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Vibro-Acoustics of Lightweight Sandwich Structures

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Preface

The purpose of this book is to present the vibration and acoustical behavior of typical sandwich structures subject to mechanical and/or acoustical loadings, which actually form a class of structural elements of practical importance in huge amounts of engineering applications, such as aircraft fuselage, ship and submarine hulls. The contents of this book has grown out of the research activities of the authors in the field of sound radiation/transmission of/through lightweight sandwich structures.

The book is organized into six chapters: Chapter 1 deals with the vibro-acoustic performance of rectangular multiple-panel partitions with enclosed air cavity theoretically and experimentally, which has accounted for the simply supported and clamp supported boundary conditions. Chapter 2 concerns with the transmission of external jet-noise through a uniform skin plate of aircraft cabin fuselage in the presence of external mean flow. As an extension, Chap. 3 handles with the noise radiation and transmission from/through aeroelastic skin plates of aircraft fuselage stiffened by orthogonally distributed rib-stiffeners in the presence of convected mean flow. Chapter 4 develops a theoretical model for sound transmission through all-metallic, two-dimensional, periodic sandwich structures having corrugated core. Chapter 5 focuses on the sound radiation and transmission characteristics of periodically stiffened structures. Ultimately, Chap. 6 proposes the sound radiation and transmission behaviors of periodical sandwich structures having cavity-filling fibrous sound absorptive materials.

This book is involving multidisciplinary subjects especially including combined knowledge of vibration, aeroelastics and structural acoustics, which pays much attention on showing results and conclusions, in addition to mere theoretical modelling. Therefore this book should be of considerable interest to a wide range of readers in relevant fields. It is hoped that the content of the book will find application not only as a textbook for a wide audience of engineering students, but also a general reference for researchers in the field of vibrations and acoustics.

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Contents

Chapter 1
Transmission of Sound Through Finite Multiple-Panel Partition

Abstract This chapter is organized as three parts: in the first part, the vibroacoustic performance of a rectangular double-panel partition with enclosed air cavity and simply mounted on an infinite acoustic rigid baffle is investigated analytically. The sound velocity potential method rather than the commonly used cavity modal function method is employed, which possesses good expandability and has significant implications for further vibroacoustic investigations. The simply supported boundary condition is accounted for by using the method of modal function, and double Fourier series solutions are obtained to characterize the vibroacoustic behaviors of the structure. Results for sound transmission loss (STL), panel vibration level, and sound pressure level are presented to explore the physical mechanisms of sound energy penetration across the finite double-panel partition. Specifically, focus is placed upon the influence of several key system parameters on sound transmission, including the thickness of air cavity, structural dimensions, and the elevation angle and azimuth angle of the incidence sound. Further extensions of the sound velocity potential method to typical framed double-panel structures are also proposed.

In the second part, the air-borne sound insulation performance of a rectangular double-panel partition clamp mounted on an infinite acoustic rigid baffle is investigated both analytically and experimentally, and compared with that of a simply supported one. With the clamped (or simply supported) boundary accounted for by using the method of modal function, a double series solution for the sound transmission loss (STL) of the structure is obtained by employing the weighted residual (Galerkin) method. Experimental measurements with Al double-panel partitions having air cavity are subsequently carried out to validate the theoretical model for both types of the boundary condition, and good overall agreement is achieved. A consistency check of the two different models (based separately on clamped modal function and simply supported modal function) is performed by

extending the panel dimensions to infinite where no boundaries exist. The significant discrepancies between the two different boundary conditions are demonstrated in terms of the STL versus frequency plots as well as the panel deflection mode shapes.

In the third part, an analytical model for sound transmission through a clamped triple-panel partition of finite extent and separated by two impervious air cavities is formulated. The solution derived from the model takes the form of that for a clamp-supported rectangular plate. A set of modal functions (or more strictly speaking, the basic functions) are employed to account for the clamped boundary conditions, and the application of the virtual work principle leads to a set of simultaneous algebraic equations for determining the unknown modal coefficients. The sound transmission loss (STL) of the triple-panel partition as a function of excitation frequency is calculated and compared with that of a double-panel partition. The model predictions are then used to explore the physical mechanisms associated with the various dips on the STL versus frequency curve, including the equivalent "mass-spring" resonance, the standing-wave resonance, and the panel modal resonance. The asymptotic variation of the solution from a finite-sized partition to an infinitely large partition is illustrated in such a way as to demonstrate the influence of the boundary conditions on the soundproofing capability of the partition. In general, a triple-panel partition outperforms a double-panel partition in insulating the incident sound, and the relatively large number of system parameters pertinent to the triple-panel partition in comparison with that of the double-panel partition offers more design space for the former to tailor its noise reduction performance.

