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HiFi Panel Speaker by Controlling the Vibration Field Using Array Actuators

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ABSTRACT

Most of panel speakers adopt a thin rectangular plate, which is excited by one or several actuators. Because of the multi-modal characteristic of the plate, the vibration field composed of many modes deteriorates the radiated sound quality inherently, in particular at the low to medium frequency range. This study deals with the control of a point-excited panel speaker to radiate HiFi and powerful sound using an actuator array enclosing the main actuator zone. The desired control is to confine the major in-phase oscillation at around the main actuator and to suppress the vibration in the remainder area, resulting a virtual circular speaker and a baffle. The gain of actuators is obtained by solving the inverse rendering problem derived from the transfer function between the actuator input and the response of field point on the plate. Two types of array configurations are investigated by simulation, which shows that the adjacent double circular array can produce the sound radiation for a wide frequency band. The experiment conducted using the double circular array reveals that the vibration field can be controlled well as desired and the present control method can improve the frequency response drastically to be flat.

Keywords: Panel speaker, inverse rendering, vibration control, array actuators, radiated sound

1. INTRODUCTION

Flat panel speakers often adopt a thin rectangular plate as the radiator excited by one or several actuators. Due to the multi-modal characteristic of the plate, however, the resultant vibration field is usually contributed from many modes. Such complicated vibration field deteriorates the radiated sound quality inherently, especially at the low to medium frequency range, and the radiation efficiency is low. So, it is important to control the vibration field of the panel speaker to make it a HiFi radiator. In this study, the desired vibration field is a system which is a combination of a circular piston and a baffle, radiating desired sound power. The basic concept is shown in Fig. 1. The control actuator array surrounding the main actuator is configured to generate the desired field. The inverse rendering technique is adopted to obtain the proper complex gain of control actuators.

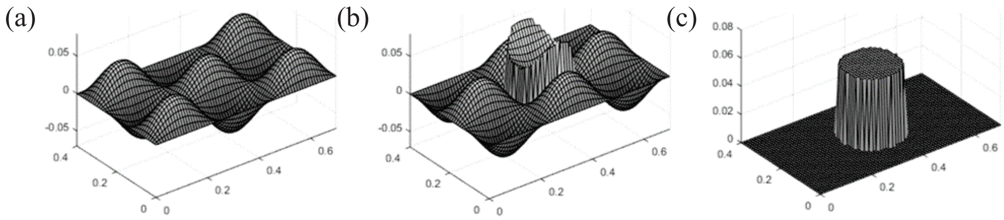

Figure 1 – Schematic of the concept of the vibration field control for the rectangular panel speaker: (a) initial field by main actuator, (b) target field for control actuators, (c) desired ideal field.

2. FIELD CONTROL USING AN ACTUATOR ARRAY IN DOUBLE LAYERS

To convert the initial field due to the main actuator into the desired field after control, the inverse problem should be set using the transfer function between the input force of each control actuator and velocity response of each field point on the plate. The vibration field of an infinite plate excited by a

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point force can be expressed by travelling wave form (1) as

$$v(r;\omega)=\frac{F_0\omega}{8Bk_B^2}\left[H_0^{(2)}(k_B r)-H_0^{(2)}(-jk_B r)\right], \tag{1}$$

where r is distance between excitation point and field point, F_0 the excitation force, B the bending rigidity, and k_B the bending wave number. On the other hand, the vibration field of a finite plate excited by a point force can be defined by the mode summation method (2, 3) as follows:

$$v(x,y;\omega)=\sum_{i=1}^{\infty}\frac{j\omega\psi_i(x,y)}{\left\{\omega_i^2(1+j\eta_i)-\omega^2\right\}\left\{\int_S\rho_p h\psi_i^2(x,y)dxdy\right\}}\int_S F_0\delta(x_n,y_n)\psi_i(x,y)dxdy\ . \tag{2}$$

Here, ψ is the mode shape function, i the index of the i th vibration mode, η the loss factor of the plate, ρ_p the plate density, and h the thickness of the plate.

In this work, Eq. 2 is to be used for deriving the transfer function, assuming that the steady-state vibration field is considered for a finite plate. Nevertheless, it is also important to look into the vibration field in travelling wave form of Eq. 1 to investigate the wave propagation invoking the controlled field. By using Eq. 2, the target vibration field to control the initial field is defined as

$$v_t(x_m,y_m;\omega)=\begin{bmatrix} T-v_0(x_m,y_m;\omega) & \text{if } (x_m,y_m)\in \text{speaker zone} \\ -v_0(x_m,y_m;\omega) & \text{if } (x_m,y_m)\in \text{baffle zone}\end{bmatrix}, \tag{3}$$

where T is the desired velocity of the speaker zone, v_0 the initial field by the main actuator, and v_d the desired vibration field to be generated on the plate. When Eq. 3 is expressed in a matrix form with respect to the field points and the input gain of control actuators, one can obtain the actuator gain to generate the desired field by solving the inverse problem (2-4) as follows:

$$\mathbf{V}_\mathrm{t}=\mathbf{GF} \quad\rightarrow\quad \mathbf{F}=\mathbf{G}^{\dagger}\mathbf{V}_\mathrm{t}\,. \tag{4}$$

Here, $\mathbf{F}$ is the vector consisting of actuator gain, $\mathbf{G}$ is the transfer matrix between the gain of actuators and the velocity response at field points, and the superscript $\dagger$ denotes the pseudo inverse operator. Tikhonov regularization is additionally applied to secure the stability of the solution (5, 6).

