DESIGN OF LARGE PLANAR DIAPHRAGM INCORPORATING MULTIPLE VIBRATORS FOR SOUND DIRECTIVITY CONTROL VIA FEM AND BEM

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ABSTRACT

We have realized a sound directivity control of a large planar diaphragm by controlling the bending vibrations using multiple vibrators. This paper proposes a method to determine the parameters such as the shape and thickness of the diaphragm and the position of the vibrators. In this method, the finite element method (FEM) is used to simulate the diaphragm vibrations and the boundary element method (BEM) is used to simulate the radiated sound. We first validate the simulation results by measuring the actual bending vibrations and sound directivity and comparing them with the simulated results. We then show that metaheuristics are effective in finding appropriate parameters because the sound directivity largely varies in a continuous manner with variations in the parameters.

Index Terms— Directional sound radiation, Sound directivity control, Acoustic transducer, FEM and BEM analysis

1. INTRODUCTION

We have constructed a system that can control the sound directivity of a large planar diaphragm by controlling the bending vibrations using multiple vibrators. In our system, multiple vibrators accelerate the diaphragm cooperatively but independently to modulate the bending vibrations at each frequency such that the directivity of the radiated sound is changed. Given, a priori, a lookup table matching accelerated vibrations with sound directivity, the system can control sound directivity by forcing the corresponding vibrations.

In [1], we showed the feasibility of this method by using a prototype system with a circular plate–one of the most typical shapes in discussions on plate vibration–as a diaphragm, and three vibrators fastened to the diaphragm at evenly spaced points. We measured the surface vibrations on the diaphragm and the radiated sound while changing the relative phases of the three vibrators, and showed visually that the surface vibrations and sound directivity change depending on the relative phases of the forced vibrations.

The controllability of the sound directivity strongly depends on the parameters–such as the shape, thickness, and material properties–of the diaphragm and also on the number and position of the vibrators. These parameters should be chosen to match the required sound directivity. In this paper, we propose a method to estimate the sound directivity for various parameters via simulation so as to choose the best parameters for the given requirements.

To prove the feasibility of this method, we show that the simulated and measured bending vibrations and sound directivity are approximately coincident. We then show that metaheuristics such as simulated annealing are effective in finding appropriate parameters because the directivity largely varies in a continuous manner with variations in the parameters.

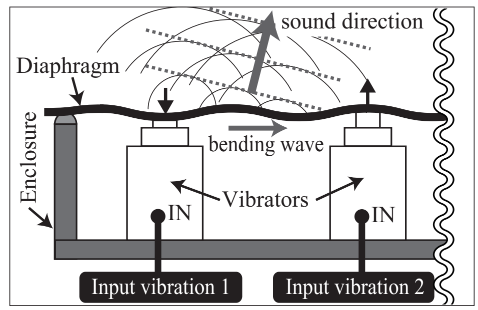

Fig. 1. Inducing bending vibration using multiple vibrators.

2. STRUCTURE AND SIMULATION OF PLANE VIBRATION AND RESULTING SOUND RADIATION

Fig.1 illustrates the concept of our directional sound reproduction system. Our system forces bending vibrations in a planar diaphragm using multiple vibrators. Because the vibrations in the diaphragm are determined by not one but all of the forcing vibrations, the vibration of the diaphragm can be controlled by modulating the forcing vibration waveforms relative to each other. As a consequence, the directivity of the radiated sound is also varied [1].

The equation of motion of a bending wave on a plate is as follows:

$$\nabla^4\eta + \frac{3\rho(1-s^2)}{Qh^2}\frac{\partial^2\eta}{\partial t^2} = F \tag{1}$$

where ρ is the density of the material; s, its Poisson ratio; and Q, its Young's modulus. h is the half-thickness of the plate; t, time; and F, the force acting on the plate. $\nabla^4 = \nabla^2\nabla^2$, where ∇^2 is the Laplacian operator.

When the plate vibrates freely, $F = 0$ in equation (1). For harmonic solutions $\eta = Z(x, y)e^{j\omega t}$, where (x, y) denotes the Cartesian coordinate system and ω is the angular velocity; then, the differential equation for Z can be written as

$$\nabla^4 Z - \frac{3\rho(1-s^2)\omega^2}{Qh^2}Z = \nabla^4 Z - k^4 Z = 0, \tag{2}$$

where

$$k^2 = \frac{\sqrt{3}\omega}{h}\sqrt{\frac{\rho(1-s^2)}{Q}}.$$

The bending waves in a plate are dispersive; that is, their velocity v depends on the frequency as

$$v(f) = \frac{\omega}{k} = \sqrt{2\pi f h}(\frac{Q}{3\rho(1-s^2)})^{-4}. \tag{3}$$

