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Active Control of the Flow-Induced Noise Transmitted Through a Panel

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Active control of sound transmitted through an elastic panel when excited by a turbulent boundary layer is investigated. The motivation is the control within an aircraft cabin of the flow-induced noise due to the wall-pressure fluctuations over the fuselage. The excitation is random and so the plate velocity and the radiated sound pressure must be described by spectral densities. These quantities can be obtained from an analysis of the response of the system to a harmonic deterministic excitation and a statistical model for the turbulent boundary layer. Criteria are discussed under which the cross-modal coupling of the structural modes can be neglected when excited by a turbulent boundary layer. When considering subsonic turbulent flows, parametric studies show that each structural mode radiates sound independently. Hence, a suitable strategy for the active structural acoustic control of the sound power transmitted through the panel would be independent feedback control of each structural mode of the system in the low-frequency domain. The performance of this active control strategy is compared to that obtained by controlling the radiation modes of the panel for different numbers of control channels.

I. Introduction

MANY types of high-speed vehicles suffer from aerodynamically induced noise caused by a turbulent boundary layer (TBL). For example, the TBL developed over an aircraft fuselage can generate high levels of noise inside the cabin during cruise conditions and, thus, directly affects the comfort of the passengers. For supersonic aircraft, the levels of sound and vibration induced by the airflow are so high that they can threaten the integrity of many electronic and mechanical instruments used for flight control. As well as aircraft, trains and even cars can have this problem because other sources of noise, due to engines, wheel-road contact, or onboard equipment, have been dramatically reduced. Another point is that the flow-induced noise increases more rapidly, with respect to the vehicle velocity, than other noise sources.[1]

The passive insulation of flow-noise transmission has been successful to a certain extent, but requires dissipative materials whose weight can be a major limitation for most applications.[2] Moreover, the passive approach is not very effective in reducing noise and vibration in the low-frequency domain.[3]

It may be possible to overcome these problems with active control systems having either acoustic or structural actuators. Indeed, both approaches have been successfully implemented for the control of tonal acoustic disturbances in propellers.[4] Also, low-frequency vibration of the fuselage structure of jet aircraft due to tonal disturbances (engine structure and airborne excitations) is often controlled by means of adaptive tunable vibration absorbers (ATVA).[5]

However, the active control of flow-induced noise is still the subject of current research.

1) There is a lack in understanding the detailed physics of the vibroacoustics phenomena involved, that is, the excitation due to a TBL, the flow noise transmission through the structure, and the flow noise radiation inside a cavity.

2) The airflow noise is a random phenomenon from which it is difficult to obtain a reasonable number of time-advanced reference signals. Therefore, active control systems based on feedback strategies are generally required.

In this paper, we have restricted ourselves to a consideration of collocated structural actuators and sensors, and so, even if a feedforward control arrangement were used with feedback cancellation, the block diagram would be identical to the internal model control architecture of a feedback controller.[6] If acoustic actuators had been used or we had considered double-panel systems, it may have been possible to obtain some time-advanced information from sensors closer to the primary disturbance than the secondary actuator, and a feedforward arrangement may have been advantageous.[7,8]

The objectives of this paper are twofold: first, to investigate the excitation and sound radiation mechanism of a panel in contact with a turbulent airflow and, second, to provide guidelines about the effectiveness of various strategies of active structural acoustic control (ASAC).

This study is organized as follows. Section II reviews the main models for boundary-layer noise in flows over smooth walls and justifies why a Corcos-like model is most suitable at high subsonic Mach numbers. Section III presents the assumptions of the problem and describes how the stochastic properties of the structural response depends on the TBL excitation through the response of the panel to a harmonic excitation scaled on the wave number of each contributing eddy. Section IV is concerned with the modal formulation of the radiated sound power. Section V describes the active control of sound radiation. Section VI presents simulation results on the active control of the sound power radiated by a vibrating panel excited by a TBL. Section VII is the conclusion.

II. Models for Turbulent Wall-Pressure Fluctuations

A large number of empirical and semitheoretical models have been developed to describe the wall-pressure fluctuations beneath a TBL on a smooth wall.[9] It is usually assumed that the surface pressure fluctuations are not modified by the vibrations of the flexible wall. The pressure developed on the structure is, therefore, the pressure that would be observed on a rigid structure, also called the blocked pressure p_b. One of the first models was introduced by Corcos,[10] and subsequent improvements were proposed, for instance, by Chase[11] and Ffowcs Williams.[12]

As shown in Fig. 1, the mean flow is assumed to travel in the y direction and the spanwise direction is denoted by x. For a fully developed TBL exciting a flat plate, the random wall-pressure field is stationary in time and homogeneous in space up to order two. Its stochastic properties can, therefore, be described in terms of the wall-pressure wave number-frequency spectrum $S_{p_b p_b}$, defined as the space–time Fourier transform of the autocorrelation function $R_{p_b p_b}$ of the turbulent excitation.

The experimental database from wind-tunnel measurements has enabled the formulation of empirical models for the excitation.

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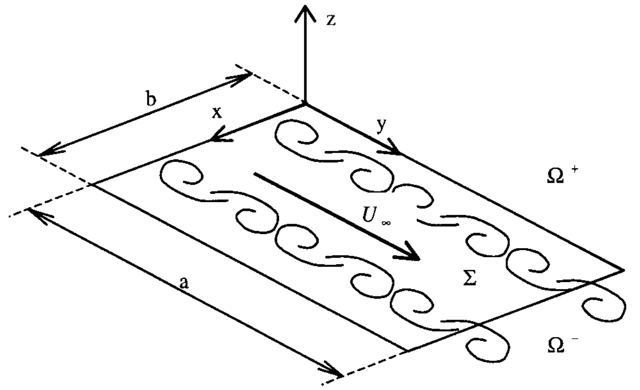

Fig. 1 Flat panel excited on one side by a turbulent airflow with the freestream velocity U_∞ .