3. EFFECTIVE FREQUENCY RANGE FOR SOUND RADIATION

3.1 Simulation set-up

A simply supported aluminum plate with a size of 700 mm x 400 mm x 2 mm is chosen as the plate for the test. A main actuator attached to the plate center excites a unit force within the frequency range of 50-1000 Hz. Control actuator arrays configured in the double layer are shown in Fig. 2. The inside area of the inner array is defined as the speaker zone and the remaining area as the baffle zone. The desired velocity of the speaker zone is determined to radiate 80 dB of sound power.

3.2 Determination of Effective Frequency Range

For the effective sound radiation, the speaker zone should have velocity field with the desired magnitude and uniform phase distribution. On the other hand, the velocity field of the baffle zone should be no larger than the tolerance value. The tolerance value is determined by a criterion on the desired difference in sound power radiation contributed from each zone. In this work, it is set to be larger than 15 dB, for the worst condition of uniform phase distribution. The velocity error magnitude compared to the desired field can be defined as

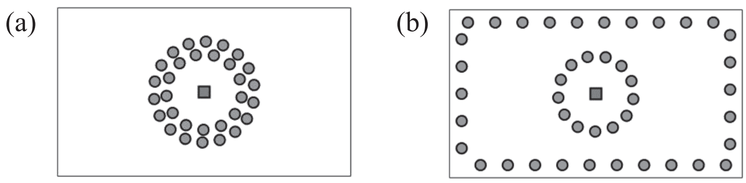

Figure 2 – Actuator array configured in double layer: (a) double circular array, (b) circular array and edge array located at the periphery of the plate.

$$e_{spk} = \left(|T| - \overline{|v_{spk}|}\right) \Big/ |T|\,, \quad e_{bff} = \left(|v_{tol}| - \overline{|v_{bff}|}\right) / |v_{tol}|\,. \tag{5}$$

Here, v_{tol} is the velocity tolerance of the baffle zone and $\bar{v}$ the spatially averaged velocity. Figure 3 shows the velocity magnitude error and the standard deviation of phase distribution in the controlled field when the diameter of speaker zone is 0.18 m. One can find that the double circular array has smaller error than the circular and edge array. The error in the baffle also reveals that the double circular array is the best in making a satisfactory baffle zone. From the baffle zone result, the lower limit of the effective frequency range can be determined. Considering the sound power radiation, it is found that the variation of the sound power is less than 3 dB when the standard deviation of velocity phase is less than 20°. Based on this criteria, the high limit of the effective frequency range is selected for each array configuration. As a result, the effective frequency range of each array configuration is determined varying the speaker zone diameter as shown in Fig. 4. One can observe that the circular and edge array has the effective frequency range only for the speaker diameter in the range of 0.24-0.28 m. If the 15 dB criterion is relieved to 10 dB, the ranges become wider.

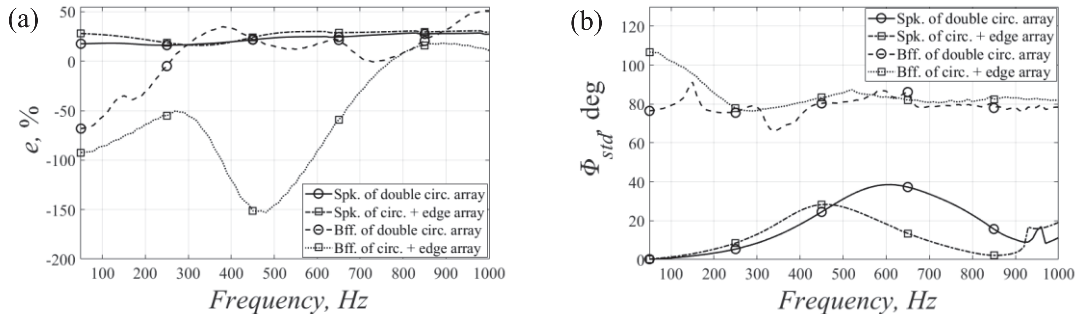

Figure 3 – Result of velocity magnitude error and phase distribution in the speaker and baffle zone: (a) velocity magnitude error, (b) phase distribution.

