Piezoelectric transducer with resonant modes control for parametric speaker

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Abstract: Parametric speakers generate narrow directive sound field using modulated ultrasonic. Small size parametric speakers have a problem in sound quality. Demodulated audible sounds by parametric speakers have poor sound pressure in low and middle frequency bands due to the theory of the finite amplitude sound. The sound pressure of a demodulated sound monotonically increases as the frequency of the demodulated sound increases. To solve these problems, an ultrasonic transducer having a wide single peak or more than two peaks must be fabricated. Our previous paper proposed to use two resonant peaks which are close each other. Two close resonant peaks were obtained by two same sized diaphragms linked each other by a specially designed rod. In this paper, a transducer was developed to better structure which has durability against strong mechanical forces and two electrical inputs to control resonant modes of vibration. Durability was improved by optimization of the junction structure. The resonant mode control was enabled by a phase difference between two electrical signals applied to piezoelectric elements on two diaphragms. We succeeded in boosting mid-range sounds, by the newly developed transducer.

Keywords: Piezoelectric transducer, Ultrasonic, Parametric speaker

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1. INTRODUCTION

Parametric speakers generate narrow directive audible sound based on finite amplitude sonic wave theory. Parametric speakers have been studied for various applications, for example, speech privacy systems, voice guidance. The governing equation of the finite amplitude sonic wave theory is presented by Westervelt (1963) [1]. The basic idea and performance of parametric speakers was presented by Yoneyama (1983) [2] as the name of “audio spotlight.”

Although parametric speakers have been used for various applications denoted above, they have high demand to be improved from point of view of the sound quality and downsizing to become more popular. Sound quality is especially poor in low and middle frequency bands. Improving sound quality and slim sizing will promote installation of parametric speakers into general equipments, such as personal computers, tablets, and telephone systems. Sound signal processing, arrangement of transducers, and nonlinear acoustical phenomena are well studied [3–9], even though there is a room for improvement of characteristics of piezoelectric transducers specialized to parametric speakers.

Our previous paper showed the design method to obtain two close resonant peaks of a piezoelectric transducer having double-linked diaphragms [10]. Design methods are derived as two ordinary differential equations (ODEs) and an electrical-mechanical equivalent circuit. These design method were very useful in optimizing the junction structure between two diaphragms. We fabricated piezoelectric transducers having two close resonant peaks in our previous paper. Two close resonant peaks of this designed transducer were expected to enable to boost of low and middle range self-demodulated sounds. This paper shows the further developed design method to improve the electro-acoustic efficiency and the sound pressure of the demodulated sound. Points of improvement are, 1: optimization of the position, the dimensions and the material of the junction rod, 2: addition of an electric signal input to enable control vibration modes. Two ODEs and the equivalent circuit are employed again to design a newly developed transducer. In Sect. 2, the basic theory to design new transducer and results of numerical calculation are denoted. In Sect. 3, the experimental results of a designed transducer are shown.
### 2. DESIGN METHODOLOGY

#### 2.1. Basic Structure of Ultrasonic Transducer Having Double-linked Diaphragms

The sound pressure of a demodulated sound of a parametric speaker monotonically increases as the frequency of the demodulated sound increases [1–3]. Therefore, it is necessary to boost the sound pressure in the vicinity of a resonant peak by improving transducers, to obtain rich sound in mid-range by parametric speakers.

More than two resonant peaks are useful in boosting the sound pressure in the vicinity of a resonant peak with maintaining the mechanical quality factor ($Q$ factor).

We proposed an ultrasonic transducer which has two diaphragms linked by a resin rod. Figure 1 shows the proposed transducer in our previous paper. In Fig. 1, diaphragm 1 is a unimorph-type piezoelectric transducer consisting of an adhered metal board and a piezoelectric element. Diaphragm 2 is a metal board. A resin rod is used to link diaphragms 1 and 2. The resin rod must be made of an appropriately stiff resin to propagate the vibration to diaphragm 2. The dimensions and materials of two diaphragms are same. Two close resonant peaks are obtained by the proposed ultrasonic transducer.

