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NOISE CONTROL FOR QUALITY OF LIFE

Flat sources for active acoustic shielding based on distributed control of a vibrating plate coupled with a thin cavity

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ABSTRACT

Air cavities between plates are often used to improve noise insulation by passive means, especially at high frequencies. Such configurations may suffer from resonances, such as due to the mass-air-mass resonance. Lightweight structures, which tend to be undamped, may suffer from structural resonances as well. Active methods have been suggested for improved noise insulation of plates, using piezoelectric patch actuators or inertial mass actuators. Other active methods for improved noise insulation can be based on acoustic control of the sound field in the cavity, using acoustic sources and acoustic sensors. Methods based on feedforward control of the sound radiated from such panels with air cavity usually suffer from an irregular frequency response of the actuators on the radiating panel and insufficient acoustic control authority at low frequencies. This paper presents a method to realize the combination of the air cavity and a radiating surface with a well controlled vibration distribution over the radiating surface. A specific distributed controller results in a smooth and well defined frequency response over a broad frequency range, enabling effective feedforward control of the radiated sound. Experimental results agree with numerical predictions.

1. INTRODUCTION

Reduction of noise transmission in partitions is often achieved by increasing the mass of the partition or by using an air gap between two panels. A larger acoustic compliance and therefore less coupling is obtained if the air gap is larger. Therefore, larger air gaps are often used to improve insulation provided the resulting mass-air-mass resonances can be dampened sufficiently. Compact partitions with narrow air gaps lead to a higher acoustic coupling and therefore the acoustic insulation performance may be less than desired. Nevertheless, at high frequencies useful improvement of the acoustic insulation can be obtained by using a relatively narrow air gap. In this paper we present an actively controlled acoustic source which is able to improve the acoustic insulation at low frequencies while providing the insulation at high frequencies from a narrow air gap.

In order to be able to control the low frequencies of arbitrary acoustic disturbances with a relatively simple active system, the active part of the structure preferably is as stiff as possible in order to reduce the modal complexity. The active part of structure consists of a sandwich plate which is perforated at

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one side in order to allow air to flow in and out of the sandwich core, and to increase the effective air layer thickness contained in the partition. This leads to two advantages: firstly, the acoustic coupling in the partition is reduced and secondly, the efficiency when used as an acoustic source becomes higher [9]. Conventional techniques to realize an acoustic source attempt to move the membrane or the panel as a single, rigid body thereby providing a pistonic movement. However, in reality, the structure not only vibrates with pistonic motion. Also the resonances due to the bending waves should be taken into account, especially at higher frequencies. These unwanted resonances of the structure lead to a coloured, uneven frequency response of the source. Furthermore, these unwanted resonances can make an active control system less effective because of the increased sensitivity to environmental changes. In order to have an even frequency response, many researchers were trying to eliminate these unwanted resonances by designing the materials, the structure, and the suspension of the membrane. Distributed Mode Loudspeakers (DMLs) appeared in 90's. Instead of trying to eliminate these unwanted resonant modes, DML uses the panels in the high-modal density regime [1-3]. DML applies filters to compensate and to equalise the uneven frequency response. This technique offers the advantage of compact dimensions. However, the resonances in the panel are complex and difficult to control, which often leads to complicated computations and insufficient low frequency response. Extended development of DML was to excite the DML panel by several exciters, which is known as Multiactuator Panels (MAPs) [4, 5]. Moreover, an array of loudspeakers is also used to improve the low frequency response [6]. At low frequencies, these small loudspeakers are driven all together, moving like a large loudspeaker. This technique extends the low frequency response by utilising multiple small loudspeakers. However, to obtain sufficient acoustic power at low frequencies, this technique needs more loudspeakers, since the radiating sound power efficiency of a small loudspeaker is too low at very low frequencies.

In this paper, a novel panel structure for the sound radiating panel is described, which has low density, high bending stiffness and low acoustic stiffness [8] and which offers the advantages of efficient space utilization and low modal density. Low modal density reduces the complexity of the panel vibration. A simple and stable control method was developed to improve the frequency response of the flat loudspeaker. The control method consists of two parts. The first part is to flatten the frequency response and the second part is to increase the response at low frequencies. Inappropriate control positions can easily lead to an unstable control system. Using a specific approach, it is shown in this paper that a stable control system and economic utilisation of sensors can be obtained, by carefully designing the positions of the sensor-actuator pairs and by using specific combinations of sensor-actuator pairs for the different control loops.

2. EXPERIMENT SET-UP AND NUMERICAL MODEL

2.1 Experiment set-up

A novel panel structure with low density and high stiffness was used as the sound radiating panel. The panel was attached to the cabinet with rubber suspension on the edges of the panel. Voice coil actuators were attached to the panel to excite the panel. The dimensions of the panel were 605*415*22 mm^3. The mode shapes and the resonant frequencies of the panel were measured while a shaker applied force on various positions of the panel (Figure 1). The microphone was put 5 cm above the panel to measure the near-field radiating sound pressure from the panel.

