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Frequency-independent radiation modes of interior sound radiation: Experimental study and global active control

C. Hesse [a,*], V. Papantoni [a], S. Algermissen [b], H.P. Monner [b]

[a] Institute of Composite Structures and Adaptive Systems, German Aerospace Center (DLR), Sportallee 54a, 22335 Hamburg, Germany
[b] Institute of Composite Structures and Adaptive Systems, German Aerospace Center (DLR), 38108 Braunschweig, Germany

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ABSTRACT

Active control of structural sound radiation is a promising technique to overcome the poor passive acoustic isolation performance of lightweight structures in the low–frequency region. Active structural acoustic control commonly aims at the suppression of the far–field radiated sound power. This paper is concerned with the active control of sound radiation into acoustic enclosures. Experimental results of a coupled rectangular plate-fluid system under stochastic excitation are presented. The amplitudes of the frequency-independent interior radiation modes are determined in real-time using a set of structural vibration sensors, for the purpose of estimating their contribution to the acoustic potential energy in the enclosure. This approach is validated by acoustic measurements inside the cavity. Utilizing a feedback control approach, a broadband reduction of the global acoustic response inside the enclosure is achieved.

1. Introduction

Most recent approaches for the active control of sound radiation of vibrating structures into enclosed spaces are based on structural mode sensing [1,2]. This requires a priori knowledge of the structural mode shapes which can necessitate a high identification effort. In Hesse et al. [3], the structural vibration is decomposed into a set of interior radiation modes. The interior radiation modes are orthogonal functions that describe vibration modes of the structure, such that the contribution from each one of them to the acoustic potential energy in the enclosed fluid is uncoupled from any other. The interior radiation modes calculated in [3] do not presume the knowledge of structural mode shapes and are independent of frequency, as they are a subset of the cavity modes at the structural interface. Since the radiation modes are orthogonal with respect to the acoustic potential energy, their reduction by active means will lead to a global reduction of the enclosed sound field.

The mechanisms of the active control of the acoustic potential energy are investigated in a series of publications by Snyder et al. [4,5] using secondary control inputs in the fluid and on the structure. Using an analytical model with vibration control sources, e.g. a point force, the two mechanisms of modal suppression and modal rearrangement can occur. The mechanism of modal suppression consists in the reduction of structural modal amplitudes, which are efficiently coupled to the fluid modes. Modal rearrangement on the other hand, is based on the fact, that one fluid mode is in general coupled to more than one structural mode. If this is the case, the structural modes are rearranged in a way, that the coupling to the

* Corresponding author.
E-mail address: christian.hesse@dlr.de (C. Hesse).

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cavity mode is less efficient.

Many approaches, regarding experimental realizations of active control of structural sound radiation into enclosed spaces, deal with the reduction of local sound pressures [6–8] or local energy density [9]. Kim and Brennan investigate the active control of harmonic and random sound transmission into an acoustic enclosure [10,11]. 11 microphones inside the cavity are used as error sensors and the acoustic potential energy is estimated based on the sum of the squared pressure amplitudes. A structural actuator is shown to be more successful in controlling the structural vibration modes than an acoustic actuator.

The approach of local pressure control does not necessarily result in a global reduction of sound radiation. Although the active control with structural error sensing based on radiation modes is investigated numerically [1,2], an experimental study using a set of radiation modes and yielding a broadband reduction of the acoustic potential energy is not known to the authors. A reason for the absence of experimental work may be found in the frequency–dependence of the resulting radiation modes, when calculated numerically. These radiation modes have been proven frequency–independent for two considered systems of fluid–structure interaction in [3] using an analytical formulation and are used in this study for experimental investigations.

This paper describes an experimental implementation of the global active control of sound radiation in an acoustic enclosure. The system of fluid–structure interaction consists of a rectangular plate coupled to a cuboid cavity. Section 2 describes the experimental setup. A set of harmonic velocities is measured by means of a scanning laser vibrometer (SLV) and included into an identified plant model. Radiation filters are then evaluated and appended to this plant. The system identification process as well as the control synthesis are described in detail in Section 3. An overview of the achieved global noise suppression as well as the control mechanisms is given in Section 4. Section 5 concludes this study and gives an outlook to future research.

2. Experimental setup

This section presents the experimental set–up of the plate–cavity system, as well as the measurement equipment and the active elements used. An exterior view of the experimental plate–cavity system is shown in Fig. 1a. The plate is made of 2 mm thick aluminium with the dimensions $(L_x \times L_y) = (0.6 \times 0.8)\mathrm{m}^2$. The plate edges are fixed in an aluminium frame of 15 mm thickness. A total of 24 accelerometers are distributed equally spaced across the plate. Additionally, a grid of 13×15 highly reflective dots are applied to the plate to facilitate the measurement of the surface velocity by a SLV.

The hardware components for the active control system are listed in Table 1. The disturbance is induced by a primary shaker while two secondary shakers are used for the active suppression of sound radiation. During the experimental identification process, broadband uncorrelated signals for the primary and secondary shakers are generated by a dSPACE® rapid control prototyping system. The low-pass filtered accelerometer outputs are measured and the transfer functions estimated. The positions of the primary and secondary shakers as well as the sensors on the plate are summarized in Fig. 2. For the primary shaker the position is chosen not to coincide with a nodal line in the frequency range up to 500 Hz, so that all modes of the plate can be excited. The positions of the secondary actuators were chosen taking into account controllability and observability criteria as stated by Gawronski [12].

Fig. 1b shows an interior view of the acoustic cavity comprised of a wooden mock-up. The acoustic boundary is built as a double-plate system with additional damping in its enclosed cavity. The interior cavity of the experimental set–up has a depth of $L_z = 0.42$ m. An array of 4×8 microphones is used for the acoustic measurements. The microphone array is shifted in six equally spaced positions along the cavity depth, so that a total of 192 pressure measurements is used for the calculation of the acoustic potential energy. It should be noted, that the microphones are not part of the control system, as they are solely used for the evaluation of the acoustic potential energy. The hardware components for the acoustic measurements are detailed in Table 2.

(a) (b)

Fig. 1. Experimental set–up of the plate-cavity system as seen from the outside (a) and the inside (b); the cavity includes loudspeakers and microphones for the evaluation of the acoustic potential energy.