Active noise reduction of a coupled rectangular cavity using active wave control

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ABSTRACT

The objective of this study is to generate a quiet space in a coupled rectangular cavity by suppressing a travelling wave caused by vibration of a flexible panel. Firstly, this study begins with description of an enclosed sound field from a wave point of view by introducing a one-dimensional transfer matrix method for three-dimensional space. Next, a concept of modal coupling method is utilized to model a coupled rectangular cavity. Furthermore, introducing multiple control sound sources in an arbitrary plane in the target space, the control law is derived that eliminates a travelling wave passing through the control plane. Finally, numerical simulations are carried out to demonstrate the validity of the proposed method. As a result, it was clarified that the proposed method enables to generate a quiet space by cancelling transmitted waves.

Keywords: Coupling system, Active wave control, Quiet zone I-INCE Classification of Subjects Number(s): 38.2

1. INTRODUCTION

Noise pollution is listed as one of the seven typical pollution in Japan since it can damage human beings physically and psychologically. Furthermore, the number of the complaints for noise is the second most and its contents are serious, so that the development of noise control techniques is very important. In addition, if disturbance sources are placed in an enclosed field, the sound inside the cavity can be amplified under the resonant condition, and hence resolution of the suppression issue is considered to be urgent. Since it is difficult to suppress such noise by passive means, active control methodology using control sources has attracted much attention in recent years.

Reviewing the literature on active control for an enclosed sound field, the main stream aims at globally suppressing the cabin noise based on the modal expression. However, there are few reports on the method which is based on the wave dynamics. Considering that the noise control is for human beings, it is not necessary to globally suppress the noise inside the cabin. In other words, if the noise energy is confined in the region where the noise is not a problem for human beings, it can be evaluated that the noise suppression is achieved. The method of realizing this control effect is active wave control method.

In this paper, generation of a quiet space in a vibro-acoustic coupling system is presented using the wave control method. Firstly, this study begins with description of an enclosed sound field from a wave point of view by introducing a one-dimensional transfer matrix method for three-dimensional space. Next, a concept of modal coupling method is utilized to model a coupled rectangular cavity. Furthermore, introducing multiple control sound sources in an arbitrary plane in the target space, the control law is derived that eliminates a travelling wave passing through the control plane. Finally, numerical simulations are carried out to demonstrate the validity of the proposed method. As a result, it is clarified that the proposed method enables to generate a quiet space by cancelling transmitted waves.

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2. THEORETICAL DEVELOPMENT

In this section, we describe the theory of the transfer matrix which is necessary in considering the wave dynamics, and then we propose a method for modeling the coupled system from the wave point of view. Finally, we derive the control law to suppress a wave that passes through the sound source plane.

2.1 Transfer Matrix

This paper put the emphasis on the wave dynamics rather than the modal one, so that the transfer matrix method is introduced which is based on the traveling-wave solution of an enclosed sound field.

First, the general solution to the equation of the three-dimensional rectangular enclosed space is described as

$$p(x,y,z)=\sum_{l,m,n=0}^{\infty} p_{x,l}(x)p_{y,m}(y)p_{z,n}(z), \quad (1)$$

where $p_{x,l}(x)$, $p_{y,m}(y)$ and $p_{z,n}(z)$ are acoustic mode components for the x, y, and z directions, respectively. Generally, the acoustic mode is referred to as (l, m, n) mode using modal indices at each spatial coordinate. Then, assuming that the acoustic mode components in the x and y directions are the periodic, the following relationship is obtained:

$$\frac{\partial^2 p_{x,l}(x)}{\partial x^2}=-\alpha_l^2 p_{x,l}(x), \quad (2)$$

$$\frac{\partial^2 p_{y,m}(y)}{\partial y^2}=-\beta_m^2 p_{y,m}(y), \quad (3)$$

where $\alpha^2{}_l$ and $\beta^2{}_m$ are constants determined by the boundary conditions. Then, substituting Eq. (1) to (3) into the Helmholtz equation $\nabla^2 p(x,y,z)+k^2 p(x,y,z)=0$ yields

$$\sum_{l,m,n=0}^{\infty} \{-\alpha_l^2 p_{z,n}(z)-\beta_l^2 p_{z,n}(z)+\frac{\partial^2 p_{z,n}(z)}{\partial z^2}\}=0\,. \quad (4)$$

