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STUDY OF ACTIVE NOISE CONTROL BY USING STRUCTURAL-ACOUSTIC COUPLED ANALYSIS

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ABSTRACT

The active noise control, which is effective for low frequency band, has been developed. In this paper, an effective active noise control method is proposed for the steady state sound field. The fundamental idea is that the unknown sound pressures are estimated by using structural-acoustic coupled analysis, i.e., the sound pressure are virtually measured at many points. As the numerical example, the rectangular prism model is considered. As the result, the proposed method is seen to be effective for the actual control.

1. INTRODUCTION

Recently, from the view point of amenity, it becomes important to control noise in the enclosed cavity of motor car, train and airplane. Because of the development of the digital signal processing technique, the active noise control is applied to many fields[1]-[4]. Active noise control is more effective than the passive noise control for the low frequency band.

The active noise control by feedforward control theory is adequate for the steady state sound field. In the method, active noise control is performed such as to minimize the sound energy at the control points and the high control effect can be attained when the sound pressures at many control points can be measured[5]. The number and location of the microphone is, however, limited from the lack of space and cost problem in actuality.

In this paper, the effective active noise control method is proposed for the steady state sound field in the enclosed cavity. In that method, the sound pressures, where no microphones are located, are estimated from both the vibration data and the measured sound pressure data. The estimation of sound pressure is based on the acoustic vibratory inverse analysis[6] and structure acoustic coupled analysis[7]. When the sound presures at all control points can be obtained, the sound energy at all control points can be suppressed by the feedforward control. As the numerical example, the rectangular prism model is considered and the validity of the proposed method is examined.

2. ANALYSIS OF STRUCTURAL-ACOUSTIC COUPLED SYSTEM

2.1 ANALYSIS OF ACOUSTIC PROBLEM

BEM is applied for analyzing the acoustic problem. Figure 1 shows a sound source model of the internal acoustic field. When it is assumed that the sound pressure behaves harmonically with the angular frequency ω, the governing equation for the sound field Ω becomes the Helmholtz equation:

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

where p is the sound pressure, k is the wave number defined as ω/c (c is the velocity of the sound).

By using BEM, the simultaneous linear equation can be obtaind as follows:

$$\{p\} = [G]\{\bar{v}\} - [H]\{\bar{p}\}, \quad [\bar{H}]\{\bar{p}\} = [\bar{G}]\{\bar{v}\}, \tag{2}$$

where $\bar{v}$ is the velocity on the surface and $(\bar{\bullet})$ denotes the boundary value of $(\bullet)$.

2.2 ANALYSIS OF STRUCTURAL VIBRATION

When it is assumed that the structure vibrates with the angular frequency ω, the equation of motion is expressed by using FEM as follows:

$$\left([K] + j\omega[C] - \omega^2[M]\right)\{\bar{x}\} = \{f_e\} + \{f_p\}, \tag{3}$$

where $\{\bar{x}\}$ is the nodal displacement, $[K]$, $[C]$ and $[M]$ are the stiffness, damping and mass matrices, respectively. The damping is assumed to be the proportional damping. $\{f_p\}$ expresses the external force caused by the sound pressure while $\{f_e\}$ expresses the other external forces.

2.3 STRUCTURAL-ACOUSTIC COUPLED ANALYSIS

In order to couple Eq.(2) which represents the sound field and Eq.(3) which represents the structural dynamics, the velocity on the boundary $\{\bar{v}\}$ is expressed by using the matrix $[T]$ which transforms from the nodal point of FEM mesh to the center point of BEM mesh as follows:

$$\{\bar{v}\} = j\omega[T]\{\bar{x}\}. \tag{4}$$

The external force of the sound pressure $\{f_p\}$ is expressed by using the matrix $[Q]$ which is composed with the area of every boundary elements as follows:

$$\{f_p\} = [T]^T[Q]\{\bar{p}\}. \tag{5}$$

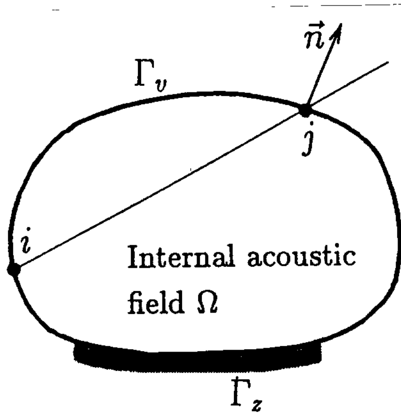


Fig.1 Model of internal acoustic problem.

