

Chapter 2

Vibroacoustics of Uniform Structures in Mean Flow

Abstract This chapter is organized as three parts: in the first part, an analytic approach is formulated to account for the effects of mean flow on sound transmission across a simply supported rectangular aeroelastic panel. The application of the convected wave equation and the displacement continuity condition at the fluid-panel interfaces ensures the exact handling of the complex aeroelastic coupling between panel vibration and fluid disturbances. To explore the mean flow effects on sound transmission, three different cases (i.e., mean flow on incident side only, on radiating side only, and on both sides) are separately considered in terms of refraction angular relations and sound transmission loss (STL) plots. Obtained results show that the influence of the incident side mean flow upon sound penetration is significantly different from that of the transmitted side mean flow. The contour plot of refraction angle versus incident angle for the case when the mean flow is on the transmitted side is just a reverse of that when the mean flow is on the incident side. The aerodynamic damping effects on the transmission of sound are well captured by plotting the STL as a function of frequency for varying Mach numbers. However, as the Mach number is increased, the coincidence dip frequency increases when the flow is on the incident side but remains unchanged when in the flow is on the radiating side. In the most general case when the fluids on both sides of the panel are convecting, the refraction angular relations are significantly different from those when the fluid on one side of the panel is moving and that on the other side is at rest.

In the second part, the transmission of external jet noise through a double-leaf skin plate of aircraft cabin fuselage in the presence of external mean flow is analytically studied. An aero-acoustic-elastic theoretical model is developed and applied to calculate the sound transmission loss (STL) versus frequency curves. Four different types of acoustic phenomenon (i.e., the mass-air-mass resonance, the standing-wave attenuation, the standing-wave resonance, and the coincidence resonance) for a flat double-leaf plate as well as the ring frequency resonance for a curved double-leaf plate are identified. Independent of the proposed theoretical

model, simple closed-form formulae for the natural frequencies associated with the above acoustic phenomena are derived using physical principles. Excellent agreement between the model predictions and the closed-form formulae is achieved. Systematic parametric investigation with the model demonstrates that the presence of the mean flow as well as the sound incidence angles affects substantially the sound transmission behavior of the double-leaf structure. The influences of panel curvature together with cabin internal pressure on jet-noise transmission are also significant and should be taken into account when designing aircraft cabin fuselages.

In the third part, a theoretical model is developed to investigate the influence of external mean flow on the transmission of sound through an infinite double-leaf panel filled with porous sound absorptive materials. The sound transmission process in the porous material is modeled using the method of equivalent fluid-structure coupling conditions that are accounted for to ensure displacement continuity at fluid-structure interfaces. Analytic solutions for the sound transmission loss of the whole structure are obtained. For validation, the model predictions are compared with existing experimental results. Numerical investigations with the model are subsequently performed to quantify how a set of systematic parameters affect the sound transmission loss. It is demonstrated that the porous material affects the STL curve in terms of both the absorption effect and the damping effect. Besides, the material loss factor and the thickness of the faceplates also have an influence on the coincidence dip of the STL curve. At frequencies below the coincidence frequency, the external mean flow increases the STL values due to the added damping effect of the mean flow while shifting the coincidence frequency upward because of the refraction effect of the mean flow. In addition, the coincidence frequency decreases with increasing azimuth angle between the sound incident direction and mean flow direction.

2.1 Finite Single-Leaf Aeroelastic Plate

2.1.1 Introduction

The vibroacoustic behavior of a flexible panel immersed in convected fluid flow is of practical importance for high-speed transportation vehicles (e.g., aircrafts, trains, and automobiles), and its prediction requires the combined knowledge of aeroelastics and structural acoustics [1–21]. For example, the interior noise of an aircraft stems mainly from the noise induced by external turbulent boundary layer (TBL) and the engine exhaust noise [6, 17, 22–25]. Therefore, the reduction of noise transmission into aircraft cabin interior has been a long-lasting issue for the airplane industry.

Earlier theoretical studies [26, 27] on the somewhat idealized but significant problem of reflection, transmission, and amplification of sound propagation into

a moving medium from a static fluid in the absence of a panel have helped to understand the physical mechanisms associated with the transmission of sound through panels immersed in convected fluids. For instance, by studying the total reflection arising from the refraction effect of mean flow and the sound amplification due to self-excited disturbance feeding, it has been found that there exist two critical angles for the occurrence of total reflection when the flow speed is more than twice that of sound [27].

Subsequently, with the focus placed upon the transmission of noise into aircraft cabin interior, Koval [1] studied theoretically the effects of airflow, panel curvature, and internal pressurization on the transmission loss of field incidence through an infinite single panel. It is found that both the mean flow and the panel curvature can enhance the sound transmission loss (*STL*), whereas the internal pressurization can lead to a slight decrease of the *STL*. Built upon Koval's work, Xin et al. [23] proposed recently a theoretical model to quantify the influence of external mean flow upon noise transmission through double-leaf aircraft fuselage into cabin interior. A series of closed-form analytic formulae were also derived from physical principles governing mass-air-mass resonance, standing-wave attenuation and resonance, and coincidence resonance, which compare favorably with model predictions.

Sgard et al. [15] developed a coupled FEM-BEM (finite element method-boundary element method) approach to study the vibroacoustic behavior of planar structures in the presence of mean flow, with formulations explicitly accounting for the effects of mean flow on the *STL* in terms of added mass, panel stiffness, and radiating damping. With due considerations of the effects of structural nonlinearities induced by in-plane forces and shearing forces due to plate bending, Wu and Maestrello [16] studied theoretically the vibroacoustic response of a finite plate supported on a rigid baffle and subjected to TBL excitations. By assuming that the aircraft fuselage is locally flat, Howe and Shah [22] solved analytically the problem of noise generation by subsonic, high-speed turbulent flow, and the reverse flow reciprocal theorem was employed to determine Green's functions for treating the scattering of boundary layer wall pressures at panel edges. Graham [24, 25, 28] proposed a theoretical model as a compromise between the requirement of simplicity and retaining the physical features of the aircraft interior noise problem and used asymptotic expressions to study the effects of mean flow on the radiation efficiency of a rectangular plate.

Recently, Frampton and Clark [2, 4, 6–8, 10, 11, 17, 19] studied systematically the influence of convected fluid loading coupling on the vibroacoustic behavior of finite-sized panels. A theoretical model combining the aerodynamic loading of panels and linearized potential flow aerodynamics was firstly developed in Ref. [4] by adopting singular value decomposition [2], and detailed analyses with the TBL-induced noise disturbance taken into account were further proposed in Refs. [6–8, 10, 17]. In particular, the effect of in-plane forces on sound radiation of convected fluid-loaded plates was theoretically analyzed [19], and it is found that the state of stress in the plate exerts a significant effect on the radiation efficiency of the plate.

There also exist typical investigations regarding that active control of vibration and noise transmission. For instance, Clark and Frampton [11] designed a static, constant-gain, output-feedback control compensators to increase the *STL* across a panel subjected to mean flow, and Maury et al. [12, 14, 29] investigated the active control of flow-induced noise transmission across single and double panels.

Although there exist numerous studies on the coupling of panel vibration with fluid flow, such as aeroelastic panel flutter and sound radiation due to flow-induced panel vibration, a few issues regarding its underlying physical mechanisms remain unclear, such as the mechanism underlying sound power flux penetration and the effect of aeroelastic coupling feedback. Furthermore, while a few studies [1, 7, 17, 23] addressed specifically the practically significant problem of sound transmission across a finite aeroelastic panel immersed in convected fluid on one side (either the incident side or the transmitted side), the more general case when the aeroelastic panel is loaded by convected fluids on both sides has not been solved. The solution of this general case not only is complementary to the existing research so that an overlook of this issue can be acquired but also has practical implications particularly in soundproofing machines that require effective cooling by convected flow. For typical instance, the nozzle of the jet engine of an aircraft works in high-temperature environment. It is well known that increasing the temperature of the jet can increase the efficiency and speed of aircraft, but few materials are able to work in such high-temperature environment particularly in hypersonic vehicles. In such cases, active cooling with convective fluid flow in between the sandwich structure (with open-celled cellular core) of the jet nozzle has emerged as a promising approach. Consequently, how the convected flow on both sides of the panel affects the sound transmission should be of concern. This chapter is mainly aimed to remedy this deficiency and to provide more details in our understanding of the interdisciplinary topic of acoustics and aeroelastics/dynamics. After presenting the modeling part in Sect. 2.1.2, in Sects. 2.1.3–2.1.5 of numerical result discussion, the refraction angular relations between the incident and transmitted sound waves are presented in terms of contour plots for three different cases (i.e., panel immersed in convected fluid on the incident side only, on the transmitted side only, and on both sides). Detailed analyses of sound power flux penetration through the convected fluid-loaded panel are also performed in terms of *STL* versus frequency characteristic curves for the three cases.

2.1.2 Modeling of Aeroelastic Coupled System

Consider a finite, rectangular aeroelastic panel simply supported in an infinite acoustic rigid baffle (as shown in Fig. 2.1a, b). The panel has length a along x -direction, width b along y -direction, and thickness h along z -direction, with $h \ll a$ and $h \ll b$ assumed. The panel divides the spatial region into two regimes, i.e., the incident field ($z < 0$) and the transmitted field ($z > 0$) which, for convenience, are numbered below by 1 and 2, respectively (Fig. 2.1b). An oblique plane sound

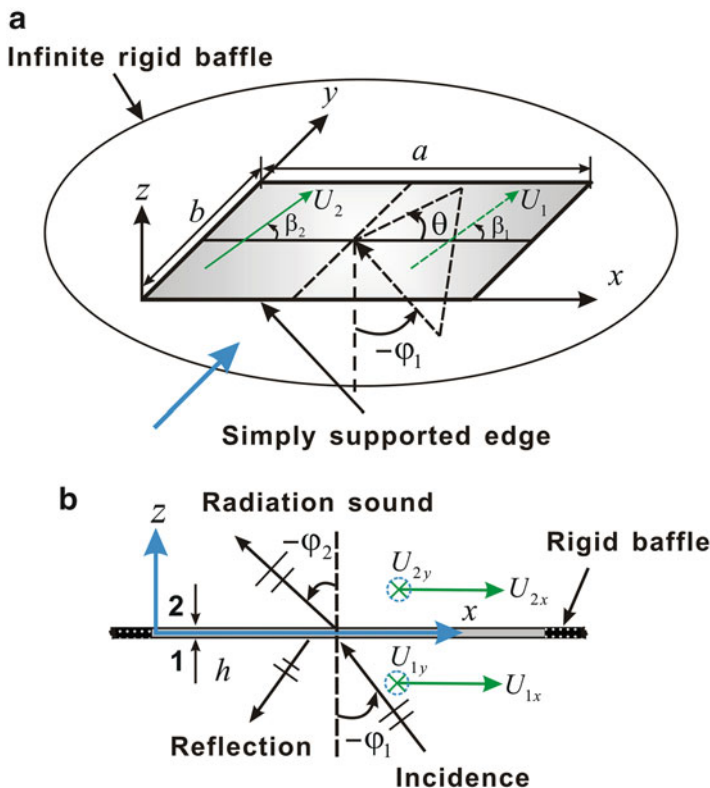


Fig. 2.1 Schematic of sound transmission through a simply supported aeroelastic panel immersed in convected fluid flow on both sides: (a) overall view; (b) view from the direction of arrow in (a) (With permission from Acoustical Society of America)

wave varying harmonically in time is incident on the bottom side of the panel, with elevation angle φ_1 and azimuth angle θ (Fig. 2.1a).

The panel is immersed in convected fluid flow on both sides which are parallel to the panel but may flow along different directions (Fig. 2.1) and have different physical properties (including density and sound speed). Without loss of generality, let the flow on the incident side ($z < 0$) have azimuth angle β_1 and speed U_1 and that on the transmitted side ($z > 0$) have azimuth angle β_2 and speed U_2 ; see Fig. 2.1b.

The panel vibration induced by the incident sound and fluid flow together also creates a pressure disturbance in the surrounding fluid media, including the reflected pressure wave $p_{\text{reflected}}^i$ and the radiating pressure wave $p_{\text{radiating}}^i$ in the incident field and the radiating pressure wave $p_{\text{radiating}}^t$ in the transmitted field. The pressure changes caused by this disturbance will in turn significantly influence the panel vibration, resulting in the so-called aeroelastic coupling [4, 6, 10, 11, 17]. In the present study, it is assumed that the panel deforms out of plane (in the z -direction), positive upward.

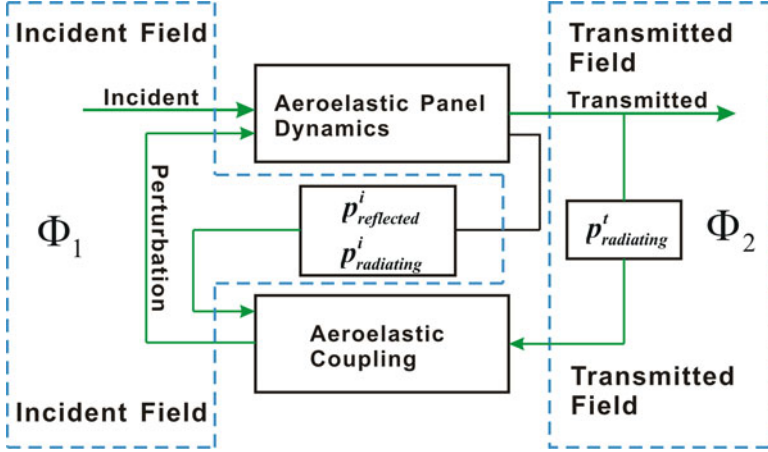


Fig. 2.2 Block diagram of sound transmission across a finite aeroelastic panel immersed in convected fluids on both sides. The *dashed frame* on the *left* indicates the incident field, including the incident sound pressure wave as well as the reflected and radiated pressure waves. The *dashed frame* on the *right* stands for the transmitted field, containing the resultant radiating pressure wave (With permission from Acoustical Society of America)

As an illustration of the sound transmission process, a block diagram (Fig. 2.2) is drawn to highlight the interaction of panel motion and fluid dynamics. Notice first that the left dashed frame represents the incident field, including the incident sound pressure wave as well as the reflected and radiating pressure waves stemming from panel vibration, and this field is characterized by the perturbation acoustic velocity potential Φ_1 . The right dashed frame for the transmitted field contains the resultant transmitted pressure wave and is characterized by the perturbation acoustic velocity potential Φ_2 . The connections between the two central solid frames in Fig. 2.2 signify essentially the aeroelastic coupling between the panel vibration and the surrounding convected fluids. Subsequent theoretical modeling will consider this aeroelastic coupling by employing the convected wave equation, fluid momentum equation, and displacement continuity condition between the proximal fluid particle and the panel particle.

The dynamic displacement of an aeroelastic panel immersed in convected fluids on both sides and subjected to a uniform, plane sound wave varying harmonically can be described by [3, 5]

$$D\nabla^4 w(x, y; t) + m \frac{\partial^2 w(x, y; t)}{\partial t^2} - j\omega [\rho_1 \Phi_1(x, y, 0; t) - \rho_2 \Phi_2(x, y, 0; t)] = 0 \quad (2.1)$$

where D and m are the flexural rigidity and surface density of the panel and ω is the angular frequency of the incident sound. With c_t denoting the speed of the

trace wave in the aeroelastic panel, the wavenumber of the trace wave is $k_t = \omega/c_t$. Hence, the displacement of the aeroelastic panel induced by the incident sound and the convected fluid flow can be expressed as:

$$w(x, y; t) = w_0 e^{-j[(k_t \cos \theta)x + (k_t \sin \theta)y - \omega t]} \quad (2.2)$$

With the assumption of idealized fluid (i.e., inviscid, irrotational, and incompressible), the convected fluid field can be regarded as full potential flow. Let Φ_i ($i = 1, 2$) denote the velocity potentials for the acoustic fields in the proximity of the aeroelastic panel, corresponding to the sound incidence and the transmitted field (fields 1 and 2 in Fig. 2.1b), respectively:

$$\Phi_1(x, y, z; t) = I e^{-j(k_{1x}x + k_{1y}y + k_{1z}z - \omega t)} + \beta e^{-j(k_{1x}x + k_{1y}y - k_{1z}z - \omega t)} \quad (2.3)$$

$$\Phi_2(x, y, z; t) = \varepsilon e^{-j(k_{2x}x + k_{2y}y + k_{2z}z - \omega t)} \quad (2.4)$$

where I and β are separately the amplitude of the positive-going wave (incident sound) and the negative-going wave (including the reflected wave and the radiating wave) in the incident acoustic field and ε is the amplitude of the positive-going wave in the transmitted field. The sound wavenumber components in (3) and (4) depend upon the incident sound elevation angle φ_1 , the azimuth angle θ and the refraction angle φ_2 according to

$$k_{1x} = k_1 \sin \varphi_1 \cos \theta, \quad k_{1y} = k_1 \sin \varphi_1 \sin \theta, \quad k_{1z} = k_1 \cos \varphi_1 \quad (2.5)$$

$$k_{2x} = k_2 \sin \varphi_2 \cos \theta, \quad k_{2y} = k_2 \sin \varphi_2 \sin \theta, \quad k_{2z} = k_2 \cos \varphi_2 \quad (2.6)$$

For the convenience of describing the modal response of the aeroelastic panel, its dynamic displacement can be rewritten by using the in vacuo orthogonal panel eigenfunctions and the generalized coordinates as

$$w(x, y; t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \varphi_{mn}(x, y) q_{mn}(t) \quad (2.7)$$

where the modal functions of a simply supported rectangular panel and the generalized coordinates are taken as

$$\varphi_{mn}(x, y) = \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b} \quad (2.8)$$

$$q_{mn}(t) = \alpha_{mn} e^{j\omega t} \quad (2.9)$$

Similarly, the acoustic velocity potentials of Eqs. (2.3) and (2.4) are expressed as

$$\Phi_1(x, y, z; t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} I_{mn} \varphi_{mn} e^{-j(k_{1z}z - \omega t)} + \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \beta_{mn} \varphi_{mn} e^{-j(-k_{1z}z - \omega t)} \quad (2.10)$$

$$\Phi_2(x, y, z; t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \varepsilon_{mn} \varphi_{mn} e^{-j(k_{2z}z - \omega t)} \quad (2.11)$$

The conversion relation between the general forms of Eqs. (2.2), (2.3), and (2.4) and the generalized forms (with modal functions) of Eqs. (2.7), (2.10), and (2.11) can be obtained by utilizing the sine Fourier transform as

$$\tilde{\lambda}_{mn} = \frac{4}{ab} \int_0^b \int_0^a \lambda e^{-j(k_{ix}x + k_{iy}y)} \sin \frac{m\pi x}{a} \sin \frac{n\pi y}{b} dx dy \quad (i = 1, 2) \quad (2.12)$$

where $\tilde{\lambda}$ refers to any of the symbols I , β , ε , and a (with w_0). Note that the expressions in terms of either traveling wave or panel modal functions are completely equivalent in physical nature when they are both subjected to the same boundary conditions.

The acoustic velocity potentials of (2.3) and (2.4) for an inviscid, irrotational, and incompressible fluid moving in a plane parallel to the aeroelastic panel should satisfy the convected wave equation [1, 30–33], given by

$$\frac{D^2 \Phi_1}{Dt^2} = \left(\frac{\partial}{\partial t} + U_1 \cdot \nabla \right)^2 \Phi_1 = c_1^2 \nabla^2 \Phi_1 \quad (2.13)$$

$$\frac{D^2 \Phi_2}{Dt^2} = \left(\frac{\partial}{\partial t} + U_2 \cdot \nabla \right)^2 \Phi_2 = c_2^2 \nabla^2 \Phi_2 \quad (2.14)$$

which, with fluid velocities expressed as $U_i = U_{ix} \hat{e}_x + U_{iy} \hat{e}_y$ ($i = 1, 2$), can be expanded as follows:

$$\left[\frac{\partial^2}{\partial t^2} + U_{1x}^2 \frac{\partial^2}{\partial x^2} + U_{1y}^2 \frac{\partial^2}{\partial y^2} + 2U_{1x} \frac{\partial^2}{\partial x \partial t} + 2U_{1y} \frac{\partial^2}{\partial y \partial t} + 2U_{1x} U_{1y} \frac{\partial^2}{\partial x \partial y} \right] \Phi_1 = c_1^2 \nabla^2 \Phi_1 \quad (2.15)$$

$$\left[\frac{\partial^2}{\partial t^2} + U_{2x}^2 \frac{\partial^2}{\partial x^2} + U_{2y}^2 \frac{\partial^2}{\partial y^2} + 2U_{2x} \frac{\partial^2}{\partial x \partial t} + 2U_{2y} \frac{\partial^2}{\partial y \partial t} + 2U_{2x} U_{2y} \frac{\partial^2}{\partial x \partial y} \right] \Phi_2 = c_2^2 \nabla^2 \Phi_2 \quad (2.16)$$

Substitution of (2.3) and (2.4) into (2.15) and (2.16) leads to

$$k_1 = \frac{k_1^*}{1 + M_{1x} \sin \varphi_1 \cos \theta + M_{1y} \sin \varphi_1 \sin \theta} \quad (2.17)$$

$$k_2 = \frac{k_2^*}{1 + M_{2x} \sin \varphi_2 \cos \theta + M_{2y} \sin \varphi_2 \sin \theta} \quad (2.18)$$

where $k_1^* = \omega/c_1$ and $k_2^* = \omega/c_2$ are the wavenumbers in the absence of airflow, while $M_1 = U_1/c_1$ and $M_2 = U_2/c_2$ are the Mach number of mean flow in the incident field and that in the transmitted field, respectively.

The sound waves traveling in fluid media proximal to the panel and the bending wave propagating in the flexural aeroelastic panel should be in coherence with each other in their wavelengths [1], that is,

$$k_{1x} = k_t \cos \theta = k_{2x}, \quad k_{1y} = k_t \sin \theta = k_{2y} \quad (2.19)$$

so that

$$\varphi_2 = \arcsin \left(\frac{c_2 \sin \varphi_1}{c_1 + [(c_1 M_{1x} - c_2 M_{2x}) \cos \theta + (c_1 M_{1y} - c_2 M_{2y}) \sin \theta] \sin \varphi_1} \right) \quad (2.20)$$

Note that the corresponding transmitted waves become evanescent waves, when

$$\left| \frac{c_2 \sin \varphi_1}{c_1 + [(c_1 M_{1x} - c_2 M_{2x}) \cos \theta + (c_1 M_{1y} - c_2 M_{2y}) \sin \theta] \sin \varphi_1} \right| > 1. \quad (2.21)$$

In such cases, total reflection of the incident sound appears, and the value of φ_2 should be taken as a complex one with real and imaginary parts (either both positive or both negative), such that the value of k_{2z} takes the form of $-j\hbar$ (where $\hbar$ is a positive real number).

