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Technical note

A note on a circular panel sound absorber with an elastic boundary condition

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A R T I C L E I N F O

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Panel sound absorbers are typically used to absorb low-frequency noise in concert halls, auditoriums, recording studios, and other architectural applications. These systems are composed of flexible panels mounted over an air space that can be either partly or completely filled with a porous material. In this paper, a theoretical model is derived for predicting the sound-absorption coefficient of a cylindrical low-frequency absorber made of a circular plate. The theory takes into account the mass, bending stiffness, damping loss and the elastic boundary condition of the circular plate. The effects of the stiffness of an air-back cavity and of partially adding a porous material into the cavity are also considered. It is observed that the low-frequency resonances of such a system are dependent upon the clamping condition, the width of the air-back cavity, and mechanical properties of the plate. There is good agreement between theoretical and experimental results.

1. Introduction

In many architectural applications there is a need to absorb low-frequency noise, which is difficult to achieve by using porous sound-absorbing materials. Panel sound absorbers are typically used to absorb low-frequency sound in concert halls, auditoriums, recording studios and other architectural applications. Basically, these systems are composed of flexible panels made of wood, metal, gypsum or plastic board, mounted over an air space that can be either partly or completely filled with a porous material. Thus, several flexural resonance modes can be excited by an incident sound wave, and maximum absorption takes place at the lowest natural frequency of the coupled panel-cavity system.

If a sufficiently large plate is fastened over its entire surface area in front of an air cavity and to an absorbing layer located in front of a rigid wall, the plate as a whole is allowed to oscillate in a piston-like manner on the equivalent spring formed by the layers. Then, the resonance frequency of the plate absorber can be determined from the mass of the plate and the stiffness of the equivalent spring formed by the elastic layers in the cavity (in this case, the plate's bending stiffness is not important). It is well-known that the heavier the plate and the softer the layer, the lower the resonance frequency. However, if the plate is clamped it will perform bending oscillations. In this case, computation of its eigenfrequencies is usually quite complex because the mechanical behavior depends on the way of clamping, the physical properties of the plate, and the air volume and filling.

1.1. Previous studies

Ford and McCormick [1] were the first to develop a theory for square panel absorbers including the effect of the plate's bending waves. This model has been used by other authors for analyzing low-frequency absorber systems. Ford and McCormick employed the virtual work principle to derive the equation of motion of a thin and homogenous square plate with clamped boundary conditions that is subjected to plane wave excitation.

Fuchs [2] discussed the acoustic behavior of sound absorbers made of square plates and established design recommendations. In his book, he showed the sound absorption of the panel for different damping conditions of the system. The maximum sound-absorption coefficient is shifted to low frequencies as the surface mass of the panel and the thickness of the air cavity are increased. Evidently, the increase in the air cavity thickness is usually limited by the available space in the room, and the increase of the mass can add to the costs of the solution. Fasold and Veres [3] have stated that the air cavity thickness does not have to be too small or too large, compared with the wavelength of the sound wave that is intended to be absorbed.

The application of a vibrating plate to absorb low-frequency noise was presented by Wang et al. [4]. They employed this

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physical principle to add a completely clamped plate with a sealed air-back cavity to the sidewall of a duct to reduce the transmitted sound. The analysis was developed for homogeneous and nonhomogeneous plates; better sound-absorption values were obtained for the latter.

Fuchs [2] reviewed other configurations of sound absorbers, such as panels completely filled with sound-absorbing material and multilayer absorbers. Other authors have developed models that included perforated and microperforated plates as vibrating panels. These plates are an inexpensive and effective option to absorb midfrequency noise. The perforations are reduced to submillimeter sizes to release sufficient acoustic resistance values and sufficiently low-mass to provide a relatively broadband sound absorption. Thus, the use of microperforated panels increases the effective frequency range while sacrificing sound absorption at lower frequencies.

Lee and Swenson [5] analyzed low-frequency sound absorbers both theoretically and experimentally using one or two square perforated panels with an air-back cavity. During the experimental work they noticed the presence of absorption peaks at very low frequencies that were associated with the panel vibration. In the work of Frommhold et al. [6], a thin vibrating panel on a light honeycomb structure was studied. This system formed small air volumes, which acted as Helmholtz resonators. They used the expression of Ford and McCormick [1] to include the effect of mechanical vibration of the plate in the theoretical model to estimate the sound absorption of the system. It was concluded that the system had two resonances, one associated with the Helmholtz resonator and the other with the stiffness of the plate.