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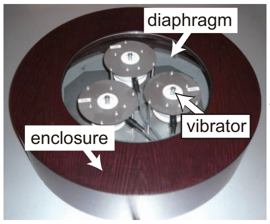
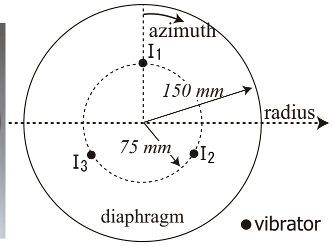


(a) Prototype system



(b) Position of vibrators on diaphragm

Fig. 2. Constructed prototype

Therefore, the velocities of the bending waves on such a thin plate are proportional to the square root of their frequencies [2].

When the plate is forced to vibrate by unequal forces, as in the proposed system, the equation of motion is difficult to solve in an accurate manner. In such cases, it is common to find approximate solutions using the finite element method (FEM). In FEM, a continuum or continuous structure is divided into small elements of finite dimensions called finite elements. The original structure is then considered as an assemblage of these elements connected to each other at a finite number of points. The properties of the elements are formulated and then combined to approximate the properties of the entire body.

Furthermore, the characteristics of the sound radiated from a vibrating plate can be calculated approximately using the boundary element method (BEM). In BEM, the domain is assumed to be uniform and only the boundary of the domain is discretized to compute the solution for the whole domain. Thus, less input data is required compared with FEM. Therefore, it is well suited for simulating acoustic radiation.

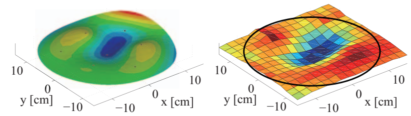
Several FEM software products are available, e.g., ABAQUS, ANSYS, MSC.MARC, and NX Nastran. We adopt ANSYS [3] for structural analysis of the plane vibration because of its usability. For sound radiation analysis, we adopt WAON [4], which is a software program designed for acoustic field analysis using BEM. This software uses the fast multiple boundary element method (FMBEM) to analyze ultra-large-scale fields in a small amount of time and using a small amount memory.

3. OVERVIEW OF PROTOTYPE SYSTEM

All the experimental data presented in this paper were measured using a prototype system constructed by us.

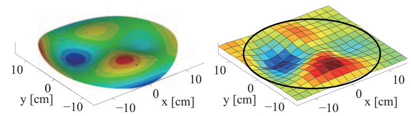
Fig. 2(a) shows a picture of the prototype. The system has a large planar diaphragm consisting of a circular glass plate of radius 150 mm and thickness 2.8 mm. Three magnetostrictive vibrators (Fostex GY-1) are tightly fastened with nuts to the diaphragm at a radial distance of 75 mm from the center and equidistant from each other, as shown in Fig. 2(b). They are driven by an audio device (Roland UA-1000) that is capable of streaming synchronous independent signals and can be controlled by a computer via ASIO. We generated forcing vibration waveforms on the computer using MATLAB; each vibrator were denoted as I_1, I_2, and I_3.

To prevent interference from the out-of-phase sound waves originating from the rear of the diaphragm, the system was placed in an enclosure that was 150 mm deep. In addition, a gap of 0.5 mm was provided between the enclosure and the edge of the diaphragm to allow free vibration. (In the future, this gap will be sealed, as in the case of a typical loudspeaker.) The diaphragm was supported only by the vibrators.



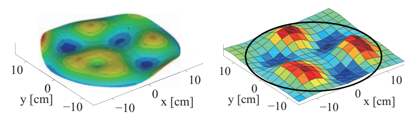
(a) Simulation (b) Measurement

(1) I_1 and I_2 were actuated at 1 kHz in phase.



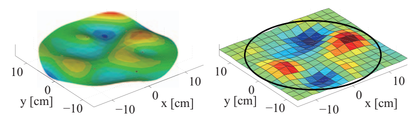
(a) Simulation (b) Measurement

(2) I_1 and I_2 were actuated at 1 kHz in reverse phase.



(a) Simulation (a) Measuring data

(3) I_1 and I_2 were actuated at 2 kHz in phase.



(a) Simulation (b) Measurement

(4) I_1 and I_2 were actuated at 2 kHz in reverse phase.

Fig. 3. Sound pressure mapping of diaphragm.

4. BENDING VIBRATIONS OF THE DIAPHRAGM

In this section, we show that the FEM simulation results are approximately coincident with the actual behavior of the diaphragm. We compare the simulation results with measurements performed using the prototype system.

