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Research on reduction of structure-borne sound using perforated plate

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ABSTRACT

It is possible to reduce the radiation of structure-borne sound by providig sound absorption to the vibrating surface of a structure. This paper focuses on reducing radiating sound by providing sound absorption using a perforated plate. Taking a vibrating rectangular structure as the object, the reduction in sound radiation achieved by using a perforated plate was determined by experiment. Sound radiation decrease at frequencies higher than the peak frequency of sound absorption of a perforated plate, and they increase at lower frequencies. And, the effect of reducing sound radiation was predicted using our original numerical analysis method. The method is an acoustic field-perforated plate coupled numerical analysis method that uses the boundary element method for acoustic fields and the finite element method for a perforated plate. The method solves an acoustic field containing a vibrating perforated plate. The effects of reducing sound radiation found by the experiment and the numeric analysis agreed with each other well. Consequently, the following facts were verified: A perforated plate is effective for reducing radiation of structure-borne sound. Our original numerical analysis method is effective for predicting sound radiation from a perforated plate, and can be used in designing a low-noise structure.

INTRODUCTION

As a way to reduce noises radiated from a vibrating structure (structure-borne sound), the method of reducing sound radiation efficiency by providing sound absorption on the surface of a structure is known. By selecting an appropriate means of providing sound absorption, this method makes it possible to reduce noises from a structure, even if the structure requires heat resistance and durability, in which case the use of a conventionally popular soundproof cover (made of a soundinsulation material and vibration proof rubber) is generally difficult.

One of the methods of providing sound absorption with excellent resistant to heat, oil, and water is to use a structure made of a metal perforated plate and air layer on the back. By optimizing the hole diameter, porosity, thickness of perforated plate and thickness of air layer, the required sound absorption can be achieved in the required frequency band [1]. However, if a perforated plate is used to reduce structure-borne sound, the vibrating force acting on the perforated plate is assumed to be greater and more complex than in the case of sound absorption. As a result, the vibration of the perforated plate, that is the sound radiation phenomena from the perforated plate, is considered to be more complex. In the past, studies in this field were made using theoretical analysis on the case where a simple vibrating force acts on a structure and a perforated plate of infinite size [2,3]. However, actual structures have a finite size and the influences of plate resonation cannot be ignored either. Therefore, use of numeric analysis is indispensable for designing an optimum structure when actually designing products.

The authors developed a numerical analysis method using the boundary element method as a technique for predicting and visualizing acoustic fields including a perforated plate [4]. However, because vibration of the perforated plate is ignored in this method, it is not effective for such complicated phenomenon as described above. For that reason, the authors developed an acoustic field-perforated plate coupled numerical analysis method that uses the boundary element method for the acoustic fields and uses the finite element method for the plate, and attempted to predict the sound radiation from a perforated plate.

This paper first describes the acoustic field-perforated plate coupled numerical analysis method, and then presents the results of using the method to predict the effects of the perforated plate.

ACOUSTIC FIELD-PERFORATED PLATE COUPLED NUMERICAL ANALYSIS METHOD

The acoustic field-perforated plate coupled numerical analysis method developed in this study, as shown in Figure 1, divides the object for analysis into a perforated plate and two acoustic fields of the front and back of the perforated plate. Then, the method applies the finite element method to the perforated plate and applies the boundary element method to the two acoustic fields, and formulates simultaneous equations for the perforated plate and the acoustic fields. The method then forms coupled vibrations of the perforated plate and acoustic fields by defining the forces acting on the perforated plate caused by the difference in sound pressures between the front and back of the perforated plate (making an