2.1.2.1 Displacement Continuity Condition at Fluid-Panel Interfaces

Let ξ_1 and ξ_2 represent the acoustic particle displacement in the incident and transmitted fluid medium, respectively. The fluid particle displacement and the acoustic pressure are related by the fluid momentum equation for inviscid, irrotational, and incompressible fluid as [22]

$$\frac{D^2 \xi_1}{Dt^2} = \left(\frac{\partial}{\partial t} + \mathbf{U}_1 \cdot \nabla \right)^2 \xi_1 = -\frac{1}{\rho_1} \frac{\partial p_1}{\partial z} \Big|_{z=0} \quad (2.22)$$

$$\frac{D^2 \xi_2}{Dt^2} = \left(\frac{\partial}{\partial t} + \mathbf{U}_2 \cdot \nabla \right)^2 \xi_2 = -\frac{1}{\rho_2} \frac{\partial p_2}{\partial z} \Big|_{z=0} \quad (2.23)$$

where the acoustic pressure can be expressed by the acoustic velocity potentials through Bernoulli's equation as [10, 30]

$$p_i = \rho_i \left[\frac{\partial \Phi_i}{\partial t} + U_{ix} \frac{\partial \Phi_i}{\partial x} + U_{iy} \frac{\partial \Phi_i}{\partial y} \right] \quad (i = 1, 2) \quad (2.24)$$

The displacements of the fluid particle adjacent to the panel can be expressed as

$$\xi_1 = \xi_{10} e^{-j(k_{1x}x + k_{1y}y - \omega t)} \quad (2.25)$$

$$\xi_2 = \xi_{20} e^{-j(k_{2x}x + k_{2y}y - \omega t)} \quad (2.26)$$

Substituting (2.24), (2.25), and (2.26) into (2.22) and (2.23) and applying the acoustic velocity potentials of (2.10) and (2.11), one can obtain

$$\xi_{10} = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} (I_{mn} \varphi_{mn} - \beta_{mn} \varphi_{mn}) \frac{\omega k_{1z}}{(\omega - U_{1x} k_{1x} - U_{1y} k_{1y})^2} e^{j(k_{1x}x + k_{1y}y)} \quad (2.27)$$

$$\xi_{20} = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \varepsilon_{mn} \varphi_{mn} \frac{\omega k_{2z}}{(\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^2} e^{j(k_{2x}x + k_{2y}y)} \quad (2.28)$$

The factual case that the aeroelastic panel immersed in a convected fluid medium requires that the displacements of the fluid particles adjacent to the panel should be the same as those of the attached panel particles. Accordingly, the displacement continuity condition can be written as [1, 31]

$$\xi_{10} = w_0, \quad \xi_{20} = w_0 \quad (2.29)$$

Together with (2.2) and (2.27) and (2.28), meanwhile utilizing the following relation between coefficients α_{mn} and w_0 ,

$$\alpha_{mn} = \frac{4mn\pi^2 w_0 \left[1 - (-1)^m e^{-jk_x a} - (-1)^n e^{-jk_y b} + (-1)^{m+n} e^{-j(k_x a + k_y b)} \right]}{(k_x^2 a^2 - m^2 \pi^2) (k_y^2 b^2 - n^2 \pi^2)} \quad (2.30)$$

One can express the coefficients in the acoustic velocity potentials by the panel displacement coefficients as

$$\beta_{mn} = I_{mn} - \frac{(\omega - U_{1x} k_{1x} - U_{2x} k_{2x})^2}{\omega k_{1z}} \alpha_{mn} \quad (2.31)$$

$$\varepsilon_{mn} = \frac{(\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^2}{\omega k_{2z}} \alpha_{mn} \quad (2.32)$$

Substituting (2.7) into (2.1) and applying the orthogonality of the modal functions, one gets

$$\ddot{q}_{mn}(t) + \omega_{mn}^2 q_{mn}(t) - \frac{j\omega}{m} \left[\rho_1 I_{mn} e^{-j(k_{1z}z - \omega t)} + \rho_1 \beta_{mn} e^{-j(-k_{1z}z - \omega t)} - \rho_2 \varepsilon_{mn} e^{-j(k_{2z}z - \omega t)} \right] = 0 \quad (2.33)$$

where the natural frequencies of the aeroelastic panel are determined by the panel properties as

$$\omega_{mn}^2 = \frac{D \iint_A \nabla^4 \varphi_{mn} \cdot \varphi_{mn} dA}{m \iint_A \varphi_{mn} \cdot \varphi_{mn} dA} \quad (2.34)$$

Equation (2.33) can be readily converted to the following form:

$$\alpha_{mn} = \frac{2j\omega\rho_1 I_{mn}}{m} \cdot \left[\omega_{mn}^2 - \omega^2 + \frac{j\omega\rho_1}{m} \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{\omega k_{1z}} + \frac{j\omega\rho_2}{m} \frac{(\omega - U_{2x}k_{2x} - U_{2y}k_{2y})^2}{\omega k_{2z}} \right]^{-1} \quad (2.35)$$

Once the panel displacement coefficients α_{mn} are known, the acoustic velocity potentials will be known, given by

$$\Phi_1(x, y, 0) = 2Ie^{-j(k_{1x}x + k_{1y}y)} - \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{\omega k_{1z}} \alpha_{mn} \varphi_{mn}(x, y) \quad (2.36)$$

$$\Phi_2(x, y, 0) = \frac{(\omega - U_{2x}k_{2x} - U_{2y}k_{2y})^2}{\omega k_{2z}} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \alpha_{mn} \varphi_{mn}(x, y) \quad (2.37)$$

2.1.2.2 Definition of Sound Transmission Loss

Different from previous researches [3, 5, 34, 35] that consider only the incident and reflected sound pressures, the proposed theoretical formulations are capable of accurately modeling all the pressure components in the incident field, and hence all the pressure components are taken into account in the present STL

calculations. Actually, this is closer to the physical nature of the factual experimental measurements [36]; thus, the power of incident sound is defined as

$$\Pi_1 = \frac{1}{2} \text{Re} \iint_A p_1 \cdot v_1^* dA \quad (2.38)$$

where the asterisk symbol denotes complex conjugate, $v_1 = p_1/(\rho_1 c_1)$ is the local acoustic velocity, and

$$\begin{aligned} p_1 &= j\rho_1 (\omega - U_{1x}k_{1x} - U_{1y}k_{1y}) \Phi_1(x, y, 0) \\ &= j\rho_1 (\omega - U_{1x}k_{1x} - U_{1y}k_{1y}) \\ &\quad \times \left[2I e^{-j(k_{1x}x + k_{1y}y)} - \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{\omega k_{1z}} \alpha_{mn} \varphi_{mn}(x, y) \right] \end{aligned} \quad (2.39)$$

is the sound pressure in the incident field. Substituting p_1 and v_1 into (2.38) yields

$$\begin{aligned} \Pi_1 &= \frac{\rho_1 (\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{2c_1} \iint_A |\Phi_1(x, y, 0)|^2 dA \\ &= \frac{\rho_1 (\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{2c_1} \\ &\quad \times \iint_A \left| 2I e^{-j(k_{1x}x + k_{1y}y)} - \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{\omega k_{1z}} \alpha_{mn} \varphi_{mn}(x, y) \right|^2 dA \\ &= \frac{\rho_1 (\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{2c_1} \\ &\quad \times \left| \begin{aligned} &4I^2 \iint_A e^{-2j(k_{1x}x + k_{1y}y)} dA - 4I \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^2}{\omega k_{1z}} \\ &\times \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \alpha_{mn} \iint_A e^{-j(k_{1x}x + k_{1y}y)} \varphi_{mn}(x, y) dA \\ &+ \frac{(\omega - U_{1x}k_{1x} - U_{1y}k_{1y})^4}{\omega^2 k_{1z}^2} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \sum_{k=1}^{\infty} \sum_{l=1}^{\infty} \alpha_{mn} \alpha_{kl} \iint_A \varphi_{mn}(x, y) \varphi_{kl}(x, y) dA \end{aligned} \right| \end{aligned} \quad (2.40)$$

In a similar manner, the radiated sound power can be defined as

$$\Pi_2 = \frac{1}{2} \text{Re} \iint_A p_2 \cdot v_2^* dA \quad (2.41)$$

where $v_2 = p_2/(\rho_2 c_2)$ is the local acoustic velocity and

$$\begin{aligned} p_2 &= j\rho_2 (\omega - U_{2x}k_{2x} - U_{2y}k_{2y}) \Phi_2(x, y, 0) \\ &= j\rho_2 \frac{(\omega - U_{2x}k_{2x} - U_{2y}k_{2y})^3}{\omega k_{2z}} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \alpha_{mn} \varphi_{mn}(x, y) \end{aligned} \quad (2.42)$$

is the sound pressure in the transmitted field. Combination of Eqs. (2.41) and (2.42) and the expression of v_2 results in

$$\begin{aligned}
 \Pi_2 &= \frac{\rho_2 \omega^2}{2c_2} \iint_A |\Phi_2(x, y, 0)|^2 dA = \frac{\rho_2 (\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^2}{2c_2} \\
 &\quad \times \iint_A \left| \frac{(\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^2}{\omega k_{2z}} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \alpha_{mn} \varphi_{mn}(x, y) \right|^2 dA \\
 &= \frac{\rho_2 (\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^2}{2c_2} \\
 &\quad \times \left| \frac{(\omega - U_{2x} k_{2x} - U_{2y} k_{2y})^4}{\omega^2 k_{2z}^2} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \sum_{k=1}^{\infty} \sum_{l=1}^{\infty} \alpha_{mn} \alpha_{kl} \iint_A \varphi_{mn}(x, y) \varphi_{kl}(x, y) dA \right|
 \end{aligned} \tag{2.43}$$

The power transmission coefficient can be obtained as

$$\tau(\varphi_1, \theta, U_1, U_2) = \frac{\Pi_2}{\Pi_1} \tag{2.44}$$

which is dependent upon the sound incident angles (φ_1 and θ) and the mean flow velocities U_1 and U_2 . Finally, the sound transmission loss across the panel, defined as the inverse of the power transmission coefficient in decibel scale, is given by

$$STL = 10 \log_{10} \left(\frac{1}{\tau} \right) \tag{2.45}$$

2.1.3 Effects of Mean Flow in Incident Field

To gain fundamental insight into the effects of mean flow on the incident side, the fluid medium on the transmitted side is taken to be at rest. The corresponding contour plot of refraction angle φ_2 versus incident angle φ_1 as well as the *STL* versus frequency plots for a series of different Mach numbers (M_1 , incident side) is presented below.

The refraction angle φ_2 is plotted in Fig. 2.3 as a function of incident angle φ_1 for selected Mach numbers. When $M_1 = 0$, the refraction angle is equal to the incident angle φ_1 (in other words, no refraction occurs), which is attributed to the fact that the fluid media considered in the present study are identical (i.e., air) on both sides of the panel. Apart from the case of $M_1 = 0$, the refraction angle is related to the incident angle in a nonlinear fashion due to the refraction effect of the convected flow stream [1, 37]. Another noteworthy point is that when $M_1 > 2$, the φ_2 - φ_1 curves are

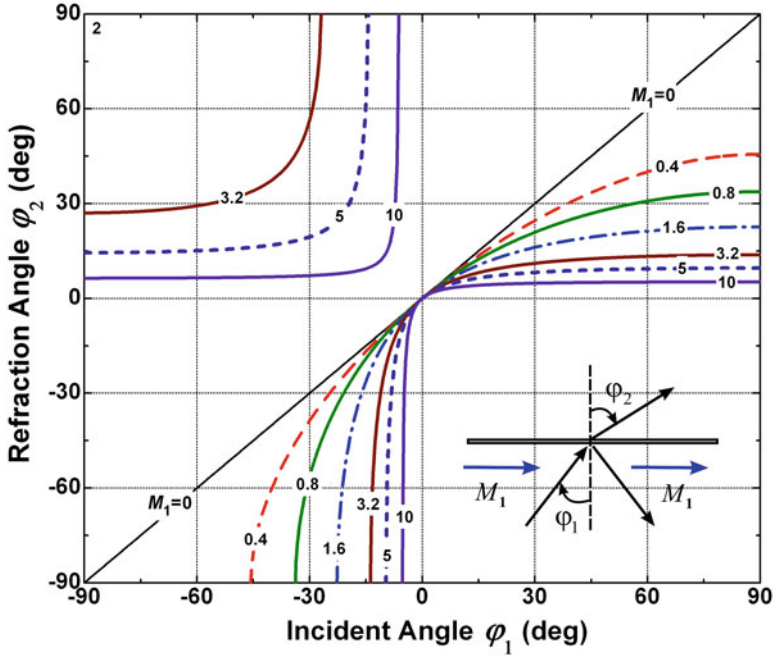


Fig. 2.3 Refraction angle φ_2 plotted as a function of incident angle φ_1 for selected values of Mach number M_1 associated with fluid medium on incident side moving parallel to panel; fluid medium on transmitted side is at rest (i.e., $M_2 = 0$) (With permission from Acoustical Society of America)

divided into two branches: the positive refraction branch and the negative refraction branch, the gap between them along the φ_1 -axis corresponding to the total reflection. In the bigger branch (positive refraction), the refraction angle has the same sign as the incident angle, whereas the reverse holds in the smaller branch (negative refraction).

It is seen from Fig. 2.3 that total reflection occurs only when the sound is incident along the upstream direction of the (incident side) mean flow. As previously mentioned by Ribner [26] and Xin et al. [23], when total reflection occurs, while a disturbance on the incident side does penetrate through the panel into the transmitted side, its amplitude attenuates exponentially over a distance on the order of $\hbar$. Furthermore, the transmitted sound carries no energy due to the fact that the pressure and the particle velocity are fully out of phase (i.e., 90°).

To demonstrate the effects of mean flow, the *STL* of an infinite aeroelastic panel is plotted in Fig. 2.4 as a function of incident frequency with incident angle $\varphi_1 = 60^\circ$ and $\theta = 0^\circ$ for different Mach numbers ($M_1 = 0, 0.4$, and 0.8). Similarly, Fig. 2.5 presents the *STL* versus frequency plots of a finite aeroelastic panel for $M_1 = 0$ and 0.8 , with their infinite counterparts included as reference. It can be observed from these results that the dip related to coincidence resonance shifts noticeably to higher frequencies as the Mach number (i.e., M_1) is increased. This shift is caused mainly

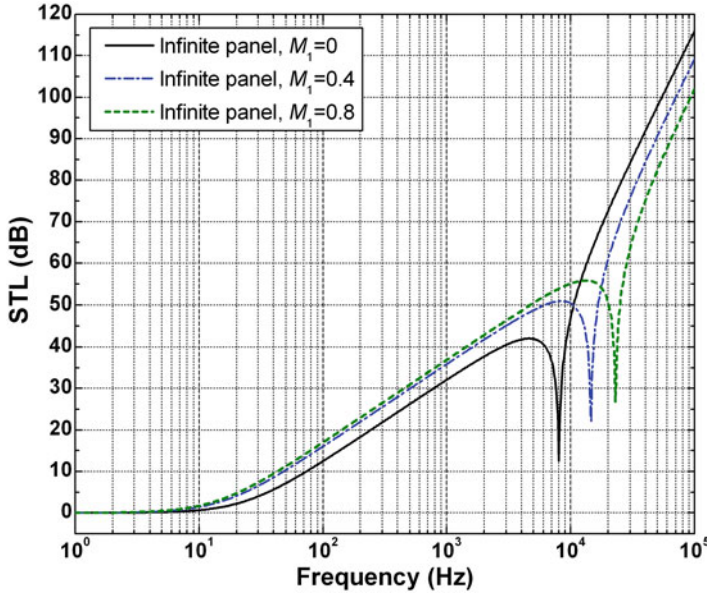


Fig. 2.4 STL plotted as a function of frequency of an infinite panel immersed in convected fluid for selected Mach numbers (*solid line* $M_1 = 0$, *dash-dot line* $M_1 = 0.4$, and *short dash line* $M_1 = 0.8$), when sound is incident with elevation angle $\varphi_1 = 60^\circ$ and azimuth angle $\theta = 0^\circ$. The mean flow on the incident side is completely aligned with the x -direction (Fig. 2.1), while the fluid on the transmitted side is at rest (i.e., $M_2 = 0$) (With permission from Acoustical Society of America)

by the refraction effect of the mean flow [1, 26, 37], i.e., the factual incident angle $\tilde{\varphi}_2$ in the proximal field of the panel has been significantly altered from its original angle of φ_1 by the moving stream on the incident side.

Note also that the factual incident angle $\tilde{\varphi}_2$ in the proximal field of the panel actually is equal to the refraction angle φ_2 when the fluid medium (i.e., air) on the incident side is the same as that on the transmitted side. This effect can be explained by considering the physical origin of the coincidence resonance phenomenon, i.e., the resonance effect between the incident sound wave and the bending wave in the panel when their phases are matched. The coincidence resonance frequency can be predicted as [23]

$$f_c = \frac{c_1^2}{2\pi h \sin \tilde{\varphi}_2} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \quad (2.46)$$

In the case when the original incident angle $\varphi_1 = 60^\circ$, one sees from Fig. 2.3 that the factual incident angle $\tilde{\varphi}_2$ ($= \varphi_2$) decreases as M_1 is increased; as a result, the coincidence resonance frequency should also increase, as predicted by Eq. (2.46).

Another observation from Fig. 2.4 is that the coincidence resonance dip divides the STL plot into two regimes within which the mean flow has opposite effects on

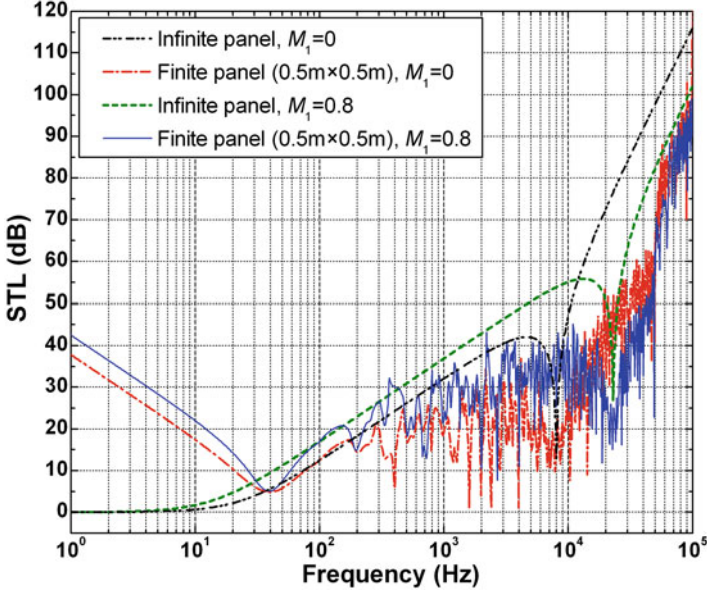


Fig. 2.5 STL plotted as a function of frequency of a finite panel immersed in convected fluid for selected Mach numbers ($M_1 = 0$ and $M_1 = 0.8$), when sound is incident with elevation angle $\varphi_1 = 60^\circ$ and azimuth angle $\theta = 0^\circ$. The mean flow on the incident side is completely aligned with the x -direction (Fig. 2.1), while the fluid on the transmitted side is at rest (i.e., $M_2 = 0$). The corresponding STL versus frequency plots of an infinite panel are included as reference (With permission from Acoustical Society of America)

the *STL*. At frequencies below the coincidence dip, a higher velocity of the mean flow enhances more significantly the *STL* value of the aeroelastic panel; in contrast, at frequencies beyond the coincidence dip, the *STL* value decreases with increasing mean low velocity.

In comparison with an infinite panel, the *STL* versus frequency plot of a finite panel immersed in convected fluid medium should reflect its modal behavior due to the boundary condition, which is affirmed by the appearance of dense peaks and dips in Fig. 2.5. With reference to the counterpart of the infinite panel, the *STL* plot of the finite panel exhibits similar tendencies beyond the (1, 1) modal frequency at approximately 40Hz, with the former setting an asymptotic maximum of the latter. However, at frequencies below the (1, 1) modal frequency, the tendency is remarkably different between the two panels as a result of the boundary effects [3, 5].

The dip associated with the coincidence resonance can also be distinguished from the *STL* plot of the finite panel, which agrees well with the dip in the *STL* plot of the infinite panel (Fig. 2.5). Moreover, similar to the infinite panel, the presence of mean flow on the incident side leads to enhanced *STL* below the coincidence resonance frequency and diminished *STL* beyond it. The present result that the *STL* increases significantly with increasing Mach number over a broad frequency range (approximately 40–10,000 Hz) is consistent with previously published results [6, 7,

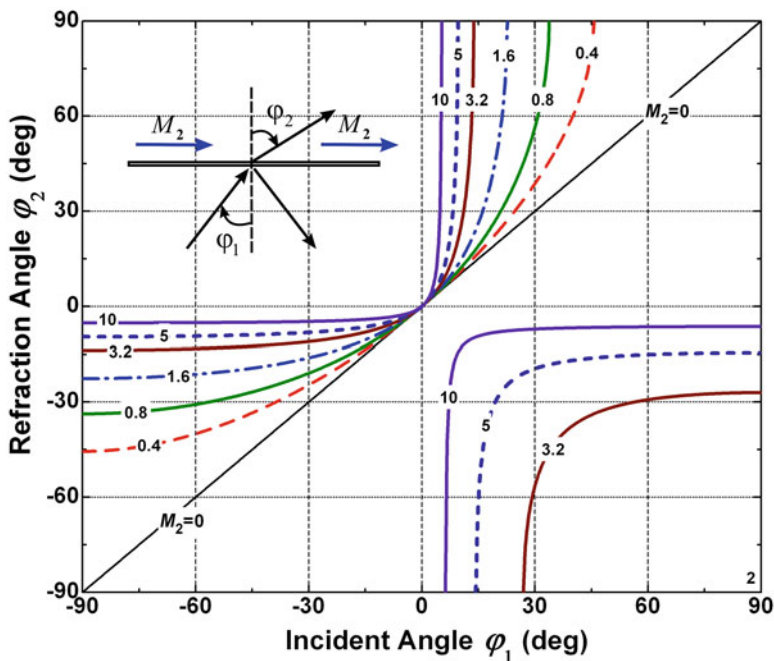


Fig. 2.6 Refraction angle φ_2 plotted as a function of incident angle φ_1 for selected Mach numbers (M_2) of moving fluid (aligned with x -direction) on transmitted side; fluid medium on incident side is at rest (i.e., $M_1 = 0$) (With permission from Acoustical Society of America)

[17, 18]. The influence of mean flow on the transmission of sound across a panel can be attributed to the so-called aerodynamic damping effect [38]. Here, the role played by aerodynamic damping is twofold: as the Mach number is increased, it not only increases the sound power radiated to the convected fluid but also modifies the panel stiffness (via convected fluid loading), leading to an overall alteration of the panel impedance [7, 17].