The general solution to the above equation is then obtained as

$$p_{z,n}(z)=c_{1,lm}e^{-k_{lm}z}+c_{2,lm}e^{k_{lm}z}, \quad (5)$$

where

$$k_{lm}=\sqrt{\alpha_l^2+\beta_m^2-k^2}\,. \quad (6)$$

As is apparent from the above equation, k_{lm} becomes imaginary if the angular frequency exceeds a certain threshold. As a result, the first term on the right-hand side of Eq. (5) becomes a positive traveling wave component, while the second term becomes the negative one. Thus, the wave in the z direction does not propagate unless the value in the square root becomes negative. The frequency corresponding to this threshold is called cut-on frequency and is defined as

$$\omega_{c,lm}=c\sqrt{\alpha_l^2+\beta_m^2}\,. \quad (7)$$

Furthermore, as is apparent from Eqs. (5) and (6), the acoustic mode component in the z direction is constant at the cut-on frequency.

Next, substituting Eq. (5) into Eq. (1), we have

$$p(x,y,z)=\sum_{l,m=0}^{\infty} p_{x,l}(x)p_{y,m}(y)(c_{1,lm}e^{-k_{lm}z}+c_{2,lm}e^{k_{lm}z})\,. \quad (8)$$

As written in the above equation, the eigenmode element of the z direction is expressed by the progressive wave solution of the (l, m) mode group. Next, sound particle velocity v_x in the z direction is determined by the sound pressure p, and then the state vector of the enclosed space is described as

$$\mathbf{z}_z(x,y,z) = (p(x,y,z) \quad v_z(x,y,z))^T = \sum_{l,m=0}^{\infty} p_{x,l}(x) p_{y,m}(y) \mathbf{z}_{z,lm} \quad . \tag{9}$$

Here, the state vector of the (l, m) mode group, $z_{z,lm}(z)$, is described as

$$\mathbf{z}_{z,lm}(z) = (p_{lm}(z) \quad v_{z,lm}(z))^T \ , \tag{10}$$

where p_{lm} is the sound pressure element of the (l, m) mode group and v_{lm} is the corresponding particle velocity element.

Next, consider the two separated x-y plane in the enclosed space. Defining the left plane as node i-1 and the right plane as node i, the relationship between the state vectors at two nodes is described as

$$_{i}\mathbf{z}_{z,lm} = {}_{i,i-1}\mathbf{T}_{lm}\ {}_{i-1}\mathbf{z}_{z,lm} \ , \tag{11}$$

where $_{i}z_{z,lm}$ and $_{i-1}z_{z,lm}$ are the state vectors at the nodes i and i-1, respectively, and $_{i,i-1}\mathbf{T}_{lm}$ is the transfer matrix between the two nodes.

2.2 Modeling of Coupling System by the Fusion of Modal Coupling and Transfer Matrix Method

In the coupling system, the structural vibration affects the sound inside the cavity and vice versa. The method to model such a loop of energy is the modal coupling approach; however, this method cannot treat the wave dynamics since the sound field is expressed in terms of the acoustical modes of rigid-walled cavity. To cope with this problem, this paper proposes to replace this modal expression by the wave solution that can be treated by the transfer matrix, that is, the fusion of modal coupling and the transfer matrix.

Consider the case where the N_c control sources are placed in the x-y plane of the coupling cavity and the disturbance force is applied to the flexible panel as shown in Fig. 1. Then, defining the left boundary (rigid wall), the control plane and the right boundary (flexible panel) as nodes 0, 1 and 2, respectively, the state vector of the (l, m) of the mode group at the flexible panel is described as

