

Experimental Realization of a Sound Radiation Filter for Feedforward Control to Improve Active Structural Acoustic Control Systems

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ABSTRACT

Due to the strengthened CO₂ and NO_x regulations, future vehicles have to be lightweight and efficient. But, lightweight structures are prone to vibrations and radiate sound efficiently. Therefore, many active control approaches are studied to lower noise radiation besides the passive methods. One active approach for reducing sound radiation from structures is the active structural acoustic control (ASAC). Since the early 90's, several theoretical studies regarding ASAC systems were presented, but only very little experimental investigations can be found for this alternative to passive damping solutions. The theoretical simulations show promising results of ASAC systems compared to active vibration control approaches.

So, for that reason in this paper an experiment is conducted to investigate the performance of an ASAC system in the frequency range up to 600 Hz. A regular sensor grid of 24 accelerometers that are interconnected to establish six radiation signals is applied to an aluminum plate. The plate is excited by a point force and the feedforward control system counteracts these vibrations with 2 inertial actuators. The radiated sound power can be reduced by 4 dB integrated over the targeted frequency band. Compared to an active vibration control system which results only in a 0.2 dB decreased sound radiation, the ASAC system is beneficial. Furthermore, the filter length and the causality aspect of the experiments are analyzed in order to investigate the digital signal processing requirements. It can be shown that short filters are sufficient for an ASAC system compared to the filter length needed for an AVC system.

In the last part of this paper some technical simplifications are investigated in order to reduce the complexity of the sound radiation filters. It is shown that the high pass filters needed to model the radiation efficiency can be neglected with a very small loss of performance.

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INTRODUCTION

Future vehicles have to be lightweight and efficient in order to reduce CO₂ and NO_x emissions. Lightweight designs consist of lightweight materials (e.g. carbon or glass fiber reinforced plastic, ultra-lightweight aluminum) and special design principles (sandwich or grid structure). These lightweight design aims for a high stiffness and a very low mass. These mechanical properties result in an efficient sound radiation of these structures. Furthermore, these structures are also prone to vibrations. So the following question arises: Can we increase the present noise levels in order to increase the efficiency of vehicles? Bearing in mind that traffic noise is the 2nd major health risk [1], the simple answer is: no. Therefore, lightweight compliant noise reduction techniques have to be further developed. Besides the passive solutions, the active solutions are seen as advantageous in the lower frequency range up to 1 kHz [2].

Known active solutions are the active noise control (ANC), the active vibration control (AVC) and the active structural acoustic control (ASAC). ANC and AVC systems are widely investigated and experimentally tested for noise control problems [3], [4]. ANC

systems have often the problem of local quiet zones which can be very small depending on the addressed frequency band. AVC systems can only indirectly influence the sound radiation of structures which results often in unsatisfying noise reduction capabilities [5]. The noise reduction performance of AVC systems can be further increased if the sensor placement is optimized for noise reduction problems [6], but the optimization potential is somehow limited.

The last solution is the design of ASAC systems. In the literature several systems are mentioned as ASAC systems. The authors of this paper want to separate systems with acoustic and structural error sensors. Using microphones as error sensors does not consequently induce the reduction of radiated sound power, because the actuated structure can act like a loudspeaker. Such applications are known as active noise vibration control systems (ANVC) and are also studied in the literature, e.g. [7] and the references therein. In this paper, ASAC systems reduce the radiated sound power of a structure by means of structural actuators and sensors, e.g. inertial shakers and accelerometers. ASAC systems are reported as the best choice for the reduction of radiated noise [8]. An ASAC system uses structural error

sensors to measure the normal velocities of the vibrating structure which are post-processed in order to generate sound power related signals. Hence, the radiated sound power of the vibrating structure can be directly influenced by the control systems. Many theoretical investigations for ASAC systems are published in the 90's [9], [10], but, to the authors best knowledge very little work has been done in order to experimentally realize ASAC systems. For feedback control systems experimental results are published in [11], where a generalized model of the vibrating structure is used to estimate the sound power. In contrast to the feedback controller, the implementation of a feedforward ASAC system requires the simultaneous measurement of a lot of sensors in a regular sensor grid. This is the reason why the experimental realization of ASAC feedforward systems is not focused so far. Bearing in mind that the cost for sensors constantly fall the realization of ASAC systems becomes more likely.