In the simulations, the vibrations of the diaphragm modeled according to the properties of the prototype system were analyzed using ANSYS, a FEM analysis software. Because the diaphragm of the prototype system was made of float glass, the density of the diaphragm material was assumed to be 2.5 g/cm^3; its Poisson's ratio, 0.20, and its Young's modulus, 7.0×10^{10} Pa. The thickness of the diaphragm was 2.8 mm. The model of the diaphragm plane was composed of 836 nodes and 1400 elements.

In the measurements, the vibration of the diaphragm of the prototype system was measured by measuring the sound pressures at 19 $\times$ 15 matrix points placed at 2-cm intervals on a plane parallel to and 1.5 cm away from the diaphragm plane. The microphones were DPA omni-directional miniature microphones (type 4060). Because the measurement plane was set extremely close to the diaphragm plane, the measurement data can be assumed to be proportional to the displacement of the corresponding part of the diaphragm plane in the direction perpendicular to the plane.

The figures on the left side of Fig. 3 show the simulated displacement maps (hereafter denoted as "DispMs") on the diaphragm. The figures on the right side of Fig. 3 show the measured sound pressure maps (hereafter denoted as "SPMs") on the measurement plane. In these figures, the circled area with the black lines corresponds to the diaphragm. The red, green, and blue areas indicate that the amplitude is positive, zero, and negative, respectively, at a given moment.

The three vibrators, I_1, I_2, and I_3, were fastened at $(x, y) = (0, 7.5)$, $(7.5 \times \sin(\pi/3), -7.5 \times \cos(\pi/3))$, $(-7.5 \times \sin(\pi/3), -7.5 \times \cos(\pi/3))$, respectively. Vibrators I_1 and I_2 were actuated and vibrator I_3 was held stationary.

Figs. 3(1) and (2) show the DispMs and SPMs at a certain moment when I_1 and I_2 were actuated with a 1-kHz sine wave in phase and in reverse phase, respectively. Figs. 3(3) and (4) show the DispMs and SPMs at a certain moment when I_1 and I_2 were actuated with a 2-kHz sine wave in phase and in reverse phase, respectively. I_3 was stationary in all cases.

As shown in the figures, the simulation and measurement results are approximately coincident. Since the coefficients of correlation between DispM and SPM are 0.57, 0.72, 0.45, and 0.73 under each condition, respectively, the simulation and measurement result have positive correlation. This confirms that the bending vibration of the diaphragm plane can be estimated by FEM analysis.

5. DIRECTIVITY OF THE RADIATED SOUND

In this section, we show that the BEM simulation results are approximately coincident with the actual behavior of the radiated sound of the prototype system. We compare the simulated and measured directivity of the radiated sound.

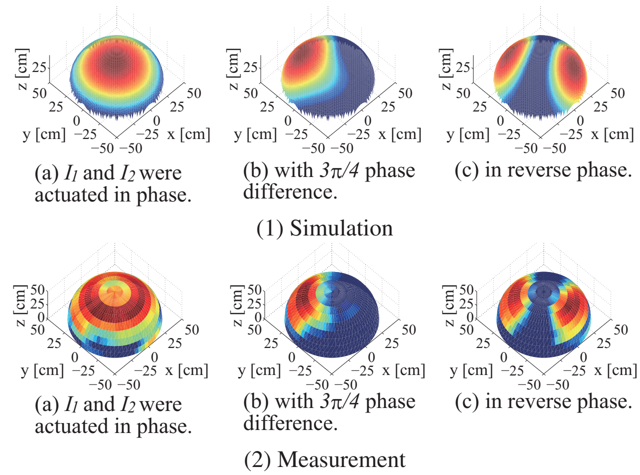
In the simulation, the radiated sound was calculated using WAON, an acoustic analysis software, on the basis of the structural analysis of the diaphragm vibration described in the previous section. The sound velocity was assumed to be 340.0 m/s and the air density, 1.225 kg/m. A total of 796 observation points were distributed on a hemispherical surface of radius 50 cm, centered at the same position as the diaphragm center.

In the measurement, the sound directivity of the prototype system was measured by measuring the sound pressures on a hemispherical surface placed at the same position as that in the simulation. The hemispherical surface contained 720 measurement points with 10° intervals in the elevation angle and 5° intervals in the azimuth. The microphones were DPA omni-directional miniature microphones (type 4060).

The figures on the first and the second line of Figs. 4 and 5 show the radiated sound pressure maps (hereafter denoted as "RSPMs") from the simulation and measurement, respectively, when vibrators I_1 and I_2 were actuated and vibrator I_3 was held stationary.