1.1 Simply Supported Finite Double-Panel Partitions

1.1.1 Introduction

Double-leaf partition structures have found increasingly wide applications in noise control engineering due to their superior sound insulation capability over single-leaf configurations. Typical examples include transportation vehicles, grazing windows and partition walls in buildings, aircraft fuselage shells, and so on [1–12].

Considerable efforts have been devoted to understanding and predicting the transmission of sound across single-leaf [13–15] and double-leaf [16–29] partitions. In fact, research about the former is often a prerequisite for studying the latter. For instance, Lomas [14] developed Green function solution for the steady-state vibration of an elastically supported rectangular plate coupled to a semi-infinite acoustic medium. An important feature of the investigation is the treatment of the elastic support boundary condition which was taken into account by assuming the rotational motion along the boundary controlled by distributions of massless rotary springs and by introducing the corresponding moments into the governing equations. The problem of sound radiation by a simply supported unbaffled panel was investigated by Laulagnet [13]. Both pressure jump and plate displacement in series of the simply supported plate models were developed.

Early sound transmission studies [16, 28–30] of double-panel structures with air cavity in between generally simplified the structure as infinite and hence did not account for the elastic boundary conditions on the periphery. For typical examples, Antonio et al. [17] gave an analytical evaluation of the acoustic insulation provided by double infinite walls and also did not take elastic boundary condition into account. Kropp et al. [19] addressed the optimization of sound insulation of double-panel constructions by dividing the frequency range into three cases, i.e., where the double wall resonance frequency is much higher (or closer or much lower) than the critical frequency of the total construction. Recently, Tadeu et al. [20] adopted an analytical method to assess the airborne sound and impact insulation properties of single- and double-leaf panels by neglecting the elastic boundary conditions. Bao and Pan [31] presented an experimental study on active control of sound transmission through double walls with different approaches, including cavity control, panel control, and room control.

For simply supported, finite rectangular double-panel structures, existing studies [3, 22–27, 32–37] concerned mainly with the loss of sound transmission across the structure, without detailed analysis about the energy transmission, the vibroacoustic coupling effects, and the physical mechanisms of sound transmission process across the structure. In particular, previous studies on double-panel partitions focus on either infinite extent or finite extent, without exploring the natural relationship between the two. The present study squarely addresses these deficiencies from the new perspectives of the integration analysis of STL, panel vibration level, and sound pressure level, with more details and the physical nature of sound penetration through double-panel partitions revealed. Since the rigid baffle bounds the cavity as well as the panel so that the cavity boundaries restrict the field to sinusoidal distributions parallel to the panel plane, analytical solutions in double Fourier series are proposed by applying the sinusoidal distributed sound velocity potential method. This method can be easily expanded to the vibroacoustic analysis of rib-stiffened double-panel structures, accounting for both the structure-borne route (i.e., structural connections between the two panels) and the airborne route (i.e., air cavity between the two panels), and hence can be regarded as an alternative of the cavity mode method in certain engineering applications. The model predictions are validated by comparing the analytical results with existing experimental data. The influences of key system parameters such as air cavity thickness, panel dimensions, and elevation angle and azimuth angle of incident sound on the sound insulation capability of the finite double-panel partition are systematically investigated. The results and conclusions of the present study should be referentially significant to others due to the similar physical nature of the vibroacoustic problem.