When the equations in the sound field and the structural system are coupled by using Eqs.(4) and (5), the following equations are obtained:

$$\begin{bmatrix} [\bar{H}] & -j\omega[\bar{G}][T] \\ -[T]^T[Q] & [A] \end{bmatrix} \begin{Bmatrix} \{\bar{p}\} \\ \{\bar{x}\} \end{Bmatrix} = \begin{Bmatrix} \{0\} \\ \{f_e\} \end{Bmatrix}. \tag{6}$$

The structural-acoustic coupled system can be analyzed using Eq.(6), and the internal sound pressure $\{p\}$ can be calculated by using the first equation in Eq.(2).

2.4 COUPLED EQUATION WITH SECONDARY SOUND SOURCE

The secondary sound source for the sound control is installed in the cavity. The sound pressure on the surface of the sound source is $\{\bar{p}_s\}$ and the velocity is $\{\bar{v}_s\}$, the expression of the secondary sound source is obtained as well as Eq.(2) as follows:

$$\{p_s\} = [G_s]\{\bar{v}_s\} - [H_s]\{\bar{p}_s\},\ [\bar{H}_s]\{\bar{p}_s\} = [\bar{G}_s]\{\bar{v}_s\}. \tag{7}$$

The external force due to the sound source $\{p_s\}$ can, then, be obtained as follows:

$$\{f_{ps}\} = [T]^T[Q]\{p_s\} \equiv [C_2]\{\bar{p}_s\}. \tag{8}$$

The coupled equation with the secondary sound source can, therefore, be obtained as follows:

$$\begin{bmatrix} [\bar{H}] & -j\omega[\bar{G}][T] \\ -[T]^T[Q] & [A] \end{bmatrix} \begin{Bmatrix} \{\bar{p}\} \\ \{\bar{x}\} \end{Bmatrix} = \begin{Bmatrix} \{0\} \\ \{f_e\} + [C_2]\{\bar{p}_s\} \end{Bmatrix}. \tag{9}$$

2.5 ESTIMATION OF INNER SOUND PRESSURE

The estimation method of the inner sound pressure which is not measured directly is shown using both some vibration data of the structure and some sound pressure data in the cavity.

The velocity on the surface $\{\bar{v}\}$ is expressed as the superposition of the vibration modes as follows:

$$\{\bar{v}\} = [\Phi]\{I\}, \tag{10}$$

where $[\Phi]$ is the modal matrix consist of the vibration modes in the required frequency band and $\{I\}$ is the influence coefficient vector. The innner sound pressure is expressed as follows:

$$\{p\} = ([G] - [H][\bar{H}]^{-1}[\bar{G}])[\Phi]\{I\} \equiv [S_M]\{I\}. \tag{11}$$

The inner sound pressures at m points and the velocities at l points are measured and the q vibration modes are adopted. From Eqs.(10), (11), the measured $\{\bar{v}\}_m$ and $\{p\}_m$ are selected and synthesized, the next equation can be obtained,

$$\begin{bmatrix} [S_M]_m \\ [\Phi]_m \end{bmatrix} \{I\} = \begin{Bmatrix} \{p\}_m \\ \{\bar{v}\}_m \end{Bmatrix}. \tag{12}$$

The influence coefficient $\{I\}$ can be identified by using the least square method when $l + m \geq q$, and the inner sound pressure can be estimated from Eq.(11).

3. ACTIVE CONTROL OF SOUND FIELD BY FEEDFORWARD CONTROL

In this chapter, the feedforward control methods are shown under two measuring conditions. One condition is that the sound pressures at all control points can be measured directly. Another is that the sound pressures at all control points cannot be measured directly and the sound pressures without measurement are estimated by the proposed method discribed in section **2.5**.