By the Euler-Lagrange equation, the vibration displacement in the thickness direction is written by two ODEs with lumped constants as [10–13].

$$m_\alpha \ddot{\Phi}_\alpha + r_\alpha \dot{\Phi}_\alpha + s_\alpha \Phi_\alpha + [s_\beta (\zeta_\alpha \Phi_\alpha - \zeta_\gamma \Phi_\gamma)] = F_\alpha$$

and

$$m_\gamma \ddot{\Phi}_\gamma + r_\gamma \dot{\Phi}_\gamma + s_\gamma \Phi_\gamma - [s_\beta (\zeta_\alpha \Phi_\alpha - \zeta_\gamma \Phi_\gamma)] = 0.$$  \hspace{1cm} (2)

Note that $\zeta_\alpha$ and $\zeta_\gamma$ are the junction coefficients, i.e. the ratio of the sum of the vibration displacement in the junction area to that of the whole diaphragms, which depends on the junction area and the position of the rod in the diaphragms. In Eqs. (1) and (2), $m_\alpha$, $m_\gamma$, $r_\alpha$, $s_\alpha$, $s_\gamma$, and $r_\gamma$ are determined by the biharmonic equation of the diaphragms. Suffixes $\alpha$, $\gamma$ and $\beta$ mean diaphragm 1 and 2, and rods, respectively.

When the values of the junction coefficients $\zeta_\alpha$ and $\zeta_\gamma$ are approximately equal, Eqs. (1) and (2) are expressed as the lumped constant equivalent circuit shown in Fig. 2.

Branches 1 and 2 represent diaphragms 1 and 2. The voltage $F_\alpha$ is the excitation force caused by the piezoelectric element of diaphragm 1. In Fig. 2, there are two resonant loops, i.e., loops 1 and 2. Loop 1 is the series resonant loop including diaphragms 1 and 2. This resonance causes the first resonant peak of the current. Loop 2 comprises two parallel resonant circuits including the rod, diaphragms 1 and 2. This resonance causes the second resonant peak of the current.

In Fig. 2, the current represents the vibration velocity of the diaphragm. At the first resonant frequency (loop 1), the currents in branches 1 and 2 flow same direction. By contrast, at the second resonant frequency (loop 2), two electrical currents flow opposite direction to each other.

#### 2.2. Basic Theory of Design to Develop Transducer

The transducer shown in Fig. 1 has two points to be improved, which are: 1: optimization of position, dimensions and material of junction rod, 2: addition of electric signal input to enable control vibration modes. Point 1 is related to the durability against mechanical pressures applied to the junction rod. Point 2 is related to the control of the vibration modes. In this subsection, we denote the basic theory of these points using electromechanical equivalent circuits and analytical formulas.

[Point 1]

The transducer shown in Fig. 1 has one resin rod. This structure is designed to decrease the junction coefficients $\zeta_\alpha$ and $\zeta_\gamma$, to bring two resonant frequencies near. Although this resin rod may not endure strong mechanical forces applied to rod when the high voltage is applied to the piezoelectric element. If some metal rods are available, the durability against mechanical forces rises. The stiffness $s_\beta$ in the equivalent circuit of metal is much larger than that of resin. Therefore, it is necessary to find a method to use metal rods with keeping a low value of the mechanical reactance of the rod transformed to the side of dia-
The vibration form of the bending disk is represented by the sum of the solution Eqs. (6) and (7). The coefficients of Eq. (6) are decided by energy equations such as the Rayleigh-Ritz method.

As shown in Fig. 3, the bending disk and the excitation area by an external force are expressed as the \( \Gamma \) and \( \eta \), respectively. The excitation area corresponds to the junction area of the transducer. The junction coefficient \( \zeta \) is defined as,

\[
\zeta = \frac{\iint_{\eta} \phi_{\delta} + \phi_{\eta} \, ds}{\iint_{\Gamma} \phi_{\delta} + \phi_{\eta} \, ds} . \tag{8}
\]

For simplification, we consider the case that a junction area is a point on the disk. When the disk has the circular shape, the integral of the solution \( \phi_{\eta} \) becomes zero by Eq. (A·8). Moreover, the particular solution of the inhomogeneous biharmonic equation Eq. (7) is rewritten as

\[
\phi_{\eta} = -\frac{1}{8k^2} \left[ Y_0(\kappa_0|\mathbf{r} - \mathbf{r}_0|) \right] ds + \frac{2}{\pi} K_0(\kappa_0|\mathbf{r} - \mathbf{r}_0|) ds, \tag{9}
\]

and

\[
\phi_{\eta}|_{\mathbf{r} = \mathbf{r}_0} = 0. \tag{10}
\]