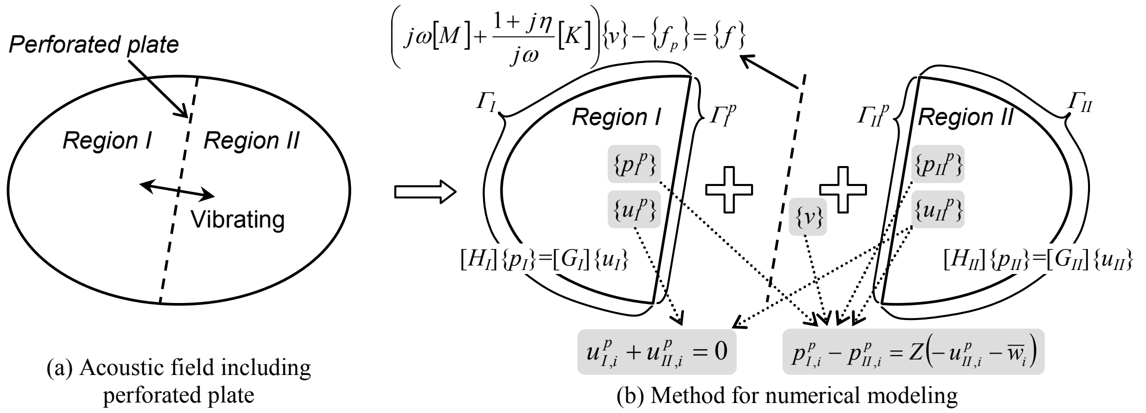

Figure 1. Numerical modeling of acoustic field including perforated plate

equation from the coupled relationship). The following describes how to form the equations for the acoustic fields, perforated plate, and coupled relationship respectively. This paper describes the case where one perforated plate and two acoustic fields exist. However, similar analysis procedures can be used for cases with more perforated plates and more acoustic fields.

Formulating equation for acoustic field by boundary element method

By applying the weighted residual method to the Helmholz equation [Equation (1)] that governs the acoustic field, boundary integral equation [Equation (2)] is given.

$$\nabla^2 p + k^2 p = 0 \tag{1}$$

$$-\frac{1}{2}p_i = \int_\Gamma \left\{ p_s \frac{\partial \phi_{is}^*}{\partial n_s} - \frac{\partial p_s}{\partial n_s}\phi_{is}^* \right\} d\Gamma \tag{2}$$

$$\phi_{is}^* = \frac{\exp(jkr_{is})}{4\pi r_{is}} \tag{3}$$

Discretising Equation (2) using the boundary element method gives the following simultaneous equations [Equation (5)]:

$$[H]\{p\} = [G]\{u\} \tag{4}$$

Where p and $\{p\}$ are the sound pressure on the boundary surface Γ of the acoustic field and its vector, u and $\{u\}$ are the particle velocity in the direction of the normal to the boundary surface Γ and its vector, k is the wave number, and $[H]$ and $[G]$ are the coefficient matrices determined by the shape and medium of the acoustic field and analysis frequency. Subscripts i and s indicate arbitrary points on the acoustic field boundary Γ, ϕ^*_{is} is the fundamental solution, n_s is the derection of outward normal on boundary Γ, and r_{is} is the distance between points i and s. Here, the sound pressure p of all acoustic field elements, and the particle velocity u of the elements on the boundary Γp of the back and front of the perforated plate are unknowns. The equation was made discrete by assuming that p and u are constant within the elements.

Formulating equation for perforated plate by finite element method

By applying the finite element method to plate vibration, the following simultaneous equation in a frequency domain is given:

$$\left(j\omega[M]+\frac{1+j\eta}{j\omega}[K]\right)\{v\}=\{f\}+\{f_p\} \tag{5}$$

Where $\{v\}$ is the vibration velocity vector of the element nodes, $[M]$ and $[K]$ are the mass matrix and stiffnes matrix, $\{f\}$ is the external force vector by mechanical excitation, $\{f_p\}$ is the force vector acting on the plate due to the difference in sound pressures between the front and back of the plate, η is the loss factor and ω is the angular frequency. Here, vibration velocity v and sound pressure excitation force f_p are unknowns. The equation was made discrete with the plane shell element made by combining the membrane element (constant strain element) and plate-bending element (element of Zienkiewicz et al).

Equation of coupled relationship between acoustic field and perforated plate

When formulating coupled relationship equations , it is assumed that the perforated plate elements and acoustic field elements on that (elements on boundaries Γ_I^p and Γ_{II}^p in Figure 1) have the same shape and size. First, assuming the method is applied to a perforated plate whose thickness is enough smaller than the wavelength of a sound wave, it is assumed that the medium in a hole moves as one, and particle velocities u_I^p and u_{II}^p facing each other are considered to be identical. That is expressed as:

$$\{u_I^p\}+\{u_{II}^p\}=0 \tag{6}$$

Next, with regard to force acting on a perforated plate caused by the difference in sound pressures between the front and back of the perforated plate, the pressure difference is expressed as follows using the viscous damping term of the Melling equation [6]:

$$p_I^P - p_{II}^P = Z\left(-u_{II}^p - \bar{v}\right) \tag{7}$$

$$\mathrm{Re}[Z] = \mathrm{Re}\left[\frac{j}{1-\dfrac{2J_1(k_s a)}{k_s a J_0(k_s a)}}\right]\omega\rho(t+d)\frac{1}{R_p} \tag{8}$$

$$\mathrm{Im}[Z] = \left\{\mathrm{Im}\left[\frac{j}{1-\dfrac{2J_1(k_s a)}{k_s a J_0(k_s a)}}\right]t + \frac{8d}{3\pi}\right\}\omega\rho\frac{1}{R_p} \tag{9}$$