2.1.4 Effects of Mean Flow in Transmitted Field

To evaluate the effects of mean flow on the transmitted side of the aeroelastic panel, the fluid on the incident side is set at rest (i.e., $M_1 = 0$), while that on the transmitted side travels along the x -direction with Mach number M_2 (Fig. 2.1). In such cases, the predicted dependence of the refraction angle φ_2 upon M_2 and the incident angle φ_1 is presented in Fig. 2.6 in the form of contour plots. The presence of mean flow on the transmitted side has an influence completely opposite to that shown in Fig. 2.3 when $M_1 \neq 0$ but $M_2 = 0$, as if the two angles φ_1 and φ_2 are reversed. Similar to Fig. 2.3, the contour plots of Fig. 2.6 are decomposed into two branches (i.e., the

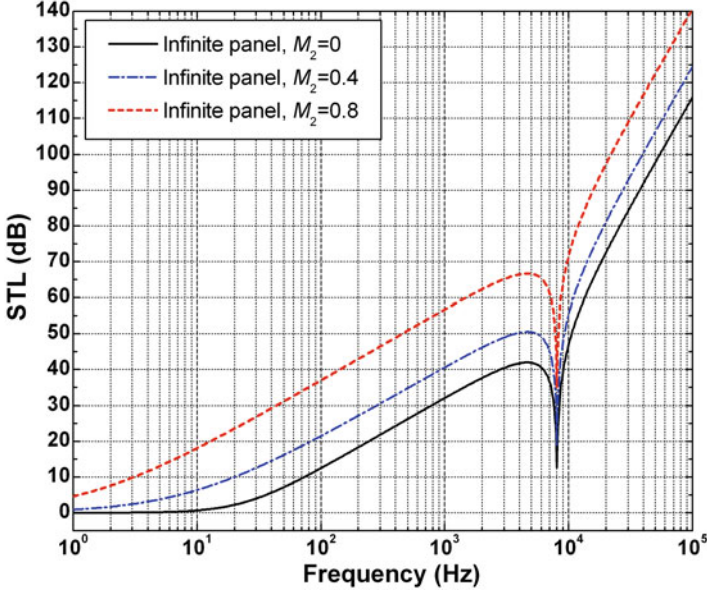


Fig. 2.7 STL versus frequency plots of an infinite panel for sound incident with elevation angle $\varphi_1 = 60^\circ$ and azimuth angle $\theta = 0^\circ$. The mean flow on the transmitted side is completely aligned with the x -direction (three cases: solid line $M_2 = 0$, dash-dot line $M_2 = 0.4$, and short dash line $M_2 = 0.8$), while the fluid on the incident side is at rest ($M_1 = 0$) (With permission from Acoustical Society of America)

positive refraction branch and the negative refraction branch) with respect to the critical Mach number value of $M_2 = 2$. Again, the gap along the φ_1 -axis is the region of total reflection, where the wavenumber has an imaginary component in the z -direction [27]. As schematically demonstrated in the inset of Fig. 2.6, the moving stream on the transmitted side exerts a significant refraction effect on the penetration of sound, just as the incident side moving stream does, which is attributable to the transfer effect of adjacent fluid particles in the moving stream. Note also that a normal incident sound penetrates straight through the panel into the other side. In other words, the refraction angle becomes zero when the incident angle is zero, whether the fluid on the transmitted (or incident) side is stationary or moving.

For a given incident angle, the refraction angle φ_2 increases with increasing Mach number M_2 until total reflection occurs. When total reflection occurs (corresponding to the gap along the φ_1 -axis between the two branches in Fig. 2.6), there is no energy flux along with the penetrated disturbance as stated previously. Interestingly, a sufficiently large incident angle together with a large Mach number generates a negative refraction angle disturbance, causing the negative refraction effect (corresponding to the smaller branch in Fig. 2.6).

Figure 2.7 plots the STL as a function of frequency for an infinite aeroelastic panel immersed in static fluid on the incident side and moving fluid on the transmitted side, with elevation angle $\varphi_1 = 60^\circ$ and azimuth angle $\theta = 0^\circ$ selected

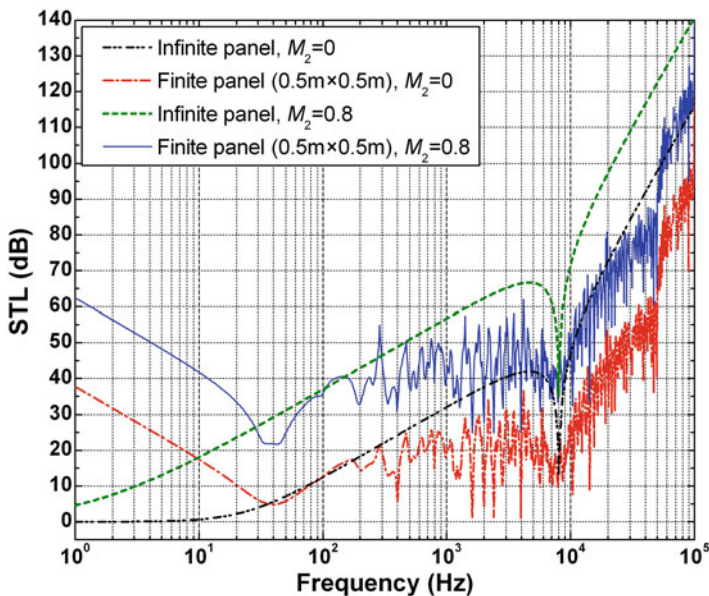


Fig. 2.8 STL versus frequency plots of a simply supported finite panel for sound incident with elevation angle $\varphi_1 = 60^\circ$ and azimuth angle $\theta = 0^\circ$. The mean flow on the transmitted side is completely aligned with the x -direction (two cases: $M_2 = 0$ and $M_2 = 0.8$), while the fluid on the incident side is at rest ($M_1 = 0$). For comparison, the corresponding plots for an infinite panel are included (With permission from Acoustical Society of America)

for the incident sound and three different Mach numbers ($M_2 = 0, 0.4$, and 0.8). The corresponding *STL* versus frequency plots of a finite aeroelastic panel are presented in Fig. 2.8.

Completely different from the tendencies shown in Fig. 2.4, there is nearly no change of the coincidence dip as Mach number M_2 is varied, while the *STL* increases dramatically in value over the whole frequency regime considered with increasing M_2 . The remarkable discrepancy between Figs. 2.4 and 2.7 suggests that the effects of incident side fluid flow on sound transmission are significantly different from those of fluid flow on the transmitted side.

As aforementioned, the dense peaks and dips appearing in the *STL* versus frequency plots of Fig. 2.8 reflect the modal behavior of a simply supported finite panel; for comparison, the corresponding *STL* plots of an infinite panel are included in Fig. 2.8. Analogous to Fig. 2.5, the *STL* curves for the two cases (i.e., infinite panel and finite panel) are in a global agreement above the (1,1) panel resonance frequency, with the infinite panel setting upper bounds for the finite bounded panel. The boundary constraint effect of the finite panel causes the discrepancy between the two cases below the (1,1) panel resonance frequency. Note also that, despite the presence of dense peaks and dips, the coincidence dip associated with the finite panel coincides with its counterpart of the infinite one. The pronounced increase of *STL* with increasing M_2 in the finite case is in accordance with the conclusion drawn previously for the infinite case.

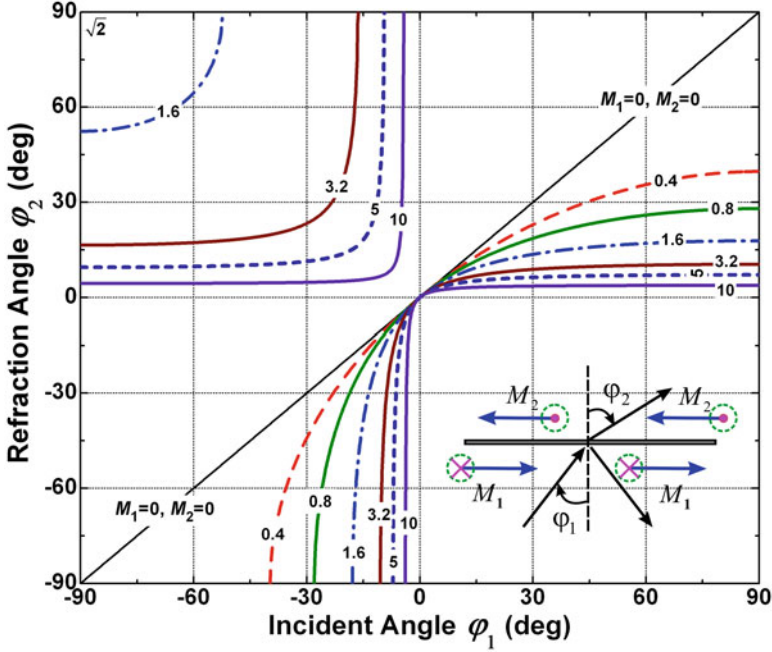


Fig. 2.9 Refraction angle φ_2 plotted as a function of incident angle φ_1 for the case when the moving fluid medium on the incident side is aligned with $\beta_1 = 45^\circ$ from the x -direction and that on the transmitted side has $\beta_2 = 45^\circ$. The two streams have the same Mach numbers, i.e., $M_1 = M_2$, but move in opposite directions. The labeled numbers denote the values of M_1 and absolute values of M_2 : for example, the curve labeled by 1.6 represents the case of $M_1 = 1.6$ and $M_2 = -1.6$ (With permission from Acoustical Society of America)

2.1.5 Effects of Incident Elevation Angle in the Presence of Mean Flow on Both Incident Side and Transmitted Side

In terms of angular contour plots as well as *STL* versus frequency plots, the above two sections have addressed the influence of mean flow residing in either the incident or transmitted side on the process of sound tunneling through an aeroelastic panel. To explore the physical nature of mean flow effects in more general cases, we consider next the case when the fluid media on both sides of an infinite panel are moving. The moving stream on the incident side is aligned with the direction of $\beta_1 = 45^\circ$ from the x -direction, while that on the transmitted side has $\beta_2 = 45^\circ$ (Fig. 2.1a). The predicted contour plots of refraction angle are shown in Fig. 2.9, where the labeled numbers denote the absolute Mach number values of the two flows; for simplicity, it is assumed that the two streams have the same Mach number but flow in opposite directions.

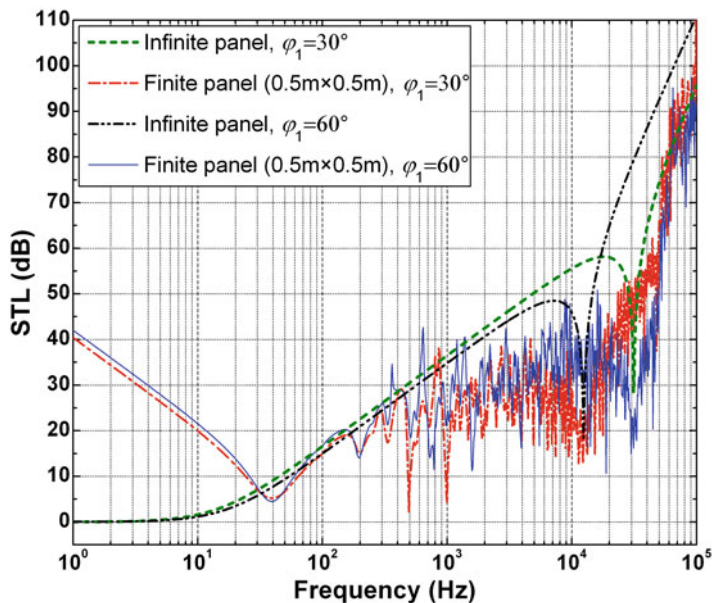


Fig. 2.10 STL plotted as a function of frequency for infinite and finite panels immersed in convected fluids, when the sound is incident with different elevation angles ($\varphi_1 = 30^\circ$ and 60°) at the same azimuth angle $\theta = 0^\circ$. The mean flow on the incident side is specified by $M_1 = 0.4$ and $\beta_1 = 45^\circ$, while that on the transmitted side by $M_2 = -0.4$ and $\beta_2 = 45^\circ$ (With permission from Acoustical Society of America)

It can be observed that the general trend exhibited by the curves of Fig. 2.9 is similar to that shown in Fig. 2.3 for the case when the fluid on the incident side is moving ($M_1 \neq 0$) and that on the transmitted side is at rest ($M_2 = 0$). However, the critical Mach number for the occurrence of negative refraction (beyond which the contour curves break into two branches) changes from 2 in Fig. 2.3 to $\sqrt{2}$ in Fig. 2.9. This is attributed to the presence of fluid flow on both sides of the infinite panel, which also leads to the significant alterations of the contour plots in Figs. 2.3 and 2.9. Moreover, when the directions of the two flows are reversed with respect to the case shown in Fig. 2.9, it has been established that the resulting contour plots are analogous to those of Fig. 2.6.

With $M_1 = 0.4$ and $\beta_1 = 45^\circ$ for flow on the incident side and $M_2 = -0.4$ and $\beta_2 = 45^\circ$ for flow on the transmitted side, Fig. 2.10 plots the STL as a function of frequency for both the infinite and finite bounded panels when the sound is incident in the downstream direction ($\varphi_1 = 30^\circ$ and 60°). The corresponding plots when the sound is incident in the upstream direction ($\varphi_1 = -30^\circ$ and -60°) are presented in Fig. 2.11.

The overall trends of the curves in Fig. 2.11 are similar to those of Fig. 2.10, implying that the intrinsic physical nature is not changed when sound is incident upstream. For example, the coincidence dip is shifted to a significantly lower

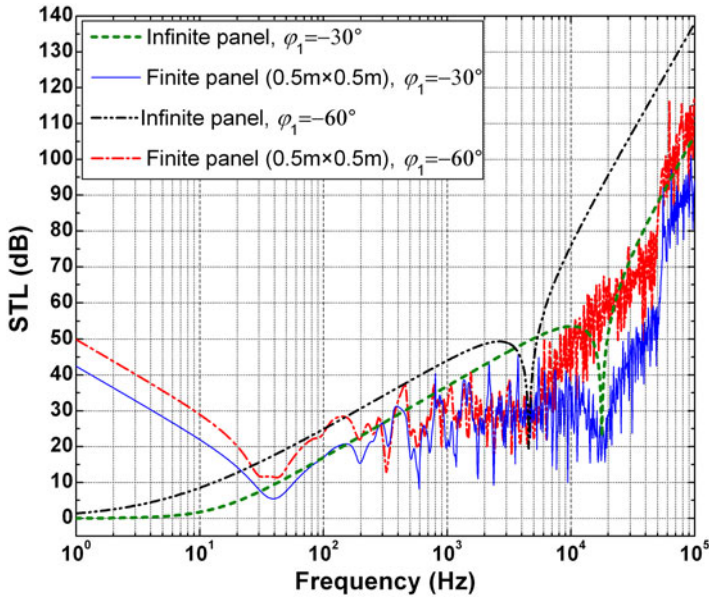


Fig. 2.11 STL plotted as a function of frequency for infinite and finite panels immersed in convected fluids, when the sound is incident with different elevation angles ($\varphi_1 = -30^\circ$ and -60°) at the same azimuth angle $\theta = 0^\circ$. The mean flow on the incident side is specified by $M_1 = 0.4$ and $\beta_1 = 45^\circ$, while that on the transmitted side by $M_2 = -0.4$ and $\beta_2 = 45^\circ$ (With permission from Acoustical Society of America)

frequency when the incident angle is increased from $\varphi_1 = 30^\circ$ to 60° in Fig. 2.10, and the same thing occurs in Fig. 2.11 upon changing φ_1 from -30° to -60° . However, the coincidence frequencies in Fig. 2.11 are considerably lower than their counterparts in Fig. 2.10, reflecting the difference between the upstream incident sound and the downstream incident sound. Moreover, the difference in STL values between the two cases of $\varphi_1 = 30^\circ$ and 60° in the lower frequency range below the coincidence dip is enlarged when the incident sound is changed from downstream in Fig. 2.10 to upstream in Fig. 2.11.

2.1.6 Conclusions

A theoretical model is proposed for the vibroacoustic behavior of a finite aeroelastic rectangular panel, which is immersed in convected fluids on both sides and simply embedded in an infinite acoustic baffle. The model accounts for the aeroelastic coupling of the convected fluid disturbance induced by panel motion by employing the convected wave equation for inviscid irrotational fluid flow and the displacement continuity condition at fluid-panel interfaces. Different from previous studies,

the general case of sound transmission across a finite panel enveloped by two-dimensional fluid flows (aligned along an arbitrary direction parallel to the panel surface) on both sides is theoretically formulated.

The influence of the incident side mean flow upon sound penetration is significantly different from that of the radiating (transmitted) side mean flow. Two branches are found in the contour plots of the refraction angle versus incident angle, corresponding to positive refraction and negative refraction, respectively. The contour plot for the case when the mean flow is on the transmitted side is just a reverse of that when the mean flow is on the incident side. The aerodynamic damping effects on the transmission of sound for both cases are well captured by plotting the *STL* as a function of frequency for varying Mach numbers. However, the frequency of coincidence dip differs significantly between the two cases: as the Mach number is increased, the coincidence dip frequency increases when the flow is on the incident side but remains unchanged when in the flow is on the radiating side.

In the most general case when the fluids on both sides of the panel are moving with mean flow, the predicted contour plots of the refraction angle versus incident angle are significantly different from those when the fluid on one side of the panel is moving and that on the other side is at rest. Furthermore, pronounced discrepancies in panel vibroacoustic behavior are found when the incident sound is changed from upstream to downstream.

2.2 Infinite Double-Leaf Aeroelastic Plates

2.2.1 Introduction

The reduction of sound transmission into aircraft interiors is a classical structural acoustic topic of paramount importance for the successful development of supersonic (or high subsonic) civil and military aircrafts [1–7, 12, 18, 39–47]. In general, the construction of such aircrafts is made by thin-walled structural elements. For example, double-leaf aeroelastic plates with a dissipative layer placed in between are commonly used for constructing the aircraft cabin fuselage [12, 24, 25, 39, 41–43, 46, 48, 49]. From the vibroacoustic point of view, the use of double-leaf partitions provides much more effective noise insulation over a wide frequency range than single-leaf plates do. The air cavity formed in between the outer panel (i.e., the source panel) and the trim panel (i.e., the radiating panel) is usually filled with high-density fiberglass blankets to improve thermal insulation and noise attenuation. However, the use of fiberglass blankets leads to increase of weight and thus offsets the above benefits to some extent.

Turbulent boundary layer (TBL)-induced noise and engine exhaust noise have been recognized as the primary sources of the interior noise of aircraft cabins [6, 7, 12, 14, 22, 25, 39, 41, 42, 45, 50–52]. A great deal of work is available

concerning the influences of convected fluid loaded on aircraft skin plates. For example, the acoustic power radiated by thin flexible panels subjected to TBL wall-pressure fluctuations was estimated by Davies [53] using a modal analysis method, in which the light fluid loading effects were considered. Also utilizing the method of modal expansion, Dowell [40] theoretically analyzed the transmission of TBL-induced noise through a flexible plate into a closed cavity by accounting for the effects of nonlinear plate stiffness and interaction between the plate and the external airflow. On the basis of the variational method for the vibration of a plate, a set of formulae for sound radiation from rectangular baffled plates having arbitrary boundary conditions were developed by Berry et al. [54]. This was later extended by Atalla and Nicolas [18] to study inviscid uniform subsonic flow, where the effects of the fluid flow were explicitly shown in terms of added mass and radiation resistance to avoid integration in the complex domain. Subsequently, by employing a suitable polynomial function to describe the displacement of a fluid-loaded plate having elastic boundary conditions, Berry [55] developed a new formulation for the vibration and sound radiation of the plate. Based upon the radiation of sound from a single, flat, elastic plate under TBL excitation, Graham [24] proposed a model to address the design problems associated with aircraft cabins, although the effect of mean flow was not taken in account. Graham [25] then developed an extended model consisting of a boundary layer-excited flat plate with its interior covered by two dissipative layers to simulate a factual aircraft cabin plate and found that the presence of the dissipative layers greatly reduces the radiation efficiency compared to a bare plate. A coupled FEM-BEM (finite element method-boundary element method) approach was adopted by Sgard et al. [15] to investigate the effects of the mean flow upon the vibroacoustic behavior of a flat plate subjected to a point force, where it was assumed that the mean flow remains undisturbed by the vibrating plate. The formulation [15] can explicitly show the effects of the mean flow in terms of added mass, damping, and stiffness, as done by Atalla and Nicolas [18]. The dynamic and acoustic responses of a finite baffled plate excited by TBL were investigated by Wu and Maestrello [56], in which the effect of structural nonlinearities induced by in-plane forces was considered. Recently, Clark and Frampton performed systematic studies on the aeroelastic structural response of a single-leaf panel excited by TBL noise and coupled with full potential flow aerodynamics [2, 4, 6–8, 10, 11]. The model accounting for the aerodynamic loading of panels and linearized potential flow aerodynamics was firstly developed in Ref. [4], and further analyses with the TBL-induced noise disturbance taken into account were presented in Refs. [6, 8, 10, 17]. In addition, numerous numerical, theoretical, and experimental investigations have been devoted to studying the transmission of airborne sound across double-panel partitions immersed in static fluid [5, 13, 21, 34, 35, 41, 48, 49, 57–68].

Most of the aforementioned investigations, however, focus either on the TBL-induced noise transmission or the transmission of sound from static fluid (irrespective of mean flow) or on the effects of the mean flow on structural stability (i.e., panel flutter, self-excited vibrations; see, e.g., Crighton [69] for a list of references).

Only a few studies [1, 7] have specifically considered the influence of the mean flow on external noise transmission, although this is of particular importance for studying jet-noise transmission into an aircraft. To squarely address this issue, we develop in the present study an aero-acoustic-elastic theoretical model to quantify the influence of external mean flow on sound transmission through a double-leaf aeroelastic plate. This chapter is organized as follows. First, an aero-acoustic-elastic theoretical model to emulate the transmission of jet-power or propeller-induced noise into cabin interior is presented for the sound transmission loss (*STL*), in which the plate dynamics and fluid-structure coupling are accounted for. Second, the physical mechanisms associated with the sound transmission process are discussed, and a set of simple closed-form formulae for the associated natural frequencies are derived from physical principles independent of the theoretical model. These formulae are then used to validate the model predictions, as no suitable experimental or numerical or theoretical results exist in the open literature that can be used to check the validity of the present model. The effects of a few relevant parameters (e.g., Mach number, direction of mean flow, sound incidence angle, panel curvature, and cabin internal pressure) on the *STL* are systematically explored. This chapter is finished with concluding remarks drawn from the obtained results.

2.2.2 Statement of the Problem

To mimic the transmission of engine exhaust noise into the interior of an airplane cabin under typical cruise conditions, a uniform plane sound wave varying harmonically in time is assumed to transmit through a double-leaf aeroelastic plate from the external mean flow side to the interior static fluid side. As shown in Fig. 2.12, the considered system consists of two infinite parallel flexural plates made of homogenous and isotropic materials and is immersed in inviscid, irrotational fluid media. The upper, middle, and bottom fluid media separated by the two plates occupy the spaces of $z < 0$, $h_1 < z < H + h_1$, and $z > H + h_1 + h_2$, respectively, and are characterized by (ρ_1, c_1) , (ρ_2, c_2) , and (ρ_3, c_3) in terms of mass density and sound speed, respectively. Here, H is the depth of the air gap, and h_1 and h_2 are the thicknesses of the two panels. The mean fluid flow with uniform speed v is assumed to move along the x -axis direction. The incident sound wave transmitting from the external mean flow side is characterized by elevation angle φ_1 and azimuth angle β with respect to the defined coordinate system (see Fig. 2.12). The incident sound is partially reflected and partially transmitted through the structure via the upper plate, middle fluid medium, and bottom plate into the static fluid medium side. The double-layer plate is modeled initially as a flat double-leaf aeroelastic partition, with both the external mean flow and the aeroelastic coupling accounted for. Subsequently, to better emulate the curved skin of an aircraft fuselage and the real process of jet-noise penetration into aircraft interior, the effects of the panel curvature and cabin internal pressurization on sound transmission are quantified.