Thus, Eq. (8) is rewritten as,

\[
\zeta = \frac{\iint_{\eta} \phi_{\delta} \, ds}{\iint_{\Gamma} \phi_{\delta} \, ds} - \frac{\phi_{\eta}|_{\mathbf{r} = \mathbf{r}_0}}{\iint_{\Gamma} \phi_{\eta} \, ds}. \tag{11}
\]

The solution \( \phi_{\delta} \) of the homogeneous equation Eq. (6) becomes zero, when the excitation area is a point which is located at nodes of the vibration displacement. On the other hand, the solution \( \phi_{\eta} \) of the inhomogeneous equation Eq. (7) is not equal to zero. Therefore, \( \zeta \) becomes zero when the junction area is located at the node of the vibration (See Appendix). The junction coefficients of points around the nodes of vibration are close to zero by the continuity of the solutions Eqs. (6) and (9). Actual junction areas can be considered as the superposition of numbers of excitation points. From these consideration, it is concluded that the junction coefficient is suppressed when the rod is located at the neighborhood of the nodes of the vibration. [Point 2]

Two resonant modes correspond to loops 1 and 2 in the equivalent circuit Fig. 2, respectively. These resonant modes can be controlled by the direction of the currents \( \Phi_{\delta} \) and \( \Phi_{\eta} \) in the equivalent circuit. To control the direction of these currents, two voltage sources are needed. Thus, we consider to add a voltage source in the equivalent circuit as shown in Fig. 4. This additional voltage source means the excitation force by the piezoelectric element on the
Two diaphragms of the transducer shown in Fig. 1 have same dimensions, thus all values of mechanical components are equal, i.e., $s_a = s_y = s, m_{a} = m_{y} = m, r_{a} = r_{y} = r,$ and $\zeta_{a} = \zeta_{y} = \zeta$. Equation (13) is simplified as,

$$
\begin{bmatrix}
    s + sp_{a} & -sp_{y} \\
    -sp_{a} & s + sp_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
+ j\omega
\begin{bmatrix}
    r + rp_{a} & -rp_{y} \\
    -rp_{a} & r + rp_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
- \omega^{2}
\begin{bmatrix}
    m_{a} & 0 \\
    0 & m_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
= \begin{bmatrix}
    (F_{a}) \\
    (F_{y})
\end{bmatrix}.
$$

Equations (1) and (12) are rewritten for a Fourier component by the matrix form as,

$$
\begin{bmatrix}
    s + sp_{a} & -sp_{y} \\
    -sp_{a} & s + sp_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
+ j\omega
\begin{bmatrix}
    r + rp_{a} & -rp_{y} \\
    -rp_{a} & r + rp_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
- \omega^{2}
\begin{bmatrix}
    m_{a} & 0 \\
    0 & m_{y}
\end{bmatrix}
\begin{bmatrix}
    (\phi_{a}) \\
    (\phi_{y})
\end{bmatrix}
= \begin{bmatrix}
    (F_{a}) \\
    (F_{y})
\end{bmatrix}.
$$

The terms of the left-hand side of Eq. (14) are the stiffness matrix, damping matrix and mass matrix, respectively. The natural frequencies and the natural vibration modes are calculated as the eigenvalues and eigenvectors of the stiffness matrix of Eq. (14), respectively.

The resonant angular frequencies $\omega_1$ and $\omega_2$ of the transducer are calculated by the eigenvalues of the stiffness matrix of Eq. (14) as,

$$
\omega_1 = \sqrt{\frac{s}{m}}, \quad \omega_2 = \sqrt{\frac{s + 2sp_{y}}{m}}.
$$