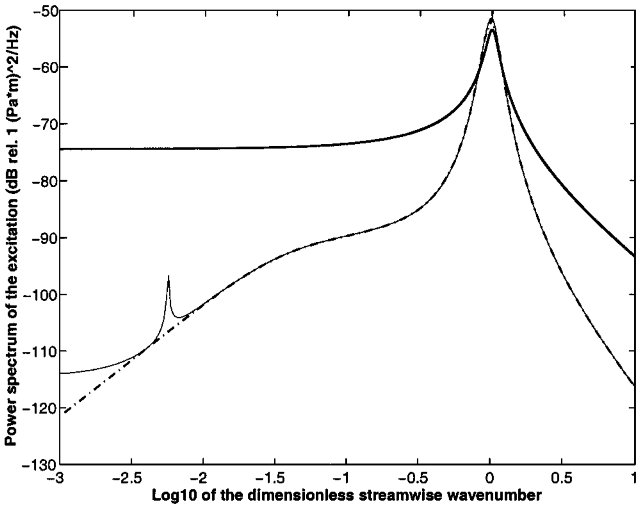

Fig. 2 Excitation spectrum $S_{p_b p_b}(0, k_y; \omega)$ as a function of the logarithm of the dimensionless longitudinal wave number $k_y U_c/\omega$; ——, Corcos model; –·–, Chase model (incompressible); and ——, Chase model (compressible).

The Corcos model assumes that the autocorrelation function can be expressed in a separable form with respect to the streamwise and spanwise directions.[10] In the hydrodynamic domain ($|\boldsymbol{k}| \gg \omega/c_0$) and in the neighborhood of the convective ridge ($k_y \sim \omega/U_c, k_x \sim 0$), the following approximation for the excitation spectrum has been proposed by Corcos:

$$S_{p_b p_b}(\boldsymbol{k}; \omega) = \Phi_0(\omega)\left\{L_y\big/\pi\left[1 + L_y^2(k_y - \omega/U_c)^2\right]\right\}$$

$$\times L_x\big/\pi\left(1 + L_x^2 k_x^2\right), \qquad L_y \approx 10U_c/\omega$$

$$L_x \approx 1.3U_c/\omega, \qquad U_c \approx (0.5 - 0.7)U_\infty \tag{1}$$

where L_y and L_x are the streamwise and spanwise correlation scales, U_c is the convection velocity, c_0 is the sound speed, and Φ_0 the point pressure spectrum, an approximation of which valid up to high Reynolds number is given by Efimtsov.[13]

A typical Corcos wave number spectrum in the streamwise direction is shown by the bold curve (Fig. 2). The peak (the convective ridge) occurs for streamwise wave numbers of order ω/U_c, that is, the principal energy fluctuations comes from the boundary-layer eddies scaled on U_c/f, where $f = \omega/2\pi$. This delimits the viscous region ($k_y \gg \omega/U_c$) from the hydrodynamic domain ($\omega/c_0 \ll k_y \ll \omega/U_c$). The major limitations of the Corcos model are, first, the assumption that the correlation scales do not depend on the boundary-layer thickness δ and, second, it fails to exhibit the theoretically required $(|\boldsymbol{k}|\delta)^2$ dependency when $|\boldsymbol{k}|\delta \ll 1$, leading to an overestimation (by more than 20 dB) of the low wave number levels for the excitation spectrum.

On the thin curve (Fig. 2), a second ridge, not described by the Corcos model, appears at the edge of the acoustic domain ($|\boldsymbol{k}| \ll \omega/c_0$) for $k_y = \omega/c_0$. It is described through the semitheoretical model derived by Chase in 1987 to account for spectral elements of the flow noise down to and near the acoustic wave number, where compressible effects cannot be neglected.[11] Though this acoustic ridge has not yet been confirmed by reliable datasets, the Chase model reasonably agrees with measurements within a wide range of subconvective wave numbers, down to $k_y \approx 0.4/\delta^*$, where $\delta^* \approx \delta/8$ is the boundary-layer displacement thickness.[14]

In summary, the Corcos and Chase models are complementary: the Corcos spectrum of the excitation provides a good estimation for the wall-pressure fluctuations levels at and near the convective peak, which is of fundamental importance for aircraft boundary layers (high subsonic Mach number) and so is used in the simulations to be described; on the other hand, the Chase spectrum provides a good description of experimental data for the subconvective domain down to the acoustic ridge. It is, therefore, more suitable for low-speed flow applications, where the strongest flow–structure interaction occurs in the low wave number region.

III. Statement and Modeling of the Problem

A. Characteristics of the Physical Model

Most models describing the vibroacoustic response of aircraft structures excited by a TBL have considered an array of uncorrelated simply supported panels vibrating individually.[15,16] This model has been confirmed as representative by extensive flight-test measurements carried out on the forward structure of an airplane fuselage.[17,18] These data have shown that, from 400 Hz to 5 kHz, the boundary layer excites the fuselage in such a way that the flow-induced vibrations are only correlated over a single fuselage bay.

A second point concerns the geometry of the modeled subsystem. The adjacent bays are set in a cylindrical fuselage, and the effects of curvature have to be considered. A recent analytical study has shown that, for subsonic applications, the influence of the panel curvature on the pressure inwardly radiated can be neglected only if the surrounding inner surface is sufficiently hard to appear as a baffle, but, at the same time, sufficiently absorbing to neglect the returned sound waves due to the curvature.[19] These competing effects seem, a priori, difficult to achieve in aeronautical applications, but the influence of curvature still appears negligible when compared with the influence of in-plane stresses acting on the boundaries of the panel.[15,20] These membrane tensions are due to the cabin pressurization and lead to an increase of the fundamental resonance frequency of each bay by a factor of up to about 7. Hence, the main physical characteristics of our problem are retained by considering the simplified, but relevant, model of a simply supported flat plate stressed by tension forces.

The third point is related to the modeling of the structural damping effects. We will introduce an equivalent damping coefficient ξ, the damping ratio, which accounts for several effects: the internal or hysteretic damping, the boundary damping due to the friction in the joint edges or due to the energy lost by the panel through its elastic boundaries, the friction damping for multilayered composite panels, etc. We will consider in our simulations two characteristic cases: an aluminum panel, which accounts both for the hysteretic damping and the energy losses through its boundaries (case a, $\xi = 0.01$), and a panel, which accounts for the damping effect of a trimmed panel, that is, the dissipation due to the insulating material placed between the interior and the exterior stringed frame (case b, $\xi = 0.05$) (Refs. 21 and 22).

B. System Under Study

Under the hypotheses discussed in the preceding sections, we consider a thin elastic plate occupying the domain Σ of the ($z = 0$) plane, as shown in Fig. 1. For simplicity, the plate Σ is assumed to be homogeneous and isotropic. Therefore, we model its motion by using the Kirchhoff thin-plate theory. The rectangular baffled plate, with thickness h, is simply supported along its boundaries and separates two half-spaces Ω^+ and Ω^- containing a perfect gas.

