Acoustic characteristics of a loudspeaker obtained by vibration and acoustic analyses
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Abstract

The acoustic characteristics of a direct radiator type loudspeaker has been studied in this paper. The natural modes of the speaker cone vibration analyzed numerically by the finite element method have been verified by comparing them with experimental results. The so-approved finite-element model has been used to calculate the vibration response of the cone excited by the voice coil. The vibration displacement of the speaker cone paper has been converted into the vibration velocity and used as a boundary condition for the acoustic analysis. The frequency characteristics, directivity, and sound pressure distribution of the loudspeaker have been calculated by the boundary element method. The numerical results have been verified by the experiments carried out in an anechoic chamber.

1 Introduction

Loudspeakers used nowadays are classified generally into two types; a direct-radiator type and a horn type.[1] This paper deals with the direct-radiator type loudspeaker, which drives a conical diaphragm (referred to as a cone hereafter) and radiates sound into the coupled air. The acoustic characteristics of the direct-radiator type loudspeaker are mostly dependent on the vibration of its cone. One of the efforts to improve the loudspeaker performance has been concentrated on the development of cone materials and the modification of design. Prediction of the loudspeaker performance in the design stage before making a prototype can be achieved by a computer simulation technique.

The cross-sectional diagram of the axisymmetric loudspeaker unit is
shown in Figure 1. The cross-section of the cone paper is not a straight line but a slightly curved one. The radius of the outer rim of the cone is much bigger than the one of the bobbin in order to enhance the radiation efficiency in the low frequency range. That may cause nonuniform vibration of the cone paper. The edge connecting the rim of the cone paper with the frame is a free-form type in this paper. The edge is made of a much softer material than the cone paper and supports the cone paper elastically on the frame. Spider is a membrane spring made of synthetic resin and guides the bobbin to move axially. The dust cap is made of the paper similar to the cone paper.

2 Vibration of the Speaker Cone

Since the cone shape is axisymmetric, the two-dimensional analysis of the axisymmetric vibration has already been reported [2,3]. In this paper, however, three-dimensional vibration of the cone has been obtained for modal analysis and for the use in the acoustic analysis.

2.1 Vibration Analysis Model

For the numerical analysis of the vibration of the loudspeaker cone, a finite-element model has been established and the material properties have been obtained.

The loudspeaker cone shown in Figure 1 has been modeled as finite elements by using MSC/PATRAN. The cone paper, dust cap, and bobbin have been modeled as shell elements (CQUAD4). The fundamental vibration mode of the loudspeaker cone is known as a piston mode [4,5]. This means that the edge and the spider play a role of a spring to support elastically the cone paper. Thus the edge and the spider have been modeled as spring elements (CELAS1) as shown in Figure 2. The largest side of the elements is 14 mm, which is
much less than 1/6 of the smallest wavelength up to 4 kHz frequency range. The boundary condition is that the edge is axially-free and the bobbin is radially fixed.

The material properties of the cone paper have been measured. The mass density \( \rho \) has been obtained by measuring the mass and volume of a rectangular sheet specimen as \( 473 \pm 23 \text{ kg/m}^3 \). Poisson's ratio \( \nu \) is difficult to measure and has been selected with some error as \( 0.3 \pm 0.05 \). Young's modulus \( E \) has been obtained as \[ E = (1 - \nu^2) \rho c^2 \] (1)

where \( c \) is the quasi-longitudinal wave speed measured in the specimen using H.M. Morgan's Dynamic Modulus Tester PPM-5R \[7\]. The measured wave speed \( c \) is \( 2,290 \pm 100 \text{ m/s} \), and Young's modulus is \( 2.26 \pm 0.20 \text{ GPa} \).

The spring constants of the edge and the spider have been obtained by measuring the static deflection under stepwise loading as \( 441 \text{ N/m} \) and \( 297 \text{ N/m} \), respectively. The spring constants have been divided to be distributed around the edge and the bobbin, respectively, as shown in Figure 2. The bobbin is mostly composed of aluminum, and its elastic constants and mass density have been cited from the known properties of aluminum \[4\] as \( E = 71 \text{ GPa}, \nu = 0.33, \) and \( \rho = 2,700 \text{ kg/m}^3 \). The total mass of the cone is \( 7.03 \text{ g} \).

### 2.2 Natural Vibration Modes

The mechanical vibration of the cone modeled above has been calculated by using MSC/NASTRAN. The natural modes of the cone vibration have been obtained by the Lanczos method, which is one of the methods to solve an eigenvalue problem \[8\]. The calculated natural frequencies are \( 54.8 \text{ Hz}, 14.1 \text{ kHz}, \ldots \), corresponding to the fundamental mode, second mode, etc. The shape of the fundamental mode represents a rigid body motion like a piston as predicted. The higher modes do not have axisymmetric mode shapes but do include the shapes with nodal diameters. This trend is different from that of the natural modes of disks and conical shells.

The calculated results of the natural frequencies and mode shapes have been verified by comparing them with experimental ones. The experiments have been carried out by the impact hammer test and by the frequency-sweeping test. The sensing of the vibration on the cone has been achieved by using an optical-fiber interferometry system. The system consists of a fiber interferometer (Polytec OFV-502), a vibration controller (Polytec OFV-3000), and a dynamic signal analyzer (HP 35665A). The excitation by the impact hammer or by the electrical signal to drive the voice coil is input to the other channel of the signal analyzer. The analyzer then computes the frequency response function, which yields modal parameters. As reported earlier \[9\], the measured natural frequencies, \( 53 \text{ Hz}, 14.3 \text{ kHz}, \ldots \), and the corresponding
mode shapes agree well to the calculated ones.

