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Reduction of the sound transmission of a periodic sandwich plate using the stop band concept

Yubao Song [a,b], Leping Feng [b], Jihong Wen [a,*], Dianlong Yu [a], Xisen Wen [a]

[a] Laboratory of Science and Technology on Integrated Logistics Support, National University of Defense Technology, 410073 Changsha, People's Republic of China
[b] The Marcus Wallenberg Laboratory for Sound and Vibration Research, KTH – The Royal Institute of Technology, SE-100 44 Stockholm, Sweden

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ABSTRACT

The sound transmission of a sandwich plate and its reduction using the stop band concept are investigated in this paper. A periodic sandwich plate consisting of a host plate and the attached structures is designed. The dispersion relation and the stop band of the periodic sandwich plate are studied first. The sound transmission of the periodic and bare sandwich plates is analysed and compared. The reduction from the stop band design (i.e., periodically adding stepped resonators) on the sound transmission of the sandwich plates is studied. The reasons for this reduction are analysed. In addition, the properties of the sandwich plate with different boundary conditions are also briefly studied. The numerical results indicate that the sound transmission is significantly reduced over the stop band of the periodic sandwich plate. The improvement can also exist in the frequency range outside the stop band.

1. Introduction

Lightweight sandwich plates have numerous applications in the civil, aerospace and automobile industries. The mechanical properties of these plates offer a distinct advantage in weight over other commonly used construction materials, such as rib stiffened plates. However, the high stiffness-to-mass ratio can result in supersonic wave propagation at relatively low frequencies and imparts poor acoustical performance [1,2].

Abundant work concerning the mechanical properties, the dynamic model, and the vibration and sound properties of sandwich structures have been conducted [1–10]. Nilsson et al. [4] and Sorokin [6] derived a sixth-order theoretical model to describe the dynamic properties of the sandwich beam. Moore and Lyon [10] investigated the sound transmission loss in the unidirectional sandwich panels with isotropic and orthotropic cores. Recently, some researchers used the finite element method in the study of the dynamic characteristics of sandwich structures [7,8,11–15]. Buehrle et al. [11] and Robinson et al. [14] developed a finite element and boundary element model to investigate the vibro-acoustic response of a curved honeycomb composite panel. The honeycomb core was modelled with solid elements, and the face sheets are modelled using linear plate elements.

Some work has also been performed to improve the vibro-acoustic performance of sandwich structures. Sun et al. [16], Ruzzene et al. [17] and Steven et al. [18] performed optimization studies. There are many research studies on suppressing the vibration and noise of a homogeneous plate or other structures [19–26], which also provide a useful reference. Fuller et al. [20] and Idrisi et al. [25] investigated the reduction of low frequency sound and vibration using the idea of heterogeneous (HG) blanket and distributed vibration absorbers (DVAs).

In the past two decades, the propagation of elastic or acoustic waves in periodic composite materials, known as phononic crystals (PCs), has received much attention [23,27–34]. PCs have elastic/acoustic band gaps (or stop bands), within which acoustic/elastic waves cannot propagate freely without attenuation. These stop bands are of great interest in the field of vibration and noise control [28,33,35–39]. Taking the 2D plates as an example, much work had been performed to suppress the vibration transmission within it [28,33,34]. The research on using the concept of metamaterials (AMs) to improve STL was also performed [31]. Xiao et al. [23] studied the sound isolation of a homogeneous plate with a stop band. To generate the stop band, one of the effective methods is to add a resonator in the periodic composites [23,28,29,32,33].

Recently, the concept of stop bands has also been introduced into the design of sandwich plates to control the wave propagation within them [40–44]. Liu et al. [43] investigated the propagation of Lamb waves in a sandwich plate with a periodic composite core by the finite element method. Chen et al. [40,41] studied the wave

* Corresponding author. Tel./fax: +86 0731 84574975.
E-mail address: wenjihong_nudt1@vip.sina.com (J. Wen).