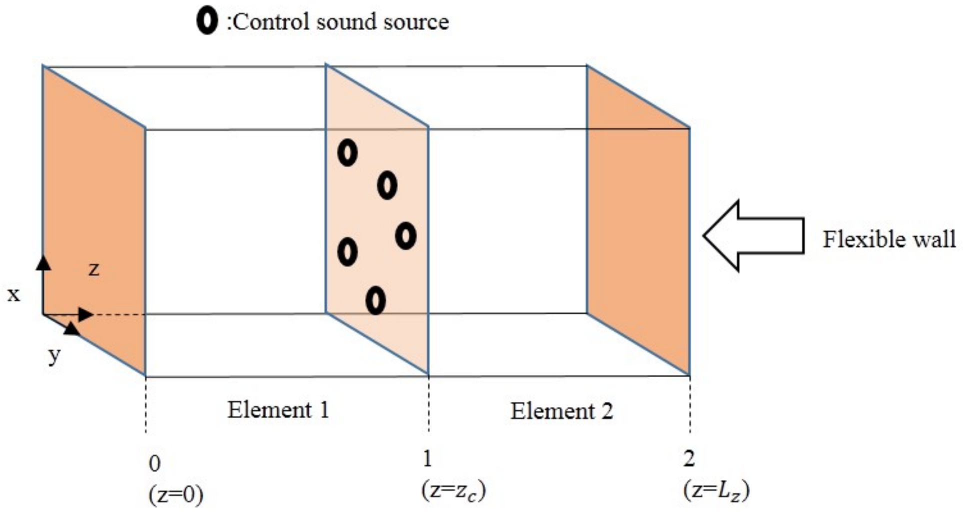

Figure 1 – Three dimensional rectangular enclosure with disturbance and control sound source

$$_{2}\mathbf{z}_{z,lm} = {}_{20}\mathbf{T}_{lm}\ {}_{0}\mathbf{z}_{z,lm} + {}_{21}\mathbf{T}_{lm}\mathbf{q}_{c,lm} + \mathbf{v}_{lm}$$
$$\begin{pmatrix} {}_{2}p_{z,lm} \\ 0 \end{pmatrix} = \begin{pmatrix} {}_{20}t_{11,lm} & {}_{20}t_{12,lm} \\ {}_{20}t_{21,lm} & {}_{20}t_{22,lm} \end{pmatrix} \begin{pmatrix} {}_{0}p_{z,lm} \\ 0 \end{pmatrix} + \begin{pmatrix} {}_{21}t_{11,lm} & {}_{21}t_{12,lm} \\ {}_{21}t_{21,lm} & {}_{21}t_{22,lm} \end{pmatrix} \begin{pmatrix} 0 \\ q_{c,lm} \end{pmatrix} + \begin{pmatrix} 0 \\ v_{lm} \end{pmatrix}, \tag{12}$$

where v_{lm} is the effect of the panel vibration to the (l,m) mode group of the cavity noise and $q_{c,lm}$ is the effect of the N_c control sources to the (l,m) group, and are given by

$$q_{c,lm} = e_{lm} \sum_{n=1}^{N_c} q_n p_{x,l}(x_n) p_{y,m}(y_n), \tag{13}$$

$$v_{lm} = e_{lm} \int_S p_{x,l}(x) p_{y,m}(y) v(x,y) dS, \tag{14}$$

where e_{lm} is a constant determined by the indices of the mode, q_n is the volume velocity of each control source. From Eq. (12), the following two equations are obtained:

$$_2p_{z,\mathrm{lm}} = {}_{20}t_{11,lm}\, {}_0p_{z,lm} + {}_{21}t_{12,lm} q_{c,lm}, \tag{15}$$

$$0 = {}_{20}t_{21,lm}\, {}_0p_{z,lm} + {}_{21}t_{22,lm} q_{c,lm} + v_{lm}. \tag{16}$$

Combining Eq. (15) and (16), the acoustic pressure at $z=L_z$ is described as

$$_2p_{z,lm} = \lambda_{1,lm} q_{c,lm} + \lambda_{2,lm} v_{lm}, \tag{17}$$

where

$$\lambda_{1,lm} = {}_{21}t_{12,lm} - \frac{{}_{20}t_{11,lm}\, {}_{21}t_{22,lm}}{{}_{20}t_{21,lm}}, \tag{18}$$

$$\lambda_{2,lm} = -\frac{{}_{20}t_{11,lm}}{{}_{20}t_{21,lm}}. \tag{19}$$

As shown later, a general solution of a simply supported panel is written as

$$v(\mathrm{x},\mathrm{y}) = \sum_{i,j=0}^{\infty} b_{ij} \sin\frac{i\pi}{L_x} x \sin\frac{j\pi}{L_y} x. \tag{20}$$

Substituting the above equation into Eq. (14), it is rewritten as.

$$\begin{aligned} v_{lm} &= \sum_{i,j=1}^{\infty} b_{ij} e_{lm} \int_S \cos\frac{l\pi}{L_x} x \cos\frac{m\pi}{L_y} y \sin\frac{i\pi}{L_x} x \sin\frac{j\pi}{L_y} y dS \\ &= \sum_{i,j=1}^{\infty} b_{ij} e_{lm} \beta_{lm,ij} \\ &= e_{lm} \gamma_{lm}^{\mathrm{T}} \mathrm{b} \end{aligned}, \tag{21}$$

where,

$$\gamma_{lm}^{\mathrm{T}} = \begin{pmatrix} \beta_{lm,11} & \beta_{lm,12} & \cdots \end{pmatrix}, \tag{22}$$

$$\mathrm{b} = \begin{pmatrix} b_{11} & b_{12} & \cdots \end{pmatrix}^{\mathrm{T}}. \tag{23}$$