The eigenvectors of the vibration displacements and velocities of two diaphragms $\phi_a$ and $\phi_y$ are written as below:

$$
(\phi_{a,1}) = A_1 \begin{bmatrix} 1 \\ 1 \end{bmatrix}, \quad (\phi_{a,2}) = A_2 \begin{bmatrix} 1 \\ -1 \end{bmatrix},
$$

and

$$
(\phi_{y,1}) = j\omega A_1 \begin{bmatrix} 1 \\ 1 \end{bmatrix}, \quad (\phi_{y,2}) = j\omega A_2 \begin{bmatrix} 1 \\ -1 \end{bmatrix},
$$

where $A_1$ and $A_2$ are the coefficients of eigenvectors. The first and second eigenvectors correspond to the first and second eigen frequencies, respectively. The subscripts 1 and 2 mean the first and second eigen modes, respectively. The excitation forces $F_a$ and $F_y$ corresponding to the natural frequencies and eigenvectors of vibration velocities are written as,

$$
(\frac{F_{a,1}}{F_{y,1}}) = A_1 B_1 \begin{bmatrix} 1 \\ 1 \end{bmatrix}, \quad (\frac{F_{a,2}}{F_{y,2}}) = A_2 B_2 \begin{bmatrix} 1 \\ -1 \end{bmatrix},
$$

where coefficients $B_1$ and $B_2$ are

$$
B_1 = \frac{s + jr\omega}{m}, \quad B_2 = \frac{s + 2sp_{y} + jr + 2pr_{y}\omega}{m}.
$$

Equations (17) and (18) mean that the first and second eigenmodes of the vibration which are expressed as loops 1 and 2, respectively, in the equivalent circuit Fig. 4. At the first natural mode, the excitation forces $F_{a,1}$ and $F_{y,1}$ are equal in magnitudes and phases. On the other hand, at the second natural mode, the excitation forces $F_{a,2}$ and $F_{y,2}$ are equal in magnitudes and opposite in phases. Note that the electrical inputs and mechanical excitation forces have in-phase relation, which is given by the configuration of the physical coordinates and direction in the equivalent circuit Fig. 4, and ODEs Eqs. (1) and (12).

All solutions of Eqs. (1) and (12) are expressed as the superposition of the natural modes Eq. (17), which are excited by the excitation force Eq. (18).

### 2.3. Detailed Design of Transducer

We denote the detailed design of the transducer having double-linked diaphragms based on two design points above in previous subsection.

Figure 7 shows the schematic and dimensions of a rod. The rod consists of a hollow bobbin-structured metal cylinder and a resin shaft. The hollow bobbin-structured cylinder and the resin shaft are made of duralumin A2017 and ABS resin, to reinforce the junction structure. Figure 8 shows the schematics of the designed transducer. Three rods are used to connect two diaphragms. The centers of three rods are arranged on the nodes of the fundamental vibration of the diaphragms, which are located 5 mm from the centers of the diaphragms. The transducer is housed in a resin box (ABS resin), which are employed to arrange the directivity of sonic fields.
Diaphragms 1 and 2 have the flap vibration mode as shown in Figs. 13(a), 13(b), 13(c) and 13(d). Therefore the sound pressure of the ultrasonic radiated from two diaphragms is dispersed to all directions. The resin box was employed to arrange the directivity of the sound pressure.

To confirm the effect of the resin box, we show the calculation results of the emission patterns of the sound pressure level, which were calculated by the finite-element-method (FEM). Figure 5 shows the plane in which the emission patterns of the sound pressure level was calculated. Figures 6(a) and 6(b) show the numerical calculation results of the emission patterns of the sound pressure for the first and second resonant modes. The distance between the top of the resin box and the observation point was 20 cm. The solid lines and dot lines in Figs. 6(a) and 6(b) show the emission patterns when the transducer has the resin box or not, respectively. As shown in Figs. 6(a) and 6(b), the sidelobes of the sound pressure are suppressed by the resin box. It was confirmed that the resin box is useful in suppressing the sidelobes by this calculation results.

Figure 9 shows the configuration of the electrical connection. Two metal boards of two diaphragms are electrically connected via the rods. In Fig. 9, \( v_x \) and \( v_y \) mean two input voltages to two piezoelectric elements. Two diaphragms are connected to the return electrodes and the opposite sides of piezoelectric elements are connected to two hot electrodes.

Table 1 shows the physical properties of metal cylinders, resin shafts, diaphragms and elastic supports. The elastic supports were fixed at a position of 5.0 mm, which is the node of the fundamental natural vibration. The piezoelectric elements were fabricated by Nihon Ceratec Co., Ltd. (Material code C). Detailed information, such as the elasto-piezo-dielectric (EPD) matrix, is listed in Table 2.