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propagation in a sandwich beam with local resonators (and periodic cores) analytically and experimentally. The previous works indicate that the design of stop bands can be a new approach for reducing the vibration transmission within a sandwich structures. However, research regarding the stop band properties of a sandwich panel is rare. In particular, there have only been a few published papers regarding the reduction of sound transmission of a sandwich plate by using stop bands.

In this paper, the sound transmission of a sandwich plate is studied, with a focus on its reduction through the use of stop bands. A theoretical model for a sandwich plate is presented. First, the dispersion relation and the stop band of a periodic sandwich plate are studied. Next, the sound transmission properties of periodic and bare sandwich plates are analysed. The improvement on sound insulation via the stop band design of the sandwich plates (i.e., periodically attached resonators) is investigated. In addition, the effect of the boundary conditions of the sandwich plate on the sound reduction is also simply studied.

2. Model and theory

The sandwich plate considered here consists of two thin aluminium faces bonded to a thick and lightweight honeycomb core. In this study, the honeycomb core is simplified to a homogeneous layer with orthotropic mechanical properties. To generate the stop band, a periodic sandwich plate consisting of a host plate and periodically attached structures is designed.

2.1. Wave propagation in an infinite periodic sandwich plate

Fig. 1(a) illustrates the schematic of the element of the periodic sandwich plate with attached stepped resonators. The stepped resonator consists of a block of soft and light material with a cap of hard and heavy material. According to Bloch's theory [28,44], the wave motion in a 2D periodic system can be described as

$$\mathbf{u}_j(x+n_xa_x, y+n_ya_y, t) = \mathbf{u}_j(x,y)\cdot \mathrm{e}^{\mathrm{i}\omega t}\cdot \mathrm{e}^{-(n_x\mu_xa_x+n_y\mu_yb_y)} \quad (1)$$

where $\mathbf{u}_j (j = x, y, z)$ represents the displacement in the x, y and z directions, (n_x, n_y) is an integer pair, $\mu_l = \delta_l + ik_l, (l = x, y)$ are the propagation constants in the x and y directions. The real and imaginary parts of the propagation constants are denoted as 'attenuation' and 'phase' constants [44], respectively. If μ_l is purely imaginary, then the waves are free to propagate; if μ_l is purely real or complex, then the waves are attenuated as they propagate from one cell to another.

For 2D periodic structures, via the determination of various combinations of propagation constants (μ_x, μ_y) in the x and y directions, the frequency values where free wave propagation occurs can be evaluated. The finite element method (FEM) combined with the Bloch periodic boundary conditions is applied to investigate the band structures, which is also one of the common methods [28,33,44]. Because of the periodicity, only one periodic element needs to be considered.

The equation of harmonic motion for the periodic element is given by

$$[\mathbf{K} - \omega^2\mathbf{M}]\mathbf{q} = \mathbf{F} \quad (2)$$

where $\mathbf{M}$ and $\mathbf{K}$ are the mass and stiffness matrices for the periodic element, $\mathbf{q}$ and $\mathbf{F}$ are the generalised displacement and forces vectors, respectively, $\mathbf{q} = [\mathbf{q}_I^T \;\; \mathbf{q}_B^T \;\; \mathbf{q}_T^T \;\; \mathbf{q}_L^T \;\; \mathbf{q}_R^T \;\; \mathbf{q}_{LB}^T \;\; \mathbf{q}_{RB}^T \;\; \mathbf{q}_{LT}^T \;\; \mathbf{q}_{RT}^T]^T$ and $\mathbf{F} = [\mathbf{F}_I^T \;\; \mathbf{F}_B^T \;\; \mathbf{F}_T^T \;\; \mathbf{F}_L^T \;\; \mathbf{F}_R^T \;\; \mathbf{F}_{LB}^T \;\; \mathbf{F}_{RB}^T \;\; \mathbf{F}_{LT}^T \;\; \mathbf{F}_{RT}^T]^T$, as denoted in Fig. 1(b). For free wave propagation, $\mathbf{F}_I = 0$.