3.1 ACTIVE CONTROL UNDER DIRECT MEASUREMENT

The active control of sound field by feedforward control is that the secondary sound source is operated such as to minimize the sound energy, which is the sum of the square values of sound pressures at the control points.

The next equation is can be obtained by solving Eq.(9),

$$\left\{ \begin{array}{c} \{\bar{p}\} \\ \{\bar{v}\} \end{array} \right\} = \left[\begin{array}{cc} R_1 & R_2 \\ R_3 & R_4 \end{array} \right] \left\{ \begin{array}{c} \{0\} \\ \{f_e\} + [C_2]\{\bar{p}_s\} \end{array} \right\}. \quad (13)$$

When the secondary sound source is not operated, i.e., $\{\bar{p}_s\} = \{0\}$, the sound pressure on the structure is obtained as follow:

$$\{\bar{p}\}_{me} = [R_2]\{f_e\}. \quad (14)$$

It is considered that the sound pressures at all control points can be measured directly. The inner sound pressures at the control points are obtained as follows:

$$\begin{aligned} \{p\}_{me} &= [G]\{\bar{v}\}_{me} - [H]\{\bar{p}\}_{me} = ([G][\bar{G}]^{-1}[\bar{H}] - [H])\{\bar{p}\}_{me} \\ &\equiv [A_1]\{\bar{p}\}_{me} = [A_1][R_2]\{f_e\}. \end{aligned} \quad (15)$$

When the secondary sound source is operated, the sound pressure on the structure is obtained as follow:

$$\{\bar{p}\}_{cn} = [R_2](\{f_e\} + [C_2]\{\bar{p}_s\}). \quad (16)$$

The inner sound pressure is, then, obtained as follows:

$$\begin{aligned} \{p\}_{cn} &= [A_1]\{\bar{p}\}_{cn} = [A_1]\{\bar{p}\}_{me} + [A_1][R_2][C_2]\{\bar{p}_s\} \\ &\equiv \{p\}_{me} + [D_2]\{\bar{p}_s\}. \end{aligned} \quad (17)$$

The objective function is defined as follows:

$$\begin{aligned} J &\equiv \{p\}_{cn}^H\{p\}_{cn} \\ &= \{\bar{p}_s\}^H[D_2]^H\{\bar{p}_s\} + \{\bar{p}_s\}^H[A_2]^H\{p\}_{me} + \{p\}_{me}^H[D_2]\{\bar{p}_s\} + \{p\}_{me}^H\{p\}_{me}, \end{aligned} \quad (18)$$

where $\{\bullet\}^H$, $[\bullet]^H$ are thr pair transpose of $\{\bullet\}$, $[\bullet]$, respectively. The sound pressure on the secondary sound source $\{\bar{p}_s\}$ such as to minimize J can be obtained by $\partial J/\partial\{\bar{p}_s\} = 0$ as follows:

$$\{\bar{p}_s\} = -([D_2]^H[D_2])^{-1}[D_2]^H\{p\}_{me}. \quad (19)$$

The optimum active control of sound field can be attained when the secondary sound source is operated under the condition that the sound pressure is set as Eq.(19).

3.2 ACTIVE CONTROL UNDER MEASUREMENT AND ESTIMATION

It is considered that the sound pressures at all control points cannnot to measured directly. The inner sound pressure $\{p\}_{es}$ without measurement can be estimated by using the influence coefficient vector $\{I\}$ from Eq.(12) as follows:

$$\{p\}_{es} = [S_M]\{I\}. \tag{20}$$

The sum of the sound pressure $\{p\}_{me}$ measured directly and $\{p\}_{es}$ are expressed as $\{p\}_{ms}$, i.e., $\{p\}_{ms}$ are the sound pressures at the control points, the sound pressure on the secondary sound source $\{\bar{p}_s\}$ such as to minimize J can be obtained as follows:

$$\{\bar{p}_s\} = -([D_2]^H[D_2])^{-1}[D_2]^H\{p\}_{ms}. \tag{21}$$

By using this method, the number of the control points can be increased without direct measurement.

