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Assessments of shear core effects on sound transmission loss through sandwich panels using a two-scale approach

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ABSTRACT

The present work proposes a modeling strategy based on a two-scale approach to deal with the flexural vibroacoustic behavior of sandwich panels notably in the low-mid frequency range. The vibroacoustic indicators of interest here are primarily the sound transmission loss and the bending-shear transition frequency of the sandwich panels. The stated indicators depend mainly on the geometrical and mechanical parameters of the studied cases. The parametric analyses, therefore, bring out the effects of the different meso-scale parameters on the macro-scale responses of the sandwich panels by performing three-dimensional model of a representative elementary cell. The two-scale approach is mainly based on a numerical method known as the wave finite element method. In the developed approach, an expression of the sound transmission loss was derived wherein the shear core effect of sandwich panels has been included. The obtained results were compared with two models and experimental data to check the accuracy of the prediction and showed a good agreement.

1. Introduction

Most industrial applications are seeking for innovative structures, which are characterized by high mechanical and chemical performances as well as they could be as light as possible. Nowadays, the most suitable structure that can combine all these properties is known as a sandwich structure. This kind of structure is widely used in several sectors especially in the high-speed transportation industries such as launcher, planes, trains and so on and so forth. This trend is thanks to their great global mechanical properties with respect to their micro mechanical properties [1–3]. Nevertheless, from the vibroacoustic side, increasing stiffness to weight ratio could have a harmful effect on the vibroacoustic performance of the structure which then might lead to unsatisfactory sound insulation performance.

Theoretically, the root cause of the poor acoustic transparency of sandwich panels is due mainly to the shear core effect at the mid frequencies. Fahy and Gardonio [4] defined three dominant zones for a typical sandwich panel: global panel bending, core shear and individual face-sheet bending. The frequency separating the first two zones, known as the transition frequency, is of great interest [5,6]. The main idea of the latter papers was to increase the transition frequency of a sandwich panel with a goal of shifting the core shear effect to the high-frequency range wherein the acoustic transparency can be easily controlled by adding a porous material for instance. Many published papers [5,7–9] propose developments of the transition frequency expression. In the present study, a survey has been carried out to reveal the influence of an increasing transition frequency on the sound transmission loss (TL) of sandwich panel with orthotropic face-sheets.

Over the past several decades, various methods have been proposed for predicting the sound transmission loss (TL) through sandwich panels. It has been proved in the Kurtze and Watters study [10] that TL is controlled by three physical characteristics, the mass per unit area, the dynamic bending stiffness and the internal damping of the structure. The latter model was improved in Ford's development [11] by introducing compressible core and dilatational motions which enables to study foam sandwich core panels. A few years later, Statistical Energy Analysis (SEA) has been used by Crocker and Price [12] to predict the sound transmission loss, the radiation resistance and the vibration amplitude of a partition. Using a software based on the SEA, De Rosa et al. [13] presented numerical evaluations of the sound transmission loss of some panel configurations. The evaluations have been compared to the theoretical and experimental data quoted from the

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literature. Renji [14] has derived an analytical transmission loss expression in terms of the coincidence frequency considering also the shear core flexibility. Griese et al. [15] carried out a parametric study on the sound transmission loss and the vibration properties of in-plane loaded honeycomb sandwich panels using a structural acoustic finite element method with varying the honeycomb cellular core geometries. Recently, Mejdi et al. [16] has developed an approach based on the discrete laminate model (DLM) [17], called a wave spectral finite element model (WSFEM), that can capture the stretch information through the thickness of the sandwich core.

On the other hand, many experimental attempts have been performed to understand and enhance the flexural vibroacoustic behavior of the honeycomb sandwich panel. Palumbo and his co-workers [18] carried out an experimental analysis by creating different forms and size of areas with reduced stiffness in the honeycomb core. The created areas led to a notable benefit on the sound transmission loss either in the low-mid or high frequency ranges but to a significant decrease in the panel stiffness as well. A year later, Peters and Nutt [19] established an experimental parametric study using a structure-borne excitation. The main purpose was to validate the influence of the shear wave speeds on the sound transmission loss which has been revealed analytically by Kurtze and Watters in [10] and Davis in [8]. Similarly, Naify et al. [20] has performed theoretical and experimental surveys in order to reduce the amount of sound energy transmitted through the honeycomb sandwich panel by attaching additional gas layers. It has been shown that the impedance mismatch of gasses is an efficient means to relatively decrease the noise transmitted through a honeycomb sandwich panel.

The main purpose of this study is to investigate the shear core and transition frequency effects of sandwich panels on the sound transmission loss (TL) through a multi-scale like numerical method. Parametric analyses were then conducted to reveal the influence of the increasing transition frequency. The proposed two-scale approach is based on a numerical method known as the wave finite element method (WFE) [21,22]. The WFE method combines the classical finite element method (FEM) and the periodic structure theory (PST) developed in [23]. The main advantage of the developed approach is that it takes into consideration the periodicity of the structure, which allows modeling typically just one elementary cell instead of the whole structure. Accordingly, the computation cost is hugely reduced. Moreover, this numerical model keeps the meso-scale parameters of the periodic cell.