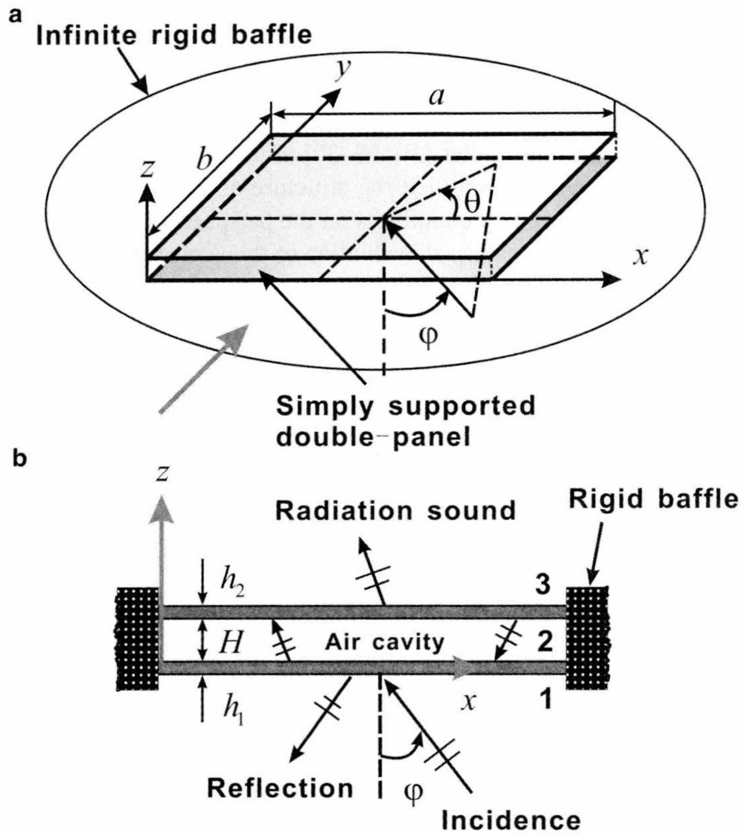

Fig. 1.1 Schematic of sound transmission through a baffled, rectangular, simply supported double-panel partition: (**a**) global view; (**b**) side view in the *arrow direction* in (**a**) (With permission from ASME)

1.1.2 Vibroacoustic Theoretical Modeling

The finite double-panel partition with enclosed air cavity is assumed to be rectangular, baffled, and simply supported along its boundaries, as shown in Fig. 1.1. The two panels are homogenous and isotropic and modeled as classical thin plate. The following geometrical dimensions are considered: the incident (bottom) panel and the radiating (top) panel have identical length a and width b, but may have different thicknesses h_1 and h_2 (Fig. 1.1b); the thickness of the air cavity is H (Fig. 1.1b). The whole configuration is mounted on an infinite acoustic rigid baffle which separates the space into two fields, i.e., sound incidence field ($z < 0$) and sound radiating field ($z > H$). A uniform plane sound wave varying harmonically in time is obliquely incident on the bottom panel, with incident elevation angle φ and azimuth angle θ (Fig. 1.1b). The vibration of the incident panel induced by the incident sound is transmitted through the enclosed air cavity to the radiating panel, which radiates sound into the acoustic medium. The vibroacoustic behaviors of the double-panel structure coupling with air cavity as well as sound transmission loss across the structure are to be solved analytically with the sound velocity potential method.

1.1.3 Mathematic Formulation and Solution

For an obliquely incident uniform plane sound wave varying harmonically in time, its acoustic velocity potential can be expressed as

$$\phi = Ie^{-j\left(k_x x + k_y y + k_z z - \omega t\right)} \tag{1.1}$$

where I is the amplitude; $j = \sqrt{-1}$; ω is the angular frequency; and k_x, k_y, and k_z are the wavenumber components in the x-, y-, and z-directions, respectively:

$$k_x = k_0 \sin\varphi \cos\theta, \quad k_y = k_0 \sin\varphi \sin\theta, \quad k_z = k_0 \cos\varphi \tag{1.2}$$

Here, $k_0 = \omega/c_0$ is the acoustic wavenumber in air, with c_0 denoting the sound speed in air.