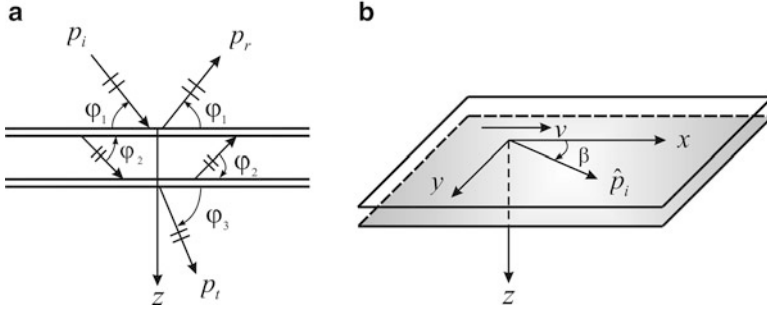


Fig. 2.12 Schematic illustration of sound transmission through a double-leaf aeroelastic plate in the presence of external mean flow: (a) side view and (b) global view

Several simplifying assumptions are adopted in the present analysis of the system shown in Fig. 2.12: (1) The two plates are modeled by the classical Kirchhoff thin plate theory. (2) The fluid media are inviscid and irrotational [2, 4, 6–8, 10, 11, 15, 18, 56]. (3) The plate surface adjacent to the mean flow is sufficiently smooth so that it is appropriate to represent the fluid-plate interface by the streamline of the fluid flow, i.e., the total flow is tangential to the acoustically deformed boundary [1, 31]. Note that several previous studies [26, 32, 70] also considered the problem of sound reflection and transmission associated with a moving fluid medium but without accounting for the interaction with a thin plate.

2.2.3 Formulation of Plate Dynamics

As shown in Fig. 2.12, the acoustic field is divided into three regimes by the two parallel flat plates which, without loss of generality, are assumed below to have the same thickness. The corresponding sound pressures for the incident field, the middle field, and the transmission field (denoted by indices 1, 2, and 3, respectively) can be expressed as

$$p_1 = P_i e^{i\omega t - i(k_{1x}x + k_{1y}y + k_{1z}z)} + \sum_n \beta_n e^{i\omega t - i(k_{1x}x + k_{1y}y - k_{1z}z)} \quad (2.47)$$

$$p_2 = \sum_n \varepsilon_n e^{i\omega t - i(k_{2x}x + k_{2y}y + k_{2z}z)} + \sum_n \zeta_n e^{i\omega t - i(k_{2x}x + k_{2y}y - k_{2z}z)} \quad (2.48)$$

$$p_3 = \sum_n \xi_n e^{i\omega t - i(k_{3x}x + k_{3y}y + k_{3z}z)} \quad (2.49)$$

The summation index n is introduced in Eqs. (2.47), (2.48), and (2.49) as a consequence of the modal decomposition of the pressure p_2 onto the standing-wave

modes of the cavity, which justifies through the continuity equations [see Eqs. (2.63) and (2.64)] the present modal formulation for the reflected and the transmitted pressure amplitudes. Note however that this index does not appear explicitly through the wavenumber components in the argument of the exponential factors. The wavenumber components in Eqs. (2.47)–(2.49) can be written as

$$k_{1x} = k_1 \cos \varphi_1 \cos \beta, \quad k_{1y} = k_1 \cos \varphi_1 \sin \beta, \quad k_{1z} = k_1 \sin \varphi_1 \quad (2.50)$$

$$k_{2x} = k_2 \cos \varphi_2 \cos \beta, \quad k_{2y} = k_2 \cos \varphi_2 \sin \beta, \quad k_{2z} = k_2 \sin \varphi_2 \quad (2.51)$$

$$k_{3x} = k_3 \cos \varphi_3 \cos \beta, \quad k_{3y} = k_3 \cos \varphi_3 \sin \beta, \quad k_{3z} = k_3 \sin \varphi_3 \quad (2.52)$$

Let c_t be the trace wave speed in the plate and the trace wavenumber be given by $k_t = \omega/c_t = 2\pi/\lambda_t$, where λ_t is the wavelength of the panel trace wave. Accordingly, the transverse deflections of the two plates induced by the incident sound can be expressed as

$$w_1(x, y; t) = w_{10} e^{i\omega t - i(k_t \cos \beta)x - i(k_t \sin \beta)y} \quad (2.53)$$

$$w_2(x, y; t) = w_{20} e^{i\omega t - i(k_t \cos \beta)x - i(k_t \sin \beta)y} \quad (2.54)$$

In the incident acoustic field, there exists a uniform flow of velocity V tangential to the acoustically deformed boundary (i.e., the fluid-plate interface). The convected wave equation for the pressure in the fluid is thence given by [1]

$$\frac{D^2 p_1}{Dt^2} = \left(\frac{\partial}{\partial t} + V \cdot \nabla \right)^2 p_1 = c_1^2 \nabla^2 p_1 \quad (2.55)$$

As stated above, for simplicity, the mean flow is aligned along the x -axis on the fluid-plate interface. Consequently, Eq. (2.55) can be simplified as

$$\left(\frac{\partial}{\partial t} + v \cdot \frac{\partial}{\partial x} \right)^2 p = c_1^2 \nabla^2 p_1 \quad (2.56)$$

Substituting Eq. (2.47) into Eq. (2.56), one obtains the wavenumber in the flowing fluid as

$$k_1 = \frac{k_1^*}{(1 + M \cos \varphi_1 \cos \beta)} \quad (2.57)$$

where $k_1^* = \omega/c_1$ is the acoustic wavenumber in the fluid at rest and $M = v/c_1$ is the Mach number of the mean flow.

Different from the incident field coupled with a mean flow, the fluid medium in between the two plates and that in the transmitted field are both static. In such cases, the propagation of sound obeys the classical wave equation, so that the wavenumbers in the two fluid media are given by

$$k_2 = \frac{\omega}{c_2}, \quad k_3 = \frac{\omega}{c_3} \quad (2.58)$$

In order for the sound waves to “fit” at the boundary (i.e., the plate), the trace wavelengths must match [1], namely,

$$k_{1x} = k_t \cos \beta = k_{2x} = k_t \cos \beta = k_{3x} \quad (2.59)$$

$$k_{1y} = k_t \sin \beta = k_{2y} = k_t \sin \beta = k_{3y} \quad (2.60)$$

Incorporating Eqs. (2.50)–(2.52) into Eqs. (2.59) and (2.60), one obtains the directions of sound propagation in the middle and the transmitted fluid media as

$$\varphi_2 = \arccos \left(\frac{c_2}{c_1} \frac{\cos \varphi_1}{1 + M \cos \varphi_1 \cos \beta} \right) \quad (2.61)$$

$$\varphi_3 = \arccos \left(\frac{c_3}{c_1} \frac{\cos \varphi_1}{1 + M \cos \varphi_1 \cos \beta} \right) \quad (2.62)$$

which also describe the refraction laws for sound transmission from one medium to another. Note that $\varphi_1 = \varphi_2 = \varphi_3$ and $c_1 = c_2 = c_3$ in the absence of the mean flow ($M = 0$). Thus, to refract the wave at the plate is one noticeable effect of the mean flow [1]. In fact, in the presence of the mean flow, the wave would be refracted (and partially reflected) even if the plates were not present.

2.2.4 Consideration of Fluid-Structure Coupling

To determine the unknown parameters appearing in the above equations, supplementary boundary conditions are needed, i.e., the displacement continuity condition between the plate particle and the adjacent fluid particle and the driving relation between the incident sound and the plate dynamic response. In general, the continuity condition for the fluid-structure coupling is described through the velocity of the particles pertaining separately to the fluid medium and the solid medium when the fluid is at rest. In the case of a moving flow, however, the transfer effect of the fluid motion needs to be considered [31], and hence the primary particle displacement continuity should be applied, as described below.

2.2.4.1 Displacement Continuity Condition

Let $\tilde{\lambda}_1$ and $\tilde{\lambda}_2$ denote separately the displacements of the fluid particles in the incident field and the middle field, both adjacent to the upper panel, and let $\tilde{\lambda}_3$ and $\tilde{\lambda}_4$ denote separately the displacements of the fluid particles in the middle field and the transmitted field, both adjacent to the bottom panel. These displacements should satisfy the Navier-Stokes equation for an inviscid and irrotational fluid, namely,

$$\frac{D^2 \tilde{\lambda}_1}{Dt^2} = -\frac{1}{\rho_1} \frac{\partial p_1}{\partial z} \Big|_{z=0}, \quad \frac{D^2 \tilde{\lambda}_2}{Dt^2} = -\frac{1}{\rho_2} \frac{\partial p_2}{\partial z} \Big|_{z=h_1} \quad (2.63)$$

$$\frac{D^2 \tilde{\lambda}_3}{Dt^2} = -\frac{1}{\rho_2} \frac{\partial p_2}{\partial z} \Big|_{z=H+h_1}, \quad \frac{D^2 \tilde{\lambda}_4}{Dt^2} = -\frac{1}{\rho_3} \frac{\partial p_3}{\partial z} \Big|_{z=H+h_1+h_2} \quad (2.64)$$

For harmonic sound wave excitation, the fluid particle displacements take the form of

$$\tilde{\lambda}_j = \tilde{\lambda}_{j0} e^{i\omega t - i(k_{1x}x + k_{1y}y)} \quad (j = 1, 2, 3, 4) \quad (2.65)$$

Substitution of Eqs. (2.47)–(2.49) and (2.65) into Eqs. (2.63) and (2.64) gives the amplitudes of the fluid particle displacements as

$$\tilde{\lambda}_{10} = -\frac{ik_{1z}}{\rho_1} \frac{\left(P_i - \sum_n \beta_n\right)}{(\omega - vk_{1x})^2} \quad (2.66)$$

$$\tilde{\lambda}_{20} = -\frac{ik_{2z} \left(\sum_n \varepsilon_n - \sum_n \zeta_n\right)}{\rho_2 \omega^2} \quad (2.67)$$

$$\tilde{\lambda}_{30} = -\frac{ik_{2z}}{\rho_2 \omega^2} \left(\sum_n \varepsilon_n e^{-ik_{2z}H} - \sum_n \zeta_n e^{ik_{2z}H}\right) \quad (2.68)$$

$$\tilde{\lambda}_{40} = -\frac{ik_{3z}}{\rho_3 \omega^2} \sum_n \xi_n e^{-ik_{3z}H} \quad (2.69)$$

In view of the wavenumber relationships, Eqs. (2.59) and (2.60), and the dynamic deflections of the two plates, Eqs. (2.53) and (2.54), the continuity condition of the particle displacements is given by

$$\tilde{\lambda}_{10} = w_{10} = \tilde{\lambda}_{20}, \quad \text{and} \quad \tilde{\lambda}_{30} = w_{20} = \tilde{\lambda}_{40} \quad (2.70)$$

Substitution of Eqs. (2.53) and (2.54) and (2.66)–(2.69) into (2.70) leads to

$$P_i - \sum_n \beta_n = \frac{\rho_1 c_1 \sin \varphi_2}{\rho_2 c_2 \sin \varphi_1} \frac{1}{1 + M \cos \varphi_1 \cos \beta} \left(\sum_n \varepsilon_n - \sum_n \zeta_n \right) \quad (2.71)$$

$$\sum_n \varepsilon_n - \sum_n \zeta_n = \frac{\rho_2 c_2}{\sin \varphi_2} \dot{w}_{10} \quad (2.72)$$

$$\sum_n \varepsilon_n e^{-ik_{2z}H} - \sum_n \zeta_n e^{ik_{2z}H} = \frac{\rho_2 c_2 \sin \varphi_3}{\rho_3 c_3 \sin \varphi_2} \sum_n \xi_n e^{-ik_{3z}H} \quad (2.73)$$

$$\sum_n \xi_n e^{-ik_{3z}H} = \frac{\rho_3 c_3}{\sin \varphi_3} \dot{w}_{20} \quad (2.74)$$

where $\dot{w}_{10} = i\omega w_{10}$ and $\dot{w}_{20} = i\omega w_{20}$.

2.2.4.2 Driving Relations

The driving relations for the pressure difference across the panel and the panel vibration response can be described as

$$P_i + \sum_n \beta_n - \sum_n \varepsilon_n - \sum_n \zeta_n = Z_{p1} \dot{w}_{10} \quad (2.75)$$

$$\sum_n \varepsilon_n + \sum_n \zeta_n - \sum_n \xi_n = Z_{p2} \dot{w}_{20} \quad (2.76)$$

Detailed derivations of Eqs. (2.75)–(2.76) are given in Sect. 2.2.6 for both the flat and curved aeroelastic plates.

For simplicity, the equivalent characteristic impedances for the three separate fluid media associated with the transmission of sound across the double-leaf plate of Fig. 2.12 are defined as

$$Z_1 = \frac{\rho_1 c_1}{\sin \varphi_1 (1 + M \cos \varphi_1 \cos \beta)} \quad (2.77)$$

$$Z_2 = \frac{\rho_2 c_2}{\sin \varphi_2} \quad (2.78)$$

$$Z_3 = \frac{\rho_3 c_3}{\sin \varphi_3} \quad (2.79)$$

It is readily seen that in addition to being strongly dependent on the intrinsic property of the fluid (i.e., density and sound speed), the equivalent characteristic impedances of the fluid media are also determined by the sound incident angle and flow velocity. Actually, these characteristic impedances reflect the close relationship between the sound pressure and the fluid particle velocity.

2.2.5 Definition of Sound Transmission Loss

The transmissivity $\tau(\varphi_1, \beta)$ is defined to quantify the sound transmission through the double-leaf plate [1] as

$$\tau(\varphi_1, \beta) = \frac{\rho_1 c_1}{\rho_2 c_2} \left| \frac{\sum_n \xi_n}{P_i} \right|^2 \quad (2.80)$$

where

$$\frac{P_i}{\sum_n \xi_n} = \frac{1}{4Z_2 Z_3 \cos(k_{2z} H)} \left[\begin{aligned} & (Z_1 Z_{p2} + Z_{p1} Z_{p2}) (e^{-i(-k_{2z} + k_{3z})H} - e^{-i(k_{2z} + k_{3z})H}) \\ & + Z_2 Z_{p2} (e^{-i(-k_{2z} + k_{3z})H} + e^{-i(k_{2z} + k_{3z})H}) + 2Z_2 Z_3 \cos(k_{2z} H) \\ & + 2i Z_3 Z_{p1} \sin(k_{2z} H) + 2i Z_1 Z_3 \sin(k_{2z} H) + 2Z_1 Z_2 e^{-ik_{3z} H} \end{aligned} \right] \quad (2.81)$$

Accordingly, the sound transmission loss is defined as a decibel scale of the transmissivity:

$$\text{STL} = -10 \log_{10} \tau(\varphi_1, \beta) \quad (2.82)$$

2.2.6 Characteristic Impedance of an Infinite Plate

2.2.6.1 Infinite Flat Plate

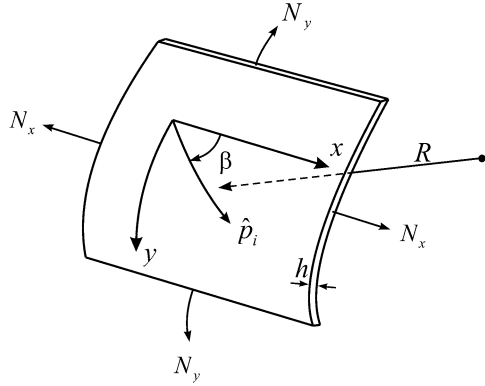
Consider an infinite plate immersed in a fluid medium and subjected to a harmonic incident sound excitation on one side. The governing equation for the deflection of the plate is given by

$$D \nabla^4 w + m \frac{\partial^2 w}{\partial t^2} = p e^{i\omega t - i(k_{2x} x + k_{2y} y)} \quad (2.83)$$

Note that, in the present study, the structural loss factor η is accounted for by introducing the complex bending stiffness D [3, 5] as

$$D = \frac{E h^3 (1 + j\eta)}{12 (1 - \nu^2)} \quad (2.84)$$

Fig. 2.13 Shallow cylindrical panel under biaxial membrane stresses



Since the incident sound is harmonic, the deflection of the plate is assumed to take the form of

$$w = w_0 e^{i\omega t - i[(k_t \cos \beta)x + (k_t \sin \beta)y]} \quad (2.85)$$

where w_0 is determined by substituting (2.85) into (2.83) as

$$w_0 = \frac{p}{D(k_{2x}^2 + k_{2y}^2)^2 - m\omega^2} \quad (2.86)$$

Accordingly, the panel impedance that determines the relation between the sound pressure and the particle velocity is given by

$$Z_p = \frac{p}{\dot{w}} = \frac{p}{i\omega w_0} = im\omega \left(1 - \frac{D\omega^2}{mc_2^4} \cos^4 \varphi_2 \right) \quad (2.87)$$

2.2.6.2 Infinite Curved Plate with Biaxial Membrane Stresses

In reality, the aircraft fuselage skin is a typical shallow cylindrical panel structure with internal pressurization during cruise condition. To better emulate the actual fuselage skin and the real process of jet-noise penetration into the aircraft interior, the Donnell-Mushtari-Vlasov shallow cylindrical shell theory [1, 71] is adopted. The geometry and coordinates of a shallow cylindrical panel of thickness h under biaxial membrane stresses (i.e., N_x in the x -direction and N_y in the y -direction) are schematically illustrated in Fig. 2.13. The governing equation for its lateral deformation w can be expressed as

$$\begin{aligned}
D \nabla^8 w + \frac{Eh}{R^2} \frac{\partial^4 w}{\partial x^4} - \nabla^4 \left(N_x \frac{\partial^2 w}{\partial x^2} + N_y \frac{\partial^2 w}{\partial y^2} \right) + m \nabla^4 \left(\frac{\partial^2 w}{\partial t^2} \right) \\
= \nabla^4 \left(p e^{i\omega t - i(k_{2x}x + k_{2y}y)} \right)
\end{aligned} \quad (2.88)$$

After a few algebraic manipulations similar to those leading to Eq. (2.86), one obtains

$$w_0 = p \cdot \left[D \left(k_{2x}^2 + k_{2y}^2 \right)^2 + \frac{Eh}{R^2} \frac{k_{2x}^4}{\left(k_{2x}^2 + k_{2y}^2 \right)^2} + \left(N_x k_{2x}^2 + N_y k_{2y}^2 \right) - m\omega^2 \right]^{-1} \quad (2.89)$$

which, together with the definition of the panel impedance, leads to

$$\begin{aligned}
Z_p = \frac{p}{\dot{w}} = \frac{p}{i\omega w_0} = \frac{1}{i\omega} \left[D \left(k_{2x}^2 + k_{2y}^2 \right)^2 + \frac{Eh}{R^2} \frac{k_{2x}^4}{\left(k_{2x}^2 + k_{2y}^2 \right)^2} \right. \\
\left. + \left(N_x k_{2x}^2 + N_y k_{2y}^2 \right) - m\omega^2 \right] \\
= i m \omega \left[1 - \frac{D\omega^2}{mc_2^4} \cos^4 \varphi_2 - \frac{Eh \cos^4 \beta}{R^2 m \omega^2} - \frac{\cos^2 \varphi_2}{mc_2^2} \left(N_x \cos^2 \beta + N_y \sin^2 \beta \right) \right] \quad (2.90)
\end{aligned}$$

2.2.7 Physical Interpretation for the Appearance of STL Peaks and Dips

Figure 2.14 plots the predicted *STL* as a function of incident sound frequency for Mach number $M = 0.05$, with the sound incidence elevation angle fixed at $\varphi_1 = 30^\circ$ and the azimuth angle at $\beta = 0^\circ$ (i.e., completely aligned with the downstream direction). In Fig. 2.14, four physical phenomena (i.e., mass-air-mass resonance, standing-wave attenuation, standing-wave resonance, and coincidence resonance) associated with the transmission of sound from the external mean flow side across the double-leaf partition can be clearly identified, which are marked with symbols in the frequency range considered (from 0 to 10,000 Hz). To predict the inherent frequencies associated with these phenomena in the presence of mean flow, a set of simple closed-form formulae are derived based purely on physical principles (see Appendix). Since the derivation of these formulae is independent of the aero-acoustic-elastic theoretical model presented in Sects. 2.2.3–2.2.6, they may be used

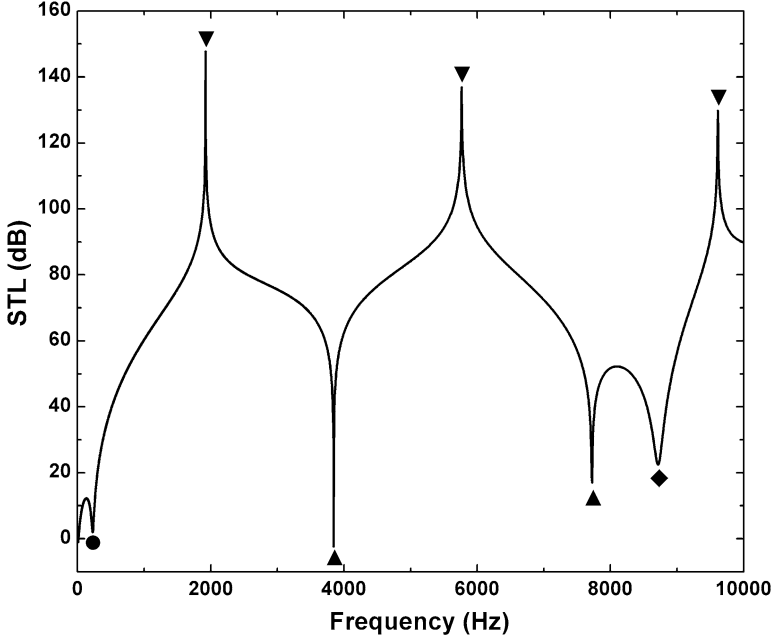


Fig. 2.14 Predicted *STL* as a function of frequency for mean flow speed $M = 0.05$, sound incident elevation angle $\varphi_1 = 30^\circ$, and azimuth angle $\beta = 0^\circ$; ● mass-air-mass resonance, ▼ standing-wave attenuation, ▲ standing-wave resonance, ◆ coincidence resonance

to check the validity of the model predictions as no other suitable experimental or theoretical work exists. In Fig. 2.14, the Mach number is selected as $M = 0.05$ only because, at this small Mach number, all the four physical phenomena can be clearly identified in the considered frequency range of 0–10,000 Hz. Of course, the present model is suitable for handling higher Mach number cases (e.g., $M = 0.4$, 0.8, and 1.2 as shown in Figs. 2.16, 2.17, 2.19, and 2.20), although the coincidence resonance occurs beyond 10,000 Hz at large Mach numbers when the sound is incident downstream. Furthermore, when the sound is incident upstream, the three phenomena (i.e., mass-air-mass resonance, standing-wave attenuation, standing-wave resonance) all disappear, leaving only the coincidence dip in the *STL* curve (see Fig. 2.18).