According to the Bloch theory, the propagation of elastic waves from one cell to another is studied by considering the interaction of the displacements and the forces between the cell and its neighbours. Further, the displacements and forces at the boundary can be expressed by the reduced degree of freedom

$$\mathbf{q} = \mathbf{R}\mathbf{q}', \quad \mathbf{F} = \mathbf{T}\mathbf{F}' \quad (3a, b)$$

with $\mathbf{q}' = [\mathbf{q}_I^T \;\; \mathbf{q}_B^T \;\; \mathbf{q}_L^T \;\; \mathbf{q}_{LB}^T]^T, \mathbf{F}' = [\mathbf{F}_I^T \;\; \mathbf{F}_B^T \;\; \mathbf{F}_L^T \;\; \mathbf{F}_{LB}^T]^T$. Substituting Eq. (3) in Eq. (2), the resulting equation in the reduced degree of freedom is given by

$$[\mathbf{K}' - \omega^2\mathbf{M}']\mathbf{q}' = 0 \quad (4)$$

where $\mathbf{K}' = \mathbf{R}^H(\mu_x, \mu_y)\mathbf{K}\mathbf{R}(\mu_x, \mu_y), \mathbf{M}' = \mathbf{R}^H(\mu_x, \mu_y)\mathbf{M}\mathbf{R}(\mu_x, \mu_y)$ are the reduced mass and stiffness matrices of the cell, and $\mathbf{R}^H$ denotes the complex conjugate transpose of $\mathbf{R}$. The values of frequency corresponding to an assigned set of propagation constants pair (μ_x, μ_y) can be obtained by solving the following eigenvalue problem

$$|\mathbf{K}'(\mu_x, \mu_y) - \omega^2\mathbf{M}'(\mu_x, \mu_y)| = 0 \quad (5)$$

When we focus on wave propagation without attenuation or dispersion relations, the attenuation constants δ_x and δ_y are set to zero. By varying the values of k_x and k_y in the irreducible

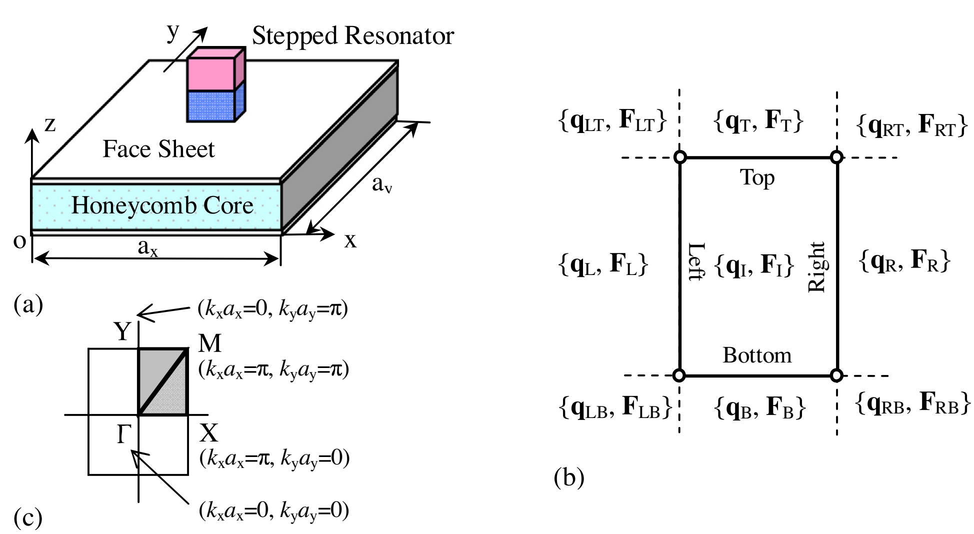

Fig. 1. (a) The element of the periodic sandwich plates with attached stepped resonators, (b) periodic element with the interactions with its adjacent elements, i.e. the forces and the displacements at the boundaries, (c) the corresponding first irreducible Brillouin zone (shaded region).