Let us consider a high subsonic mean flow in the exterior domain Ω^+: A TBL develops at the interface between the fluid and the baffled plate and is assumed fully developed when exciting the plate. The wall-pressure fluctuations due to this TBL induce the vibration of the plate, which radiates acoustic energy in both fluid domains.

Because the acoustic fluid load mainly depends on the mechanical properties of the fluid in the immediate neighborhood of the plate, where the fluid flow velocity is close to zero, we can reasonably assume that sound pressure inwardly radiated is not significantly affected by the external fluid flow. Hence, the plate is excited by the turbulent blocked pressure p_b and by the acoustic pressure fluctuations generated by the plate motion in the absence of turbulence. The vibroacoustic problem, thus, reduces to evaluating the response of the plate surrounded by a gas at rest and excited by the turbulent wall-pressure fluctuations.

The half-space Ω^- corresponds to the interior of the aircraft and is assumed anechoic, which is a reasonable assumption above a few hundred hertz in a well-damped enclosure such as an aircraft passenger cabin, but acoustic resonances would be important below this frequency.[23]

C. Derivation of the Spectral Densities Involved in the Vibroacoustic Problem

The plate is excited by a wall-pressure field, which depends randomly in both the time and space variables, and so its response is a random process characterized by a power spectral density. Two quantities of interest are the kinetic energy of the plate and the sound power inwardly radiated. It can be shown that the spectral densities of both the kinetic energy S_V and the sound power radiated in Ω^-, S_{P^-}, are given by[24]

$$S_V(\omega) = \frac{\rho_p h}{2} \iint_\Sigma S_{v_\omega v_\omega}(x, y, 0; \omega)\,\mathrm{d}x\,\mathrm{d}y \tag{2}$$

$$S_{P^-}(\omega) = \frac{1}{2}\Re\left[\iint_\Sigma S_{p_\omega^- v_\omega}(x, y, 0; \omega)\,\mathrm{d}x\,\mathrm{d}y\right] \tag{3}$$

where $S_{v_\omega v_\omega}(x, y, 0; \omega)$ is the power spectral of the velocity on the plate and $S_{\bar{p}_\omega v_\omega}(x, y, 0; \omega)$ is the cross-spectral density between the plate velocity and the pressure radiated immediately in front of the plate in Ω^-. These are related to the wave number transforms of the corresponding quantities by

$$S_{\alpha\beta}(\boldsymbol{x}; \omega) = \frac{1}{4\pi^2}\int_{-\infty}^{+\infty}\int_{-\infty}^{+\infty} \alpha_\omega(\boldsymbol{x}; \boldsymbol{k}) S_{p_b p_b}(\boldsymbol{k}; \omega)\beta_\omega^*(\boldsymbol{x}; \boldsymbol{k})\,\mathrm{d}^2\boldsymbol{k} \tag{4}$$

so that

$$S_{\alpha\beta}(\boldsymbol{x}; \omega) = \iint_\Sigma \iint_\Sigma \hat{\alpha}_\omega(\boldsymbol{x}-\boldsymbol{x}') S_{p_b p_b}(\boldsymbol{x}'-\boldsymbol{x}''; \omega) \times \hat{\beta}_\omega^*(\boldsymbol{x}-\boldsymbol{x}'')\,\mathrm{d}^2\boldsymbol{x}'\,\mathrm{d}^2\boldsymbol{x}'' \tag{5}$$

where $\alpha_\omega(\boldsymbol{x}; \boldsymbol{k})$ and $\beta_\omega(\boldsymbol{x}; \boldsymbol{k})$ are the contributions to the pressure and/or the velocity response of the fluid-loaded plate at a point $\boldsymbol{x} = (x, y, 0)$ of Σ from a boundary layer eddy of wave number $\boldsymbol{k}$ and frequency ω. Here $\hat{\alpha}_\omega$ and $\hat{\beta}_\omega$ are the wave number Fourier transform of α_ω and β_ω. In Eq. (5), $\hat{\alpha}_\omega(\boldsymbol{x}-\boldsymbol{x}')$ and $\hat{\beta}_\omega(\boldsymbol{x}-\boldsymbol{x}'')$ are the harmonic system response of the plate at a point $\boldsymbol{x}$ when excited by point forces at $\boldsymbol{x}'$ or $\boldsymbol{x}''$, the amplitudes of which are correlated by the separation scales in the streamwise and spanwise directions as follows:

$$S_{p_b p_b}(\boldsymbol{x}-\boldsymbol{x}'; \omega) = \boldsymbol{\Phi}_0(\omega)\exp(-|x-x'|/L_x) \times \exp(-|y-y'|/L_y)\exp[j\omega(y-y')/U_c]$$

Spatial Fourier transform of this expression leads to the excitation spectrum given in Eq. (1).

At this stage, a choice has to be made between the space–frequency or the wave number–frequency formulation of the problem. Using the wave number approach, recent models for turbulence can easily be taken into account.[9] With a Corcos-like model,[10] for the excitation, it even leads to an analytical expression for the response of a simply supported panel.[24] Furthermore, with this approach, the response of the structure excited by a turbulent boundary layer can be interpreted as a shaped signal $S_{p_b p_b}$ passed through a wave vector filter, whose transfer function is Green's kernel of the system response to a harmonic excitation.[24] A direct physical interpretation of the results can, therefore, be obtained in terms of a wave number filtering model.

In this section, we have shown that part of the problem is to determine the response of the system to a set of harmonic excitations. An expansion of the harmonic response of the fluid-loaded panel as series of its in vacuo eigenmodes (modal method) is one of the most efficient tools for solving this problem because the in vacuo eigenmodes of the structure are rather close to the light fluid-loaded structure ones. In the case of a general fluid loading, however, a resonance mode series expansion of the response would represent the most suitable solution.[25]

IV. Modal Formulation

A lower limiting frequency has been estimated above which the fluid-loading cross-coupling terms can be neglected in the governing modal equations (satisfied by Green's kernel v_ω) with respect to the diagonal terms that are dominated by the structural properties of the panel.[24] This criterion leads to a minimum frequency that is lower than the fundamental frequency of the panel used in our configuration. Under the so-called diagonal approximation, an explicit expansion for the harmonic response of the fluid-loaded panel as a series of its structural modes is readily obtained.[24] The velocity $v_\omega(x, y, 0; \boldsymbol{k})$ of the simply supported panel is then specified in terms of the amplitudes v_{mn} of an infinite set of structural modes, which are due to a pressure field of wave number $\boldsymbol{k}$, as follows:

$$v_\omega(x, y, 0; \boldsymbol{k}) = j\omega \sum_{m=1}^{+\infty}\sum_{n=1}^{+\infty} v_{mn}(\boldsymbol{k}; \omega) W_{mn}(x, y)$$

$$v_{mn}(\boldsymbol{k}; \omega) = \frac{F_{mn}(\boldsymbol{k})}{\rho_p h\left[\omega_{mn}^2 - \omega^2 + 2j\xi\omega_{mn}\omega\right]} = F_{mn}(\boldsymbol{k}) a_{mn}(\omega)$$

$$W_{mn}(x, y) = \frac{2}{\sqrt{ab}}\sin\left(\frac{m\pi x}{b}\right)\sin\left(\frac{n\pi y}{a}\right) \tag{6}$$

where W_{mn} are the normalized structural modes of the plate, ω_{mn} is the corresponding natural frequency, and ρ_p is the density of the plate material.