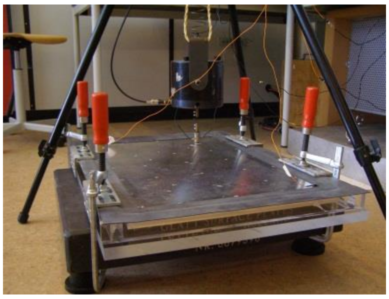

Figure 1 - Mode shapes and resonant frequencies measurement of the panel attached to the cabinet

2.2 Numerical Model

Lumped element model

A lumped element model of the perforated plate coupled with the cavity and moving coil actuators was built. The lumped element model assumes that the plate moves as a rigid body. The different acoustic impedances representing the acoustic load at the back side of the perforated plate is shown in Figure 2. The sound pressure level with feedback control and combined feedback and feedforward control is shown in Figure 3. The feedback control is based on current drive for the moving coil actuators and collocated velocity sensing. In Figure 3 it can be seen that in this example a smooth response is possible over a frequency range of 40 Hz to at least 1 kHz. Practical panels can not be modeled with lumped elements because they will suffer from resonances due to the finite bending stiffness of the panel. Experiments were carried out to evaluate different control strategies for the control of the higher order resonances of the system. Experiments with decentralized control of the multiple actuator-sensor combinations in order to be able to control higher order resonant mode was not successful because of the vulnerability for low frequency instability. Therefore, a finite element model was made, which enabled the design of the controller based on a model of the system.

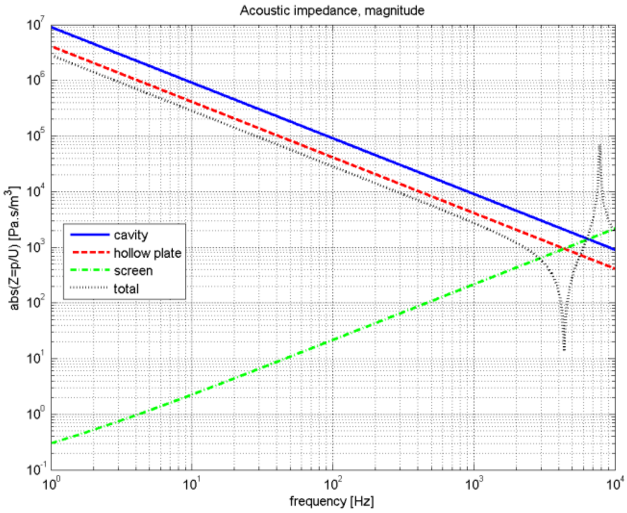

Figure 2 – Lumped acoustic impedance load as seen by the back of the radiating structure.

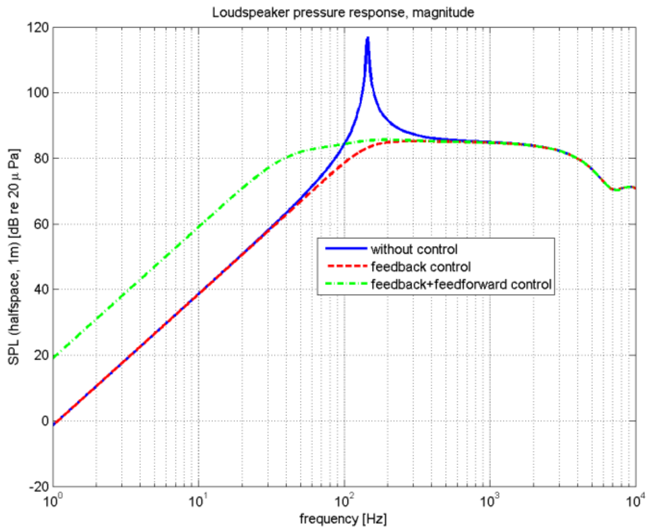

Figure 3 – Sound pressure level at 1m distance, 1W nominal input power, half-space radiation

Finite Element model

We used the finite element method (FEM) with the COMSOL Multiphysics 4.2 (COMSOL, Inc., Burlington, MA 01803, USA) to model our system. To accurately model the characteristics of the system, the acoustic and structural properties must be considered simultaneously. There were two domains in our model: the fluid domain and the solid domain. The physical quantities on the fluid-solid interacting boundaries of these two domains were coupled. On the fluid-solid interacting boundaries: The fluid pressure in the fluid domain produced the normal force to the structure in the solid domain. In the meanwhile, the normal acceleration to the acoustic pressure in the fluid domain can be derived from the acceleration of the structure in the solid domain. To validate our numerical model, we compared the resonant frequencies between the simulation and experiment for the panel without the cavity. Table 1 shows that our numerical model can estimate the resonances of the system with reasonable error.

Table 1 - Resonant frequencies of the panel

Mode	Experiment [Hz]	Simulation [Hz]	Error
1	221	218	1.3%
2	256	243	5.0%
3	506	510	7.9%
4	571	541	5.2%
5	611	643	5.2%

3. CONTROL PERFORMANCE

3.1 Control and Excitation Positions

We applied voice coil actuators on the panel to excite and control the panel. However, the position of the actuators affects the vibration response of the flat loudspeaker system. Figure 4 shows that the kinetic energy response of the panel varies when the excitation position changes. The energy of the near field sound pressure wave is related to the kinetic energy of the radiating panel at lower modes [7]. Therefore, the kinetic energy of the radiating panel can represent the near field sound.