4. NUMERICAL EXAMPLES

4.1 ANALYTICAL MODEL

In this paper, the accuracy of the proposed method is examined by the numerical examples. As a structural model, the rectangular prism model is considered as shown in Fig.2 and the internal sound field is generated in the model. The model is assumed to be composed of six flexible plates individually supported rigidly along the edge line. Table 1 shows the specifications of the model.

The control points are shown as No.1 $\sim$ 9 in Fig.3, where the center of the model is No.9 and dx, dy and dz are 1/6 length of the side, respectively. The sound pressures are measured at three points, No.1, 7, 9 (m=3). Three secondary sound sources are located in the model, whose shapes are triangle. The magnitude of the external force is 1.0[N] and applied at the excitation point. The frequency of the external force is 55[Hz]. This frequency is selected because it is between the 1st and 2nd natural frequencies of 500×600 and 400×600 plate and between the 2nd and 3rd ones of 400×500 plate. The vibrations are measured at two points on each plate, so that the number of vibration measuring points is 12 (l=12). The adopted vibration modes in Eq.(10) are the 1st and 2nd ones for 500×600 and 400×600 plate and 2nd and 3rd ones for 400×500 plate, so that the number of adopted vibration modes 12 (q=12). The number of unknown parameters in Eq.(12) is 12 and the number of equations is 15, so that $\{I\}$ can be identified.

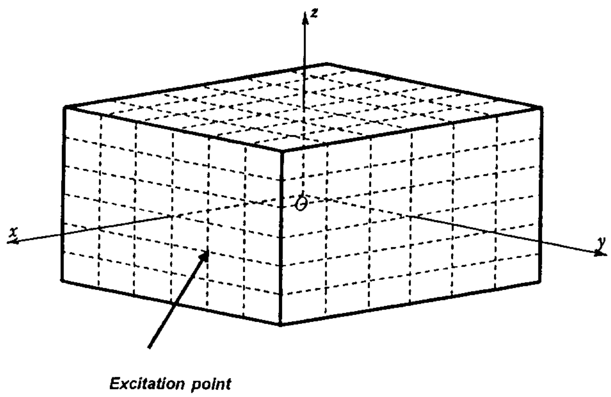


Fig.2 Model for numerical example.

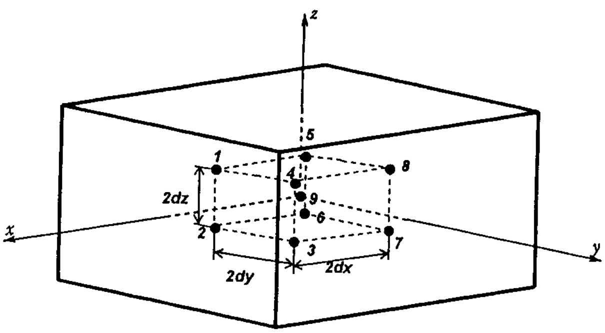


Fig.3 Control points of sound pressure.

The proposed control of sound field is that to minimize the objective function which is the sound energy of nine control points when three inner sound pressures and 12 vibration data on the surface are measured and the six inner sound pressures are estimated.

4.2 ACCURACY OF ESTIMATED INNER SOUND PRESSURE

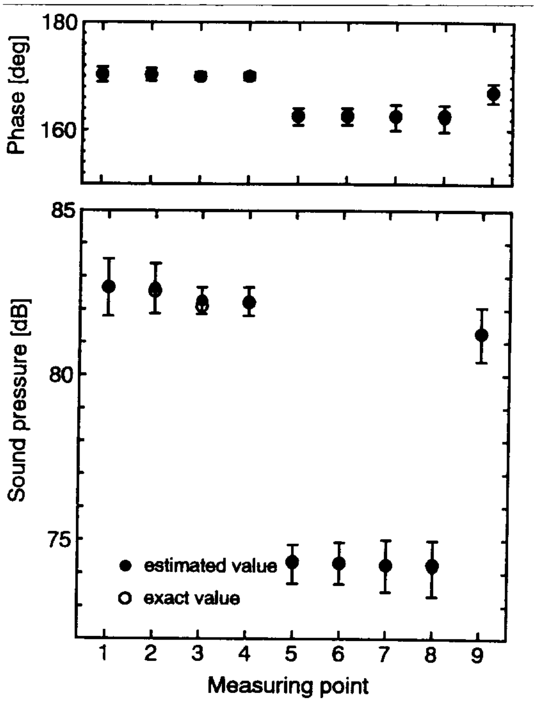
The accuracy of the estimated inner sound pressure is examined. In the measured data, there are some noise caused by the uncertainty of the specifications of the model and so on. To simulate the actual measurement, the random numbers are added to the calculated data, which are assumed to be the measured data. The Monte Carlo simulation, in which the number of samples is 1000, is carried out. The results are shown in Fig.4. The horizontal axis shows the measuring point number. The character ○ shows the exact sound pressure and ● shows the estimated sound pressure. The bar shows the standard deviation. From Fig.4, it is seen that the mean values of the estimated values agree with the exact ones and the bigger the coefficient of variation is, the variation of the estimated value is big.