Therefore, this paper investigates the experimental realization of a feedforward ASAC system on an aluminum plate. In order to realize the ASAC system, a regular grid of 24 accelerometers is interconnected to generate 6 radiation signals. By reducing these radiation signals, the feedforward controller directly reduce the radiated sound power. Beside the sound power reduction, this paper focuses on the required filter length needed to implement the finite impulse response (FIR) filters and the influence of causality on the sound power reduction. Towards the technical realization of such an ASAC feedforward system, some simplifications of the ASAC system are studied (e.g. violation of the baffled condition, the need of high-pass filter to represent singular values).

The study is structured as follows: First of all, the experimental test-bed is described, followed by a description of the design method of the sound radiation filter. In the next part of the paper the validity of the sound radiation filter is demonstrated for the test-bed. Afterwards, the results of the experiments are described in detail. The paper ends with a summary and an outlook on future research.

EXPERIMENTAL TEST-BED

In order to experimentally investigate the sound radiation filter, a wooden mock-up is used. In the wooden mock-up an aluminum plate with dimension $800\text{mm} \times 600\text{mm} \times 3\text{mm}$ is mounted on its four corners with an offset of 35mm in x- and y- direction. A model picture of the test-bed is shown in [Figure 1](#).

The wooden frame is connected via shock mounts to an aluminum frame in order to decouple the test-bed from the ground. The plate is fixed between two shims and M6 nuts on a threaded steel rod on each corner. The nuts are tightened with a momentum of 15 Nm. It has to be noticed, that this mounting induces a small slit between the plate and the wooden frame which is a violation of the theory used for the implementation of the sound radiation filters. The effect of this violation is discussed later on.



Figure 1. Model of the test bed

The aluminum plate is equipped with 24 accelerometers in a 6×4 regular grid. Furthermore, for the sound radiation filter design, a fine grid, consisting of 25×15 reflective points, is glued to the plate which is scanned with a laser scanning Doppler vibrometer (LSDV). The radiated sound power of the plate is measured with a PP (pressure-pressure) sound intensity probe in a semi-anechoic room. A picture of the equipped test plate is shown in [Figure 2](#).

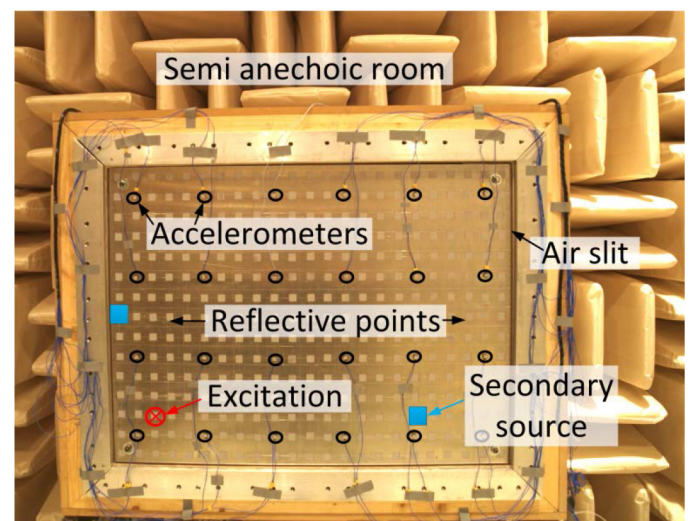


Figure 2. Picture of the equipped aluminum plate with coarse sensor grid (accelerometers) and fine sensor grid (reflective points)

In future studies, the plate will be excited with sound fields which can be generated with the speakers shown in Figure 1. But for this preliminary study, the excitation of the plate is realized with a classical shaker and the secondary sources are realized with inertial shakers which are glued to the rear-side of the plate. The placement of the inertial actuators is done with a simple method. The absolute values of vibration patterns (surface velocities) are measured with the LSDV and are overlaid for the frequencies from 100 Hz to 600 Hz. Finally, two inertial shakers are placed in the areas of maximum velocity. The sensors and actuators are low-pass filtered and are connected to the real-time processor board. The frequency range addressed in this study is 100 Hz up to 600 Hz. The lower frequency range is set by the cut off frequency of the anechoic room. Below this frequency no free field conditions are achieved. The upper bound of 600 Hz is reasoned by the modal density of the plate. They are 35 vibration modes present up to 600 Hz which is a large modal density compared to other studies [8], [12]. Detailed information of the used experimental hardware can be found in Table 1.