As shown in Fig. 2.14, the first dip is associated with the mass-air-mass resonance that is particularly marked by the filled circle. The mass-air-mass resonance in the absence of the mean flow usually occurs when the two panels move in opposite phases [3, 5, 34, 59, 65], with the air gap behaving like an elastic spring. With the influence of the mean flow accounted for, the frequency of the mass-air-mass resonance is (see Appendix for detailed derivations)

$$f_\alpha = \frac{1}{2\pi \sin \varphi_2} \sqrt{\frac{\rho_2 c_2^2}{H} \frac{(m_1 + m_2)}{m_1 m_2}} \quad (2.91)$$

In view of the expression for φ_2 in Eq. (2.61), the above relation indicates that f_α is dependent of the Mach number M as well as sound incidence angles φ_1 and β , which is different from that of a double-leaf plate immersed in static fluid.

The second phenomenon relates to the three peaks on the *STL* versus frequency curve (marked by the inverted and filled triangle ▼ in Fig. 2.14), corresponding separately to the first-order, second-order, and third-order standing-wave attenuations. Wave attenuation occurs when the distance difference between the routes that the two intervening waves pass through is the odd number of the one-quarter wavelength of the incident sound. Thus, the frequencies for these standing-wave attenuations are given by (Appendix)

$$f_{p,n} = \frac{(2n-1)c_2}{4H \sin \varphi_2}, (n = 1, 2, 3 \dots) \quad (2.92)$$

which are again dependent upon the Mach number and the sound incidence angles. When standing-wave attenuation occurs, the destructive interference between the positive- and negative-going waves causes the wave amplitude to significantly decrease before the sound is transmitted across the partition, resulting in maximum sound reduction.

In contrast to the standing-wave attenuation, the third phenomenon (i.e., the standing-wave resonance) occurs when the distance difference between the routes that the two intervening waves pass through is the multiple of the half wavelength of the incidence sound (denoted by the filled triangle ▲ in Fig. 2.14). In such circumstances, the constructive interference of the positive- and negative-going waves leads to an enhanced sound transmission through the partition. The standing-wave resonance as a result of the enhanced effect occurs at the following frequencies (Appendix):

$$f_{d,n} = \frac{nc_2}{2H \sin \varphi_2}, (n = 1, 2, 3 \dots) \quad (2.93)$$

The fourth phenomenon (i.e., the coincidence resonance) occurs when the wavelength of the flexural bending wave in the panel matches the trace wavelength of the incidence sound. The corresponding dip is marked in Fig. 2.14 by the filled diamond ♦. Due to the influence of the mean flow, the resonance frequency differs from that in the static fluid case, given by (Appendix)

$$f_c = \frac{c_2^2}{2\pi h \cos^2 \varphi_2} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \quad (2.94)$$

In the absence of mean flow ($M = 0$), according to Eq. (2.61), the angle φ_2 is equal to the incident angle φ_1 for the case considered here (i.e., $c_1 = c_2$, both panels immersed in air). Consequently, in this limiting case (static fluid), the frequencies for the above four phenomena are simply obtained by replacing φ_2 with φ_1 in Eqs. (2.91), (2.92), (2.93), and (2.94).

Table 2.1 Comparison between theoretical model predictions and simple closed-form formulae for *STL* peaks and dips ($M = 0.05$, $\varphi_1 = 30^\circ$, $\beta = 0^\circ$)

Mass-air-mass resonance f_a (Hz)		Standing-wave attenuation $f_{p,n}$ (Hz)		Standing-wave resonance $f_{d,n}$ (Hz)		Coincidence resonance f_c (Hz)	
Theory	Eq. (2.91)	Theory	Eq. (2.92)	Theory	Eq. (2.93)	Theory	Eq. (2.94)
227.73	231.69	1922.2	1922.2	3844.3	3844.3	8714.9	8726.2
		5766.5	5766.5	7720.5	7688.6		
		9610.8	9610.8	11533	11533		

The existence of the four distinct acoustic phenomena influences significantly the shape of the *STL* versus frequency curves, as evidenced by the intense peaks and dips appearing in Fig. 2.14. The frequencies of the four phenomena predicted from the theoretical model, i.e., Eqs. (2.47), (2.48), (2.49), (2.50), (2.51), (2.52), (2.53), (2.54), (2.55), (2.56), (2.57), (2.58), (2.59), (2.60), (2.61), (2.62), (2.63), (2.64), (2.65), (2.66), (2.67), (2.68), (2.69), (2.70), (2.71), (2.72), (2.73), (2.74), (2.75), (2.76), (2.77), (2.78), (2.79), (2.80), (2.81), (2.82), (2.83), (2.84), (2.85), (2.86), and (2.87), are compared in Table 2.1 with the closed-form formulae, i.e., Eqs. (2.91), (2.92), (2.93), and (2.94). Excellent agreement is achieved, which in a way validates the theoretical model since the closed-form formulae are derived completely independent of the model. Of course, it would be more desirable to validate the present model predictions with other theories or experimental measurements, but, unfortunately, none exists in the open literature.

2.2.8 Effects of Mach Number

As discussed in the previous section, all the four acoustic phenomena associated with the *STL* peaks and dips depend on the Mach number of the mean fluid flow. It is thus expected that the Mach number plays an important role in the transmission process of sound through a double-leaf partition. Two typical cases for sound incidence in the downstream direction and in the upstream direction, respectively, are studied below to explore further the Mach number influence.

2.2.8.1 Sound Incidence Along the Downstream Direction

Consider first the case of sound incidence having an elevation angle of $\varphi_1 = 30^\circ$ and an azimuth angle of $\beta = 45^\circ$, with its wave vector component in the downstream direction being positive. Figure 2.15 presents the predicted *STL* versus frequency curves for selected Mach numbers, $M = 0, 0.4, 0.8$, and 1.2 . It is seen from Fig. 2.15 that changes in the Mach number lead to noticeable shifts of the *STL* curves: as the Mach number is increased, the *STL* peaks and dips are all shifted to lower frequencies, resulting in an increase of the *STL* value over a relatively broad

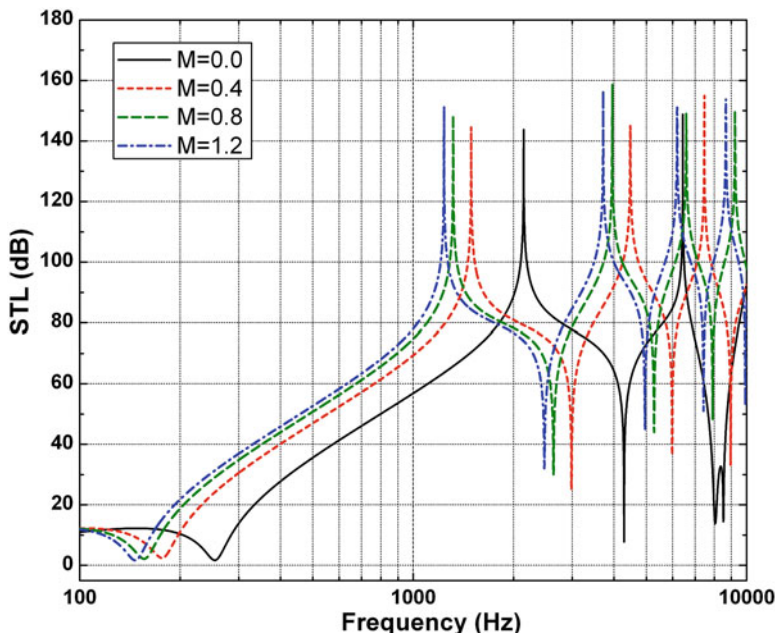


Fig. 2.15 Predicted *STL* plotted as a function of frequency for selected Mach numbers with sound incidence elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 45^\circ$

frequency range. While the noticeable decrease of the *STL* peaks and dips can be attributed to the added-mass effect of the convected fluid loading, the increase of the *STL* value in the frequency range considered agrees well with existing results [6, 7]. One may expect that the *STL* value corresponding to the peaks and dips should also increase as the Mach number is increased, because the aerodynamic damping effect increases when the convected flow becomes more turbulent [7]. However, the increase of the *STL* value related to the peaks and dips is not as remarkable as anticipated (Fig. 2.15). This may be attributed to the fact that the present study assumes irrotational, inviscid potential flow, which is much different from the turbulent boundary layer considered by Frampton and Clark [7]. Another possible reason may be that as the Mach number is increased, the complex fluid-structure coupling effects overwhelm the aerodynamic damping effect when the *STL* peaks and dips move considerably away from their original locations.

By extracting the frequencies associated with the *STL* peaks and dips for different Mach numbers, the dependence of these frequencies on the Mach number is obtained, as shown in Fig. 2.16a–d using different symbols (e.g., hollow circles, hollow diamonds, and hollow squares). For comparison, predictions from the closed-form formulae are also included in Fig. 2.16b, c, denoted by different lines (e.g., solid line, dash line, and dash-dot line). Note that the first three orders of the frequencies are plotted for the standing-wave attenuation and standing-wave resonance in Fig. 2.16. Similar to the results of Table 2.1, it is seen from Fig. 2.16

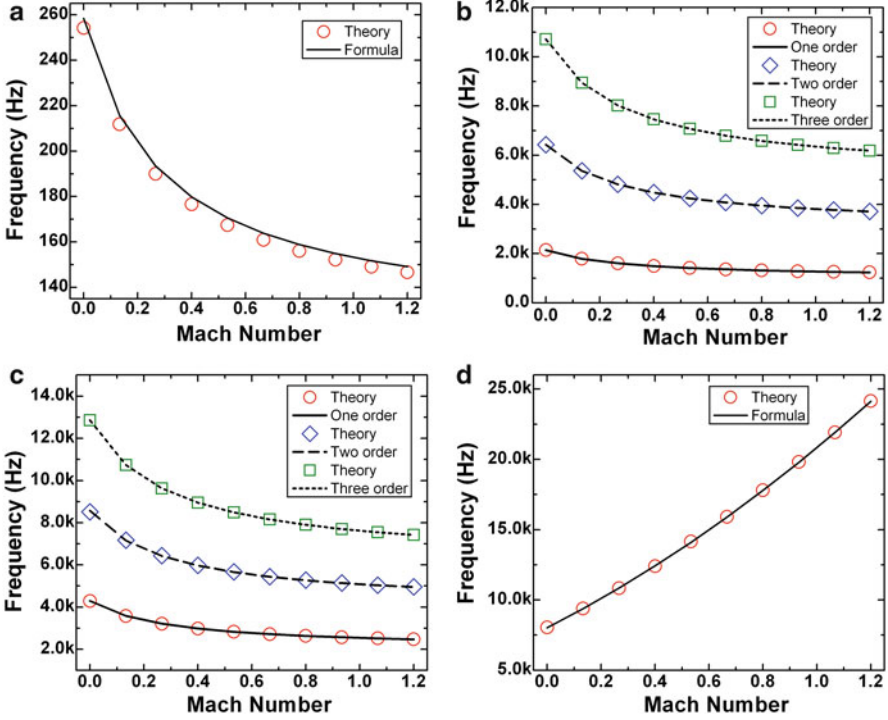


Fig. 2.16 Effects of Mach number on the frequencies of *STL* peaks and dips for sound incidence with elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 45^\circ$: (a) mass-air-mass resonance, (b) standing-wave attenuation, (c) standing-wave resonance, and (d) coincidence resonance. Symbols (e.g., *hollow circles*, *hollow diamonds*, and *hollow squares*) refer to theoretical predictions. Lines (e.g., *solid line*, *dash line*, and *dash-dot line*) denote the calculated results from Eqs. (2.91), (2.92), (2.93), and (2.94)

that the model predictions agree very well with Eqs. (2.91), (2.92), (2.93), and (2.94), and the same can be said regarding the other cases shown in Figs. 2.17, 2.19, and 2.20. The results of Fig. 2.16 demonstrate that, except for the coincidence resonance frequencies, the frequencies for the mass-air-mass resonance, standing-wave attenuation, and standing-wave resonance decrease as the Mach number increases, due mainly to the added-mass effects of the convected fluid loading. The exception of the coincidence resonance is attributed to the fact that the significant refraction effect of the mean flow has overwhelmed the added-mass effect on the coincidence resonance.

2.2.8.2 Sound Incidence Along the Upstream Direction

Consider next the case of sound incidence in the upstream direction, with elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 135^\circ$. The effects of the Mach number

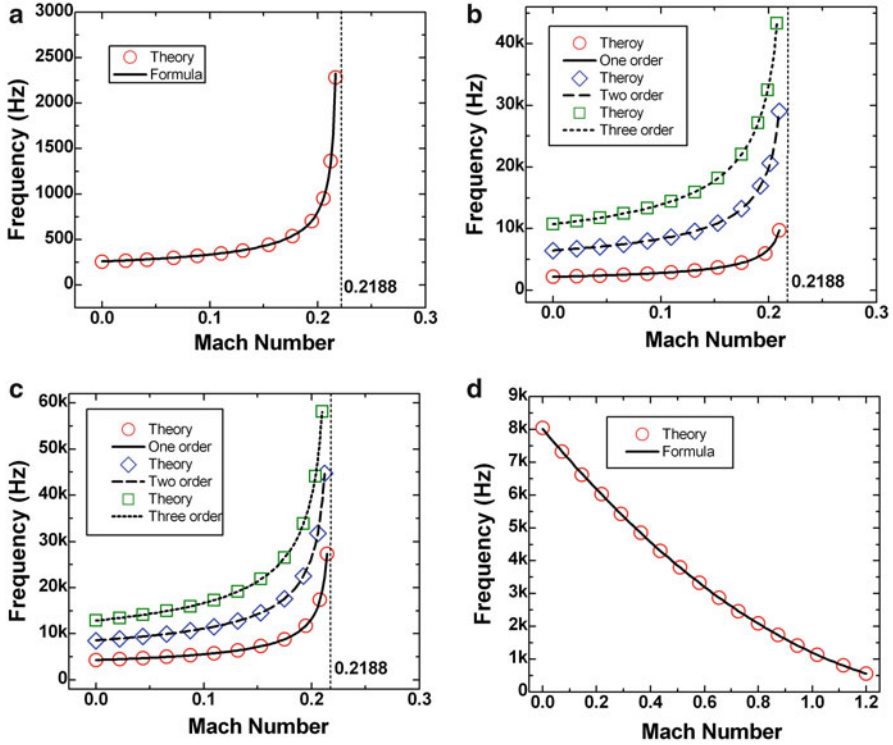


Fig. 2.17 Effects of Mach number on the frequencies of *STL* peaks and dips for sound incidence with elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 135^\circ$: (a) mass-air-mass resonance, (b) standing-wave attenuation, (c) standing-wave resonance, and (d) coincidence resonance

on the mass-air-mass resonance, the standing-wave attenuation, the standing-wave resonance, and the coincidence resonance frequencies are displayed in Fig. 2.17a–d. The frequencies for the first three acoustic phenomena increase with increasing Mach number until a critical value of $M = 0.2188$ is reached, beyond which these phenomena all disappear. The existence of the critical Mach number can be explained as follows. Note from Eqs. (2.91), (2.92), and (2.93) that $\sin \varphi_2$ is present in each of the denominators and φ_2 depends on the Mach number through Eq. (2.61). It is seen from (2.61) that for sound incidence with $\varphi_1 = 30^\circ$ and $\beta = 135^\circ$, $\sin \varphi_2$ approaches zero as the Mach number approaches 0.2188, thus giving rise to infinitely large values of the frequencies. Consequently, the mass-air-mass resonance, the standing-wave attenuation, and the standing-wave resonance do not exist for $M > 0.2188$. On the other hand, the transmitted evanescent wave for coincidence resonance is enhanced by the coincidence effect of the bottom panel. Accordingly, the corresponding frequency in the present upstream case (see Fig. 2.17d) decreases as the Mach number is increased, a feature that is opposite to that in the downstream case shown in Fig. 2.11d.

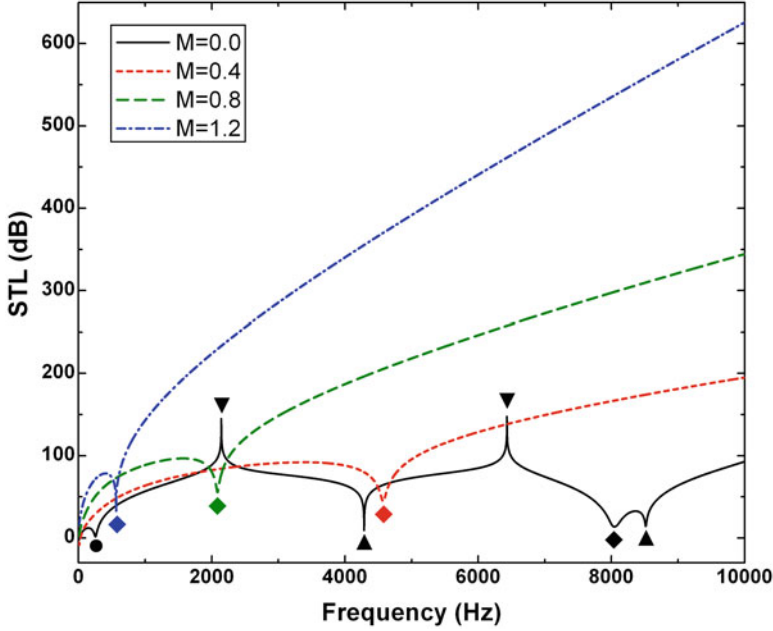


Fig. 2.18 Variations of *STL* with incident frequency for selected Mach numbers, with elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 135^\circ$; $\bullet$ mass-air-mass resonance, $\blacktriangledown$ standing-wave attenuation, $\blacktriangle$ standing-wave resonance, $\blacklozenge$ coincidence resonance

To illustrate further the shift of the *STL* peaks and dips with varying flow velocity, Fig. 2.18 plots the predicted *STL* versus frequency curves for selected Mach numbers, with elevation angle $\varphi_1 = 30^\circ$ and azimuth angle $\beta = 135^\circ$. It is seen that all the four acoustic phenomena appear in the case of static fluid ($M = 0$), which are marked by different symbols in Fig. 2.18. As the Mach number increases, the mass-air-mass resonance, the standing-wave attenuation, and the standing-wave resonance gradually disappear, consistent with the results of Fig. 2.17a–c. This also backs the selection of a small Mach number ($M = 0.05$) for plotting the results in Fig. 2.14, so that all the four different acoustic phenomena can be clearly identified within the considered frequency range. In addition to the disappearing of *STL* peaks and dips with increasing Mach number, a dramatic increase of the *STL* value over a broad frequency range is observed in Fig. 2.18. It should be clarified that the significant increase of the *STL* value (up to 600 dB) is caused not only by the structural damping but also by the total reflection phenomenon occurring in the specific case associated with Fig. 2.18. When total reflection occurs, a disturbance penetrates through the double-leaf panel into the transmitted side fluid medium, while the wavenumber component k_z in the z -direction takes the form of $-j\hbar$ ($\hbar$ being a positive real number), resulting in a rapid exponential decay of the wave amplitude in the form of $\exp(-\hbar)$. Although the physical nature of the total reflection has been addressed in detail by Ribner [26], we believe that its influence on *STL* may have been quantified for the first time in Fig. 2.18.

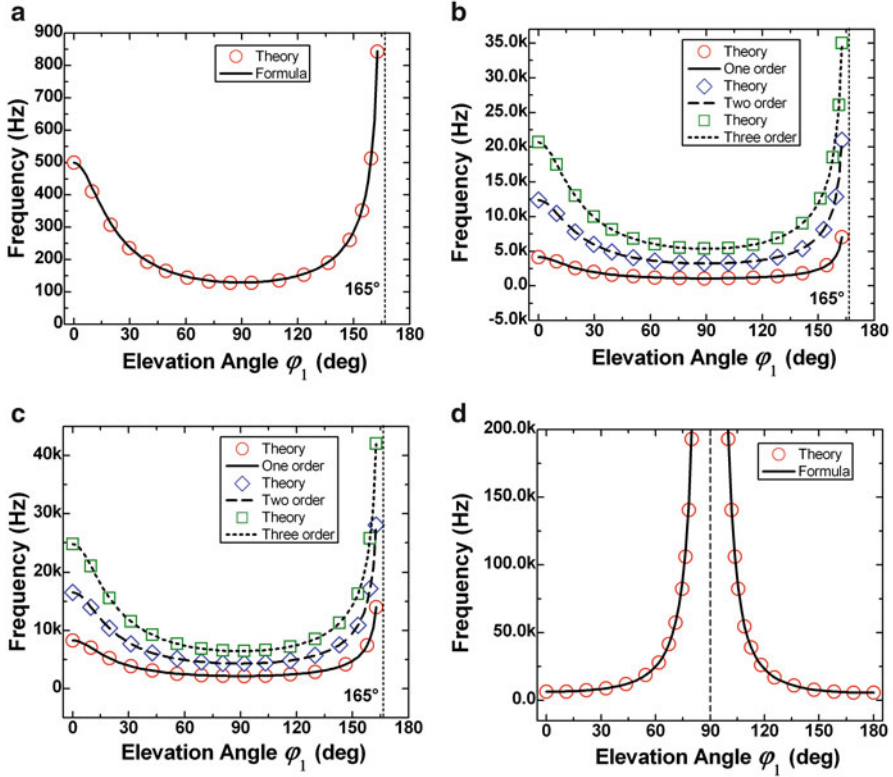


Fig. 2.19 Effects of incident elevation angle on the frequencies of STL peaks and dips for sound incidence with azimuth angle $\beta = 45^\circ$ and Mach number $M = 0.05$: (a) mass-air-mass resonance, (b) standing-wave attenuation, (c) standing-wave resonance, and (d) coincidence resonance

2.2.9 Effects of Elevation Angle

It has been reported that the sound incidence elevation angle has a noticeable effect on the transmission of sound through a partition immersed in static fluid [65]. For the problem considered here, its influence in the presence of mean flow is quantified. Obtained results for a fixed azimuth angle of $\beta = 45^\circ$ and a fixed Mach number of $M = 0.05$ are presented in Fig. 2.19, where it is seen that the four distinct acoustic phenomena all exhibit significant dependence on the elevation angle. It is interesting to find that for the first three phenomena (i.e., the mass-air-mass resonance, the standing-wave attenuation, and the standing-wave resonance), critical values of the elevation angle exist, beyond which all three phenomena instantaneously vanish. For example, for the specific case of $\beta = 45^\circ$ and $M = 0.05$, the critical value is found to be $\varphi_1 = 165^\circ$, whereas for the case of $\beta = 135^\circ$ and $M = 0.05$ (results not shown here for brevity), the three phenomena are suppressed when the elevation angle lies within the range between 0 and 15° . Moreover, the coincidence resonance frequency becomes infinitely large when the elevation angle approaches $\pi/2$, and the system is

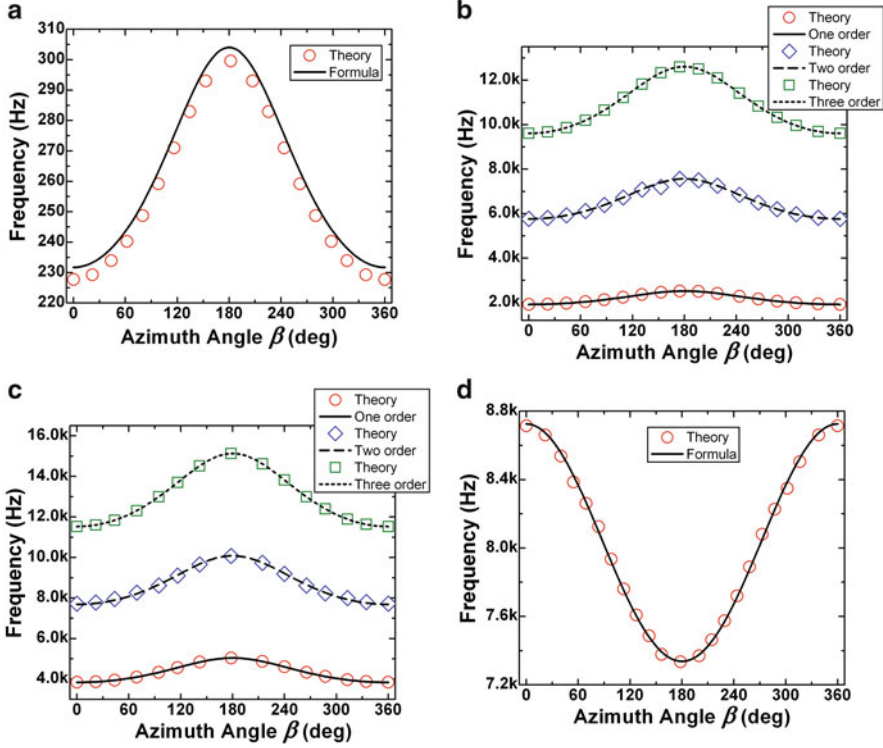


Fig. 2.20 Effects of incident azimuth angle on the frequencies of *STL* peaks and dips for sound incidence with elevation angle $\varphi_1 = 30^\circ$ and Mach number $M = 0.05$: (a) mass-air-mass resonance, (b) standing-wave attenuation, (c) standing-wave resonance, and (d) coincidence resonance

symmetrical with respect to the sound incidence angle and the flow direction (e.g., the dependence of the four critical frequencies upon the elevation angle for the case of $\beta = 135^\circ$ is simply obtained by inverting Fig. 2.19 for the case of $\beta = 45^\circ$).