4.3 ACTIVE CONTROL BY PROPOSED METHOD

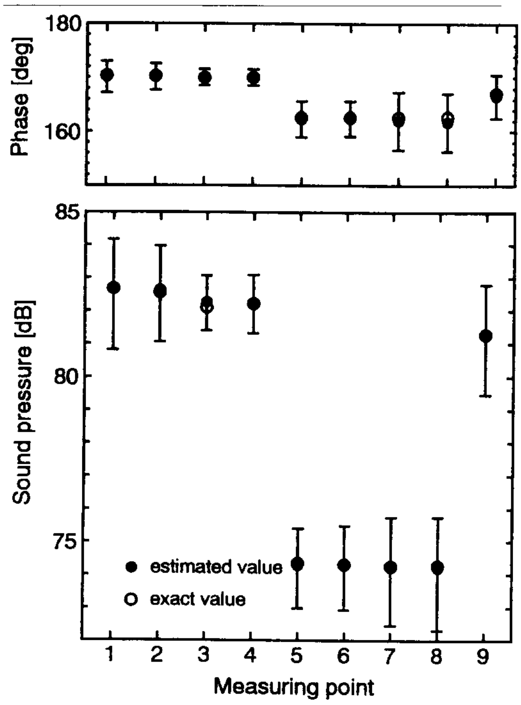
The validity of the proposed method is examined. The active control of sound field under noisy measurement as well as section **4.2** is simulated. The results are shown in Fig.5. The character ○ shows the exact sound pressure without control, ◇ shows the suppressed sound pressure by the control using nine measured directly data, and ■ shows the one by the proposed control under the measurement and estimation. From Fig.5, it is seen that the mean values of the suppressed sound pressures by the proposed control agree with the ones by the control under direct measurement.

4.4 COMPARISON OF CONTROL EFFECT BY DIFFERENCE OF CONTROL POINT NUMBER

In the previous section, the objective function is the sound energy of nine control points, so that when the sound pressures are only measured at three points, the six sound



(a) Coefficient of variation : 0.1



(b) Coefficient of variation : 0.2

Fig.4 Accuracy of estimated sound pressure.

pressures are estimated by the proposed method. In that case, it is natural that the objective function is set as the sound energy of three control points where the sound pressures are measured. The control effect is, therefore, compared by the following way. The active control are carried out in case of

(a) the sound pressures at three control points are measured and the objective function for active control is $J_1 = \sum_{i=1,7,9} p_i^H \cdot p_i$,

(b) the sound pressures at nine control points are measured and the objective function for active control is $J_2 = \sum_{i=1}^{9} p_i^H \cdot p_i$.