Using Kirchhoff thin-plate theory, the natural frequencies, for a simply supported tensioned plate, are given by[26]

$$\omega_{mn}^2 = \frac{1}{\rho_p h}\left\{\frac{Eh^3}{12(1-\upsilon^2)}\left[\left(\frac{m\pi}{b}\right)^2 + \left(\frac{n\pi}{a}\right)^2\right]^2 + N_m\left(\frac{m\pi}{b}\right)^2 + N_n\left(\frac{n\pi}{a}\right)^2\right\}$$

E and υ are, respectively, Young's modulus and the Poisson coefficient of the plate, and N_m and N_n are the lateral and longitudinal plate tensions.

Because the panel is driven, for each wave number $\boldsymbol{k}$, by a unitary pressure distribution $e^{j\boldsymbol{k}\cdot\boldsymbol{x}}$, the modal excitation term F_{mn} can be defined as

$$F_{mn}(\boldsymbol{k}) = \frac{2}{\sqrt{ab}}\int_0^b\int_0^a \sin\left(\frac{m\pi x}{b}\right)\sin\left(\frac{n\pi y}{a}\right) \times \exp[j(k_x x + k_y y)]\,\mathrm{d}x\,\mathrm{d}y = \left\langle e^{j\boldsymbol{k}\cdot\boldsymbol{x}}, W_{mn}^*(\boldsymbol{x})\right\rangle_\Sigma$$

In practice, a finite number of modes will be used in Eq. (6) to describe the response of the plate, and it is assumed that N^2 modes are required to describe the response with a sufficient accuracy.

By substituting series (6) into expression (4) for the spectral characteristics of the system response, we can then derive an algebraic formulation for the power spectral densities we aim to compute.

Let us first introduce some notation. All of the matrices we will define in this section are $N^2 \times N^2$ matrices. $\boldsymbol{D}_\omega(\boldsymbol{a})$ will stand for a diagonal matrix whose elements are the modal coefficients $a_{mn}(\omega)$ that appear in series (6). $\boldsymbol{\Phi}(\boldsymbol{x})$ is the matrix for the basis functions of the solution, that is, $\boldsymbol{\Phi}_{\{i,j;k,l\}}(\boldsymbol{x}) = W_{ij}(\boldsymbol{x})W_{kl}(\boldsymbol{x})$.

The algebraic expression of the integral (4) in the case of the spectral density function of the panel velocities ($\alpha_\omega = \beta_\omega = v_\omega$) is then given by

$$S_{v_\omega v_\omega}(\boldsymbol{x};\omega) = \omega^2 \operatorname{tr}\left[\boldsymbol{D}_\omega^H(\boldsymbol{a})\boldsymbol{\Psi}_\omega \boldsymbol{D}_\omega(\boldsymbol{a})\boldsymbol{\Phi}(\boldsymbol{x})\right] \tag{7}$$

where the trace operator applied to a matrix denotes the sum of its diagonal elements and the superscript H stands for the Hermitian (conjugate transpose). The matrix $\boldsymbol{\Psi}_\omega$ that appears in expression (7) is the matrix of modal excitation terms defined as

$$\boldsymbol{\Psi}_{\omega,\{i,j;k,l\}} = \frac{1}{4\pi^2}\int_{-\infty}^{+\infty}\int_{-\infty}^{+\infty} F_{ij}(\boldsymbol{k}) S_{p_b p_b}(\boldsymbol{k};\omega) F_{kl}^*(\boldsymbol{k})\,\mathrm{d}\boldsymbol{k} \tag{8}$$

Each coefficient of $\boldsymbol{\Psi}_\omega$ quantifies how the structural modes are coupled by the turbulent excitation. A criterion has been derived on the correlation lengths to neglect, above a certain frequency, the coupling between the structural modes due to the turbulent forcing field, that is, the cross terms of the modal excitation matrix $\boldsymbol{\Psi}_\omega$ (Ref. 27). For the simulations carried out in this paper, this lower limiting frequency is well below the first natural frequency of the plate, and we have checked that, for the frequency range of interest, the structural modes of the plate are independently excited.[24]

Using the wave number approach, the spectral density S_V of the kinetic energy in the structure can easily be deduced from Eqs. (2) and (7) as

$$S_V(\omega) = \left(\rho_p h\omega^2/2\right) \operatorname{tr}\left[\boldsymbol{D}_\omega^H(\boldsymbol{a})\boldsymbol{\Psi}_\omega \boldsymbol{D}_\omega(\boldsymbol{a})\right] \tag{9}$$

A similar method leads to an algebraic expression for the spectral density of the sound power S_{P^-} inwardly radiated by the plate:

$$S_{P^-}(\omega) = \tfrac{1}{2}\Re\left\{-j\omega \operatorname{tr}\left[\boldsymbol{D}_\omega^H(\boldsymbol{a})\boldsymbol{M}_\omega^- \boldsymbol{D}_\omega(\boldsymbol{a})\boldsymbol{\Psi}_\omega\right]\right\} \tag{10}$$

with

$$\boldsymbol{M}^-_{\omega,\{i,j;k,l\}} = \left\langle P_{ij}^-(\boldsymbol{x};\omega),\, W_{kl}^*(\boldsymbol{x})\right\rangle_\Sigma$$

where P_{ij}^- is the acoustic field inwardly radiated by the ijth structural mode. $\boldsymbol{M}_\omega^-$ is the matrix of intermodal coupling coefficients. A priori, the cross terms of this matrix are not zero valued. This means that the pressure field radiated by any structural mode of the plate is coupled with any other structural mode. As presented in the work of Elliott and Johnson,[28] it is possible to define a new basis for the solution that diagonalizes $\boldsymbol{M}_\omega^-$. The corresponding frequency-dependent eigenvectors or radiation modes do not depend on the mechanical properties of the plate.