2.2.10 Effects of Azimuth Angle

For sound transmission through sandwich panels with corrugated cores immersed in static fluid, it has been found that the sound incidence azimuth angle β plays a negligible role due to the symmetrical property of the considered system [5, 65]. This, however, is no longer valid if the system is immersed in a flowing fluid. For the present double-leaf plate, Fig. 2.20 plots the predicted critical frequencies as functions of the azimuth angle β in the specific case of $M = 0.05$ and $\varphi_1 = 30^\circ$. The azimuth angle is seen to have a significant effect on all the four acoustic phenomena. Note that varying the azimuth angle from 0 to 2π corresponds to four processes in

sequence, i.e., first downstream phase from 0 to $\pi/2$, first upstream phase from $\pi/2$ to π , second upstream phase from π to $3\pi/2$, and second downstream phase from $3\pi/2$ to 2π . The symmetry of the results shown in Fig. 2.20 with respect to $\beta = \pi$ is therefore readily understandable. Actually, alteration of these frequencies with respect to the incident azimuth angle confirms the refraction effect of the mean flow.

2.2.11 Effects of Panel Curvature and Cabin Internal Pressurization

As for the investigation of external jet-noise penetration through fuselage skin into aircraft interior, it would be of great interest to estimate the effects of panel curvature and cabin internal pressurization on the noise transmission. To this end, we extend Koval's work [1] regarding the effects of panel curvature and cabin internal pressurization on sound transmission through a single-leaf aeroelastic plate, and for computational simplicity (assuming that the apparent contradiction of an unlimited curved panel can be overlooked), an infinite slightly curved double-leaf panel is considered. However, we believe that the results obtained with this somewhat idealized model are reasonable because all the physical phenomena have been well captured (as shown below), including the mass-air-mass resonance, the ring frequency resonance, the standing-wave attenuation and resonance, and the coincidence resonance. Particularly, the predicted ring frequency resonance is consistent with existing results [1, 72] concerning single-leaf curved panels.

Again, to clearly demonstrate all the significant acoustic phenomena associated with a curved double-leaf panel for frequencies below 10,000 Hz, a small Mach number of $M = 0.05$ is selected, with the incident elevation angle arbitrarily fixed at $\varphi_1 = 30^\circ$. Under such conditions, the effects of panel curvature and internal pressurization are shown in Fig. 2.21 using a set of combinations, i.e., a flat panel $R = \infty$ with and without internal pressure $P = 0.1$ MPa and a curved panel $R = 6$ m with and without internal pressure $P = 0.1$ MPa. It is seen from Fig. 2.21 that, irrespective of the panel curvature or internal pressurization, all the four acoustic phenomena can be distinctively identified. In the two cases concerning curved double-leaf panels, i.e., ($P = 0$ Pa, $R = 6$ m) and ($P = 0.1$ MPa, $R = 6$ m), a newly added ring frequency resonance dip (i.e., the first dip) occurs, which is absent in flat double-leaf panels. In the absence of internal pressure, the ring frequency of the curved panel can be predicted by [1, 72]

$$f_R = \frac{1}{2\pi R} \sqrt{\frac{Eh}{m}} \quad (2.95)$$

In addition to generating the ring frequency resonance, the panel curvature also shifts the mass-air-mass resonance dip to a higher frequency, which can be seen by comparing the second dip in the curve of ($P = 0$ Pa, $R = 6$ m) with the first dip

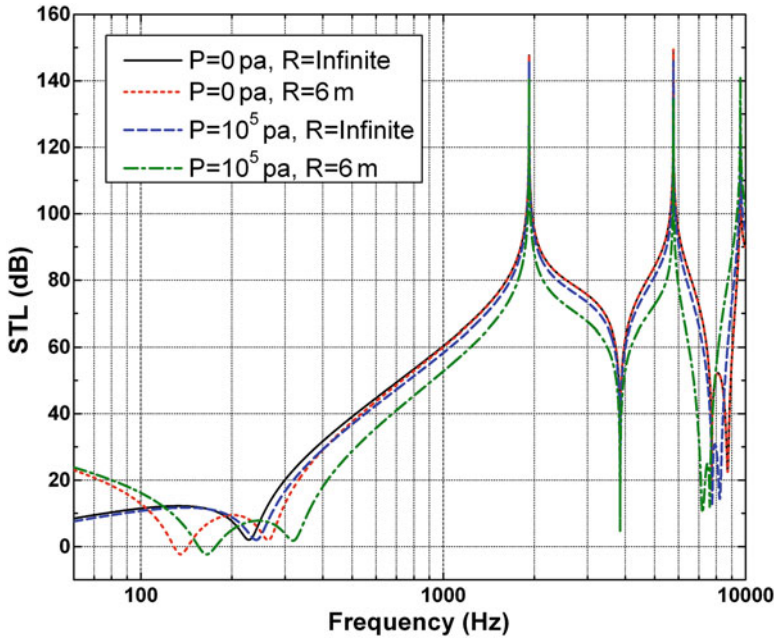


Fig. 2.21 Effects of panel curvature and cabin internal pressurization on *STL* for sound incidence with elevation angle $\varphi_1 = 30^\circ$ in the presence of external mean flow ($M = 0.05$)

in the curve of ($P = 0$ Pa, $R = \infty$). Otherwise, the general trend of the *STL* versus frequency curve of the flat panel agrees well with that of the curved panel (Fig. 2.21).

By comparing the two cases of ($P = 0$ Pa, $R = \infty$) and ($P = 0.1$ MPa, $R = \infty$), it is seen that the internal pressurization shifts the mass-air-mass resonance dip to a higher frequency but only has a small influence on the second-order standing-wave resonance and coincidence dips. In comparison, the panel curvature has a noticeable influence on all of these phenomena. Therefore, when designing a practical aircraft fuselage, the noticeable combination effects of the panel curvature and internal pressurization on noise transmission need to be carefully considered, especially in the relatively low-frequency range where the ring frequency resonance occurs.

2.2.12 Conclusions

The effects of external mean flow on sound transmission through double-leaf aeroelastic plates have been quantified analytically, with the intention to simulate the transmission of engine exhaust noise through typical aircraft fuselage skin panels into cabin interiors. Theoretical formulations have been developed for the analysis of the fluid-plate coupling problem, and the *STL* versus frequency curves for various specific cases (Mach number, direction of mean flow, sound incidence angle, panel

curvature, and internal pressurization) are obtained, with the added-mass effects of the convected fluid loading well captured. Four distinct acoustic phenomena (i.e., the mass-air-mass resonance, the standing-wave attenuation, the standing-wave resonance, and the coincidence resonance) for flat double-leaf plates as well as the ring frequency resonance for curved double-leaf plates are clearly identified. Simple closed-form formulae for predicting the natural frequencies associated with these acoustic phenomena in the presence of mean flow are subsequently derived from physical principles, which are completely independent upon the theoretical model. In the absence of other relevant theoretical or experimental work, the excellent agreement between these formulae and the model predictions serve to validate the two theories against each other.

Systematic parametric studies are subsequently conducted to quantify the effects of Mach number, the direction of mean flow, the sound incidence elevation and azimuth angles, the panel curvature, as well as the internal pressurization on the *STL*. As the Mach number is increased, in the case of sound incidence along the downstream direction, the *STL* values increase over a broad frequency range, and the natural frequencies for the associated acoustic phenomena (except for the coincidence resonance) are shifted considerably to the lower frequency range due to the added-mass effects of the mean flow. The exception of the coincidence resonance is attributed to its strong dependence on the refraction angle φ_2 but not on the convected fluid loading.

For sound incidence along the upstream direction, the corresponding frequencies increase until the Mach number is increased up to a critical value, except again for the coincidence resonance. Further increase of the Mach number beyond the critical value results in the disappearance of the mass-air-mass resonance, the standing-wave attenuation, and the standing-wave resonance, but the coincidence resonance is always existent. The increase of the Mach number induces a noticeable increment of the *STL* value over a relatively wide range of frequency due to the total reflection effects.

In the presence of external mean flow, the transmission of sound is significantly influenced by the sound incidence elevation angle and azimuth angle, and the noticeable combination effects of the panel curvature and internal pressurization should be taken into account in the practical design of aircraft fuselages.

2.3 Double-Leaf Panel Filled with Porous Materials

2.3.1 Introduction

Improving the sound transmission insulation performance of aircraft cabin fuselage panels is helpful for reducing aircraft interior noise. As different kinds of sandwich panels have been applied to construct cabin fuselages to provide more effective noise isolation, such as double-leaf thin panels filled with porous absorbent materials [39, 42, 43], it is important to evaluate the sound transmission characteristics of

such sandwich constructions. Further, since there exists high-speed airflow outside the cabin in typical cruise condition, it is even more significant to investigate the influence of external airflow upon sound transmission.

For double-leaf panels filled with porous absorbent materials, before exploring the sound insulation capability of the whole structure, it is necessary to accurately model the propagation process of sound in the porous material. Two main approaches have been developed to address the issue. One is the well-known Biot theory [73, 74], which assumes that stress wave propagation in a fluid-saturated porous material can be described by four nondimensional parameters and a characteristic frequency, and there exist two dilatational waves and one rotational wave in the material. In addition, the theory provides a set of functions of characteristic parameters to govern the acoustic medium. However, when the pore diameter is equal to the quarter wavelength of the acoustic wave, the Biot theory breaks down as stated by Lighthill [75]. The Biot model has been extensively applied to investigate the acoustic properties of different materials including saturated sand [76], rectangular and triangular pores [77], and catalytic converters [78]. The other approach is the semiempirical model [25, 79–81]. Delany and Bazley [79] showed empirically that the characteristic impedance and the propagation coefficient are functions of frequency f divided by static flow resistance R , i.e., f/R , and provided a method to estimate R of a material from its bulk density. Bies and Hansen [82] presented static flow resistance information for typical porous materials and suggested the most common applications of the flow resistance. Graham [25] used this approach to examine the characteristic impedance of dissipative materials for cabin inside treatment. Allard and Champoux [80] proposed a set of new semiempirical equations for sound propagation in rigid frame fibrous materials by modeling the porous material as an equivalent fluid with dynamic density and dynamic bulk modulus. They also suggested that the equations could be used when f/R is smaller than 1 kg/m^3 for most cases.

The transmission loss of sound across double-leaf panels separated by porous materials has been extensively investigated. For typical example, Lauriks et al. [83] developed a transfer matrix model to study the transmission loss through panels with solid porous layers, with the Biot model employed to describe sound propagation through the porous material, while Brouard et al. [84] proposed a general method to model sound propagation in layered systems such as fluid-saturated porous layers. Panneton et al. [34] presented a three-dimensional (3D) finite element model to calculate the loss of sound transmission through multilayer structures containing porous absorbent materials. The structures considered vary according to whether the filling porous material is bonded or not to the faceplates. Making use of two-dimensional (2D) elasticity theory, Chonan and Kugo [85] presented a model to examine the sound transmission characteristics of a three-layered panel excited by plane waves. Kang et al. [86] employed the method of Gauss distribution function for incident energy to predict the sound transmission loss (*STL*) of multilayered panels such as double-plate structures embedded with porous materials. Bolton et al. [87] calculated random incidence transmission loss through double-leaf panels lined with elastic porous materials.

As for the influence of external flow interaction on acoustic characteristics of structures, numerous works exist in the open domain. For example, Clark and Frampton [2, 4, 7] systematically investigated the structural response of panels excited by turbulent boundary layer (TBL)-induced noise, and Graham [24, 25] developed an extended theoretical model to investigate aircraft structure response excited by TBL-induced noise. Davies [53] theoretically estimated acoustic power radiation of TBL excited panels, while Koval et al. [1] examined the mean flow effect on *STL* of a single plate and found that the mean flow increased the transmission loss in the whole frequency range. Xin and Lu [23, 88] studied theoretically the effect of external mean flow on the transmission loss of double-leaf panels and identified four different kinds of acoustic phenomena in the sound transmission process through a double-leaf structure. Accounting for the mean flow effect, Sgard et al. [15] developed a coupled FEM-BEM (finite element method-boundary element method) model to investigate the vibroacoustic behavior of planar plates. However, in the presence of external mean flow, the structures studied in the aforementioned investigations are relatively simple and do not consider the presence of additional porous sound absorptive materials.

To more accurately examine the effects of mean flow on jet-engine noise transmission through aircraft fuselages, we propose a theoretical model for the aero-acoustic problem of jet-engine noise transmission through infinite double-leaf panels filled with fibrous sound absorptive materials in the presence of uniform external mean flow. Although a uniform flow across the panel might be an ideal case that would be not happening in reality, the assumption of a uniform mean flow can be warranted as has been done by many researchers [1, 22]. To describe sound propagation in the fibrous material, the equivalent fluid model is employed, and the fluid momentum equations are applied to ensure displacement continuity at fluid-structure interfaces. Upon validating the model predictions against existing experimental results, systematic parameter investigations are carried out using the proposed model, with guidance conclusions for practical aircraft fuselage designs obtained.

2.3.2 Problem Description

Consider the double-leaf panel structure shown schematically in Fig. 2.21, which contains two parallel thin elastic plates filled with fibrous sound absorptive materials in between. The two plates are made of homogenous and isotropic material. For simplicity, the double-leaf panel is taken as infinitely large in plane. A harmonic plane sound wave penetrates through the sandwich panel, accompanied by a uniform external mean flow that moves along the x -axis with uniform speed V on one side of the panel (Fig. 2.22). The fluid on the other side is motionless. The sound wave is incident upon the panel with elevation angle φ_1 and azimuth angle β relative to the coordinate axes of Fig. 2.22. The thicknesses of the two thin plates are h_1 and h_2 , respectively, while H is the thickness of the filled porous material. In general,

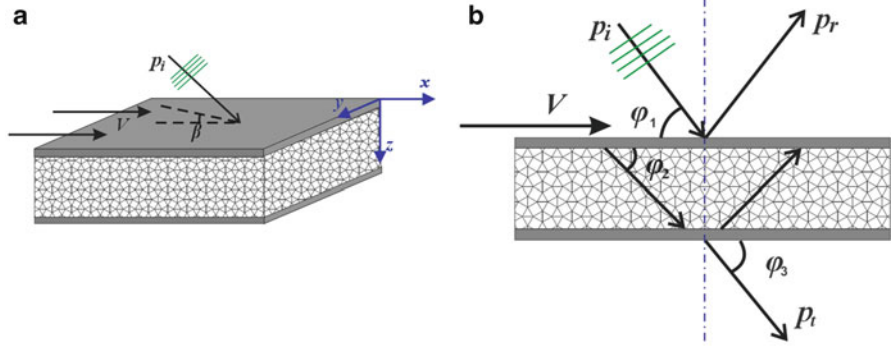


Fig. 2.22 Schematic of sound transmission through a *double-leaf panel* filled with porous material in the presence of uniform external mean flow: (a) global view; (b) side view

the acoustic field may be divided into three parts: incident field $z < 0$, middle field (i.e., sound absorbent field) $h_1 < z < h_1 + H$, and transmitted field $z > h_1 + H + h_2$. The acoustic media in the three fields are defined by mass densities (ρ_1, ρ_2, ρ_3) and sound wavenumbers (k_1, k_2, k_3), respectively.

Two further assumptions are introduced to develop the theoretical model: (1) The porous sound absorbent material is bonded to both faceplates, with no gap between the porous material and either of the faceplates. The equivalent fluid model is employed to describe the acoustic behaviors of the porous material. According to Rebillard et al. [89], when a porous material is bonded to a screen, the equivalent fluid model is applicable only when the skeleton of the porous material is very limp. Consequently, the porous material considered here is assumed to be limp enough to apply the equivalent fluid model, which is usually true for commonly used porous sound absorbent materials. (2) The fluid media are inviscid and irrotational.

2.3.3 Theoretical Model

2.3.3.1 Acoustic Field

The acoustic field is divided into three parts as previously described. Accordingly, sound pressures in the three acoustic fields can be expressed as

Incident field:

$$p_1 = P_{1i} e^{j\omega t - j(k_{1x}x + k_{1y}y + k_{1z}z)} + P_{1r} e^{j\omega t - j(k_{1x}x + k_{1y}y - k_{1z}z)} \quad (2.96)$$

Middle field:

$$p_2 = P_{2i} e^{j\omega t - j(k_{2x}x + k_{2y}y + k_{2z}z)} + P_{2r} e^{j\omega t - j(k_{2x}x + k_{2y}y - k_{2z}z)} \quad (2.97)$$

Transmitted field:

$$p_3 = P_{3t} e^{j\omega t - j(k_{3x}x + k_{3y}y + k_{3z}z)} \quad (2.98)$$

where P_{1i} is the amplitude of the incident wave, P_{1r} is the amplitude of the reflected wave, P_{2i} and P_{2r} are separately the amplitude of the positive and negative waves, P_{3t} is the amplitude of the transmitted wave, (k_1, k_2, k_3) are the wavenumbers of the three acoustic fields, while $(k_{ix}, k_{iy}, k_{iz}, i = 1, 2, 3)$ are the components of the corresponding wavenumbers along (x, y, z) -directions, respectively. Thus, (k_{ix}, k_{iy}, k_{iz}) can be written as

$$k_{ix} = k_i \cos \varphi_i \cos \beta, \quad k_{iy} = k_i \cos \varphi_i \sin \beta, \quad k_{iz} = k_i \sin \varphi_i \quad (2.99)$$

Due to the refraction effect of external mean flow [1], the wavenumber in the incident field is

$$k_1 = \frac{\omega/c_1}{(1 + M \cos \varphi_1 \cos \beta)} \quad (2.100)$$

The equivalent fluid model [80, 90, 91] is employed to calculate the sound wavenumber in the middle field. According to this model, sound propagation in fibrous materials can be described by equivalent dynamic density and dynamic bulk modulus. The dynamic density $\rho(\omega)$ is given by

$$\rho(\omega) = \rho_0 \left[1 + \frac{1}{j2\pi} \left(\frac{R}{\rho_0 f} \right) G_1 \left(\frac{\rho_0 f}{R} \right) \right] \quad (2.101)$$

and the dynamic bulk modulus is given by

$$K(\omega) = \gamma_s P_0 \left[\gamma_s - \frac{\gamma_s - 1}{1 + (1/j8\pi N_{Pr}) (\rho_0 f/R)^{-1} G_2 (\rho_0 f/R)} \right]^{-1} \quad (2.102)$$

where

$$G_1 (\rho_0 f/R) = \sqrt{1 + j\pi (\rho_0 f/R)} \quad (2.103)$$

$$G_2 (\rho_0 f/R) = G_1 [(\rho_0 f/R) 4N_{Pr}] \quad (2.104)$$

Here, R is the static flow resistivity, γ_s is the specific heat ratio of air, ρ_0 is the air density, P_0 is the standard atmospheric pressure, N_{Pr} is the Prandtl number, and f is the frequency of the sound wave. The dynamic density considers inertial and viscous forces per unit volume of air in the material and is related to the averaged molecular displacement of air and the averaged variation of pressure. With k_{cav} denoting the

complex wavenumber of the porous material, it can be expressed by the dynamic density and dynamic bulk modulus as

$$k_{cav} = 2\pi f \sqrt{\frac{\rho(\omega)}{K(\omega)}} \quad (2.105)$$

According to Morse and Ingard [92], the complex density of the porous material may be written as

$$\rho_2 = \rho_{cav} = \frac{k_2^2 \rho_0}{k_0^2 \gamma_s \sigma} \quad (2.106)$$

where σ is the porosity of the porous material.

In the transmitted field, the wavenumber is

$$k_3 = \frac{\omega}{c_3} \quad (2.107)$$

where $c_3 = c_0$.

Let the transverse deflection wavenumber in the plate be denoted as k_p . For the sound waves to fit at the boundary, the trace wavelengths must match [1, 23, 88], namely,

$$\begin{aligned} k_{1x} &= k_p \cos \beta = k_{2x} = k_p \cos \beta = k_{3x} \\ k_{1y} &= k_p \sin \beta = k_{2y} = k_p \sin \beta = k_{3y} \end{aligned} \quad (2.108)$$

Substitution of Eqs. (2.100) and (2.105) into Eq. (2.108) yields

$$\cos \varphi_2 = \frac{k_1 \cos \varphi_1}{k_2} = \frac{\cos \varphi_1}{c_1 (1 + M \cos \varphi_1 \cos \beta)} \sqrt{\frac{K(\omega)}{\rho(\omega)}} \quad (2.109)$$

$$\cos \varphi_3 = \frac{k_2 \cos \varphi_2}{k_3} = \frac{c_3 \cos \varphi_1}{c_1 (1 + M \cos \varphi_1 \cos \beta)} \quad (2.110)$$

2.3.3.2 Fluid-Structure Coupling

To determine the unknown parameters associated with the three acoustic fields, two fluid-structure coupling conditions must be supplemented: (a) displacement continuity between plate particle and adjacent fluid particle and (b) driving relation between incident sound and plate dynamic response.