### 2.4. FEM Simulation

The detailed dimensions are decided by numerical simulation by finite-element-method (FEM). FEM calculation was executed by FEMTET (Murata Software Co., Ltd.).
Figures 10(a) and 10(b) show the frequency response of the electrical conductance and susceptance of the transducer of diaphragms 1 and 2. The frequency response of the electrical conductance represents the sum of the vibration velocity of each diaphragm. As shown in Figs. 10(a) and 10(b), both of two diaphragms have two resonant peaks. The first resonant frequency is 51.6 kHz, and the second resonant frequency is 52.8 kHz. Due to the location of the rods around the vibration nodes of the disk, two resonant frequencies become close in 1.2 kHz.

Figures 11(a) and 11(b) show the frequency response of the vibration velocity at the center of diaphragm 2. Figure 11(a) shows the frequency response of the vibration velocity of diaphragm 2 when the signal voltage is applied to either one of two diaphragms. Figure 11(b) shows the frequency response of the vibration velocity when two input voltages are applied to both of two diaphragms. The phase difference of two electrical inputs in Fig. 11(b) is 0, 90 and 180 deg.

As mentioned in previous subsection, all vibration modes are expressed as the superposition of the eigenvectors of the vibration velocities Eq. (17), which are excited when the phase difference of two electrical inputs is 180 deg and 0 deg. In Fig. 11, it is confirmed that the resonant peaks of the vibration velocity of transducer are varied, i.e. the first and second resonant peaks are enhanced when the phase difference of two electrical inputs is 180 and 0 deg.

If the phase difference of two electrical inputs is gradually changed from 0 deg to 180 deg with maintaining equal vibration magnitude, the magnitude of the first resonant peak increases and that of the second resonant peak decreases. When the phase difference of two diaphragms is not equal to either of 0 deg or 180 deg, two resonant peaks appear. For example, when the phase difference of two electrical inputs is 90 deg, the frequency response has two resonant peaks as shown in Fig. 11(b). This transition is shown as the transition from the line of 0 deg through 90 deg to 180 deg in Fig. 11(b).

Moreover, if the ratio of the magnitudes of two electrical inputs gradually changes from one to zero with maintaining the phase difference as 0 deg or 180 deg, the frequency response shifts from the single peak response to the double peaks response. This transition is shown as a shift from the line of 0 deg or 180 deg in Fig. 11(b) to the line of diaphragm 2 in Fig. 11(a).

Figures 12(a) and 12(b) show the frequency response of the sound pressure level. The distance between the top of the resin box and the observation point was 20 cm. Figure 12(a) shows the frequency response of the sound pressure level when the signal voltage is applied to either one of two diaphragms. Figure 12(b) shows the frequency response of the sound pressure level when two input voltages are applied to both of two diaphragms. The phase difference of two input voltages in Fig. 12(b) is 0, 90 and 180 deg. As shown in Figs. 12(a) and 12(b), the frequency response of the sound pressure level has the same behaviour of the vibration velocity at the center of

| Table 1 Physical properties of rod, diaphragm and elastic support. |
|------------------|-----------------|-----------------|
| Part             | Material        | Mass density (kg/m³) | Young’s modulus (GPa) |
| resin rod        | ABS resin       | 1.20 × 10³          | 2.65                |
| metal cylinder   | duralumin       | 2.70 × 10³          | 69.9                |
| diaphragm        | phosphor bronze| 8.80 × 10³          | 110.0               |
| elastic support  | polychloroprene | 1.20 × 10³          | 0.10                |

| Table 2 Physical properties of the piezoelectric board. |
|-----------------|-----------------|-----------------|-----------------|
| Dielectric constant | Coupling factor | Piezoelectric strain constant (×10⁻¹² m/V) | Elastic constant (×10⁻¹² m²/N) |
| ε₃₃/ε₀ = 4.500      | K₃₁ = 0.61     | d₃₁ = −160       | S₁₁ = 15.2      |
| ε₁₁/ε₀ = 4.700      | K₃₃ = 0.35     | d₃₃ = 280        | S₃₃ = 15.5      |
| K₃₃ = 0.65          | d₁₅ = 450      |                 |                 |

Fig. 10 Numerical calculation results of frequency response of electrical admittance.
diaphragm 2. The magnitude of the second resonant peak is much larger than that of the first peak. This difference of the magnitude of the resonant peaks is due to the vibration form of diaphragms. The vibration form of two diaphragms are different, as shown in Figs. 13(a), 13(b), 13(c) and 13(d). Thus the sound pressure levels at two resonant frequencies are different each other. Figures 13(a), 13(b), 13(c), and 13(d) show the schematic of the calculation results of vibration mode.