By definition, the radiation modes radiate sound independently. When this new set of basis functions is used, the sound power radiated by the plate can, thus, be seen as the sum of independent contributions due to each radiation mode. This means that the sound pressure radiated by the plate vibrating with a surface velocity proportional to one of the radiation modes will not reexcite the other (radiation) modes. As we have noticed, this is not the case with the structural modes because the sound power radiated by each structural mode is partially reabsorbed by the other (structural) modes.

At this stage, an important conclusion for control strategies can be drawn: We are certain to reduce the total sound power radiated by the plate if we manage to reduce the amplitude of any radiation mode. Previous studies have shown, however, that, under a general harmonic excitation, reducing the amplitude of a structural mode does not necessarily guarantee a decrease in the sound power radiated.[29] Nevertheless, when the plate is excited by a TBL, the sound power radiated by the plate can be expressed as a sum of uncorrelated structural modes. In contrast to harmonic excitation, reducing the amplitude of any structural mode should also guarantee an attenuation of the sound power radiated while reducing the vibration or near-field pressure levels because the amplitudes of the other structural modes have not been affected. This observation is verified in Sec. VI.B.

The shape of the first radiation mode of the panel has been plotted in Fig. 3 for two analysis frequencies. The first radiation mode can be considered as a pistonlike mode up to 600 Hz, and its amplitude can, thus, be approximated by the volumetric contribution of the

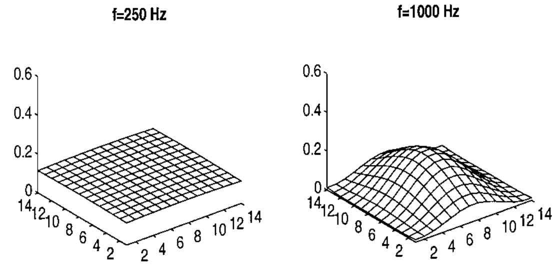

Fig. 3 Shape of the first radiation mode on the panel for two analysis frequencies.

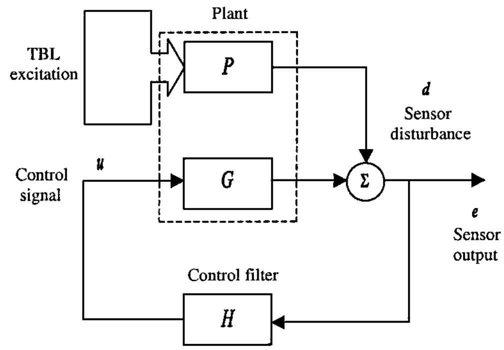

Fig. 4 Equivalent block diagram of the feedback control system to reject the disturbance *d* through a plant *G*.

structure velocity. Up to 1 kHz, this uniform shape is slowly transformed into a dome shape. It would be difficult to design a sensor whose sensitivity could detect the frequency-dependent contribution of the first radiation mode over a broad frequency range. However, at low frequencies where the first radiation mode is dominant, its amplitude can be well approximated by the net volume velocity of the panel.[30,31]

In the next section, we will describe the cancellation of the net volume velocity as a control strategy to reduce the radiated sound power.

V. Feedback Control of TBL-Induced Sound Radiation Using a Volume Velocity Sensor

The wall-pressure fluctuations due to a TBL over an aircraft fuselage is a broadband random signal from which it is difficult to extract a reasonable number of time-advanced reference signals. A feedback control system must generally be used to reduce TBL-induced sound radiation for the single panel arrangement discussed here. Figure 4 shows a block diagram describing how a feedback control system using a controller H is able, through a plant G, to compensate for the disturbance d at the sensor.

A. Cancellation of Net Volume Velocity

Our goal is to reject the volumetric response of the system to a TBL excitation. This can be done by canceling the total volume velocity of the panel expressed as a combination of contributions from the primary source (TBL excitation) and the secondary source (control signal).

Let us denote the power spectral density of the net volume velocity of the panel due to the primary excitation as S_{dd}, an algebraic expression for which is given by

$$S_{dd}(\omega) = \omega^2 \operatorname{tr}\left[\boldsymbol{D}_\omega^H(\boldsymbol{a})\boldsymbol{\Psi}_\omega \boldsymbol{D}_\omega(\boldsymbol{a})\boldsymbol{P}\right] \tag{11}$$

where

$$\boldsymbol{P}_{\{i,j;k,l\}} = F_{ij}(\boldsymbol{0})F_{kl}(\boldsymbol{0})$$

$\boldsymbol{P}$ is a sparse matrix, any of its nonzero coefficients represents the volumetric contribution of an odd–odd structural mode.

To cancel the total volume velocity, the spectral characteristics of the secondary source strength must be given by

$$S_{uu}(\omega) = [1/|G(\omega)|^2]S_{dd}(\omega) \qquad [\Leftrightarrow S_{ee}(\omega) = 0]$$

where S_{ee} is the power spectral density of the total volume velocity measured at the sensor output. This last expression shows clearly that the control filter H has to compensate for the behavior G of the plant, that is, the transfer function between the sensor and the actuator. Hence, perfect rejection of the disturbance can only be achieved if the plant is perfectly invertible, independently of the nature of the disturbance. In practice, the inversion of the plant response is often altered by time delays in the system. We will show in the next section under which conditions we can design a transducer that prevents these effects.

B. Properties of a Matched Actuator/Sensor Pair

To cancel the net volume velocity, it is required to design a sensor that provides an accurate measure of the volume velocity or average velocity of a surface. This could be done using a sheet of piezofilm made in polyvinylidene fluoride (polymer) attached to the surface of the panel. The reciprocal transducer is a distributed actuator that generates a uniform force over the surface when driven by a voltage. A feedback control system could then be achieved using a volume velocity sensor matched with a uniform force actuator. Such a configuration will guarantee a decrease in the sound power radiated by the structure without involving an increase (control spillover) in the vibration or near-field pressure levels.[30]

However, the point is to design a controller such that the system remains stable whatever the magnitude of the feedback gain, to achieve good performances. Such a property is satisfied by matched actuator/sensor pairs the transfer function of which is minimum phase, that is, with both poles and zeros in the left-hand side of the Laplace complex plane or s plane.[32]

Indeed, the frequency response G between the sensor and the actuator may be written as

$$G(\omega) = \sum_{m=1}^{+\infty}\sum_{n=1}^{+\infty} F_{mn}^a F_{mn}^s a_{mn}(\omega)$$

$$F_{mn}^a = \langle \eta^a, W_{mn}^* \rangle_\Sigma, \qquad F_{mn}^s = \langle \eta^s, W_{mn}^* \rangle_\Sigma$$

where η^a and η^s are, respectively, the sensitivity functions of the actuator and the sensor. For a volume velocity sensor and a uniform force actuator, we would have $\eta^a(\boldsymbol{x}) = \eta^s(\boldsymbol{x}) = 1(\boldsymbol{x})$. In this case [or more generally when $\eta^a(\boldsymbol{x}) = \eta^s(\boldsymbol{x})$], the transfer function is minimum phase because its residues $F_{mn}^a F_{mn}^s$ are all positive, each zero being, therefore, located between two poles in the left-hand side of the s plane and, thus, will have the additional property that the real part of $G(\omega)$ is always positive whatever the angular frequency ω.

