alpha_2 = \pm \sqrt{\frac{N_x + D_x D_y \beta_2^2}{2D_x D_y} \beta_2^4 + \frac{N_x D_y - N_y D_x}{D_x D_y} \beta_2^2 + \frac{N_y}{2D_y} \beta_2^2 + \frac{R^4}{D_y}} \]  

\[ \lambda_{1,2} = \pm i \beta_1 = \pm i \sqrt{\frac{N_y + D_x D_y \alpha_1^2}{2D_x D_y} \alpha_1^4 + \frac{N_x D_y - N_y D_x}{D_x D_y} \alpha_1^2 + \frac{N_y}{2D_y} \alpha_1^2 + \frac{R^4}{D_y}} \]  

\[ \lambda_{3,4} = \pm \beta_2 = \pm \sqrt{\frac{N_y + D_x D_y \alpha_1^2}{2D_x D_y} \alpha_1^4 + \frac{N_x D_y - N_y D_x}{D_x D_y} \alpha_1^2 + \frac{N_y}{2D_y} \alpha_1^2 + \frac{R^4}{D_y}} \]  

According to the characteristic roots in Eq. (23), the eigenfunctions in the \( x \) and \( y \) directions can be written as

\[ \phi(x) = A_1 \cosh(\beta_x x) + A_1 \sinh(\beta_x x) + A_1 \cos(\alpha_x x) + A_1 \sin(\alpha_x x) \]  

\[ \phi(y) = B_1 \cosh(\beta_y y) + B_1 \sinh(\beta_y y) + B_1 \cos(\beta_y y) + B_1 \sin(\beta_y y) \]

Obviously, the mode function can be obtained by \( W(x, y) = \phi(x)\phi(y) \).

There are two boundary conditions at each edge of the laminated plate, which can be utilized to calculate the integral constants in Eqs. (23) and (24) and derive the frequency eigenvalue equations. Subsequently, the closed form equations can be established by combining the characteristics root equations and frequency eigenvalue equations. In order to simplify the specification of the boundary conditions, \( S \) and \( C \) are used to denote simply supported and clamped boundary conditions in this paper, respectively. To make it more clear, the formulation of frequency equations, mode functions and closed form solutions for a laminated heated plate under \( S-S-C-C \) boundary condition is presented in Appendix A.


3.2 Forced vibration

With the mode functions obtained in the previous subsection, the mode superposition method can be utilized to calculate the dynamic response of the laminated heated plates with the external excitation. A concentrated harmonic force \( q_0 \delta(x-x_0, y-y_0) e^{i\omega t} \) at point \((x_0, y_0)\) is considered in this paper. \( q_0 \) and \( \omega_0 \) denote the amplitude and the circular frequency of the load, respectively. \( \delta \) represents the Dirac function. Thus, Eq. (18) can be rewritten as

\[
D_1 \frac{\partial^4 w}{\partial x^4} + 2D_1 \frac{\partial^4 w}{\partial x^2 \partial y^2} + D_2 \frac{\partial^4 w}{\partial y^4} - N_i \frac{\partial^2 w}{\partial x^2} - N_j \frac{\partial^2 w}{\partial y^2} + \rho \frac{\partial^2 w}{\partial t^2} = q_0 \delta(x-x_0, y-y_0) e^{i\omega t} \quad (25)
\]

According to the mode superposition method, the solution for Eq. (25) can be obtained

\[
w(x, y, t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{q_0 W_{mn}(x_0, y_0)}{\omega_{mn}^2 - \omega_0^2} W_{mn}(x, y) e^{i\omega t} \quad (26)
\]

where \( W_{mn} \) and \( \omega_{mn} \) represent the mode function and natural frequency for the \((m, n)\) mode of laminated heated plates, respectively.