3. EXPERIMENT

3.1. Ultrasonic Characteristics

Figure 14 shows a picture of a manufactured prototype of transducer. The prototype was manufactured based on Figs. 7 and 8. In Fig. 14, the edges of two diaphragms were coated by adhesives to prevent chipping of piezoelectric elements.

Figures 15(a) and 15(b) show the electrical conductance and susceptance of prototype transducer, respectively. The electrical admittance was measured the impedance analyzer IM3570 (HIOKI E.E. CORPORATION). Input voltage was 1.0 V_{rms}.

Diaphragms 1 and 2 have the first and second resonant peaks at 51.3 kHz and 52.5 kHz, respectively.

Figure 16 shows the measuring system of the vibration velocity of diaphragm 2. The vibration velocity at the center of diaphragm 2 was measured by Laser doppler vibrometer (LDV). Figures 17(a) and 17(b) show the experimental data of vibration velocity by LDV. Figure 17(a) shows measured vibration velocity at the center of diaphragm 2 when the input voltage is applied to
either of two diaphragms. Figure 17(b) shows the vibration velocity of diaphragm 2 when the input voltages are applied to both of two diaphragms (phase difference is set to 0, 90 and 180 deg). As shown in Fig. 17(b), the first and second resonant peaks are boosted when the phase difference of input voltages is set to 180 and 0 deg, respectively. Due to the production tolerance, the vibration velocity does not become the single peak response.

Figure 18 shows the measuring system of the sound pressure level of the transducer. PXI platform (National Instruments Corporation) was employed as the central measuring system. The distance between the microphone Type 4936-A-011 and the top of the box of the transducer was 20 cm. Experimental conditions were;

- Measuring system: Fig. 18
- Room: Anechoic chamber in Waseda University
- Input signal: Sweep tone (50.0–55.0 kHz, 5 Vrms), Microphone and conditioning amplifier: Type 4936-A-011 and Type 2690-A
- Distance between the microphone and the top of the resin box: 20 cm.

Figures 19(a) and 19(b) show the experimental data of the sound pressure level of ultrasonic. As shown in Fig. 19(b), both peaks are boosted as with the vibration velocity of diaphragm 2.

3.2. Audible Sound Characteristics

The frequency response of the demodulated audible sound was measured using two ultrasonic signals, i.e. a pure tone signal and a chirp signal. The audible sounds are demodulated in the air as the difference frequency sounds of them. To maximize the sound pressure level of the difference sounds, the frequency of the pure tone must be set to one of the two resonant frequencies of the prototype transducer. We set the frequency of the pure tone to the first resonant frequency 51.5 kHz. The frequency of the chirp signal was set around 52.5 kHz. Thus, it is expected that the peak of the sound pressure level of the demodulated audible sound is located around $52.5 - 51.5 = 1.0 \text{[kHz]}$. The sound pressure level of the demodulated sound was measured with changing the phase difference of input voltages applied to two diaphragms. Sound level meter NL-32 (RION Co., Ltd.) was employed as the microphone for audible sounds.

The frequency response of the demodulated audible sound was measured as the experimental conditions as below;

- Measuring system: Fig. 18
- Room: Anechoic chamber in Waseda University
- First signal = pure tone (frequency = 51.5 kHz, phase difference = 0, 180 deg, 5 Vrms)
- Second signal = chirp signal (frequency = 52.0–61.5 kHz, phase difference = 0, 180 deg, 5 Vrms)
- Microphone: Sound level meter NL-32
- Distance between the microphone and the top of the resin box: 20 cm.

Figures 20(a) and 20(b) show the measured frequency response of the sound pressure level of the demodulated sound.
audible sound. Figure 20(a) shows the frequency response of the sound pressure level when voltage is applied to either of two diaphragms. Figure 20(b) shows the frequency response of the sound pressure level when input voltages are applied to both of two diaphragms. All of the frequency responses of the sound pressure level have the peaks around 1 kHz. In Fig. 20(b), the peaks of the sound pressure level were maximized when the phase differences of two pure tones and two chirp signals are set to 180 deg and 0 deg, respectively. These results are as expected by the ultrasonic characteristics shown in Figs. 19(a) and 19(b).