If we choose a controller that perfectly compensates for the plant response G while including a large feedback gain α, it can easily be shown that the controller transfer function $H(\omega) = -\alpha G^{-1}(\omega)$ is also minimum phase. This property involves two major consequences. First, the alternative distribution of poles and zeros leads to a control system that does not accumulate phase when the frequency increases, that is, no time delays. Second, the control system remains stable (all of the poles are on the left-hand side of the s plane) and causal, whatever the magnitude of the feedback gain.

Hence, a feedback control system using a matched volume velocity sensor/uniform force actuator could achieve perfect cancellation of the net volume velocity response of the panel to an arbitrary excitation while ensuring unconditional stability because the feedback gain can be made arbitrarily large.

VI. Results and Discussion

In this section, simulation results will be given for attenuation achieved in the radiated sound power when canceling either the radiation modes of the panel or its structural modes. First, however, we examine the influence of the main physical parameters of a TBL-excited plate on its vibroacoustic response.

Table 1 Geometrical and physical parameters for a typical aircraft panel

Parameter	Value
Freestream velocity	$U_\infty = 225$ m/s
Boundary-layer thickness	$\delta = 0.1$ m
Panel thickness	$h = 0.001$ m
Panel Young's modulus	$E = 7.24 \times 10^{10}$ Pa
Panel longitudinal tension	$N_n = 29.3 \times 10^3$ N·m
Panel lateral tension	$N_m = 62.1 \times 10^3$ N·m
Panel Poisson's ratio	$\nu = 0.33$
Panel mass density	$\rho_p = 2800$ kg/m^3
Panel damping ratio	
Case a	$\xi = 0.01$
Case b	$\xi = 0.05$
Panel dimensions	$a = 0.414$ m, $b = 0.314$ m
Sound speed	
External fluid	$c_+ = 300$ m/s
Internal fluid	$c_- = 340$ m/s
Mass density	
External fluid	$\rho_+ = 0.44$ kg/m^3
Internal fluid	$\rho_- = 1.42$ kg/m^3

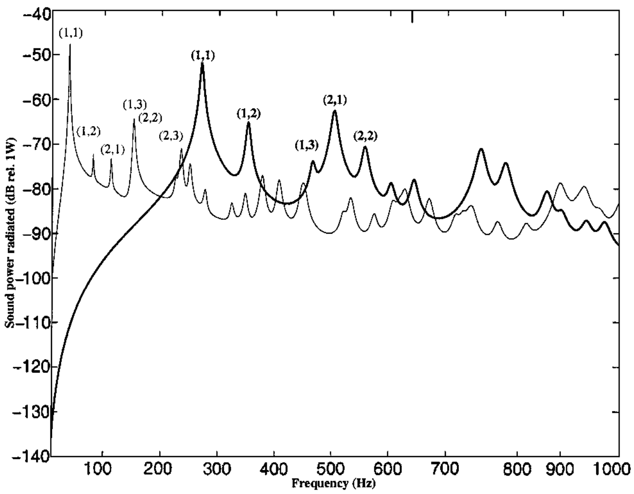

Fig. 5 Sound power inwardly radiated by a tensioned plate (——) and an untensioned plate (——).

A. Vibroacoustic Response of TBL-Excited Plate

Figure 5 shows the result of a calculation of the internal sound power radiated from a flat untensioned panel subject to TBL excitation, as well as that radiated from a plate tensioned by membrane stresses due to the cabin pressurization for the typical parameters given in Table 1. It can be seen that the fundamental frequency of the panel is increased from about 39 to 270 Hz when in-plane tensions are taken into account. Moreover, the order of the corresponding resonance frequencies for the individual modal values m and n associated with each structural mode are modified. The corresponding mode shapes are, however, still the same as the untensioned plate. It is clear that we cannot neglect the influence of the in-plane tension, even in our simplified model.

In Fig. 6, the effect of changing the structural damping ratio from 1 to 5% has been considered. As expected, it is the levels at the resonant peaks that are mainly affected, and a reduction of about 10 dB has been achieved in the overall total sound power radiated. Moreover, we notice that the influence of the structural damping increases with frequency. Although this increase in damping has a beneficial influence on the sound power radiated, further increases in the structural damping for highly dissipative panels do not seem to affect significantly the radiated sound power levels.[24]

We have noted the influence of the structural damping on the sound radiated by each panel. However, two other fundamental mechanisms underlie the physical features concerning the acoustic response of an airflow excited panel: the hydrodynamic coincidence effect and the radiation efficiency of each structural mode.

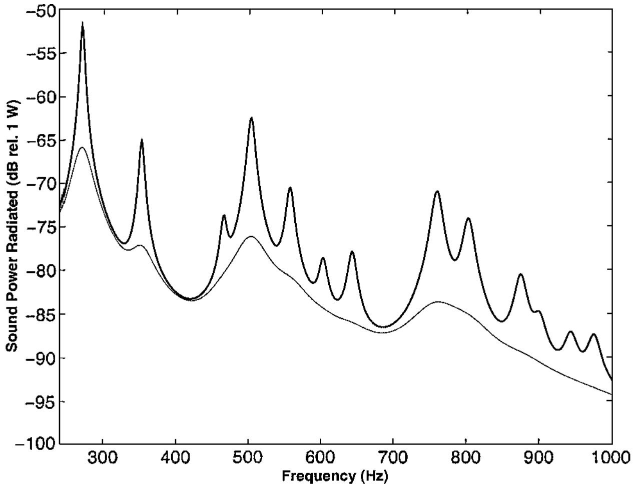

Fig. 6 Sound power radiated by a tensioned aluminum plate for two values of the damping ratio: 1% (——) and 5% (——).

