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Noise reduction analysis for a stiffened finite plate

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ABSTRACT

This paper presents an analytical solution for the vibration and acoustic responses of a finite stiffened plate that is covered with decoupling layers and subjected to external excitation. The theory of elasticity is used for the decoupling layer, and the stiffened plate is modeled by the plate theory and Euler–Bernoulli beam equation. Equations are constructed by the boundary conditions at the plate/coating and coating/fluid interfaces. The problem can be solved by the proposed method in this paper. Test verification shows that a good correlation exists between theoretical and test results. Thus, the theoretical study in this paper is correct. Numerical results show that shear waves insignificantly affect the structural vibration level difference (VR) under low frequencies. The noise reduction of the stiffened plate covered with decoupling layers is greatly influenced by the decoupling layer loss factor. A failure region of the vibration level difference is present in the low frequency band of the decoupling layer. Furthermore, the thickness of the decoupling layer significantly affects noise reduction.

1. Introduction

Changes in structural thickness and rib size are influenced by the static strength and stability limitations of the structure. Thus, reducing the structural vibration and radiated noise by changing the structural stiffness and mass poses a great challenge. To solve this problem, viscoelastic damping materials can be used on the plate to reduce structural vibration and radiated noise [1].

The differential equation of the thin plate theory was established in the 18th century [2]. Mindlin [3] revised thin plate theory by considering the influence of shearing deformation and rotary inertia. To analyze vibration wave propagation in ideal elastic solid media, 3D Navier equations can be used [4]. For the acoustic radiation of an infinite plate covered with damping layers, the structure insertion loss under mechanical excitation can be derived by Hankel transform and transfer matrix method [5]. Maidanik [6] established the vibro-acoustic coupled relation of elastic plates, upper and lower damping layers, and two semi-infinite subsystems by borrowing ideas from coupled dynamic systems. Gonzalez [7] used Fourier transform to obtain the sound radiation of an infinite three-layer composite plate under external force or acoustic excitation. Sandman [8] studied the noise reduction performance of three-layer composite rectangular plates. According to the kinetic equation of the three-layer composite plate, Sandman [8] used the modal superposition method to derive the analytic solution of radiated sound pressure. The influence of the damping loss factor of a sandwich layer on the quantity and directivity of radiated noise was analyzed. Moreover, Sandman [8] also indicated that the stiffness and damping loss factor

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of the sandwich layer are the major factors that control the resonant response and sound radiation of composite plates and that the radiated noise can be reduced by increasing the stiffness and damping loss factor. Keltie [9] researched the normal and tangential vibration velocity distribution in a damping layer when a plane acoustic wave incident from water occurs on the elastic coating attached to a submerged plate. Plate theory was applied to the plate, and elastic theory was used to describe the damping layer. The results show that the normal velocity remains the same, whereas the shear wave velocity changes obviously with increasing frequency. Minimum normal vibration velocity depends on compression resonance vibration, whereas maximum shear wave velocity is correlated with shear deformation. Cummings [10] investigated the sound radiation characteristics of an elastic plate covered with porous material, which is equivalent to a liquid layer. The effect of the thickness of porous materials on sound radiation efficiency was numerically analyzed. Ko [11,12] applied elastic theory and the double Fourier transform method on the study of structure-borne noise reduction for an infinite plate covered with a damping layer. Yong [13] researched the vibration and scattering sound field of a rectangular elastic–viscoelastic composite plate that is simply supported at the edges according to the extensive theory of flexural vibration of a composite plate system. Yong [13] also expanded the composite plate into a series of normal mode terms. Berry [14] analyzed the vibro-acoustic response of an elastic baffled rectangular plate covered with decoupling layers by regarding the decoupling layer material as a locally reacting model. By using the 3D theory of elasticity and modal superposition method, Berry [15] successfully investigated the vibration and sound radiation characteristics of a rectangular plate covered with decoupling layers and immersed in heavy fluid.