$$k_s^2 = -\frac{j\omega\rho}{\mu} \tag{10}$$

Then, the force acting on the perforated plate caused by the pressure difference is expressed as follows:

$$\{f_p\} = [N]\left(\{p_I^P\} - \{p_{II}^P\}\right) \tag{11}$$

Where $\bar{v}$ is the in-element average vibration velocity of the perforated plate elements, $[N]$ is the coefficient matrix composed of the shape functions of the plate-bending elements, t is the thickness of the perforated plate, a and d are the radius and diameter of the hole, R_P is the porosity, ρ is the density of the medium in the hole, μ is the viscosity coefficient of the medium in the hole, and J_0, and J_1 are the Bessel functions of order 0 and 1 respectively.

As described above, using the simultaneous equations for two acoustic fields [Equation (4)] and the simultaneous equation for the perforated plate [Equation (5)] and coupled relationship equations [Equation (6) through (11)], the acoustic field-perforated plate coupled problem can be solved.

PREDICTING REDUCTION EFFECT OF SONUD RADIATION BY PERFORATED PLATE

The analysis method described in the previous section was used to predict the effect of a perforated plate sound insulation cover. In addition, the analysis results were compared with the results of an equivalent experiment to verify the validity of the analysis method.

Object of analysis

Figure 2 shows an outline of a numerical model of the analysis object. Figure 2 shows only 1/9 area of the object. A perforated plate was installed 40mm from the surface of a vibrating structure (shaded area in Figure 2). The size of the perforated plate was 45mm x 30mm. The walls surrounding the air space between the structure and the perforated plate were rigid. The surface of the structure was vibrated with an amplitude of 1m/s in the same phase on all its surfaces. Assuming that the perforated plate was connected rigidly to the surface of the structure, the rim of the perforated plate was vibrated at the same amplitude and phase as the structure. In addition, it was assumed that the surrounding area of the perforated plate was covered with an infinite baffle plate. The perforated plate was made of aluminum with a thickness of 2mm. Nine holes with a diameter of 2mm were made on the surface (porosity: 2%). When this structure is used for sound absorbing structure, it has a peak of sound absorbing at approximately 600Hz.

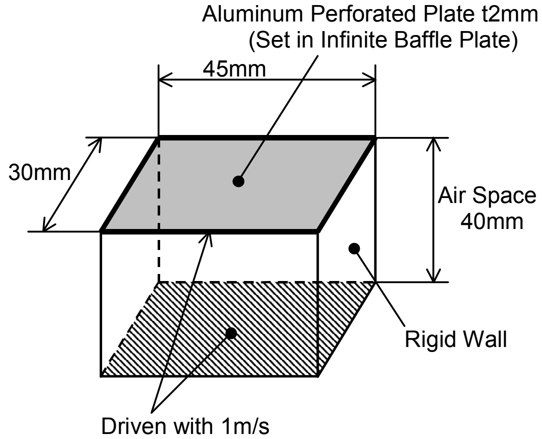

Figure 2. Object for numerical analysis (1/9 area)

Taking the analysis procedures described in the previous section, the finite element method is applied to the perforated plate, and the boundary element method is applied to the closed air space between the structure's surface and the perforated plate and the infinite outer sound field to which sound radiated from the perforated plate.

Analysis results

The effects of the perforated plate were evaluated with changes in sound pressure level at a point 10mm above the center of the sound radiating surface. Figure 3 shows the reduction in sound presuure level by installing the perforated plate, compared with the case where only the surface of the structure is vibrating and radiating sound (without any perforated plate and rigid walls surrounding the air space). When no perforated plate was installed, analysis was made assuming that the surrounding area of the surface of the structure was covered with an infinite baffle plate.

According to Figure 3, the effect of reducing sound pressure is observed in frequencies higher than approximately 560Hz (roughly identical to the peak frequency of sound absorbing). A maximum of 12dB was achieved and stable effects were obtained in frequencies higher than that. On the other hand,

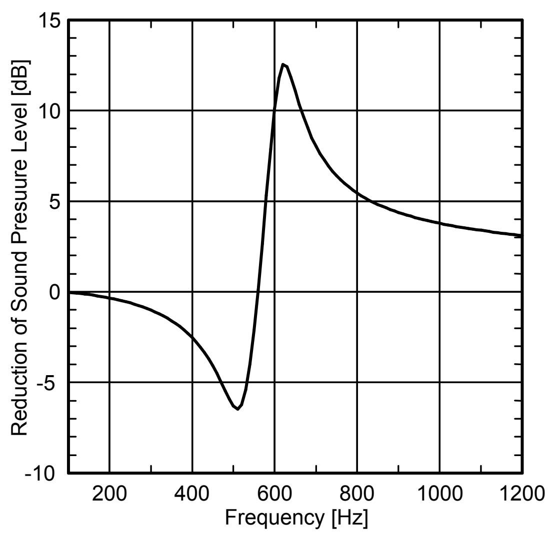

Figure 3. Reduction of sound presuure level at 10mm point from sound radiation surfece using perforated plate by numerical analysis

in lower frequencies, sound presuure level increased (became worse). After a maximum of increasing by 6.5dB, the amount of worsening decreased in even lower frequencies.