The case of an infinite elastic panel with an air-back cavity has been analyzed by Sakagami et al. [7,8] for an arbitrary angle of incidence. Their results indicated that the contribution of the air cavity is dominant and that the contribution to the sound absorption of the panel's damping loss factor is negligible. Obviously, the effect of structural resonance of the panel was not included because this effect appears only when the panel is finite and reflections occur at the boundaries. Subsequently, Sakagami et al. [9] included the effect of the vibration of the microperforated panel in a simplified manner, considering the surface density of the plate for inclusion into the impedance equations of the equivalent circuit. Thus, the reactance of the panel's mass was added in parallel to the impedance of the microperforations. The results showed that a very light panel reduced the amplitude of the absorption peak and slightly increased the resonance frequency. However, for panels having a surface mass greater than 2 kg/m^2, the difference was very small and the maximum value of absorption was almost the same as in the case of a immobile panel.

Another study analyzed the case of a microperforated panel with an air-back cavity [10], considering the effect of the structural vibration of the panel. The theory involved a modal analysis of the plate's classical equation coupled with the acoustic wave equation. They observed that if the frequency of the incident sound is less than the first structural resonance, then the structural vibration degrades the sound absorption and, conversely, a frequency greater than the first resonance causes an improvement.

In later work, Lee et al. [11] also studied theoretically the case of a microperforated panel absorber, considering the effects of the microperforation, the resonance of the panel and the Helmholtz resonator. The analysis included an equivalent circuit: Lee and colleagues considered that the impedances of the three effects are connected in parallel and then connected in series to the air cavity. They observed the presence of two characteristic peaks in the curve of the sound-absorption coefficient of the system.

Tayong et al. [12] analyzed the microperforated system with an air-back cavity in the case of very high sound pressure levels incident on the panel where the nonlinear behavior should be considered. Their results suggested the existence of an absorption maximum that depends upon the flow velocity in the perforations, which was experimentally confirmed. Interestingly, they used the model of Ford and McCormick [1] to incorporate the effect of the structural vibration of the panel, even though their samples were circular and the boundary conditions were approximately clamped.

Sakagami and Morimoto [13] studied the characteristics of a sound-absorb-ing panel coupled with an air-back cavity, which had a nonuniform depth. This case can be found in some particular architectural applications. The theoretical results indicated that the absorption peak can be wider than in the case of a cavity of uniform depth.

In general, a panel absorber is used to cover a relatively large area in a room. Thus, characterization of its sound-absorbing properties can be made in a reverberation chamber. However, for design purposes, measurements using an impedance tube are often employed to obtain the normal-incident sound-absorption coefficient. Thus, small samples of panel absorbers having circular areas are tested in the tube. Obviously, in such a case the shape geometry and boundary conditions of the circular plate will be important for determining both the eigenfrequencies and mode shapes of the system and, consequently, the acoustic behavior of the sound absorber.

It seems that the case of a sound absorber made of a circular plate has been considered only by Hiraizumi et al. [14] in a much forgotten paper. In their work, the problem is analyzed for a system composed of a clamped circular plate, an air cavity and eventually a sound-absorbing material partially filling the cavity. The theoretical analysis was based on solving the classical wave equation in cylindrical coordinates, considering each layer as an isolated medium. Unlike Ford and McCormick, they did not use electrical analogies in the derivation.

1.2. Statement of the problem

In the formulation presented by Hiraizumi et al. [14] they considered a plate with a clamped rim and they did not consider the general case of an elastic boundary condition. Thus, the main objective of this note is to present a sound absorption formula for a circular panel sound absorber that includes the effect of the panel' structural vibration when subjected to a general elastic boundary condition. The absorber is composed of a circular plate and backing air layer, which is partially filled with a porous sound-absorbing material. Then, when the system is excited by airborne sound, the flexural oscillation of the plate on the spring formed by the air cavity and the sound attenuation inside the porous material are the main sound-absorption mechanisms. The next section presents the derivation of the theoretical equations. The following sections describe the experimental setup and the results. Concluding remarks are presented in the final section.

2. Theory

The low-frequency sound absorber is represented in Fig. 1. A plane sound wave of amplitude p_0 coming from the top strikes an impervious layer formed by a thin plate with radius a and thickness h. This sound wave induces a vibration that is propagated through the air cavity of thickness d_1 and through a sound-absorbing layer of thickness d_2, which is resting in front of a rigid backing surface with ideal infinite acoustic impedance. Complex exponential convention $e^{j\omega t}$ has been used to represent harmonic time dependence for all the theory presented in this paper.

Because the study focuses on the problem of a circular sample placed inside of an impedance tube, the analysis has to be done