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Objective evaluation of interior trim effects on sound quality and noise reduction of a coupled plate cavity system

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ABSTRACT

In this study, the impedance mobility and psychoacoustic analysis methods are combined to develop a structural-acoustic model of a plate-cavity coupling system. The objective is to evaluate the effect of interior trim materials on sound loudness and sharpness of a plate-cavity coupling system. The impedance mobility method is applied to calculate the pressure frequency responses of the interior acoustic field for the plate-cavity coupling system. The sound pressure results calculated by the impedance mobility method are then directly used to calculate the psychoacoustic metrics using psychoacoustic analysis method. A good agreement was found between the experimental and analytical results. The results show that the interior trim has a large influence on the distribution of the sound loudness and sharpness inside the cavity in the middle and high frequency ranges.
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1. Introduction

Sound quality of vehicle cabin interior noise has recently become more and more important as passengers demand a more comfortable ride. Among the various interior noise problems, structure-borne noise such as booming, which occur due to strong structural-acoustic modal coupling, was investigated [1–4]. In order to control the interior noise, it is very important to carry out a vibro-acoustic analysis rather than an acoustic or structural analysis alone. The interaction between a flexible boundary structure and an acoustic cavity is a complex process that has attracted attentions of researchers especially in different fields of engineering and technology. Coupled structural acoustic systems are found in different practical applications, for instance, aircraft fuselages, industrial machine elements and vehicular cabins. Dowell et al. [5] presented a comprehensive theoretical model for coupled response of structural-acoustic system. The coupled response was obtained from the modal characteristics of the uncoupled systems. Kim and Brennan [6] introduced the impedance mobility method to solve the same system as in [5]. The advantage of this method comes from its formulation simplicity and results in a compact matrix which can be easily solved in numerical computer software. Pan and Bies [7] studied the interaction between the cavity boundaries and its effect on the contained steady state and transient sound fields theoretically and experimentally from the room acoustics point of view. The plate-cavity coupled system consists of a rectangular box with slightly absorptive walls and simply supported rectangular panel on the top. Yamamoto et al. [8] employed the concept of the density approach in topology optimization to find the optimum thickness distribution of a rectangular cavity-backed multilayered structure containing a poroelastic core layer, aiming at minimization of sound pressure level inside the cavity.

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Akl et al. [9] used finite element model to optimize the topology of a flexible rectangular plate coupled with an open-ended and five rigid walled acoustic cavity in an attempt to minimize the fluid structure interactions at different structural frequencies by redistribution of the material of the flexible plate. In the previous research on objective evaluation methods for the sound quality of the booming noise, it was found that the booming noise was dominated by the loudness and sharpness of interior noise [10,11].

Different from previous studies, the present study focus on investigation of the effect of interior trim materials on sound loudness and sharpness of a plate-cavity coupling system. Based on the impedance mobility approach, the pressure frequency responses are calculated for the structural-acoustic coupling system under a simply supported condition. Zwicker's sound loudness and Bismarck's sharpness then were evaluated at different point locations inside the plate-cavity system. The research work provides an analytical method for predicting and designing sound quality of the structural-acoustic coupling system in terms of the psychoacoustic metrics.

2. Structural–acoustic coupling theory

2.1. Flexible plate

The structural vibration velocities at any point on the flexible plate can be expressed as the product of the structural mode shape vector $\mathbf{\Phi}$ and the complex vibration velocity modal amplitude, $\mathbf{b}$ as:

$$u_s = \Phi^{\mathrm{T}}\mathbf{b} \tag{1}$$

where, superscript T denotes transpose, each column of Φ^{T} consists of M structural mode shape-functions, $\phi_m(\mathbf{y})$ at a specified location, (x_i, y_i) on the flexible plate.

Therefore, the structural vibration velocity, u_s at any point $(\mathbf{y})$ on the flexible plate can be written as:

$$u_s(\mathbf{y},\omega) = \sum_{m=1}^{M} \phi_m(\mathbf{y}) \cdot b_m(\omega) = \mathbf{\Phi}^{\mathrm{T}}\mathbf{b} \tag{2}$$

where, the M length column vectors $\mathbf{\Phi}$ and $\mathbf{b}$ consist of the array of uncoupled vibration mode shape functions, $\phi_m(\mathbf{y})$ and the complex amplitude of the vibration velocity mode, $b_m(\omega)$ respectively.

The mode shape function $\phi_m(\mathbf{y})$ satisfies the orthogonal property in uncoupled structural system, and is normalized as follows:

$$S_f = \int_{S_f} \phi_m^2(\mathbf{y}) \mathrm{d}S \tag{3}$$

where S_f is the surface area of the flexible plate. The complex vibration velocity of the mth structural mode of the flexible plate in Fig. 1 for an isotopic thin plate can be expressed as:

$$b_m(\omega) = \frac{1}{\rho_s h S_f} \cdot B_m(\omega) \cdot \left(\int_{S_f} \phi_m(\mathbf{y}) \cdot f(\mathbf{y},\omega)\mathrm{d}S - \int_{S_f} \phi_m(\mathbf{y},\omega) \cdot p(\mathbf{y},\omega)\mathrm{d}S \right) \tag{4}$$

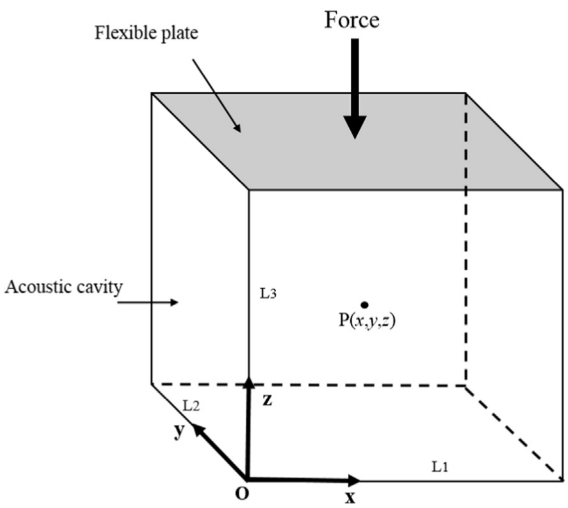

Fig. 1. A flexible plate-acoustic cavity system.

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