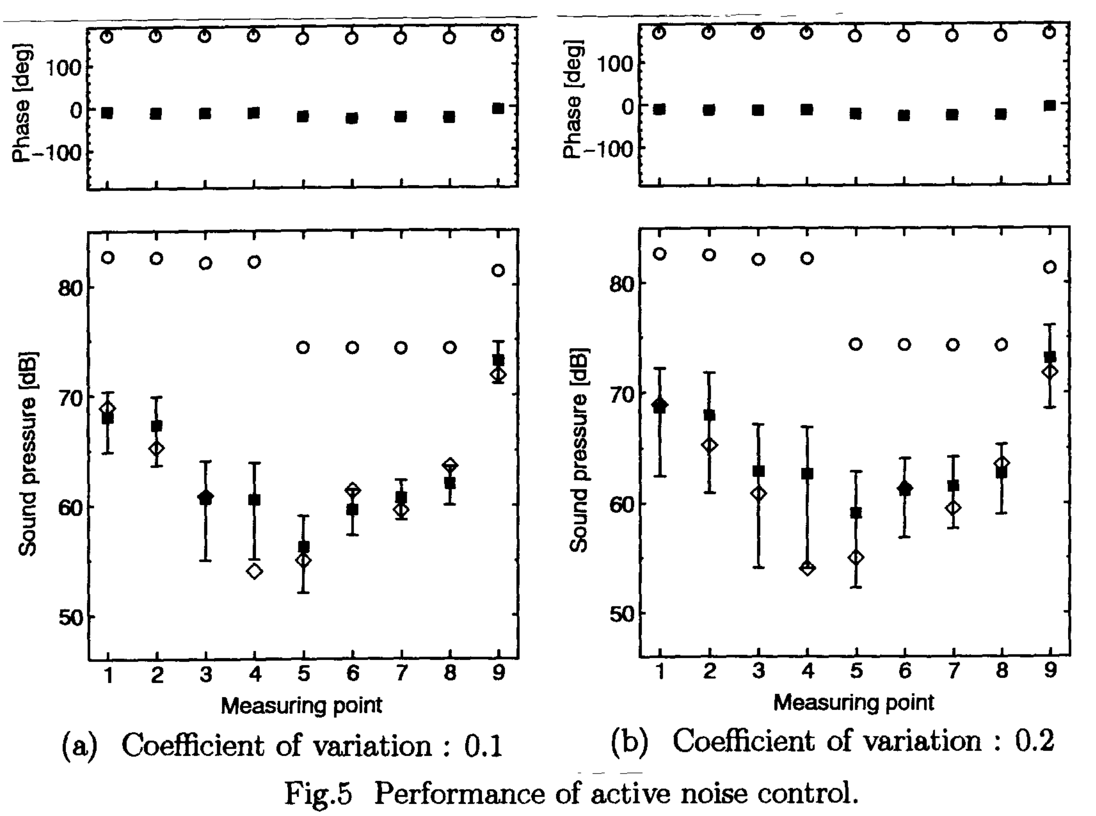
The control effect is conpared by using the functions,

$$F_1 = \sum_{i=1,7,9} p_i^H \cdot p_i, \quad F_2 = \sum_{i=1}^{9} p_i^H \cdot p_i \tag{22}$$

The results are shown in Table 2. From Table 2, the value of F_2 for control (a) is relatively big, i.e., the control effect for nine control points is small by control (a). Control (b) is, therefore, effective for control of the large area.

5. CONCLUSIONS

In this paper, the effective active noise control method was proposed for the steady state sound field in the enclosed cavity. In that method, the sound pressures, where no microphones are located, are estimated from both the vibration data and the measured sound pressure data based on the acoustic vibratory inverse analysis and structure acoustic coupled analysis. By using the method, the number of the control points can be increased without direct measurement. As the numerical example, the rectangular prism model was



(a) Coefficient of variation : 0.1 (b) Coefficient of variation : 0.2

Fig.5 Performance of active noise control.

Table 1 Specification of example model

Length of model	[mm]	600 ×500× 400
Thickness of plates	[mm]	1.0
Mass density	[kg/m^3]	7.86 ×10^3
Modulus of elasticity	[Pa]	2.05 ×10^{11}

Table 2 Comparison of objective function

	Initial value	Control (a)	Control (b)
F_1[Pa^2]	0.2770	1.0370×10^{-26}	7.1330×10^{-3}
F_2[Pa^2]	0.7483	7.8387×10^{-1}	0.2614×10^{-1}

considered and the validity of the proposed method was examined. As the results, it was seen that the mean values of the estimated sound pressures agree with the exact ones and the mean values of the suppressed sound pressures by the proposed control agree with the ones by the control under direct measurement. It was, therefore, seen that the proposed method is effective for the actual control.

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