Moreover, the configuration of the sensors and actuators also affects the control stability and performance. Therefore, we have analysed the response and the mode shapes of the flat loudspeaker to obtain the optimal control configuration. Figure 5 shows the near-field sound pressure level response of the control results of two sensor-actuator configurations. With control, the first resonant peak was reduced remarkably by both of these two configurations. However, the resonant peak at 420 Hz was hardly reduced by applying sensor-actuator configuration 2. This peak was even increased by applying sensor-actuator configuration 1. In order to control this resonant peak, a further sensor-actuator configuration design is necessary. Therefore, we selected different detection and excitation positions based on a mode shape analysis.

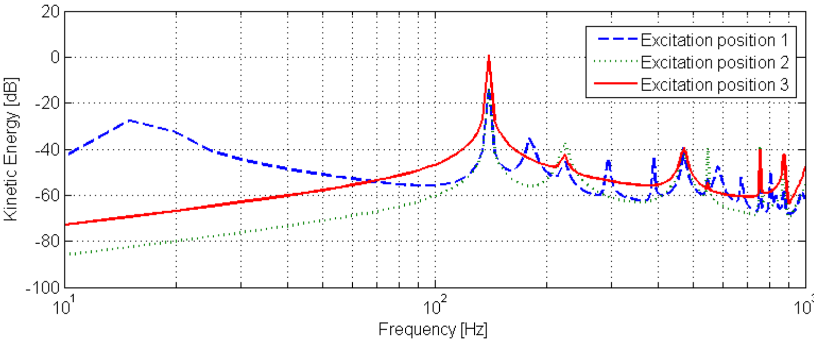

Figure 4 - Simulation of the kinetic energy response of the panel with various excitation positions

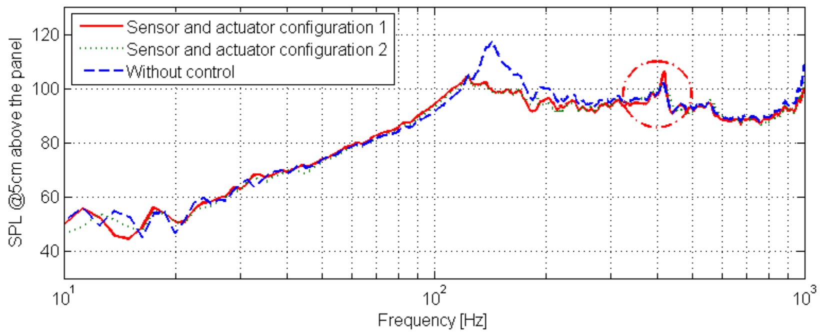

Figure 5 - Sound pressure level (SPL) response of the flat loudspeaker with various sensor-actuator configurations

3.2 Combined Control Performance

With further mode shape analysis, the optimised control configuration 3 was obtained. This configuration did not only reduce all the resonant peaks below 1 kHz, it also enabled a reduction of the number of sensors. Figure 6 shows that the control method equalizes the frequency response of the flat panel. With the combination of method 1 and 2, the attenuated response of the loudspeaker at low frequencies can be compensated.

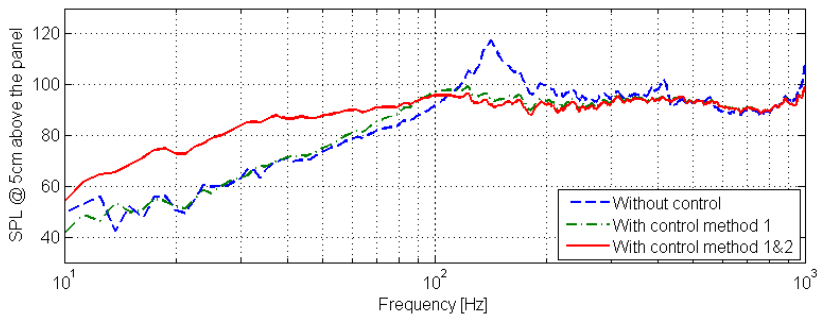

Figure 6 - SPL response of the flat loudspeaker with control sensor-actuator configuration 3

4. CONCLUSIONS

A flat loudspeaker comprising a novel panel structure with a stable and simple control method has been studied in this paper. This novel panel structure offers low density, high stiffness, and efficient space utilization. Our work shows the frequency response of the flat panel can effectively be flattened. The insufficient response of the flat panel loudspeaker at low frequencies can be increased. Furthermore, appropriately designed positions of sensors and actuators can stabilize the control system and minimize the amount of sensors. The flat loudspeaker in this paper obtains a very even frequency response from 30 Hz to 1kHz and offers compact dimensions, a simple algorithm, stable control, and a small number of sensors.

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