Verifying experiment

Figure 4 shows an outline of the test specimen and equipment used in the experiment, which was conducted to verify the validity of the analysis results in the previous section. The photo on the left of Figure 4(a) shows a view of the test specimen without a perforated plate. The areas were partitioned with 3mm-thick aluminum plates, and the outer sides were made of 6mm-thick aluminum plates. The bottom plate of the test specimen, which corresponds to the surface of the structure, was made of 20mm-thick aluminum plate. The area surrounding the test specimen was covered with a steel plate. For the case without a perforated plate, in a similar manner to analysis, partitions and outer walls were not installed, and the bottom plate surface was installed so that it had the same height as the surrounding steel plate.

Verifying validity of numerical analysis

Figure 5 shows a comparison of the numerical analysis and experiment regarding the effects of a perforated plate. In the experiment also, greater effects were observed in frequencies higher than approximately 600Hz, and sound radiation were worsened in lower frequencies. The effects of reducing sound radiation by the experiment and the numeric analysis agreed with each other well.

CONCLUSION

This paper presented a method of acoustic field- perforated plate coupled numerical analysis for use in designing a structure-borne sound reduction structure by providing sound absorption using a perforated plate installed on the surface of a vibrating structure. As the first step for this purpose, the presented method was used to predict the effects in reducing structure-borne sound under simple vibrating conditions. The analysis results well agreed with the results of the experiment.

On the structure under study in this paper, both numerical analysis and experiment proved that noises were reduced in higher frequencies than a point where sound absorption reaches a maximum level, and noises increase in lower frequencies on the contrary. The results suggested how this technique could be used to optimally design products.

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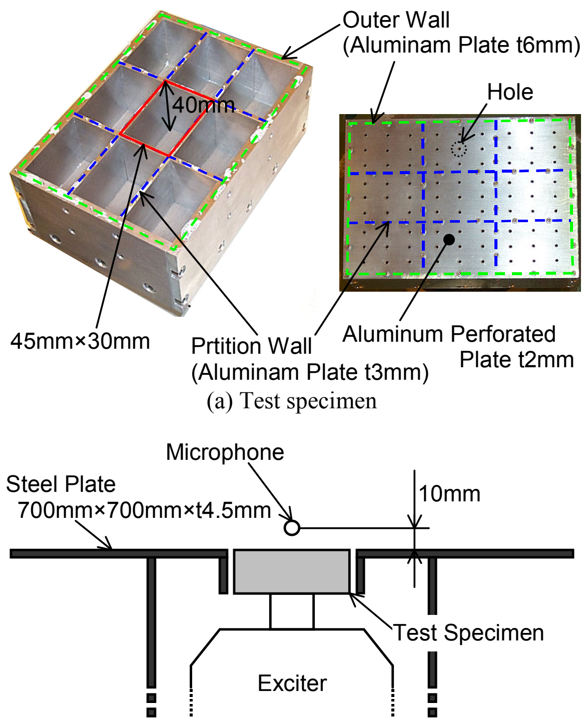

(b) Experiment system
Figure 4. Verification test

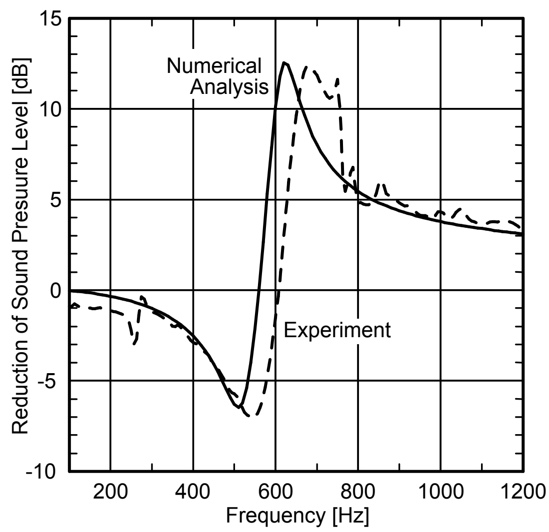

Figure 5. Reduction of sound pressure level at 10mm point from sound radiation surfece using perforated plate by numerical analysis and experiment