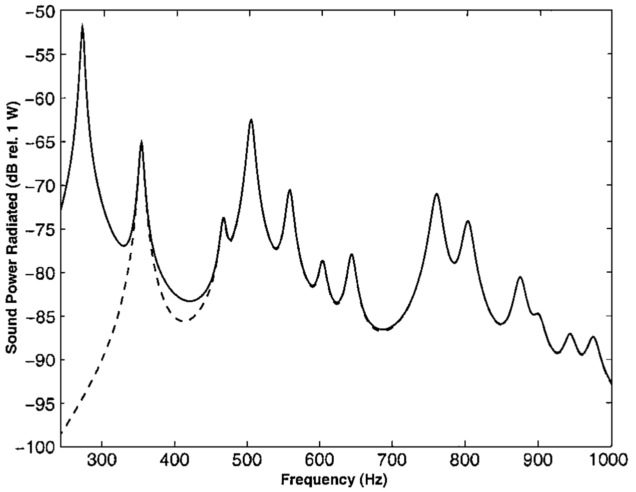

Fig. 7 Sound power radiated by a tensioned aluminum plate before control (——) and after cancellation of the first structural mode (- - -).

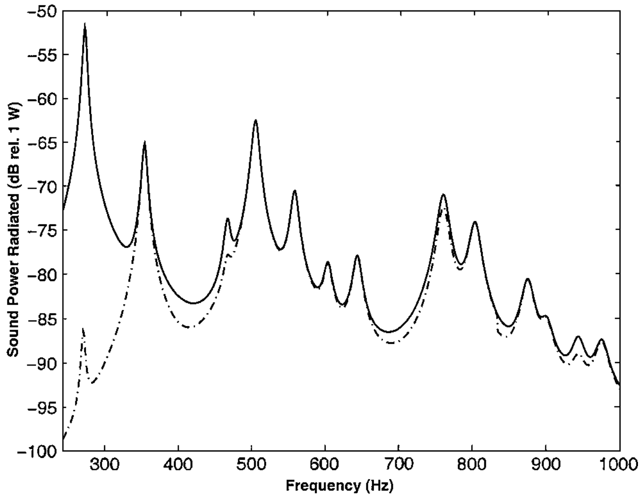

Fig. 8 Sound power radiated by a tensioned aluminum plate before control (——) and after cancellation of the first radiation mode (–·–).

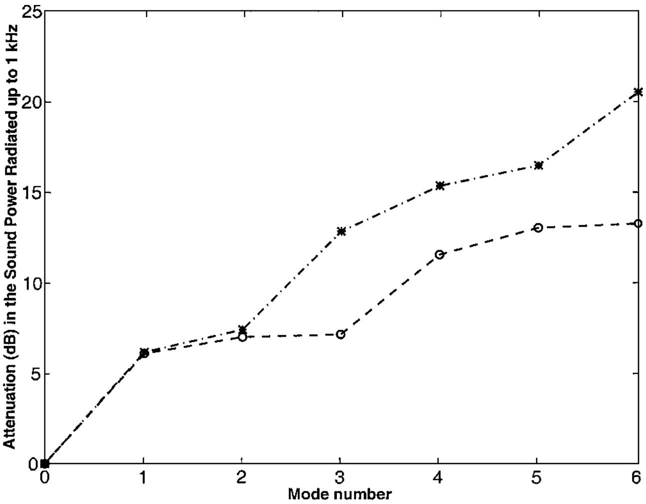

Fig. 9 Attenuation in the sound power radiated by a TBL-excited panel (case a): after cancellation of the first radiation modes (–·–), and after cancellation of the first structural modes (——).

The first major effect occurs within a frequency range (up to 1 kHz in our configuration) over which the structural modes are highly excited by the wall-pressure fluctuations due to the TBL. This strong flow–structure interaction exactly appears when the convective scale of the turbulent excitation, where the main fluctuating energy lays, matches the scale of a structural mode.

The second major effect that governs the amplitudes of the resonant peaks for the sound radiated concerns the radiation efficiency of each structural mode. Most of the modes that contribute to the response of the system are inefficient radiators. This means that, below their critical frequency, these modes have large differences in their contribution to the far-field radiated pressure, the less efficient modes being the even–even modes. Above their critical frequency, however, these modes radiate sound independently with the same efficiency. In Fig. 5, for example, below 500 Hz, the resonant modes with the highest contribution to the sound power radiated are odd–odd inefficient modes. However, the (2,2) mode that does not contribute to the sound power radiated by the untensioned panel at its natural frequency of 154 Hz, because it is so inefficient, begins to radiate efficiently as for the tensioned panel when its natural frequency is raised to 556 Hz.

B. Sound Power Attenuation: Structural Modes vs Radiation Modes Cancellation

To define a suitable control strategy, we first examine how the first structural modes or radiation modes of the plate contribute to the vibroacoustic response by canceling their participation in the modal expression [Eqs. (9) and (10)] for both the kinetic energy and the sound power radiated.

Figures 7 and 8 show the effect of canceling, respectively, the first structural mode and the first radiation mode on the sound power radiated by the lightly damped tensioned panel. The attenuation achieved in the total sound power radiated up to 1 kHz by canceling a limited number of the higher-order structural or radiation modes is summarized in Fig. 9. The corresponding influence on the kinetic energy of the panel is shown in Fig. 10. Figure 11 shows the effect of canceling either the first structural or radiation mode on the sound power radiated by the tensioned panel with a damping ratio of 5%.

From Figs. 7–9, it can be seen that approximately the same level of attenuation (about 6.5 dB) is observed by canceling the first radiation mode or structural mode of the plate. When the higher-order modes are canceled, however, sound power reductions are achieved more efficiently when suppressing the contribution of the radiation modes compared with the structural modes. It can be seen, from Fig. 9, that canceling the first three radiation modes leads to 13 dB of attenuation, whereas six structural modes must be canceled to achieve the same level of attenuation.

