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Letter to the Editor

Some physical insights for active control of sound radiated from a clamped ribbed plate

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ABSTRACT

An analytical study of mechanisms of active control of sound radiation from a rib stiffened plate is presented in this letter. Using the well-known modal expansion method to solve the beam/plate coupling vibration response, the physical mechanisms are interpreted in an analytical way similar as that used in the unribbed plate case. But some special rules are discovered for the ribbed plate case. The primary characteristic for the ribbed plate is that its new resonant modes are essentially the superposition of a couple of specific base plate modes that possess the same vibration pattern along the rib. These modes should be all suppressed in controlled condition to achieve on-resonances noise reduction. A large number of base plate modes contribute to off-resonances sound radiation such that multi-point forces are needed to rearrange their amplitudes and phases to guarantee their sound radiation being canceled each other. The imperfect control effects when control force acting on nodal areas of the mode of the ribbed plate are due to the opposite vibration states of the corresponding base plate modes where the control force applied for suppressing one mode may consequentially excite another.

1. Introduction

Beams stiffened thin plates have been considered as classical structural elements used in various engineering applications, such as aircraft fuselages, and ship hulls. Since aerospace and marine vehicles are continually subjected to uncertain dynamic loads, the ribbed plates may be frequently excited with excessive vibration levels which consequently induce high noise levels within the cabin. Traditional passive methods for suppressing noise radiation from vibrating structures work well in high and middle frequency ranges. However, its performances deteriorate rapidly in low frequency range where active methods [1] must be introduced to remedy this insufficiency.

Though the vibration characteristics of the ribbed plate are different from the unribbed case, but it is pertinent to predict that low frequency noise radiation from ribbed plate also needs to be controlled effectively by active method. Such topic of introducing active means to complex ribbed structure has attracted little attention until now and in turn deserves to be investigated specially in light of practical engineering requirements. Thus the purpose of this letter is planning to explore physical mechanisms inherent in the control process, which is hoped to provide some help for ongoing work of maximally improving low frequency performance of active ribbed plate structure.

An amount of approximate analytical and numerical methods have been proposed for confident prediction of free vibration of ribbed plates. These proposed approaches include the Rayleigh–Ritz energy method [2–4], finite difference method [5], semi-modal decomposition method [6], and differential quadrature method [7] and so on. Numerical methods such as the finite element method [8] (FEM) are extensively used nowadays in industries due to their high accuracy and versatility. These works gain a clear understanding of the vibration characteristics of complex ribbed plates and have offered great guidance in preventing their excessive vibration levels [9,10] and noise radiation [11] by passive methods. Recently Dozio [12] and Lin [13,14] use the well-known modal expansion method to predict free vibration and input mobility characteristics of finite ribbed plates for the requirement of estimating input power into the hull structure of a ship due to the exterior excitations. The beam/plate interface is modeled as a nonslip line connection in their model. Though the assumption is only valid when the beam width is not greater than the plate thickness, but the research is also significant since this type of structure is widely adopted in aeronautic and marine industry. Considering the feasibility of the analytical study, a simplified model of a clamped single beam stiffened plate applying controllable point forces to suppress low frequency noise radiation

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is established on the basement of Dozio and Lin's works. The single beam stiffened plate is used as an example so that attention can be concentrated on understanding fundamental physical nature and summarizing some preliminary rules. Clamped boundary edges [15] are also chosen to meet practical requirement. The mechanism is investigated using modal analysis method similar as that used in the unribbed plate case. But the physical nature has distinctive particularities that differ from the base plate. Due to the beam's coupling effects, the primary characteristic for the ribbed plate lies in the fact that each resonant mode is the superposition of several base plate modes. Thus modal suppression and modal rearrangement mechanisms summarized in excited literatures [16] cannot adequately and drastically explain physical natures of this case and should be modified for the ribbed plate. Some further deepened and detailed physical rules are summarized to offer helps for designing so called active quiet structure.

2. Theoretical model

A clamped ribbed plate baffled in an infinite wall and the associated coordinate system are shown in Fig. 1. The ribbed plate consists of a clamped base plate and a clamped beam, and the beam/plate interface is considered as a nonslip line connection. As a result of plate bending, the beam is subject to bending and torsion deformations where the coupling force F and coupling moment M should be considered at the beam/plate interface. Then the governing equation of motion of each substructure is reinforced by this pair of coupling loads.

Under an oblique incident plane wave excitation, the governing equations of the bending displacement (w) of the base plate and the beam flexural and torsional displacements (U, θ) have the same structures similar as these listed in [12]. The primary excitation $p_i(x,y,t) = p_0 e^{j(\omega t - kx\sin\theta\cos\alpha - ky\sin\theta\sin\alpha)}$ and the controllable point force $f_s = F_s\delta(x - x_s, y - y_s)e^{j\omega t}$ should be added into the right side of the bending displacement equation of the plate. In light of the modal superposition theory, these displacements (w, U, θ) can all be expressed as the superposition of a finite number of mode shapes. Further applying the two compatibility conditions at the beam/plate interface $[U(y) = w(x_b, y)$ and $\theta(y) = \partial w/\partial x(x_b, y)]$ can educe the following matrix equation that the modal amplitudes of the base plate satisfy,

$$(\mathbf{K} - \omega^2\mathbf{M})\mathbf{w} = \mathbf{Q}_p + F_s\mathbf{Q}_s. \tag{1}$$

where $\mathbf{K}$ and $\mathbf{M}$ are the stiffness and mass matrix, and the expression of their (mn, pq) element can be referenced in [12]. $\mathbf{w} = (w_{11}, w_{12}, \ldots w_{MN})^T$ is the modal amplitudes vector of the clamped base plate. $\mathbf{Q}_p$ is the generalized primary modal force vector, $Q_{p,mn} = \int_A p_i(x,y)\phi_m(x)\psi_n(y)dA$. $\mathbf{Q}_s$ is the generalized secondary modal force vector, $Q_{s,mn} = \phi_m(x_s)\psi_n(y_s)$, where (x_s, y_s) is the location on which the control point force acts.