Let (δ_1, δ_4) represent the fluid particle displacements adjacent to the outer side of the two faceplates and (δ_2, δ_3) represent the porous material particle displacements adjacent to the inner side of the plates, respectively. With the transfer effect of the

mean flow accounted for, the fluid-structure coupling is described using the particle displacement continuity condition [31]. These particle displacements satisfy the Navier-Stokes equation for inviscid and irrotational fluid as

$$\frac{D^2 \delta_1}{Dt^2} = -\frac{1}{\rho_1} \frac{\partial p_1}{\partial z} \Big|_{z=0_1} \quad (2.111)$$

$$\frac{D^2 \delta_2}{Dt^2} = -\frac{1}{\rho_2} \frac{\partial p_2}{\partial z} \Big|_{z=h_1} \quad (2.112)$$

$$\frac{D^2 \delta_3}{Dt^2} = -\frac{1}{\rho_2} \frac{\partial p_2}{\partial z} \Big|_{z=h_1+H} \quad (2.113)$$

$$\frac{D^2 \delta_4}{Dt^2} = -\frac{1}{\rho_3} \frac{\partial p_3}{\partial z} \Big|_{z=h_1+H+h_2} \quad (2.114)$$

For an infinite uniform flat plate, the vibration displacement excited by a harmonic wave is given by [1]

$$w_j = w_{j0} e^{j\omega t - j[(k_p \cos \beta)x + (k_p \sin \beta)y]} \quad (2.115)$$

where $k_p = \omega/c_p$ denotes the transverse deflection wavenumber in the plate and c_p is the trace velocity of the transverse wave in the plate.

Since the porous material and the adjacent fluid are in good contact with the two faceplates, the displacements of the fluid particle and the porous material particle adjacent the two plates take the same form as that of the plates:

$$\delta_i = \delta_{i0} e^{j\omega t - j(k_{ix}x + k_{iy}y)} \quad (2.116)$$

Combining Eqs. (2.111), (2.112), (2.113), (2.114), (2.115), and (2.116) yields

$$\delta_{10} = -\frac{jk_{1z}(P_{1i} - P_{1r})}{\rho_1(\omega - vk_{1x})^2} \quad (2.117)$$

$$\delta_{20} = -\frac{jk_{2z}(P_{2i} - P_{2r})}{\rho_2\omega^2} \quad (2.118)$$

$$\delta_{30} = -\frac{jk_{2z}}{\rho_2\omega^2} (P_{2i}e^{-jk_{2z}H} - P_{2r}e^{jk_{2z}H}) \quad (2.119)$$

$$\delta_{40} = -\frac{jk_{3z}}{\rho_3\omega^2} P_{3i}e^{-jk_{3z}H} \quad (2.120)$$

Continuity of particle displacements dictates

$$\delta_{10} = w_{10} = \delta_{20}, \delta_{30} = w_{20} = \delta_{40} \quad (2.121)$$

Substitution of (2.115), (2.116), (2.117), (2.118), (2.119), and (2.120) into (2.121) leads to

$$P_{1r} = P_{1i} - \frac{j\rho_0(\omega - vk_{1x})^2 w_{10}}{k_{1z}} \quad (2.122)$$

$$P_{2i} = \frac{\rho_2 \omega^2}{2k_{2z} \sin(k_{2z}H)} (w_{10} e^{jk_{2z}H} - w_{20}) \quad (2.123)$$

$$P_{2r} = \frac{\rho_2 \omega^2}{2k_{2z} \sin(k_{2z}H)} (w_{10} e^{-jk_{2z}H} - w_{20}) \quad (2.124)$$

$$P_{3t} = \frac{j\rho_3 \omega^2 w_{20}}{k_{3z}} e^{ik_{3z}H} \quad (2.125)$$

The driving relation between the pressure difference across the plate and the plate response is

$$\begin{aligned} P_{1i} + P_{1r} - P_{2i} - P_{2r} &= Z_{p1} \dot{w}_{10} \\ P_{2i} + P_{2r} - P_{3t} &= Z_{p2} \dot{w}_{20} \end{aligned} \quad (2.126)$$

where Z_{p1} and Z_{p2} represent separately the plate impedances for the upper and lower faceplates (Fig. 2.22) and $\dot{w}_{j0} = i\omega w_{j0}$ ($j=1,2$). For infinite plates as considered in the present study, the plate impedance can be calculated as follows.

The governing equation of an infinite plate immersed in a fluid medium excited by a harmonic wave is given by

Upper faceplate:

$$(D_1 \nabla^4 - m_1 \omega^2) w_1(x, y) = p_1(x, y, 0) - p_2(x, y, h_1) \quad (2.127)$$

Lower faceplate:

$$(D_2 \nabla^4 - m_2 \omega^2) w_2(x, y) = -p_3(x, y, H + h_1 + h_2) + p_2(x, y, H + h_1) \quad (2.128)$$

The complex Young's modulus of the plate material is modified by the structure loss factor η as

$$E'_n = E_n (1 + j\eta_n), \quad n = 1, 2 \quad (2.129)$$

Accordingly, the complex bending stiffness is given by

$$D'_n = D_n (1 + j\eta) = \frac{E_n h^3 (1 + j\eta_n)}{12(1 - \nu^2)}, \quad n = 1, 2 \quad (2.130)$$

Substitution of (2.115) into (2.127) and (2.128) yields

$$w_{10} = \frac{p_1(x, y, 0) - p_2(x, y, h_1)}{D_1 k_p^4 - m_1 \omega^2} \frac{1}{e^{j\omega t - j(k_{1x}x + k_{1y}y)}} \quad (2.131)$$

$$w_{20} = \frac{p_2(x, y, H + h_1) - p_3(x, y, H + h_1 + h_2)}{D_2 k_p^4 - m_2 \omega^2} \frac{1}{e^{j\omega t - j(k_{2x}x + k_{2y}y)}} \quad (2.132)$$

Finally, the impedances of the upper and lower faceplates are obtained as

$$Z_{p1} = \frac{p_1(x, y, 0) - p_2(x, y, h_1)}{i\omega w_{10}} = jm_1\omega \left(1 - \frac{D_1\omega^2 \cos^4 \varphi_2}{m_1} \sqrt{\frac{\rho(\omega)}{K(\omega)}} \right) \quad (2.133)$$

$$\begin{aligned} Z_{p2} &= \frac{-p_3(x, y, H + h_1 + h_2) + p_2(x, y, H + h_1)}{j\omega w_{20}} \\ &= jm_2\omega \left(1 - \frac{D_2\omega^2 \cos^4 \varphi_2}{m_2} \sqrt{\frac{\rho(\omega)}{K(\omega)}} \right) \end{aligned} \quad (2.134)$$

2.3.3.3 Sound Transmission Loss

The transmission loss of sound across a double-leaf panel filled with porous material is defined as

$$STL = -10 \log_{10} \tau(f, \varphi_1, \beta) \quad (2.135)$$

where $\tau(\omega, \varphi_1, \beta)$ is the acoustic transmissivity given by

$$\tau(f, \varphi_1, \beta) = \frac{\rho_1 c_1}{\rho_2 c_2} \left| \frac{P_{3t}}{P_{1i}} \right|^2 \quad (2.136)$$

with

$$\begin{aligned} \frac{P_{1i}}{P_{3t}} &= \frac{-j\rho_3 k_{2z} \tan(k_{2z}H)}{2\rho_2(\rho_3 + k_{3z}(Z_{p2}/\omega) e^{-jk_{3z}H}) Z_{p1}} \\ &\quad \left[\left(\frac{\rho_2 \cos(k_{2z}H) + \rho_2 \frac{Z_{p1}}{Z_{p2}}}{\rho_2 \rho_3} \right) \left(\frac{Z_{p1} k_{1z} \omega + \rho_0(\omega - vk_{1x})^2}{jk_{2z} k_{1z} \omega \sin(k_{2z}H)} \right) \right. \\ &\quad \left. + \frac{\rho_2 \rho_0 \cos(k_{2z}H) (\omega - vk_{1x})^2}{jk_{2z} k_{1z} \omega \sin(k_{2z}H)} \right] \end{aligned} \quad (2.137)$$

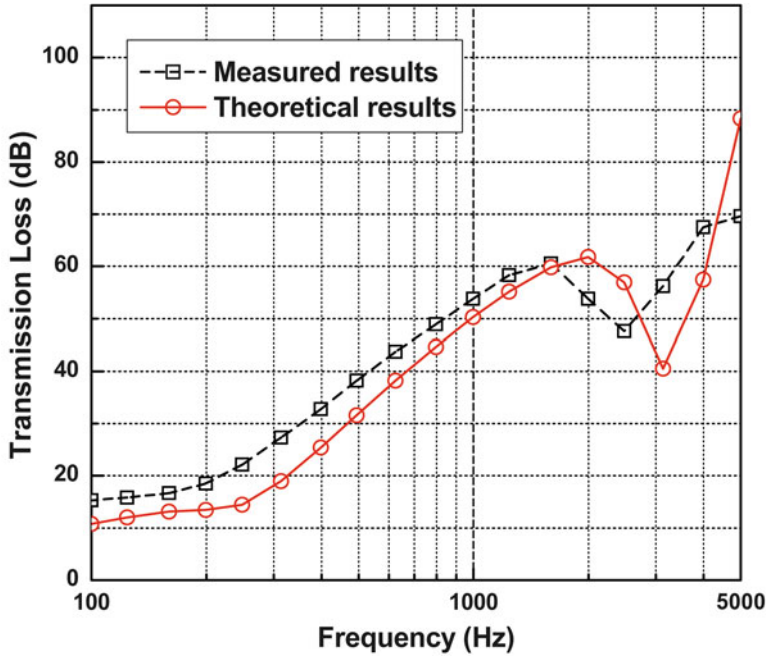


Fig. 2.23 Sound transmission loss of double-leaf panel separated by foam material in the absence of mean flow ($M = 0$): comparison between the present model predictions and experimental results by Pellicier et al. [93]

2.3.4 Validation of Theoretical Model

To validate the theoretical model presented in the previous section, the model predictions are compared with existing experimental results [93] in Fig. 2.23 for the case of no external mean flow. For experimental measurement, the sandwich structure is composed of two plywood plates with thickness of 8.5 mm and separated by an open-celled foam with thickness of 50 mm. The plywood has Young's modulus of 4.25 GPa, Poisson ratio 0.5, density 650 kg/m^3 , and loss factor 0.03. The foam material has a porosity of 93 % and a static flow resistivity of $55,000 \text{ Ns/m}^4$. The comparison is conducted by applying these parameters into our theoretical model and setting the Mach number to 0. Since the experimental results were obtained in a diffuse sound field, the model should be adjusted to calculate the *STL* in a diffuse sound field. The azimuth angle makes no sense to the *STL* in the absence of mean flow; in such case, Eq. (2.136) becomes

$$\tau(f, \varphi_1) = \frac{\rho_1 c_1}{\rho_2 c_2} \left| \frac{P_{3t}}{P_{1i}} \right|^2 \quad (2.138)$$

The diffuse transmissivity is then calculated by

$$\tau_{diff}(f) = \frac{\int_0^{\varphi_{lim}} \tau(f, \varphi) \sin \varphi \cos \varphi d\varphi}{\int_0^{\varphi_{lim}} \sin \varphi \cos \varphi d\varphi} \quad (2.139)$$

Accordingly, the sound transmission loss becomes

$$STL = -10 \log_{10} \tau_{diff}(f) \quad (2.140)$$

The results of Fig. 2.23 show that the theoretical model predictions for diffuse incidence ($\theta_{lim} = 78^\circ$) exhibit the same trend as that of the experimental measurements and, overall, the agreement is reasonable. However, there does exist some discrepancy between theory and experiment (e.g., the shift of the coincidence dip location in Fig. 2.23), which is attributed to the fact that finite-sized boundary conditions hold in the experiment measurements [93], while infinite size is assumed in the present model. Moreover, the imperfect connection between the plywood plates and foam material may increase the damping effect of the whole structure, causing the higher measured *STL* values than those predicted over the frequency range considered. Notice also that as the value of *STL* at the coincidence dip is also greatly affected by the damping of the whole structure, the measured value that includes the effect of structure material damping and boundary constraints damping is expected to be larger than the prediction which only considers the effect of structure material damping.

2.3.5 Influence of Porous Material and the Faceplates

Figure 2.24 compares the predicted transmission loss of a double-leaf panel filled with porous material with that of a double-leaf panel with air cavity having the same thickness as that of the porous material, for Mach number $M = 0.05$ and sound incident elevation angle $\varphi_1 = 30^\circ$, middle layer thickness $H = 0.08$ m, and azimuth angle $\beta_1 = 0^\circ$ (Fig. 2.22). As previously mentioned in the Introduction, the necessary condition for using the equivalent fluid theory is $f/R < 1$ kg/m³ [80]. Therefore, the highest frequency of the present analysis is limited to 24,000 Hz. In the following, the model predictions are presented within the frequency range of 0–10,000 Hz, for all the acoustic phenomena of interest can be shown within this range.

The results of Fig. 2.24 show that the presence of the filling porous material significantly weakens the plate-air-plate resonance dip of the system, due to the damping effect (or the absorption effect) of the porous material. Further, in the frequency range of 1,000–8,000 Hz, two kinds of acoustic phenomena exist for double-leaf panel with air cavity: antiresonance and standing-wave resonance. As illustrated schematically in Fig. 2.25, the plane sound wave propagates from point

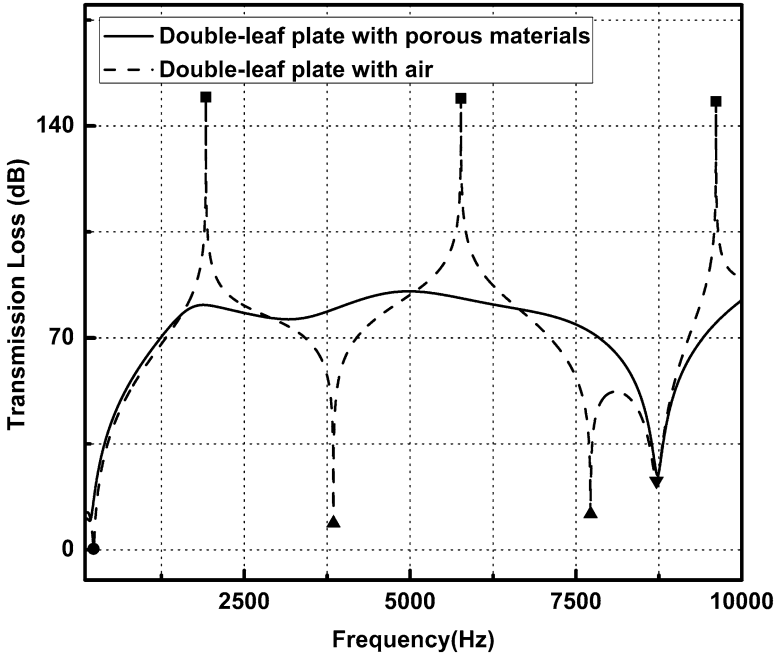
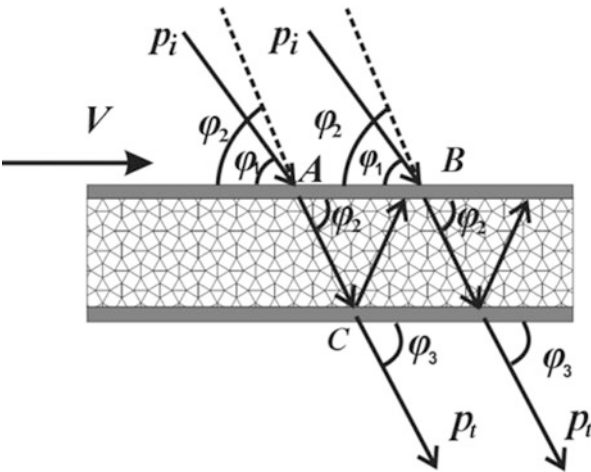


Fig. 2.24 Sound transmission loss of double-leaf panel for Mach number $M = 0.05$, loss factor $\eta = 0.01$, incident elevation angle $\varphi_1 = 30^\circ$, middle layer thickness $H = 0.08$ m, and azimuth angle $\beta_1 = 0^\circ$: ■ antiresonance, • “plate-air-plate” resonance, ▲ standing-wave resonance, ▼ coincidence dip

Fig. 2.25 Schematic of standing-wave resonance and antiresonance



A on the upper plate to point B on the lower plate; it then reflects to point C on the upper plate where it meets another incident sound wave. If the distance between the incident wave and the reflected wave is the odd number of the one-fourth wavelength of the sound wave, the interference between the two waves causes antiresonance. Alternatively, if the distance is the multiple of half the wavelength, the interference becomes standing-wave resonance. In comparison, for a double-leaf panel filled with porous material, since the energy intensity of the reflected wave has been significantly weakened due to the absorption of the porous material, both the standing-wave resonance and antiresonance almost disappear. Accordingly, the curve is relatively smooth in the frequency range of 1,000–8,000 Hz, as shown in Fig. 2.24. The absorption effect of the porous material exists in both the presence and absence of mean flow, which will be discussed in detail in Sect. 2.3.7 of this section.

It can also be observed from Fig. 2.24 that coincidence dip, at which the wavelength of the incident sound wave matches with the wavelength of the flexural bending wave in the plate, exists on both *STL* and frequency curves at the same frequency. This agrees well with Xin et al. [23, 94] and Wang et al. [59], both pointing out that the coincidence frequency is given by

$$f_c = \frac{c_2^2}{2\pi h \cos^2 \varphi_2} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \quad (2.141)$$

Combining Eqs. (2.99) and (2.108) with Eq. (2.141) yields

$$f_c = \frac{c_2^2}{2\pi h \cos^2 \varphi_2} \sqrt{\frac{12\rho(1-\nu^2)}{E}} = \frac{\omega^2}{2\pi h k_1^2 \cos^2 \varphi_1} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \quad (2.142)$$

which means that the frequency of the coincidence dip for the whole structure is independent of the presence of the porous material. However, the value of the *STL* at the coincidence dip is dependent on the panel damping, as shown in Fig. 2.26, which increases when the loss factor of the faceplate increases. Since the effective value of the loss factor is related with the damping of the attached porous material [25], it deduced that the extent of coincidence dip is significantly affected by the damping of the porous material.

The influence of faceplate thickness on structure transmission loss is shown in Fig. 2.27. As the thickness of the faceplates is increased, the frequency at which coincidence dip occurs decreases significantly, consistent with Eq. (2.141). However, the extent of the coincidence dip does not change significantly with increasing faceplate thickness.

2.3.6 Influence of Porous Material Layer Thickness

Since the presence of a porous material layer affects considerably the transmission of sound across a double-leaf panel, the effect of its thickness is further explored

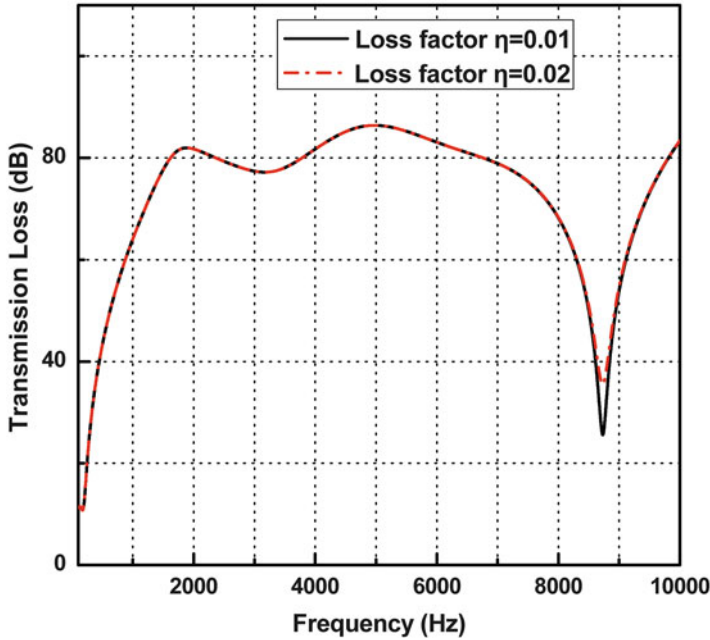


Fig. 2.26 Sound transmission loss of *double-leaf panel* filled with porous material for selected loss factors, with $\varphi_1 = 30^\circ$, $\beta_1 = 0^\circ$, $M = 0.05$, and $H = 0.08$ m

below. Figure 2.28 plots the predicted transmission loss as a function of frequency for selected porous layer thicknesses, with Mach number fixed at $M = 0.05$, sound incident elevation angle at $\varphi_1 = 30^\circ$, and azimuth angle at $\beta_1 = 0^\circ$. It is seen from Fig. 2.28 that at frequencies between the coincidence resonance and the plate-air-plate resonance, both the standing-wave resonance and antiresonance are increasingly weakened and the value of *STL* decreases as the porous layer thickness increases. This should remarkably be affected by the increase of the gap between the panels, which is also affected by the damping loss effect of the porous material within this frequency range [95]. In contrast, at frequencies near or above the coincidence resonance, since the porous material has a strong sound absorption capability at high frequencies, the *STL* may have reached its utmost value with a certain thickness of the porous material. Consequently, at relatively high frequencies, increasing the porous layer thickness further has negligible influence upon the transmission loss, as shown in Fig. 2.28.

2.3.7 Influence of External Mean Flow

Figure 2.29 presents the influence of external mean flow on *STL* across double-leaf panels with filling porous materials for $\varphi_1 = 30^\circ$, $H = 0.08$ m, and $\beta_1 = 0^\circ$. The

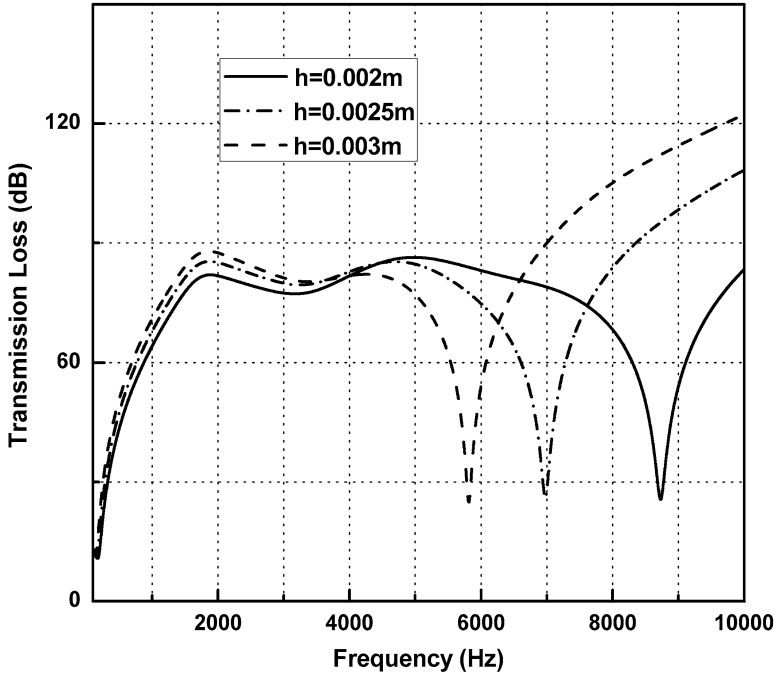


Fig. 2.27 Sound transmission loss of double-leaf panel filled with porous material for selected faceplate thicknesses, with $\varphi_1 = 30^\circ$, $\beta_1 = 0^\circ$, $M = 0.05$, and $H = 0.08$ m

results demonstrate that increasing the Mach number leads to noticeable changes of the *STL*: as the Mach number increases, the *STL* increases over a wide frequency range below the coincidence dip frequency. This is mainly caused by the added damping effect of the mean flow as suggested by Sgard et al. [15]. Therefore, the radiated acoustic power decreases with increasing Mach number of the mean flow, resulting in the increase of the *STL* value.