The stiffened plate structure has been studied by several scholars. Mukherjee et al. [16] analyzed the dynamic behavior of stiffened plate. Mead et al. [17] studied the harmonic response of reinforced viscoelastic sandwich plate. Hull [18] obtained the elastic analytic solution of reinforced thick plate. However, only few scholars have researched the dynamic behavior of a stiffened plate covered with viscoelastic material.

Constrained layer damping (CLD) treatment provides a method to control the vibration and noise in structures. Alam and Asnani [19,20] established the governing equation of CLD. The equation is suitable for arbitrary multilayers. Baz and Ro [21,22] controlled the bending mode by using a simple proportional control arrangement. Chantalakhana and Stanway [23] conducted a numerical and experimental study on the application of active constrained layer damping (ACLD) to a clamped–clamped plate. Passive viscoelastic constrained layer damping is augmented with an active scheme that employs a piezoelectric patch as the actuator.

In this paper, the noise reduction performance of a finite stiffened plate coated with decoupling layers in semi-infinite water is investigated. Section 2 presents the approximate analytic solution to the underwater acoustic radiation of a stiffened plate coated with decoupling layers. We present the analytic model of a stiffened plate with decoupling layer treatment by applying elasticity theory and mode superposition method to describe the decoupling layer and construct response functions, respectively. By using the deformation compatibility conditions of the interface between the stiffened plate and decoupling layer and the surface continuity conditions of the fluid–structure system, the vibration equations coupled by the sound–fluid–structure system are established. The proposed method is verified by the consistency between the calculated and model test results. The vibration and sound radiation characteristics of the stiffened plate are also researched.

2. Theoretical model of the stiffened plate with decoupled layers

The analytical model shown in Fig. 1 is a finite stiffened plate (dimensions a, b, and h) covered by a decoupling coating, which is immersed in semi-infinite water. E_1, υ_1, ρ_1, and η_1 are Young's modulus, Poisson's ratio, density, and structural loss factor of the stiffened plate, respectively. E_2, υ_2, ρ_2, and η_2 are Young's modulus, Poisson's ratio, density, and structural loss factor of the decoupling layer, respectively. The dynamic modulus is denoted by $\tilde{E}_j = E_j(1+i\eta_j)$, and $j=1, 2$. ρ_3 is the density of the fluid, and υ_3 is the sound velocity. The dynamic response of the thin plate is represented by the displacement w in the z direction at a neutral surface. The edges of the finite stiffened thin plate are observed as infinite baffle, and the coordinate system used in this theoretical model is shown in Fig. 1.

2.1. Model of the decoupling damping layer mechanism

The Navier equation can be expressed as follows:

$$(\lambda+\mu)\nabla(\nabla\cdot u)+\mu\nabla^2 u=\rho\frac{\partial^2 u}{\partial t^2}, \tag{1}$$

where λ,μ are the Lame constants: $\lambda=\tilde{E}_2\nu_2/(1+\nu_2)(1-2\nu_2)$, $\mu=\tilde{E}_2/2(1+\nu_2)$, and $\tilde{E}_2=E_2(1+i\eta_2)$. $u=u\mathbf{i}+v\mathbf{j}+w\mathbf{k}$ is the displacement vector, and ρ is the density of the decoupling layer. The solution of the equation is provided by $u=\nabla\Phi+\nabla\times H$, where Φ is the scalar potential, and $\mathbf{H}$ is the vector potential. Φ is a motor component that is unrelated to the rotation (i.e., irrotational), and $\nabla\times\nabla\Phi=0$. The pure rotational (tangential) motion is illustrated by $\mathbf{H}=H_x\mathbf{i}+H_y\mathbf{j}+H_z\mathbf{k}$. The divergence of $\mathbf{H}$ is zero. Therefore the vector wave equation can be decomposed into the following:

$$\nabla^2\Phi=\frac{1}{c_1^2}\frac{\partial^2\Phi}{\partial t^2}, \tag{2a}$$