Fig. 4 show the RSPMs when I_1 and I_2 were actuated with a 1-kHz sine wave (a) in phase, (b) with a phase difference of $3\pi/4$, and (c) in reverse phase. Fig. 5 show the RSPMs when I_1 and I_2 were actuated with a 2-kHz sine wave (a) in phase, (b) with a phase difference of $3\pi/4$, and (c) in reverse phase. I_3 was stationary in all cases. In these figures, the colored areas correspond to the regions of the hemispherical surface used for measurement. The three vibrators, I_1, I_2, and I_3, were located at $(x, y, z) = (0, 7.5, 0)$, $(7.5 \times \sin(\pi/3), -7.5 \times \cos(\pi/3), 0))$, and $(-7.5 \times \sin(\pi/3), -7.5 \times \cos(\pi/3), 0)$, respectively. The red, green, and blue areas are regions where the sound pressure in decibels was high, average, and low, respectively. The sound pressure in the stark red area exceeded that in the stark blue area by more than 10 dB.

As shown in the figures, the simulated and measured results seem to be roughly coincident. Since the coefficients of correlation between the RSPMs of the simulation and measurement are 0.78, 0.82, and 0.80 at 1-kHz and 0.12, 0.18, and 0.31 at 2-kHz under each condition, the simulated and measured results have positive correlation at 1 kHz, though not at 2 kHz. At higher frequencies, the vibration mode pattern of the diaphragm becomes finer and the sound radiated by the diaphragm vibration forms a beam. Therefore, the model parameters must be configured more precisely to obtain better coincidence between the simulation and measurement results. Nevertheless, the figures show that the simulation results are similar



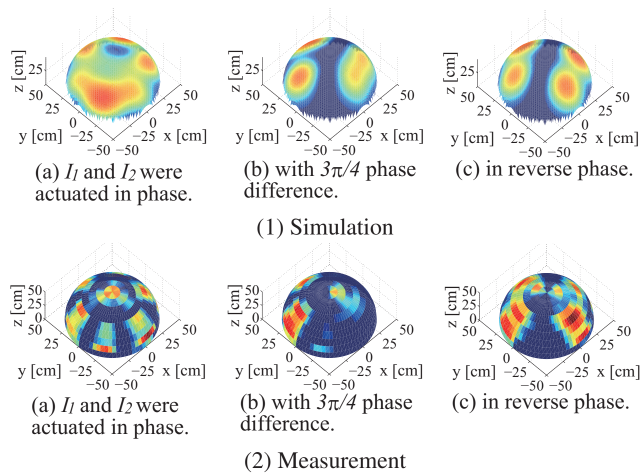
(a) I_1 and I_2 were actuated in phase. (b) with $3\pi/4$ phase difference. (c) in reverse phase.

(1) Simulation

(a) I_1 and I_2 were actuated in phase. (b) with $3\pi/4$ phase difference. (c) in reverse phase.

(2) Measurement

Fig. 4. Directivity of the sound radiated by the prototype system when I_1 and I_2 were actuated at 1 kHz.



(a) I_1 and I_2 were actuated in phase. (b) with $3\pi/4$ phase difference. (c) in reverse phase.

(1) Simulation

(a) I_1 and I_2 were actuated in phase. (b) with $3\pi/4$ phase difference. (c) in reverse phase.

(2) Measurement

Fig. 5. Directivity of the sound radiated by the prototype system when I_1 and I_2 were actuated at 2 kHz.

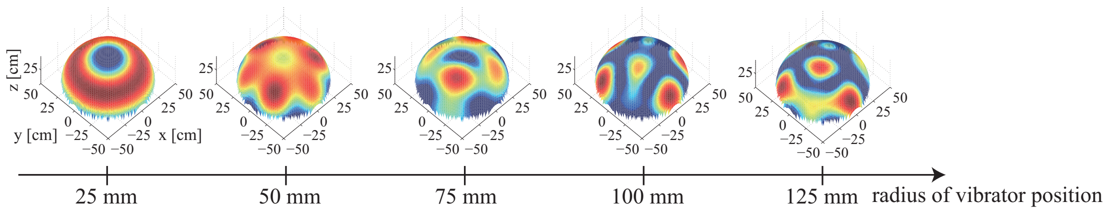
to the measurement results at any frequency. Thus, the simulation results may serve as a useful reference for deciding the system parameters.

