NOISE RADIATION OF PERIODICAL SANDWICH STRUCTURES

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Summary The vibroacoustic behavior of rib-stiffened double-leaf structures is of growing interest due to their wide range applications in civil and transport engineering. An analytical model is developed in this paper for sound radiation of an infinite orthogonal rib-stiffened double-leaf structure subjected to a point force excitation. The governing equations for panel vibration are solved in the wavenumber space by utilizing the Fourier transform technique. In terms of dynamic panel deflections, the acoustic radiation of the sandwich structure is derived by integrating the radiation pressure intensity over a hemisphere. Obtained results for both a uniform single panel and the sandwich structure provide fundamental insight into how the vibroacoustic performance of the double-leaf partition is affected by orthogonal rib stiffeners.

INTRODUCTION

The vibroacoustic behavior of periodically rib-stiffened structures is of growing interest as these structures are increasingly used in civil and transport engineering (e.g., as the cabin skin of aircrafts, ships and high speed trains) [1-5]. In this article, we aim to study analytically the vibration and acoustic radiation of the more general 2D periodical structure consisting of two parallel plates reinforced by orthogonal extended ribs (i.e., sandwich panel with orthogonally rib-stiffened core); due to space restrictions, only a simplified outline of our theoretical work is presented.

MATHEMATICAL FORMULATION

Governing Equations for Panel Vibration

An infinitely large double-panel structure having periodic square lattice rib connections is considered. As shown in Figure 1, one facesheet of the structure (lying in the plane of $z=0$) is connected with the other facesheet (lying in the plane of $z=H$) by a set of orthogonal parallel rib connections with a periodic uniform spacing l_x and l_y in the x-and y-directions, respectively. The vibroacoustic behavior of the sandwich structure subjected to a point force excitation at (x_0, y_0) can be described by [6]:

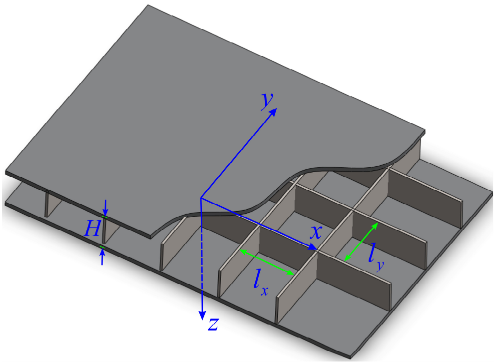

Figure 1: A typical sandwich structure with orthogonally rib-stiffened core

$$D_1\nabla^4 w_1(x,y;t)+m_1\frac{\partial^2 w_1(x,y;t)}{\partial t^2}+\sum_m\left[K_{Fx}(w_1(x,y;t)-w_2(x,y;t))-K_{Mx}\frac{\partial^2}{\partial x^2}(w_1(x,y;t)-w_2(x,y;t))\right]\cdot\delta(x-ml_x)$$
$$+\sum_n\left[K_{Fy}(w_1(x,y;t)-w_2(x,y;t))-K_{My}\frac{\partial^2}{\partial y^2}(w_1(x,y;t)-w_2(x,y;t))\right]\cdot\delta(y-nl_y)+q_0\cdot\delta(x-x_0)\delta(y-y_0)=0 \quad (1)$$

$$D_2\nabla^4 w_2(x,y;t)+m_2\frac{\partial^2 w_2(x,y;t)}{\partial t^2}+\sum_m\left[K_{Fx}(w_2(x,y;t)-w_1(x,y;t))-K_{Mx}\frac{\partial^2}{\partial x^2}(w_2(x,y;t)-w_1(x,y;t))\right]\cdot\delta(x-ml_x)$$
$$+\sum_n\left[K_{Fy}(w_2(x,y;t)-w_1(x,y;t))-K_{My}\frac{\partial^2}{\partial y^2}(w_2(x,y;t)-w_1(x,y;t))\right]\cdot\delta(y-nl_y)=0 \quad (2)$$

where $\delta(\cdot)$ is the Dirac delta function, m, $n=0$, ±1, $\pm2,\ldots$ for an infinite structure, M and N being the number of the rib connections parallel to the y- and x-axis, respectively, (K_{Fx}, K_{Mx}) and (K_{Fy}, K_{My}) are separately the translational stiffness and rotational stiffness per stiffener for the orthogonal rib connections parallel to the y- and x-axis, and (D_1, m_1, w_1) and (D_2, m_2, w_2) are the stiffness, mass density and deflection of the two facesheets. More details of the translational stiffness and rotational stiffness of the rib stiffeners will be presented in a separate study.

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Radiation Sound Power

The displacement of each facesheet can be solved in wavenumber domain by adopting the Fourier transform, hence, the radiating sound power P can be obtained by integrating the radiation intensity $\left|p(r,\theta,\varphi)\right|^2/2\rho_0 c_0$ over a hemisphere of radius r [7,8], as:

$$P=\frac{2\pi^2\rho_0\omega^4}{c_0}\int_0^{2\pi}\int_0^{\pi/2}\left|\tilde{w}(\alpha,\beta)\right|^2\sin\theta d\theta d\varphi \tag{3}$$

NUMERICAL RESULTS AND ANALYSIS

Figure 2 shows the predicted results of sound power radiated by the present sandwich structure with orthogonal rib connections. It is observed from Figs. 2 that within the frequency range of 1,200-1,800 Hz, the PWL of the sandwich structure has a significantly higher value. This is associated with the passband of wave propagation, owing to the particular configuration of the orthogonal rib connections in between the two facesheets. The denser peaks and dips in the high frequency range of Fig. 2 can be regarded as a reflection of the complex interference (i.e., constructive or destructive interference) between the traveling waves stemming from the vibrating facesheet.

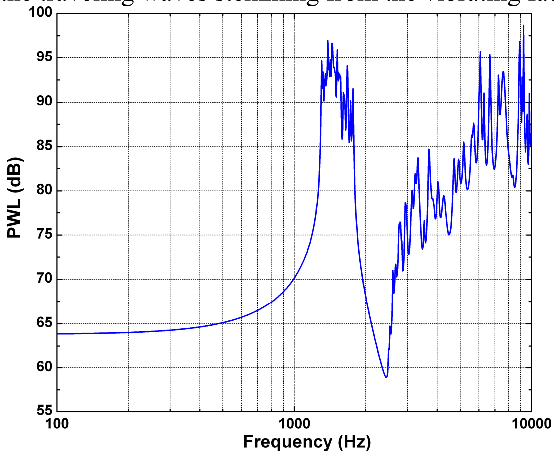

Figure 2: Sound power level (PWL in dB scale *re* 10^{-12} W) radiated by a sandwich panel with orthogonal rib connections plotted as a function of frequency

CONCLUSIONS

A theoretical model is developed for the radiation of sound from an infinite double-leaf partition periodically stiffened by orthogonal rib connections under a point force excitation. The technique of Fourier transform is employed to solve the governing equations for panel vibration in the wavenumber space. Once the dynamic deflections of the panels are obtained, the acoustic radiation of the sandwich structure is derived by integrating the radiation pressure intensity over a hemisphere. Numerical results show the characteristics of the PWL curve tendency for the sandwich structure and demonstrate that, different from that of a uniform single panel, there exists a passband for wave propagation. The wave propagation characteristics resulting purely from the sandwich configuration have been well captured.

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