From Fig. 10, we note that the cancellation of the first radiation mode does not involve a significant reduction in the kinetic energy of the panel at very low frequencies compared with the cancellation of the first structural mode. Indeed, the cancellation of the volumetric or space-averaged velocity of the structure, which is a good approximation of the first radiation mode (Fig. 3), does not necessarily guarantee the cancellation of the velocity of the structure. An important consequence for passengers close to the aircraft panel is

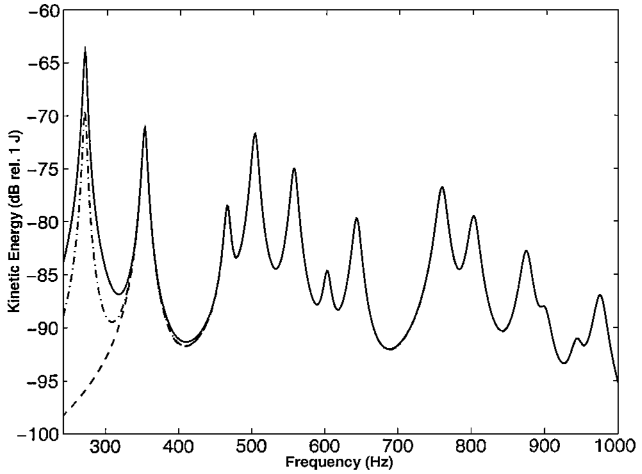

Fig. 10 Kinetic energy of the tensioned aluminum plate (case a) before control (——), after cancellation of the first radiation mode (–·–), and after cancellation of the first structural mode (– – –).

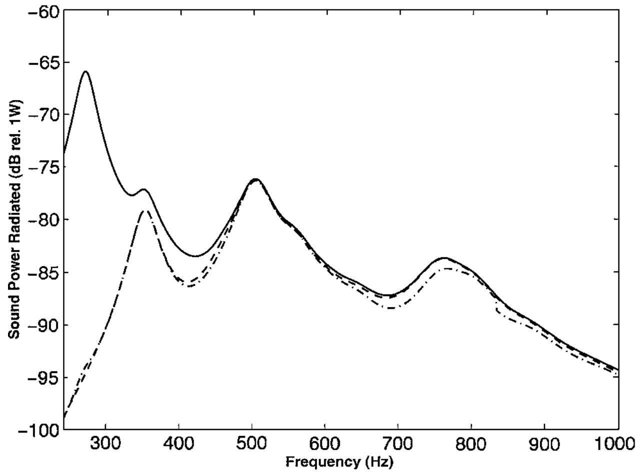

Fig. 11 Sound power radiated by a tensioned aluminum plate (case b) before control (——), after cancellation of the first structural mode (– – –) and after cancellation of the first radiation mode (–·–).

that, even when canceling the volumetric contribution of the panel velocity, the reduction in the near-field pressure levels is not significant. However, as expected in Sec. IV, the near-field levels do not increase at higher frequencies (no spillover effect).

Comparing Fig. 11 with Figs. 7 and 8, we note that increasing the structural damping from $\xi = 1$ to 5% does not affect significantly the attenuation of the overall sound power radiated after cancellation of the first structural or radiation mode. However, as expected, the performances of both strategies is slightly degraded when increasing the damping ratio.

C. Active Control of the Sound Power Radiated by a TBL-Excited Plate When Canceling the Net Volume Velocity

For the simulations in Fig. 12, we have considered a feedback control with a uniform force actuator controlling a matched volume velocity sensor. In this simulation, we have reduced the sound power radiated up to 1 kHz by canceling the net volume velocity, except that two extra peaks appear, near 460 and 720 Hz. They correspond to a minimum value for the sound power radiated by the plate when only excited by the uniform force actuator, that is, to minimum values of the plant transfer function G. These extra peaks also appear in the plant response when we attempt to reduce the sound power radiated by a plate excited by a harmonic plane wave, but they are balanced by the plate antiresonance, which, as expected, occurs at the same frequencies.

As a final remark, we note that the total sound power radiated up to 1 kHz has been reduced by 5.9 dB when canceling the net volume velocity. This value is very close to the attenuation predicted in

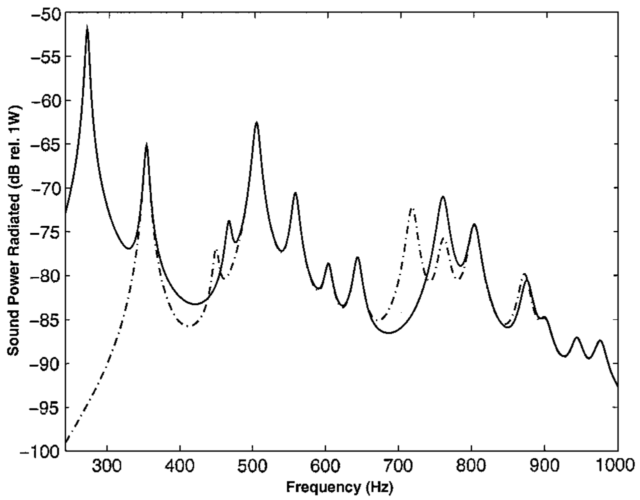

Fig. 12 Sound power radiated by a tensioned aluminum plate excited by a TBL before control (——), after cancellation of the net volume velocity (–·–) as a function of frequency and using a uniform force actuator.

Fig. 9, when suppressing the contribution of the first radiation mode or canceling the first structural mode.

VII. Conclusions

A simple model has been developed for predicting both the kinetic energy and the acoustic radiation of a thin, baffled plate, simply supported along its boundaries, stressed by in-plane tensions and excited by a TBL. This simplified modeling describes the dynamic behavior of the cylindrical part of an aircraft fuselage as a network of adjacent panels excited by wall-pressure fluctuations under subsonic cruise conditions and vibrating individually in the frequency range of interest (up to 1 kHz). The important conclusion we have drawn is that the structural modes of each panel radiate sound independently and that a suitable strategy for the active structural acoustic control of the sound power inwardly radiated by the panels would be independent feedback control of each structural mode of each panel in the low-frequency domain. We have shown that this strategy could provide attenuation up to 13 dB with six independent controllers.

As an alternative, we could use a strategy based on the cancellation of the radiation modes. We have shown that canceling the first structural mode or the first radiation mode provides similar attenuation in the total sound power radiated up to 1 kHz. We have also noticed that the cancellation of a limited number of the higher-order radiation modes was more efficient than the cancellation of the same number of structural modes for the control of the sound power radiated by such panels. Surprisingly, both strategies are almost independent of the structural damping effect in the range $\xi = 1$–5%.

Because perfectly matched sensors and actuators have been assumed in this paper, the performance has not been degraded by any spillover effects. In practice, with discrete actuators and sensors, these effects will always occur and degrade the results. However, the object here was to derive the performance limitations due to the physical properties of the TBL-excited panel, rather than a particular actuator/sensor arrangement, and so the assumption of idealized matched transducers is justified.

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