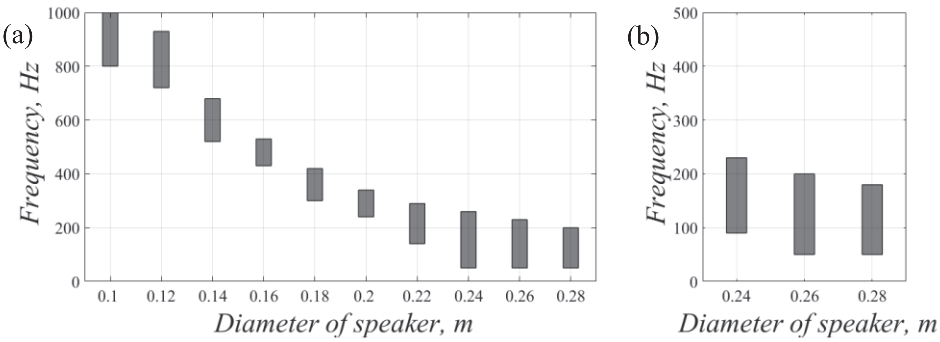

Figure 4 – Frequency range for effective sound radiation: (a) double circular array, (b) circular and edge array.

4. MEASUREMENT OF TRANSFER FUNCTION

4.1 Experimental Set-up

To validate the vibration field control method suggested in this work, an experiment is conducted for the test plate under the same condition with the simulation. However, the actual boundary condition would be a bit different from the ideal simply-supported one. The double circular array is configured with 31 control actuators (Tectonic, TEAX14C02-8) and a main actuator (Dayton Audio, DAEX25FHE-4), which is attached to the geometric center of the plate. The vibration field is measured using laser vibrometer (Polytec, OFV 056) at 325 observation points on the plate. The transfer function for the control is determined by using the measured velocity response at each observation point and the input voltage signal of each control actuator.

4.2 Result of Vibration Field Measurement

The measured frequency range of interest is 200-570 Hz considering the resonance characteristic of the actuator and modal overlap of the test plate. To investigate how much the speaker and baffle zones are generated as intended, the *sectioning performance (SP)* is defined as follows:

$$SP = E_{spk} / E_{total} = \left(\int_{S_{spk}} \left| v(\omega) \right|^2 dS \right) \Big/ \left(\int_S \left| v(\omega) \right|^2 dS \right). \tag{6}$$

Eq. 6 means the ratio of the vibration energy concentrated in the speaker zone to that of the total plate and the result is shown in Fig. 5(a). Compared with the initial field by a main actuator only, one can see that the most of vibration energy is concentrated in the speaker zone by the control. Figure 5(b) shows the sound power radiated from the plate. After the field control by the actuator array, the power spectrum become flattened and the magnitude is enhanced satisfying the desired value of 80 dB.

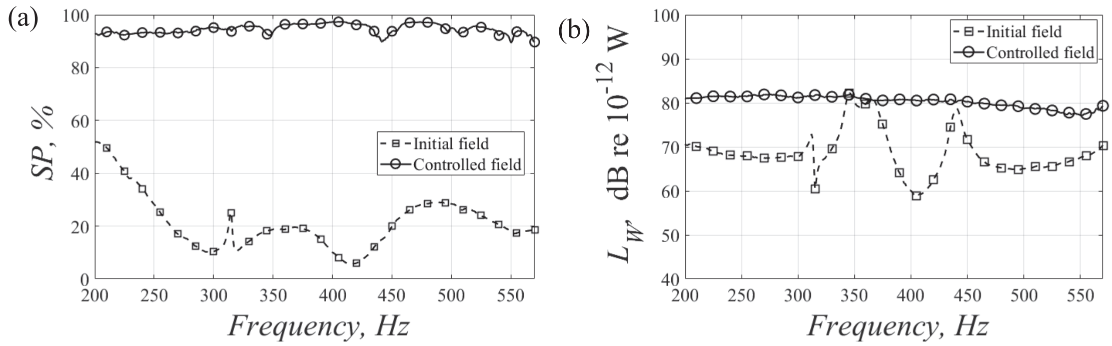

Figure 5 – Experimental result of vibration field control: (a) sectioning performance, (b) estimated sound power radiation from the controlled vibration field

5. CONCLUSIONS

In this study, the vibration field of a panel speaker excited by a main actuator is controlled by using an array of control actuators arranged in the double layered configurations. The control performance of two types of control actuator array is simulated, and the effective frequency range for the sound radiation is determined for each array. It is noted that the double circular array has much wider effective frequency range than the circular and edge array. An experiment to validate the proposed control method shows that the vibration energy is concentrated within the speaker zone as desired by the control. The sound power radiated from the controlled vibration field reveals that the severe fluctuation in sound spectrum of the initial field is flattened and the overall magnitude is also enhanced to fulfill the desired value. Consequently, it is shown that the resultant sound field does not include the distortion, resulting an improved sound quality, owing to the control by control array.

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