Let the excitation terms in the right side of Eq. (1) equal to zero, the resulting equation is the generalized form of eigenproblem for the coupled beam/plate structure. This homogeneous equation

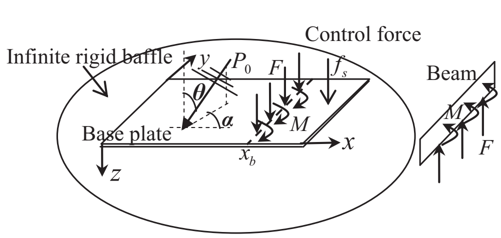

Fig. 1. The ribbed plate model and the illustration of the associated coupling force and moment excitations.

which poses a non-standard eigenvalue problem represents the free vibration characteristic of the coupled rib and plate structure. The lth eigenvalue λ_l and eigenvector $\mathbf{Q}_l$ can be solved from the sparse eigenequation by easily performing matrix transformations. The eigenvectors $(\mathbf{Q}_1, \mathbf{Q}_2 \ldots, \mathbf{Q}_l \ldots)$ are actually a set of orthogonal base functions that represent the resonant vibration patterns of the rib/plate coupled structure. The elements in each vector are the ratio coefficients of these base plate modes that participate in the coupled ribbed plate mode. And the eigenvalue λ_l accordingly represents the resonant frequency of the lth ribbed plate mode. Then the lth resonant mode shape of the ribbed plate $w_{\text{rib-plate},l}(x,y)$ and the resonant frequency $f_{\text{rib-plate},l}$ can be recovered as,

$$w_{\text{rib-plate},l}(x,y) = \mathbf{S}(x,y)^T\mathbf{w}_l, \tag{2-a}$$

$$f_{\text{rib-plate},l} = \frac{\sqrt{\lambda_l}}{2\pi}. \tag{2-b}$$

where $S = [\phi_1\psi_1, \phi_1\psi_2, \ldots, \phi_1\psi_N, \ldots, \phi_M\psi_N]^T, s_{mn}(x,y) = \phi_m(x)\psi_n(y)$ is the mode shape function of the base plate. M and N are the upper modal index considered in the simulation along the x and y axis, respectively.

According to the discrete elemental approach, the power output of the ribbed plate can be expressed as follows [17] when it is evenly divided into N_e elements,

$$W = \mathbf{V}^H\mathbf{R}\mathbf{V}. \tag{3}$$

where H denotes complex conjugate transpose, $\mathbf{V}$ is the velocity vector for an array of N_e elemental radiators. $\mathbf{R} = \Delta S\text{Re}(\mathbf{Z})/2, \mathbf{Z}$ is the $N_e \times N_e$ transfer impedance matrix. The modal amplitudes w of the clamped base plate can be solved from Eq. (1), and then the power output can be reexpressed as

$$W = (\mathbf{a} + \mathbf{b}F_s)^H\mathbf{R}(\mathbf{a} + \mathbf{b}F_s), \tag{4-a}$$

$$\mathbf{a} = j\omega\mathbf{\Phi}(\mathbf{K} - \omega^2\mathbf{M})^{-1}\mathbf{Q}_p, \tag{4-b}$$

$$\mathbf{b} = j\omega\mathbf{\Phi}(\mathbf{K} - \omega^2\mathbf{M})^{-1}\mathbf{Q}_s. \tag{4-c}$$

where $\mathbf{\Phi}$ is $N_e \times (M \times N)$ matrix that is consisted of the values of the mode shape functions of the base plate on N_e elements. Given that matrix $\mathbf{R}$ is real, symmetrical and positive definite, Eq. (4-a) has a Hermitian quadratic form. Using the linear quadratic optimal method, Eq. (4-a) will have a unique minimal value when the optimal control force amplitude $F_s = -(\mathbf{b}^H\mathbf{R}\mathbf{b})^{-1}\mathbf{b}^H\mathbf{R}\mathbf{a}$.

3. Active control results

In the simulation, assume that the plate and beam are both made of aluminum with density $\rho = 2790$ kg/m^3, Young's modulus $E = 7.2 \times 10^{10}$ N/m^2, and Poisson's ratio $v = 0.34$. The plate has a surface area of 0.6×0.42 m^2 and thickness of 0.003 m, while the beam is 0.42 m long with a flat rectangular cross section of $A = 0.003 \times 0.02$ m^2 which is located at $x_b = 0.15$ m. It is further assumed that the plate and beam both have a constant internal loss factor $\eta = 0.01$ which is added into the complex Young's modulus $E(1 + \eta i)$ to make simulation results further close to the practical situation [13]. The primary excitation is a plane wave with amplitude $P_0 = 1$ Pa, incident at $\theta = \pi/4$ and $\alpha = \pi/4$. Two secondary point forces are located at (0.5, 0.32) and (0.1, 0.32), respectively. The solution for the first sets of modes is well converged with acceptable accuracy when $M = N = 10$. With these parameters the first six eigenfrequencies and mode shape functions of the ribbed plate are calculated as shown in Table 1 and Fig. 2, which