Here, $\beta_{lm,ij}$ is a coupling coefficient between the (l,m) mode group of the cavity and the (i,j) mode of the panel. Considering all mode groups, Eq. (15) is rewritten in the matrix form as

$$\begin{aligned} \begin{pmatrix} {}_2p_{z,00} \\ {}_2p_{z,10} \\ {}_2p_{z,01} \\ \vdots \end{pmatrix} &= \begin{pmatrix} \lambda_{1,00} & & & 0 \\ & \lambda_{1,10} & & \\ & & \lambda_{1,01} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} q_{c,00} \\ q_{c,10} \\ q_{c,01} \\ \vdots \end{pmatrix} + \begin{pmatrix} \lambda_{2,00} & & & 0 \\ & \lambda_{2,10} & & \\ & & \lambda_{2,01} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} v_{00} \\ v_{10} \\ v_{01} \\ \vdots \end{pmatrix} \\ &= \begin{pmatrix} \lambda_{1,00} & & & 0 \\ & \lambda_{1,10} & & \\ & & \lambda_{1,01} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} q_{c,00} \\ q_{c,10} \\ q_{c,01} \\ \vdots \end{pmatrix} + \begin{pmatrix} \lambda_{2,00}e_{00} & & & 0 \\ & \lambda_{2,10}e_{01} & & \\ & & \lambda_{2,01}e_{01} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} \gamma_{00}^{\mathrm{T}} \\ \gamma_{10}^{\mathrm{T}} \\ \gamma_{01}^{\mathrm{T}} \\ \vdots \end{pmatrix} \mathrm{b}. \\ \mathbf{p}_z(L_z) &= \mathbf{\Lambda}_1 \mathbf{q}_c + \mathbf{\Lambda}_2 \mathbf{C} \mathbf{b} \end{aligned} \tag{24}$$

Next, consider the dynamics of a flexible panel. The equation of motion can be given as

$$D\nabla_s^4 w(x,y,t) + \rho_s h\ddot{w}(x,y,t) = f(x,y,t) - p(x,y,L_z,t)\,, \tag{25}$$

where D is bending rigidity of the panel, ρ is density and f is external force. Assuming the harmonic excitation at an angular frequency ω, the equation of motion with respect to the velocity can be written as

$$D\nabla_s^4 v(x,y) - \rho_s h\omega^2 v(x,y) = j\omega(f(x,y) - p(x,y,L_z))\,. \tag{26}$$

The general solution to the above equation can be given as

$$v(x,y) = \sum_{i,j=0}^{\infty} b_{ij} \sin\frac{i\pi}{L_x}x \sin\frac{j\pi}{L_y}y\ , \tag{27}$$

where

$$b_{ij} = \frac{j\omega}{M_s(\omega_{ij}^2 - \omega^2)}(f_{ij} - g_{ij})\,, \tag{28}$$

$$f_{ij} = \int_S f(x,y)\sin\frac{i\pi}{L_x}x\sin\frac{j\pi}{L_y}ydS\,, \tag{29}$$

$$g_{ij} = \int_S p(x,y,L_z)\sin\frac{i\pi}{L_x}x\sin\frac{j\pi}{L_y}ydS\ , \tag{30}$$

$$M_s = \frac{\rho h L_x L_y}{4}\,. \tag{31}$$

Here, f_{ij} is the generalized force to the (i,j) mode of the panel, g_{ij} is the effect of the sound pressure in cavity to the (i,j) mode of the panel. The acoustic pressure that acts on the panel is described as

$$p(x,y,L_z) = \sum_{l,m=1}^{\infty} p_{x,l}(x)p_{y,m}(y)p_{z,lm}(L_z)\,. \tag{32}$$