The sound pressure radiated from the vibrating plate at the observation point can be expressed as

\[
p(x_1, y_1, z_1, t) = \frac{i \alpha_0 \rho_0 e^{i\omega t}}{2\pi} \int_{S} v(x, y, z) e^{-ikR} ds \quad (27)
\]

4. Numerical Results and Discussion

4.1 Validation of the presented method

In this part, a considerable number of numerical examples are carried out to validate the accuracy and efficiency of the presented method. A MATLAB code is developed based on the proposed formulation and the FEM/BEM models are built in ABAQUS and Virtual Lab, respectively. A three-layer rectangular laminated plate with the same composite material for each layer is considered, as shown in Figure 2. The orientations of three layers from top to bottom are 90/0/90, and the thicknesses of these layers are 0.001m, 0.002m and 0.001m. The geometric and material parameters of each single layer are listed in Table 1. The initial temperature \( T_0 \) is defined as 293.15K and the structural damping ratio is 0.001 in this paper.

![Figure 2. The laminated plate with three orthotropic and symmetrical layers.](image)

<table>
<thead>
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<th>( L_1(\text{m}) )</th>
<th>( L_2(\text{m}) )</th>
<th>( \rho \text{(Kg/m}^3) )</th>
<th>( E_1\text{(GPa)} )</th>
<th>( E_2\text{(GPa)} )</th>
<th>( G_{12}\text{(GPa)} )</th>
<th>( v_{12} )</th>
<th>( \alpha_1 \text{(K}^{-1}) )</th>
<th>( \alpha_3 \text{(K}^{-1}) )</th>
<th>( \alpha_5 \text{(K}^{-1}) )</th>
</tr>
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<td>175</td>
<td>32</td>
<td>12</td>
<td>0.25</td>
<td>1.2e-6</td>
<td>2.3e-6</td>
<td>0</td>
</tr>
</tbody>
</table>

Firstly, the developed method is tested on the free vibration problems of laminated plates under thermal environment. The temperatures are taken into account in this section: \( T_1 = 303.15\text{K} \). The natural frequencies of the laminated plate with various boundary conditions obtained by the proposed approach are compared with those by ABAQUS, as shown in Table 2. Excellent agreements can be observed between the obtained results and those from FEM, which means the introduced approach is efficient to predict free vibration of laminated heated plates. Moreover, the first four mode shapes of the laminated heated plate under C-C-C-C boundary condition are presented in Figure 3.
Then, the forced vibration analysis of laminated heated plates subjected to a unit harmonic concentrated excitation is carried out by the proposed method and FEM. The location of external force is defined at the point \( P (x=0.4m, y=0.25m) \).

In the harmonic response analysis, The current temperature is selected as 303.15K here. Displacement responses measured at the point \( P \) of the laminated heated plate with S-S-C-C boundary conditions by proposed method and FEM are illustrated in Figure 4. The acoustic response in air problems of the vibrating laminated heated plate are researched. As shown in Figure 4, sound pressure response at the observation point \( Q (0m, 0m, 300m) \) of the vibrating laminated plates under S-S-C-C boundary conditions obtained by the developed approach are compared with those from BEM. In both cases, excellent agreements can be found between the obtained results from the developed formulation and those from FEM and BEM.

**4.2. Parameter study**

In order to evaluate the effects of the variation of temperature on the natural frequencies of laminated plates, the natural frequencies of C-C-C-C laminated plates with different temperatures are calculated as shown in Figure 5. It is obvious that natural frequencies of laminated plates decrease with increment of the temperature. The relationship between natural frequencies and temperature change is nonlinear. Figure 5 illustrates the displacement responses at the point \( P \) of laminated plates under different temperatures, respectively.
5. Conclusions

In this paper, a closed form formulation is developed to study the structural and acoustic responses of laminated heated plates with multiple orthotropic and symmetrical layers. Firstly, the governing equation of the laminated plate with thermal loads has been derived by applying the Hamilton’s principle. Next, the separation of variables approach is utilized to solve the governing equation. The accuracy and applicability of proposed method have been proved by comparing the obtained results including natural frequencies, dynamic and sound radiation responses with those from numerical simulations. The excellent agreement observed demonstrates the present model is efficient to evaluate vibration and sound radiation problems of laminated heated plates.

It is found that with the thermal loads increased, the stiffness softening phenomenon will occur, which leads to the decrement of the natural frequencies of laminated plates. The elevation of temperature will make the resonance peaks of dynamic and acoustic radiation responses shift to lower frequency range, but the global characteristics stay the same.

REFERENCES