2.3 Vibration Response of Voice Coil Excitation

The vibration response of the cone excited by the voice coil has been calculated to simulate the motion of the cone in the real situation. The exciting force $F$ has been obtained as follows:

$$F' = B_{\text{gap}} \times l \times i,$$

where $B_{\text{gap}}$ is the electromagnetic field in the gap, $l$ is coil length, and $i$ is the current whose magnitude varies with frequency. The frequency-dependent force obtained from Eq.(2) with the measured current is shown in Figure 3. The axial force has been divided to be distributed around the bobbin.

The vibration of the cone for the single frequency excitation has been calculated, and the results at 125 Hz, 1 kHz, and 4 kHz are shown in Figure 4. The vibration response is axisymmetric, that is not surprising because the axisymmetric body has been considered to be excited axisymmetrically.

![Figure 3](image1.png) Excitation force on the bobbin driven by the voice-coil.

(a) 125 Hz

(b) 1 kHz

(c) 4 kHz

![Figure 4](image2.png) Vibration response of the speaker cone excited by the voice coil.
3 Acoustic Characteristics of the Speaker

Once the vibration of the cone is calculated, the acoustic characteristics of the sound radiated by the loudspeaker can be calculated through the coupled vibro-acoustic analysis [10].

3.1 Acoustic Analysis Model

Since the speaker box has a air hole and the bobbin has a free edge and junctions, the analysis has been carried out by the indirect boundary element method based on the variational method. The boundary element model used for the analysis is shown in Figure 5. The modeling and analysis have been performed by using SYSNOISE [11].

The element size in the cone paper is less than 1/6 of the minimum wavelength in the frequency range from 30 Hz to 8 kHz. The field point meshes are in the horizontal plane of 2 m x 2 m as shown in Figure 5. In the air, the sound speed is 343 m/s and the mass density is 1.21 kg/m³.

3.2 Vibro-acoustic Analysis

The vibration displacement $u$ obtained in Section 2.3 has been converted to the vibration velocity component $v_n$ normal to the cone paper as follows.

$$v_n = j \omega u \cdot n,$$

where $n$ is the unit vector normal to the cone paper. The acoustic matrix equation constructed by the indirect boundary element method with the boundary condition of Eq.(3) is as follows [11].

(a) speaker cone and box  
(b) field point mesh

Figure 5  Acoustic analysis model.
\[
\begin{bmatrix}
B & C \\
C^T & D
\end{bmatrix}
\begin{bmatrix}
\sigma \\
\mu
\end{bmatrix} = \begin{bmatrix}
f \\
g
\end{bmatrix}
\] (4)

where \(B\), \(C\), \(D\) are the area integrals expressed in terms of Green’s function \(G\) and wave number \(k\). In Eq.(4) \(\mu\) is the difference between interior and exterior sound pressures, \(\sigma\) is the difference between interior and exterior sound pressure derivatives, and \(f\) and \(g\) are the excitation vectors exerted by the boundary conditions of the acoustic pressure and vibration velocity, respectively.

The acoustic pressure \(p\) at a certain point is obtained as follows once \(\mu\) and \(\sigma\) at the boundary elements are obtained from Eq.(4).

\[
p = \oint_S \left( p^+ - p^- \right) \frac{\partial G}{\partial n} - G \left( \frac{\partial p^+}{\partial p^-} - \frac{\partial p^-}{\partial n} \right) \right] dS
\] (5)

In addition, vibration velocity of a particle is obtained by

\[
v = - \frac{1}{j \omega \rho} \nabla p
\] (6)

3.3 Analysis Results

The acoustic characteristics of the loudspeaker calculated for the model shown in Figure 5 are the frequency characteristics, directivity, and sound pressure distribution. The frequency characteristics display the sound pressure at 1 m away in front of the center of the cone as a function of the frequency. The result is shown in Figure 6(a) and shows a good agreement with the curve in Figure 6(b), which is the experimental result discussed in the next section.
The directivity is the sound pressure at 1 m away from the center of the cone on the horizontal plane in Figure 5 as a function of the propagation direction at a certain frequency. Among many results only three at 125, 500, and 2000 Hz are shown in Figure 7 (a)-(c). The calculated results of the directivity in Figure 7 (a)-(c) are compared in good agreement with the measured ones in Figure 7 (d)-(f).

The sound pressure distributions in the horizontal plane shown in Figure 5 for the frequencies of 63, 250, 1000, and 4000 Hz are displayed in Figure 8. This result yields the directional characteristics and the propagation pattern of the sound radiated from the speaker.

### 3.4 Experiments

In order to verify the calculated results, the experiment has been carried out in an anechoic chamber. The sound pressure magnitude at 1 m away in front of the cone center has been measured by sweeping single-frequency sine signal. The result is shown in Figure 6(b) and is compared with the calculated one.

The sound pressure magnitude at 1 m away from the cone center has also been measured as rotating the speaker installed on a turn table. The directional characteristics measured in this way are shown in Figure 7 (d)-(f) and are compared with the calculated ones.

![Sound pressure distributions](image)

**Figure 8** Sound pressure distribution on the horizontal plane around the speaker.
Figure 7 Acoustic directivity at each frequency.
4 Conclusion

The acoustic characteristics of a loudspeaker, such as the frequency characteristics and the directivity, calculated by the vibro-acoustic analysis for the numerical model have shown good agreements with the experimental results. The numerical model can be used to predict the acoustic characteristics in the design stage for the change of the design parameter values.

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References

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