Due to the excitation of the incident sound wave, the double-panel configuration with enclosed air cavity vibrates and radiates sound. The vibroacoustic behaviors of the structure are governed by

$$D_1\nabla^4 w_1 + m_1\frac{\partial^2 w_1}{\partial t^2} - j\omega\rho_0\left(\Phi_1 - \Phi_2\right) = 0 \tag{1.3}$$

$$D_2\nabla^4 w_2 + m_2\frac{\partial^2 w_2}{\partial t^2} - j\omega\rho_0\left(\Phi_2 - \Phi_3\right) = 0 \tag{1.4}$$

where ρ_0 is the air density and (w_1, w_2), (m_1, m_2) and (D_1, D_2) are the transverse displacements, surface densities, and flexural rigidities of the incident and radiating panels, located at $z = 0$ and $z = H$, respectively (Fig. 1.1). By introducing the loss factor of the panel material, the flexural rigidity of the panel D_i ($i = 1, 2$) can be written in terms of the complex Young's modulus $E_i(1 + j\eta_i)$ as

$$D_i = \frac{E_i h_i^3 \left(1 + j\eta_i\right)}{12\left(1 - \nu_i^2\right)} \tag{1.5}$$

The hard-walled cavity modal function $\phi_{mnl}^c = \cos(m\pi x/a)\cos(n\pi y/b)\cos(l\pi z/c)$ can only accurately model the sound field in a rigidly bounded cavity volume. It will therefore deviate somewhat from the precise results when the hard-walled cavity modal function is employed here to model the cavity bounded by two large flexural panels. In order to avoid this drawback, the sound velocity potential method is adopted, which is completely different from previous investigations based on cavity modal function. Let Φ_i $(i = 1, 2, 3)$ denote the velocity potentials of the three acoustic fields, i.e., sound incidence field, air cavity field, and structure radiating field (Fig. 1.1b), respectively. The velocity potential for the incident field can be defined as

$$\Phi_1(x, y, z; t) = Ie^{-j(k_x x + k_y y + k_z z - \omega t)} + \beta e^{-j(k_x x + k_y y - k_z z - \omega t)} \tag{1.6}$$

where the first and second terms represent separately the velocity potential of the incident and the reflected plus radiating sound waves and I and β are the amplitudes of the incident (i.e., positive-going) and the reflected plus radiating (i.e., negative-going) waves, respectively. Similarly, the velocity potential in the air cavity can be written as

$$\Phi_2(x, y, z; t) = \varepsilon e^{-j(k_x x + k_y y + k_z z - \omega t)} + \zeta e^{-j(k_x x + k_y y - k_z z - \omega t)} \tag{1.7}$$

where ε is the amplitude of positive-going wave and ζ is the amplitude of negative-going wave. In the radiating field, there exist no reflected waves; thus, the velocity potential is only for radiating waves:

$$\Phi_3(x, y, z; t) = \xi e^{-j(k_x x + k_y y + k_z z - \omega t)} \tag{1.8}$$

where ξ is the amplitude of radiating (i.e., positive-going) wave. The local acoustic velocities and sound pressure are related to the velocity potentials by

$$\widehat{\mathbf{u}}_i = -\nabla\Phi_i, \quad p_i = \rho_0 \frac{\partial \Phi_i}{\partial t} = j\omega\rho_0\Phi_i \quad (i = 1, 2, 3) \tag{1.9}$$

For simply supported boundary condition, the transverse displacement and the transverse force are constrained to be zero at the periphery of the panel. Given that the double-panel structure is rectangular, the boundary conditions can be expressed as

$$x = 0, a: \quad w_1 = w_2 = 0, \frac{\partial^2 w_1}{\partial x^2} = \frac{\partial^2 w_2}{\partial x^2} = 0 \tag{1.10}$$

$$y = 0, b: \quad w_1 = w_2 = 0, \frac{\partial^2 w_1}{\partial y^2} = \frac{\partial^2 w_2}{\partial y^2} = 0 \tag{1.11}$$

At the air-panel interface, the normal velocity should be continuous, yielding the following velocity compatibility equations:

$$z = 0: \quad -\frac{\partial \Phi_1}{\partial z} = j\omega w_1, \ -\frac{\partial \Phi_2}{\partial z} = j\omega w_1 \tag{1.12}$$

$$z = H: \quad -\frac{\partial \Phi_2}{\partial z} = j\omega w_2, \ -\frac{\partial \Phi_3}{\partial z} = j\omega w_2 \tag{1.13}$$