DESIGN OF THE SOUND RADIATION FILTER

In order to design a sound radiation filter the method developed by Gibbs et al. called "Radiation modal expansion" is used [13]. It is based on the elemental radiator formulation where the radiated sound power of a baffled plate can be estimated with the radiation resistance matrix $\underline{\underline{R}}(\omega)$ and the vector of surface velocities $\underline{v}_e$ of the plate. Here only a very short summary is presented, further details can be found in [14].

$$P(\omega) = \underline{v}_e^H(\omega) \cdot \underline{\underline{R}}(\omega) \cdot \underline{v}_e(\omega)$$

(1)

Table 1. Experimental hardware

Hardware	Type	Additional Information
Sound intensity probe	Brüel & Kjær 3599 and Modul 3050-A-060	-
Accelerometers	PCB Piezotronic 352A24/NC	Mass: 0.8 g
Secondary sources	Visaton EX45S 8 Ohm	Mass: 60 g Power: 10 W
LSDV	Polytec PSV 400	-
Real-time system	dSpace DS1006	Sampling frequency: 1200 Hz
Excitation shaker	LDS V201	-
Low-pass filter	KEMO Card Master 255G	-

The radiation resistance matrix (RRM) is calculated by the geometric properties of the sensor grid

$$\underline{\underline{R}}(\omega) = \frac{\omega^2 \rho_0 A_E^2}{4\pi c_L} \begin{bmatrix} 1 & \frac{\sin(kd_{12})}{kd_{12}} & \dots & \frac{\sin(kd_{1N})}{kd_{1N}} \\ \frac{\sin(kd_{21})}{kd_{21}} & 1 & & \vdots \\ \vdots & & \ddots & \vdots \\ \frac{\sin(kd_{N1})}{kd_{N1}} & \dots & \dots & 1 \end{bmatrix}, \quad (2)$$

where ω is the circular frequency, ρ_0 and c_L are the density of and the speed of sound in air, A_E is the area of an elemental radiator, k is the wave number, and, d_{xy} is the distance between the x and y elemental radiator. Now, singular value decomposition can be applied to the real and positive matrix $\underline{\underline{R}}(\omega)$.

$$\underline{\underline{R}}(\omega) = \underline{\underline{Q}}^T(\omega) \underline{\underline{\Delta}}(\omega) \underline{\underline{V}}(\omega) \quad (3)$$

The RRM is decomposed to the matrix of singular values $\underline{\underline{\Delta}}$ and the matrices of the left $\underline{\underline{Q}}$ and right $\underline{\underline{V}}$ eigenvectors (so called radiation modes (RM)). Plots of the first ten singular values $\underline{\underline{\Delta}}$ of the aluminum plate for the fine sensor grid are presented in Figure 3.

It can be easily seen, that the first five singular values dominating the response up to 600 Hz. Therefore, only the first five radiation modes, summarized in $\underline{\underline{\hat{Q}}}$, are used to estimate the radiated sound power. In order to ease the implementation of the radiation resistance matrix, the RME [13] method removes the frequency dependency of the radiation modes and uses the radiation modes at a fixed frequency ω_k .

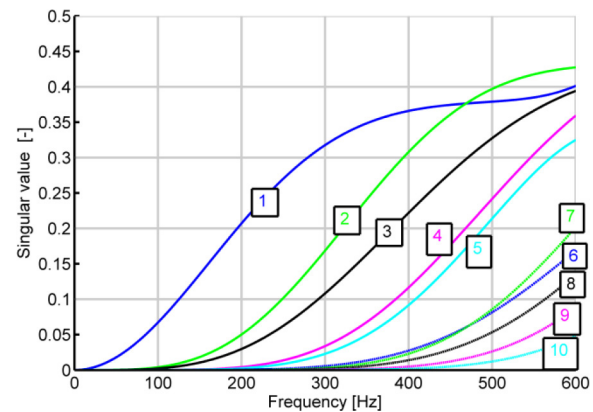


Figure 3. First ten singular values of the radiation resistance matrix

To ensure the accuracy of the sound power prediction, the residual singular values ψ are calculated by

$$\psi^2(\omega) = \text{trace}(\underline{\underline{\hat{Q}}}^H \underline{\underline{R}}(\omega) \underline{\underline{\hat{Q}}}). \quad (4)$$

The approximated sound power $\hat{P}(\omega)$ of the sound radiation filter is then calculated by

$$\hat{P}(\omega) = \underline{v}_e^H(\omega) \underline{\underline{\hat{Q}}}^T \psi^2(\omega) \underline{\underline{\hat{Q}}} \underline{v}_e(\omega).$$