The paper is then structured as follows: Section 2 is devoted to develop the 3D numerical model, based on the 2D WFE method. Then, the numerical prediction of the bending-shear transition frequency has been briefly presented. The transition has been identified using the numerical techniques developed in [5,6]. In Section 3, the expression of the sound transmission loss (TL) for a sandwich panel has been presented in detail. The formula of TL was derived using Renji's analytical model [24] and the equivalent wavenumbers obtained by the WFE method. In the last section, the validation of the proposed modeling strategy has been checked and then a parametric survey has been carried out for different topologies of the sandwich core.

2. Wave propagation through a sandwich panel

2.1. 2D wave finite element finite formulation

A rectangular or parallelogram periodic cell of the honeycomb sandwich panel is hereby considered (see Fig. 1) which represents the meso structural scale in the present paper. The topology of the both periodic cell is generally determined by six conventional geometric parameters: the cell angle α, the vertical member length h, the angled member length l, the wall thickness t, the core thickness h_c, and the face-sheet thickness h_f. The wall thickness in the vertical member length h is double the one in the angled member length l. In the present study, the honeycomb sandwich panel was modeled as 3D geometric architecture without using any homogenization techniques.

The dynamic characteristics of the sandwich panels were predicted by applying the wave finite element method. The numerical method comprises the reformulation of the motion equation by using the dynamic stiffness matrix (DSM). The DSM involves the reduced mass $\widetilde{M}$ and stiffness $\widetilde{K}$ matrices of a periodic cell of the sandwich panel. The wave motion through the sandwich structure is expressed in terms of the eigenvalues and the eigenvectors of the dynamic stiffness matrix.

$$D^d = \widetilde{K} - \omega^2 \widetilde{M}. \tag{1}$$

The reduced mass $\widetilde{M}$ and stiffness $\widetilde{K}$ matrices were obtained by applying the reduced computation processes derived for 2D WFE formulations. The reduction has been proposed by C. Droz et al. in [22,25]. The proposed strategy develops a suitable numerical technique to reduce the computation time of the wave dispersion in 2D periodic systems by combining the Component Mode Synthesis (CMS) approach and the model order reduction (MOR).

CMS is based on the Craig-Bampton method which enables to reduce the interior dofs of the structure by transforming the interior physical displacement U_I into the local modes representing the structural dynamic information. The CMS approach defines the reduced stiffness and mass matrices of the periodic cell as follows:

$$\widetilde{M} = R^T M R \quad and \quad \widetilde{K} = R^T K R \quad with \quad R = \begin{pmatrix} I & 0 \\ \Phi_R & \Phi_L \end{pmatrix}, \tag{2}$$

where M and K are respectively the full mass and stiffness matrices of the periodic cell extracted using a classical finite element package, which is in the present work Ansys software. Φ_R and Φ_L are matrices given in detail in [26]. I is the identity matrix with the same size as the dofs of the boundary nodes.

Using the 2D WFE formulation, the motion equation for periodic structural waveguides can be rewritten by expressing Eq. (1) in terms of block matrices $\left(D^d_{bdI}, D^d_{bdbd}, D^d_{Ibd}, \text{ and } D^d_{II}\right)$ as follows:

$$\begin{pmatrix} D^d_{bdI} & D^d_{bdbd} \\ D^d_{Ibd} & D^d_{II} \end{pmatrix} \begin{pmatrix} U_{bd} \\ U_I \end{pmatrix} = \begin{pmatrix} f_{bd} \\ 0 \end{pmatrix}, \tag{3}$$

where the indexes I and bd stand for the internal and boundary nodes, respectively. The degrees of freedom (dofs) of the displacement vector related to interior nodes U_I are condensed to the dofs of the displacement vector related to boundary nodes by substituting the bellow expression (4) into Eq. (3).

$$U_I = -\left(D^d_{II}\right)^{-1} D^d_{Ibd} U_{bd}. \tag{4}$$

The dofs of the boundary nodes U_{bd} comprise the dofs of the edge nodes (U_B, U_L, U_R, U_T) and the corner nodes (U_1, U_2, U_3, U_4). The present numerical approach assumes that each opposite edge and corner nodes must have exactly the same number of dofs. However, there are no mesh constraints for the interior nodes of the periodic cell. The FEM mesh inside the waveguide periodic cell can then be applied arbitrarily.

By applying the Floquet-Bloch theory and assuming a time harmonic response, the displacements at each edge and corner can be written as a function of the displacements at one single corner U_1 and edges U_B and U_L.