6. METHOD FOR SYSTEM PARAMETER SEARCHING

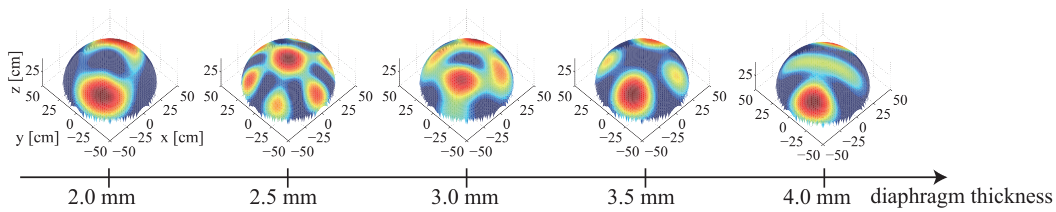
The parameters for the given requirement could be searched effectively using metaheuristics because the sound directivity largely varies in a continuous fashion with variations in the parameters.

Fig. 6 shows how the simulated RSPM varies with variations in parameters such as "the positions of the vibrators, " "thickness of the diaphragm," and "shape of the diaphragm." The original model of the diaphragm is the same as that described in sections 4 and 5, except that the thickness of the diaphragm plane is 3.0 mm. The simulation method is also same as that described in sections 4 and 5. Vibrators I_1 and I_2 were actuated with a 2-kHz sine wave in phase and vibrator I_3 was held stationary.

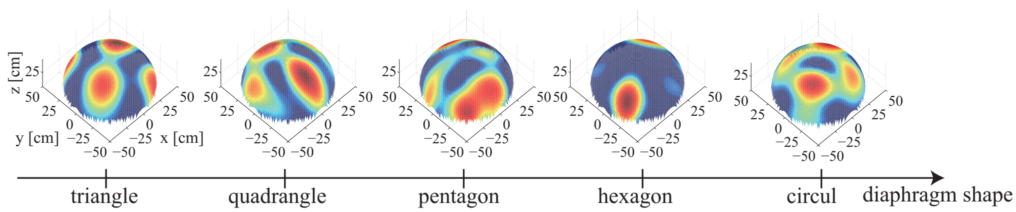
In Fig. 6(a), the radial distance of the three vibrators from the center of the diaphragm changes from 25 mm to 125 mm in steps of 25 mm. In Fig. 6(b), the thickness of the diaphragm plane changes from 2.0 mm to 4.0 mm in steps of 0.5 mm. In Fig. 6(c), the num-



(a) When the positions of the vibrators are changed.



(b) When the thickness of the diaphragm plate is changed.



(c) When the number of vertices of the diaphragm is changed.

Fig. 6. Simulation: Variation in radiated sound with changes in parameters (I_1 and I_2 were actuated at 1 kHz in phase).

ber of vertices of the diaphragm changes from three (triangle) to six (hexagon) and finally to infinity (circle).

As shown in the figures, although the RSPM changed in various ways with a change in even one parameter value, similar sets of parameter values produced similar RSPMs. In practice, though the RSPM sometimes changes drastically, the changes are largely smooth with variations in the parameters. This shows that appropriate parameters for a given requirement can be identified using metaheuristics such as simulated annealing.

7. CONCLUSION

We have constructed a system that can control the sound directivity of a large planar diaphragm by controlling the bending vibration using multiple vibrators. In our system, multiple vibrators accelerate the diaphragm cooperatively but independently to modulate the bending vibrations at each frequency such that the directivity of the radiated sound is changed.

Rebillat et al. proposed MAP (Multi-Actuator Panel) loudspeaker that has also multiple vibrators fasten to a single panel and controlled sound directivity using WFS (Wave Field Synthesis) technique[5]. Meanwhile, our system controls sound directivity by controlling the bending vibration of the panel with fewer vibrators.

The controllability of the sound directivity strongly depends on the parameters–such as the shape, thickness, and material properties–of the diaphragm and also on the number and position of the vibrators. The proposed method can be used to find the best parameters for the system to obtain the required sound directivity by simulating the sound directivity for various parameters.

First, to prove the feasibility of this method, we showed that the simulated and measured bending vibrations and sound directivities were approximately coincident. The diaphragm vibrations were simulated using ANSYS, one of the most popular software packages for FEM analysis. The radiated sound was simulated using WAON, a software program designed for acoustic field analysis using BEM. The experimental diaphragm vibration and directivity were measured on a prototype system with a diaphragm consisting of a circular glass plate of radius 150 mm and thickness 3 mm. Three magnetostrictive vibrators were tightly fastened to the diaphragm.

Second, we show that metaheuristics such as simulated annealing are effective in finding appropriate parameters because the directivity largely varies in a continuous manner with variations in the parameters. In the simulations, although the RSPM changed in various ways with a change in even one parameter value, similar sets of parameter values produced similar RSPMs.

In the future, we will evaluate the proposed method by implementing a parameter search algorithm based on metaheuristics and constructing an actual system according to the search results.

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