It is interesting to see from Fig. 2.29 that the amplitudes of the standing-wave resonance and antiresonance for the case of $M = 0.05$ do not change significantly in comparison with those when there is no mean flow. This implies that the absorption effect of the porous material is almost independent of the mean flow.

The effect of Mach number on the coincidence frequency of double-leaf structures is quantified in Fig. 2.30, for $\varphi_1 = 30^\circ$, $H = 0.08$ m, loss factor $\eta = 0.01$, and $\beta_1 = 0^\circ$. With increasing Mach number, the coincidence dip frequency increases almost proportionally. This is attributed to the fact that the refraction effect of external mean flow is noticeably enhanced as the Mach number is increased, which is confirmed by

$$f_c = \frac{\omega^2}{2\pi h k_1^2 \cos^2 \varphi_1} \sqrt{\frac{12\rho(1-\nu^2)}{E}} = \frac{c_1^2(1 + M \cos \varphi_1)^2}{2\pi h \cos^2 \varphi_1} \sqrt{\frac{12\rho(1-\nu^2)}{E}} \quad (2.143)$$

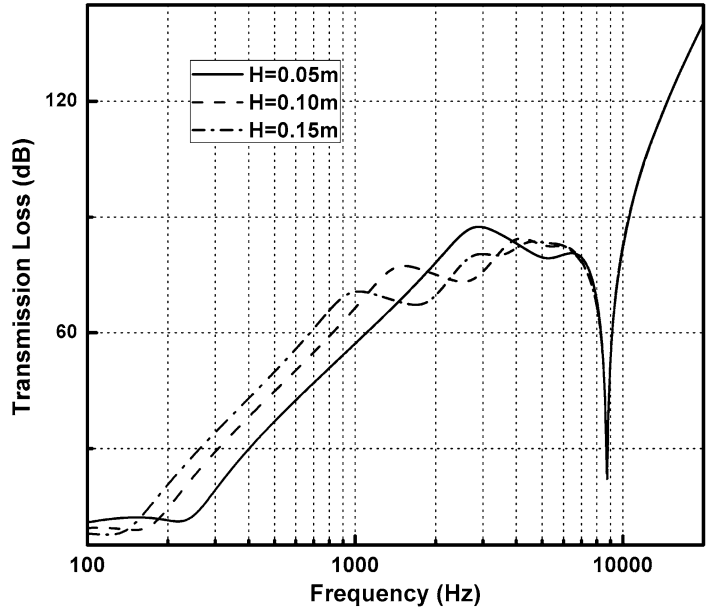


Fig. 2.28 Sound transmission loss of double-leaf panel plotted as a function of frequency for selected porous layer thicknesses, with Mach number $M = 0.05$, loss factor $\eta = 0.01$, incident elevation angle $\varphi_1 = 30^\circ$, and azimuth angle $\beta_1 = 0^\circ$

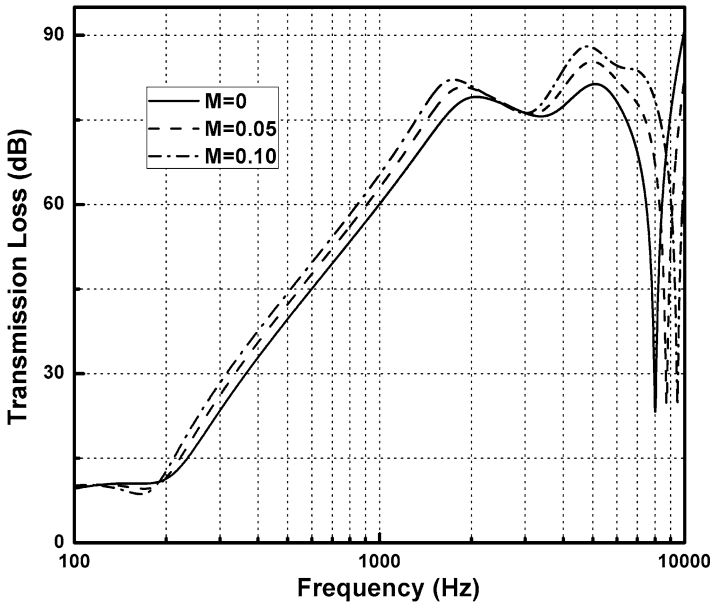


Fig. 2.29 Sound transmission loss of double-leaf panel filled with porous material for selected Mach numbers, with $\varphi_1 = 30^\circ$, loss factor $\eta = 0.01$, $\beta_1 = 0^\circ$, and $H = 0.08\text{ m}$

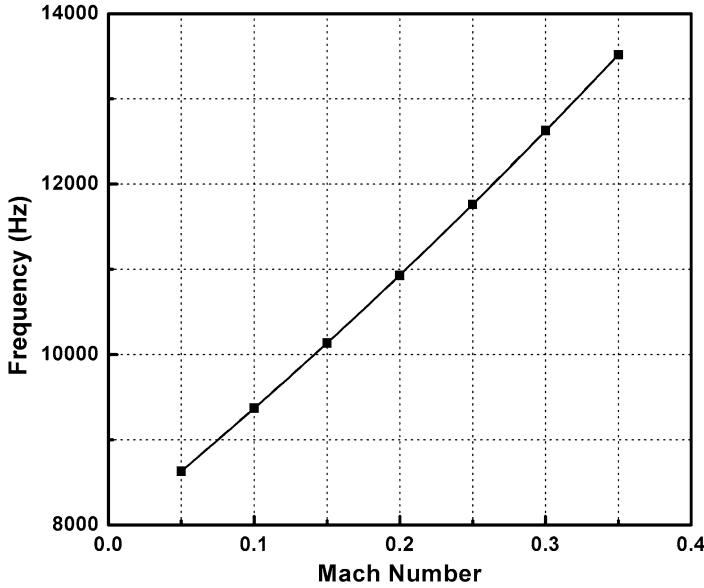


Fig. 2.30 Effect of Mach number on coincidence frequency of double-leaf panel filled with porous material, with $\varphi_1 = 30^\circ$, loss factor $\eta = 0.01$, $\beta_1 = 0^\circ$, and $H = 0.08$ m

Since the value of M considered is small, Eq. (2.143) may be transformed into the form below:

$$f_c = \frac{c_1^2(1 + M \cos \varphi_1)^2}{2\pi h \cos^2 \varphi_1} \sqrt{\frac{12\rho(1 - \nu^2)}{E}} \approx \frac{c_1^2(1 + 2M \cos \varphi_1)}{2\pi h \cos^2 \varphi_1} \sqrt{\frac{12\rho(1 - \nu^2)}{E}} \quad (2.144)$$

Equation (2.144) exhibits the same trend as that of the present model predictions.

2.3.8 Influence of Incident Sound Elevation Angle

Figure 2.31 plots the *STL* of a double-leaf panel with filling porous material as a function of frequency for selected incident elevation angles, with $M = 0.05$, $\beta_1 = 0^\circ$, and $H = 0.08$ m. The incident angle is seen to have an obvious effect on the transmission loss and the position of coincidence frequency. Between the plate-porous material-plate resonance and the coincidence dip frequency, the *STL* value increases when the angle from the axis normal to the plane of the panel decreases (i.e., the elevation angle increases). However, since the coincidence frequency increases with increasing elevation angle, the *STL* decreases in the frequency range above the coincidence frequency. Besides, when the incident angle is relatively

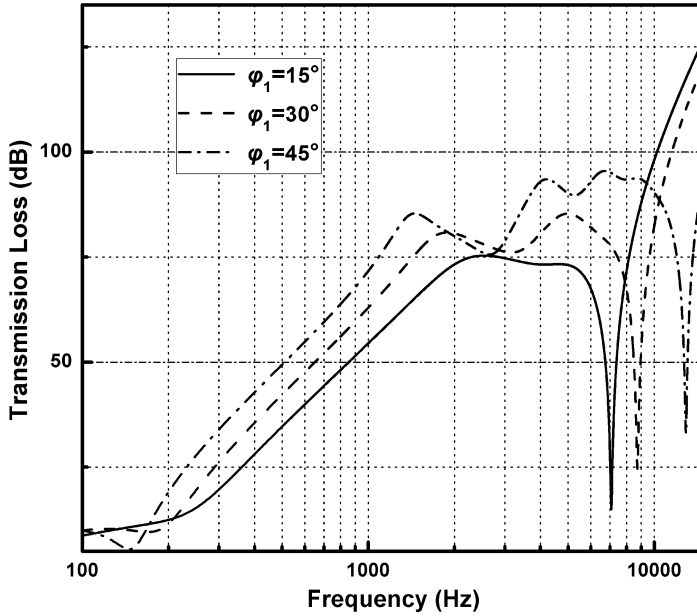


Fig. 2.31 Sound transmission loss of double-leaf panel filled with porous material for selected elevation angles, with $M = 0.05$, $\beta_1 = 0^\circ$, loss factor $\eta = 0.01$, and $H = 0.08$ m

small, the effect of plate-porous material-plate resonance can be ignored since it does not cause an apparent dip in the *STL* versus frequency curve. In sharp contrast, the influence of plate-porous material-plate resonance on transmission loss will be reinforced as the incident angle is increased.

The predicted effect of sound incidence angle on the coincidence frequency is illustrated further in Fig. 2.32. The coincidence frequency increases remarkably with increasing incident elevation angle, agreeing well with the coincidence frequency formula of Eq. (2.142). Also, it can be seen from Fig. 2.32 and Eq. (2.142) that the coincidence dip tends to infinity when the incident angle approaches $\pi/2$, implying that the structure considered here has the best sound insulation ability for normal incident sound.

2.3.9 Influence of Sound Incident Azimuth Angle

It can be seen from Fig. 2.33 that the sound transmission loss of the panel is affected by the azimuth angle of the incident sound, particularly in the high-frequency regime. The coincidence frequency increases as the azimuth angle is increased. The influence of azimuth angle on sound transmission loss is mainly caused by the refraction effect of the mean flow. The results presented in Table 2.2 illustrate

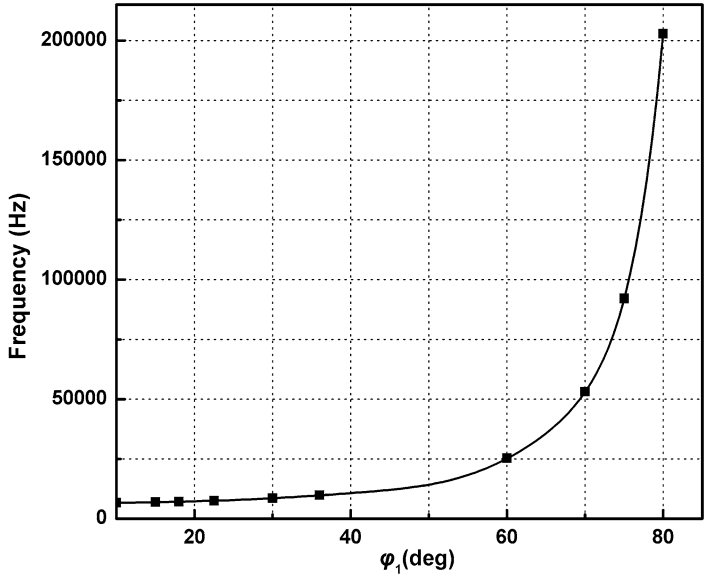


Fig. 2.32 Effect of sound incident angle on coincidence frequency of double-leaf panel filled with porous material, with $M = 0.05$, loss factor $\eta = 0.01$, $\beta_1 = 0^\circ$, and $H = 0.08$ m

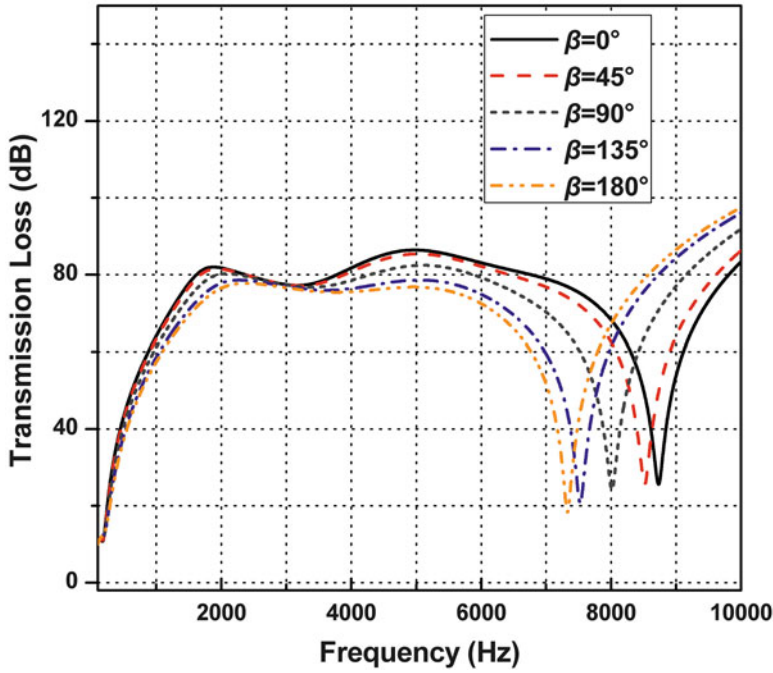


Fig. 2.33 Sound transmission loss of double-leaf panel filled with porous material for selected azimuth angles, with $M = 0.05$, $\varphi_1 = 30^\circ$, loss factor $\eta = 0.01$, and $H = 0.08$ m

Table 2.2 Comparison of coincidence frequency for different azimuth angles and Mach numbers

	$M = 0$	$M = 0.05$	$M = 0.10$	$M = 0.15$
f_c (Hz) for $\beta = 0^\circ$	8,020	8,730	9,641	10,230
f_c (Hz) for $\beta = 45^\circ$	8,020	8,517	9,026	9,553
Difference	0	213	615	677

that when the Mach number is increased, the coincidence frequency difference between different azimuth angles is enlarged, which also proves that the refraction effect is enhanced by the increase of Mach number as mentioned in Sect. 2.3.7. As anticipated, since the double-leaf panel is isotropic and homogenous, the sound transmission loss should not change with the azimuth angle when the fluid media are static, as warranted by the results of Table 2.2.

2.3.10 Conclusion

A theoretical model has been developed for sound transmission across infinite double-leaf panels filled with porous absorptive materials in the presence of external mean flow. Based on the model predictions, the following conclusions are drawn:

1. The presence of external mean flow has negligible influence upon the sound absorption ability of the filling porous material. Besides, the damping effect of the porous material affects significantly the magnitude of *STL* at coincidence dip, and the same can be said for the material loss factor of the faceplate material.
2. The added damping effect of the external mean flow prevails at frequencies below the coincidence dip frequency, enhancing the transmission loss as the Mach number is increased. Also, since the radiated acoustic power decreases with increasing Mach number, the peak value of the *STL* increases with increasing Mach number. In addition, the refraction effect of the external mean flow causes an upward shift of the coincidence frequency when the Mach number is increased.
3. The transmission loss of double-leaf panels filled with porous materials is considerably affected by sound incident elevation angle. Increasing the incident elevation angle (i.e., decreasing the angle from the axis normal to the plane of the panel) leads to enhanced transmission loss at frequencies below the coincidence dip as well as reduced transmission loss above the coincidence dip. The azimuth angle of the incident sound also has an influence on the sound transmission loss of the panel. The coincidence frequency decreases with increasing azimuth angle due to the refraction effect of the mean flow.

Appendix

As stated in Sect. 2.2.7, the simple closed-form formulae, i.e., Eqs. (2.91), (2.92), (2.93), and (2.94), are applied to validate the proposed aero-acoustic-elastic theoretical model in Sect. 2.2, because these formulae are developed independent of the theoretical model. The physical nature of the four acoustic phenomena based on which the closed-form formulae are derived is presented below (Fig. 2.34).

Mass-Air-Mass Resonance

In the absence of the external mean flow, the formula for predicting the mass-air-mass resonance frequency of a double-leaf aeroelastic panel has been presented in Refs. [3, 5, 47, 73]. The radially outspreading bending wave in the panel caused by the incident sound leads to the highly directional sound radiation. As a result, the mass-air-mass resonance strongly depends on the incident angle. When the mass-air-mass resonance occurs, the two panels with the air cavity in between behave like a mass-spring-mass system, with the stiffness of the air cavity given by $\rho_2 c_2^2 / (H \sin^2 \varphi_1)$. However, due to the refraction effect of the mean flow as shown in Fig. 2.34, the incident angle φ_1 in the presence of the mean flow will be changed to φ_2 before the sound penetrates through the incident panel. In other words, the mean flow case is equivalent to the case when the sound is incident on the panel with angle φ_2 in the absence of the mean flow (denoted by the dash lines in Fig. 2.34). Accordingly, the stiffness of the air cavity is changed to $\rho_2 c_2^2 / (H \sin^2 \varphi_2)$, and the eigenvalue equation of the equivalent mass-spring-mass vibration system becomes

$$\begin{vmatrix} \frac{\rho_2 c_2^2}{H \sin^2 \varphi_2} - \omega^2 m_1 & \frac{-\rho_2 c_2^2}{H \sin^2 \varphi_2} \\ \frac{-\rho_2 c_2^2}{H \sin^2 \varphi_2} & \frac{\rho_2 c_2^2}{H \sin^2 \varphi_2} - \omega^2 m_2 \end{vmatrix} = 0 \quad (2.145)$$

Solving Eq. (2.145) leads to the mass-air-mass resonance given in Eq. (2.91).

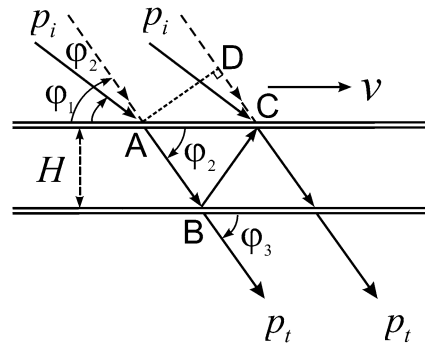


Fig. 2.34 Sketch of sound transmission through double-leaf aeroelastic panel in the presence of external mean flow

Standing-Wave Attenuation

In the presence of external mean flow, the phenomenon of standing-wave attenuation in a double-panel system is caused by the destructive interference between two meeting waves, which occurs only when the distance difference between the routes that the two intervening waves pass through is the odd number of one quarter of the incident sound wavelength. Under such conditions, the interference between the positive-going wave and the negative-going wave tends to be destructive when the sound is transmitting through the partition, resulting in maximum sound reduction. As shown in Fig. 2.34, the plane sound ray comprised of a set of harmonic sound waves with the same vibration phase obliquely impinges on the incident panel. There indeed exists a case when one sound ray is incident on the upper panel at point *A* and transmits through the air cavity onto the bottom panel at point *B* and its reflected portion back to the upper panel at point *C* meets another sound ray from the external incident side, i.e., the interference between the two sound waves has occurred.

As is well known, the destructive effect between two harmonic waves occurs only when their vibration phases differ by odd numbers of $\pi/2$. In the case considered here (Fig. 2.34), the distance difference between the routes that the two intervening waves pass through is the only cause of the phase difference. For convenience, the equivalent case of sound incident with angle φ_2 in the absence of mean flow (dash lines in Fig. 2.34) is used to represent the case of sound incident with angle φ_1 in the presence of mean flow (solid lines in Fig. 2.34). The distance difference between the routes that the two sound waves pass through is such that

$$\overline{AB} + \overline{BC} - \overline{CD} = \frac{2H}{\sin \varphi_2} - 2H \cot \varphi_2 \cos \varphi_2 = 2H \sin \varphi_2 \quad (2.146)$$

The occurrence of the destructive effect requires that

$$H \sin \varphi_2 = \frac{(2n-1)\lambda}{4} \quad (n = 1, 2, 3 \dots) \quad (2.147)$$

where $\lambda = c/f$ is the wavelength in air. Equation (2.92) for the standing-wave attenuation frequency follows from Eq. (2.147).

Standing-Wave Resonance

As the counterpart of the standing-wave attenuation, the standing-wave resonance is also caused by the interference effect between two meeting waves, which is however constructive rather than destructive. It appears when the distance difference between the routes that the two intervening waves pass through is the multiple of

the half wavelength of the incidence sound wave. In such cases, the constructive interference between the positive-going and negative-going waves leads to an enhanced transmission through the double-panel partition. The condition for the appearance of the standing-wave resonance is given by

$$H \sin \varphi_2 = \frac{n\lambda}{2} \quad (n = 1, 2, 3 \dots) \quad (2.148)$$

With the relation $\lambda = c/f$, Eq. (2.148) can be readily converted to Eq. (2.93).

Coincidence Resonance

Similar to the standing-wave attenuation and resonance, the coincidence resonance is also an interference effect between two intervening waves by physical nature. The difference between the two different types of acoustic phenomenon is that the coincidence resonance is caused by wave interference between the sound wave and the panel flexural bending wave. As shown in Fig. 2.34, there exists a situation when the panel flexural bending wave at point A (excited by the incident sound ray at this point) rapidly outspreads to point C, where it matches another sound ray from the external incident side. The premise for this matching is that the flexural bending wave in the aeroelastic panel propagates faster than sound propagation in air, requiring that the former transmits a larger distance than the latter during the same time period (i.e., $\overline{AC} > \overline{DC}$ in Fig. 2.34), and hence

$$\frac{\overline{AC}}{c_t} = \frac{\overline{DC}}{c} \quad (2.149)$$

This premise is in general satisfied in practice. Consequently, the constructive interference between the two waves results in the coincidence resonance. From the governing equation of panel vibration, we have

$$c_t = \sqrt[4]{\frac{D\omega^2}{m}} \quad (2.150)$$

Substitution of Eq. (55) and the relation $\overline{DC} = \overline{AC} \cdot \cos \varphi_2$ into Eq. (2.149) leads to Eq. (2.94) for the coincidence resonance frequency.

Note that the coincidence resonance will occur in the bottom panel as well when the upper panel generates coincidence resonance. In other words, the incident sound will be enhanced twice, initially by the upper panel and followed by the bottom panel, when the condition for the occurrence of the coincidence resonance is satisfied.

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