4. CONCLUSIONS

In this paper, we design the ultrasonic transducer having double linked diaphragms for parametric speakers, which enables to boost mid-range audible sounds. The transducer was designed based on the basic theory, i.e. point 1: design to make close the two resonant frequencies by the control of the junction coefficient between two diaphragms, point 2: design to enhance the resonant peaks by the control of the phase difference of two input signals. About point 1, it was derived that two resonant frequencies become close when junction rods are placed around the vibration nodes of the natural mode of bending disks. About point 2, it was derived that the phase difference of two input voltages is able to control vibration mode of two diaphragms. We arrived at the conclusion of theoretical consideration that the first and second resonant peaks become close when junction rods are placed around the vibration nodes of the natural mode of bending disks.

The mid-range sounds were enhanced when the phase differences of two pure tones and two chirp signals are set to 180 deg and 0 deg, respectively. Based on the theoretical consideration, prototype ultrasonic transducer was designed by these design points. It was verified by FEM and experiments that the detailed design of the ultrasonic transducer fulfills the characteristics required to boost the mid-range demodulated sounds. The frequency responses of the demodulated sounds were measured using two ultrasonic signals, i.e. pure tones and a chirp signals. The mid-range sounds were enhanced when the phase differences of two pure tones and two chirp signals are set to 180 deg and 0 deg, respectively.

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APPENDIX: THE SUM TOTAL OF THE BENDING DISPLACEMENT OF CIRCULAR DISK

It is proved that the sum total of the bending displacement of circular disk becomes zero under the plane stress condition.

Integrate the both sides of Eq. (3) in the face of the circular disk $\Gamma$,

$$\int \int_{\Gamma} \nabla^4 \phi_\gamma dS - k_0^2 \int \int_{\Gamma} \phi_\gamma dS = 0. \quad (A\cdot1)$$

When the solution $\phi_\gamma$ is axisymmetric, Eq. (A\cdot1) is written as,

$$\int \int_{\Gamma} \phi_\gamma dS = \frac{2\pi}{k_0^2} \int_0^R r^4 \phi_\gamma dr, \quad (A\cdot2)$$

where $dS$ is the surface element, $r$ is the radial coordinate, $R$ is the radius of the circular disk.

If $\phi$ and $\psi$ are both twice continuously differentiable on $\Gamma$, Green’s second identity is represented as,
\[
\int \int_{\Gamma} (\psi \nabla^2 \phi - \phi \nabla^2 \psi) dS = \oint_{\partial \Gamma} \left( \psi \frac{\partial \phi}{\partial n} - \phi \frac{\partial \psi}{\partial n} \right) dl, \tag{A-3}
\]

where \(dl\) is the circumference element, \(n\) is the normal vector of the boundary, and \(\partial \Gamma\) is the boundary of the region \(\Gamma\). When \(\psi = 1\) and \(\phi\) is axisymmetric, Eq. (A-3) is written as,

\[
2\pi \int_{0}^{R} r \nabla^2 \phi dr = 2\pi R \left[ \frac{\partial \phi}{\partial r} \right]_{r=R}. \tag{A-4}
\]

Let \(\phi = \nabla^2 \phi_x\), Eq. (A-4) becomes as,

\[
2\pi \int_{0}^{R} r \nabla^4 \phi_x dr = 2\pi R \left[ \frac{\partial}{\partial r} \nabla^2 \phi_x \right]_{r=R}. \tag{A-5}
\]

Substitute Eq. (A-5) into Eq. (A-2),

\[
\int \int_{\Gamma} \phi dS = \frac{2\pi R}{\kappa_0^2} \left[ \frac{\partial}{\partial r} \nabla^2 \phi_x \right]_{r=R}. \tag{A-6}
\]

Here, by the boundary condition that the resultant share force is zero at the free edge, i.e.,

\[
Q_{gr} = -D \left[ \frac{\partial}{\partial r} \nabla^2 \phi_x \right]_{r=R} = 0, \tag{A-7}
\]

where \(Q_{gr}\) is the share force in radial direction and \(D\) is the bending stiffness.

By Eqs. (A-6) and (A-7), the sum total of the bending displacement becomes

\[
\int \int_{\Gamma} \phi_x dS = 0. \tag{A-8}
\]