Substitutng the above equation into Eq. (30), it is rewritten as

$$\begin{aligned} g_{ij} &= \int_S \rho(x,y,L_z)\sin\frac{i\pi}{L_x}x\sin\frac{j\pi}{L_y}ydS \\ &= \sum_{l,m=1}^{\infty} p_{z,lm}(L_z)\int_S \cos\frac{l\pi}{L_x}x\cos\frac{m\pi}{L_y}y\sin\frac{i\pi}{L_x}x\sin\frac{j\pi}{L_y}ydS \\ &= \sum_{l,m=0}^{\infty} p_{z,lm}(L_z)\beta_{lm,ij} \\ &= \boldsymbol{\beta}_{ij}^{\mathrm{T}}\mathbf{p}_z(L_z) \end{aligned}\ , \tag{33}$$

where

$$\boldsymbol{\beta}_{ij}^{\mathrm{T}} = (\beta_{00,ij} \quad \beta_{10,ij} \quad \beta_{01,ij} \quad \cdots)^{\mathrm{T}}\,. \tag{34}$$

Next, substitutng Eqs (28) and (34) into Eq (23), it is rewritten as

$$\begin{pmatrix} b_{11} \\ b_{12} \\ \vdots \end{pmatrix} = \begin{pmatrix} \dfrac{j\omega}{M_s(\omega_{11}^2 - \omega^2)} & & 0 \\ & \dfrac{j\omega}{M_s(\omega_{12}^2 - \omega^2)} & \\ 0 & & \ddots \end{pmatrix} \left(\begin{pmatrix} f_{11} \\ f_{12} \\ \vdots \end{pmatrix} - \begin{pmatrix} g_{11} \\ g_{12} \\ \vdots \end{pmatrix} \right) \tag{35}$$

$$\mathbf{b} = \mathbf{B} \left(\mathbf{f} - \begin{pmatrix} \beta_{11}^{\mathrm{T}} \\ \beta_{12}^{\mathrm{T}} \\ \vdots \end{pmatrix} \mathrm{p}_z(L_z) \right) = \mathbf{B}\left(\mathbf{f} - \mathbf{C}^{\mathrm{T}} \mathrm{p}_z(L_z)\right).$$

Simultaneously solving Eqs. (24) and (25), $\mathbf{p}_z(L_z)$ and $\mathbf{b}$ are separately obtained as

$$\mathbf{p}_z(L_z) = (\mathbf{I} + \mathbf{\Lambda}_2\mathbf{CBC}^{\mathrm{T}})^{-1}(\mathbf{\Lambda}_1\mathbf{q}_c + \mathbf{\Lambda}_2\mathbf{CBf}), \tag{36}$$

$$\mathbf{b} = (\mathbf{I} + \mathbf{BC}^{\mathrm{T}}\mathbf{\Lambda}_2\mathbf{C})^{-1}\mathbf{B}(\mathbf{f} - \mathbf{C}^{\mathrm{T}}\mathbf{\Lambda}_1\mathbf{q}_c). \tag{37}$$

Thus, the modeling of the coupling system was established in the vector form as written in the above equations.

2.3 Derivation of the Control Law

In this subsection, the control law to confine the sound energy around the vibrating panel is derived. The control target is the negative travelling waves between the left boundary and the control plane, that is, the waves passing through the control plane are cancelled with control sources. Since the nullification of the sound pressure on a rigid wall is equivalent to that of the traveling wave, this subsection begins with derivation of the sound pressure at z=0.

From Eq. (15), the sound pressure of the initial state vector is written as

$${}_0p_{z,lm} = \frac{1}{{}_{20}t_{11,lm}}\, {}_2p_{z,lm} - \frac{{}_{21}t_{12,lm}}{{}_{20}t_{11,lm}} q_{c,lm}. \tag{38}$$

Considering all the target modes, the above equation can be expressed in the matrix form as

$$\begin{pmatrix} {}_0p_{z,00} \\ {}_0p_{z,10} \\ {}_0p_{z,01} \\ \vdots \end{pmatrix} = \begin{pmatrix} \frac{1}{{}_{20}t_{11,00}} & & & 0 \\ & \frac{1}{{}_{20}t_{11,10}} & & \\ & & \frac{1}{{}_{20}t_{11,01}} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} {}_2p_{z,00} \\ {}_2p_{z,10} \\ {}_2p_{z,01} \\ \vdots \end{pmatrix} + \begin{pmatrix} -\frac{{}_{21}t_{12,00}}{{}_{20}t_{11,00}} & & & 0 \\ & -\frac{{}_{21}t_{12,10}}{{}_{20}t_{11,10}} & & \\ & & -\frac{{}_{21}t_{12,01}}{{}_{20}t_{11,01}} & \\ 0 & & & \ddots \end{pmatrix} \begin{pmatrix} q_{c,00} \\ q_{c,10} \\ q_{c,01} \\ \vdots \end{pmatrix}. \tag{39}$$

$$\mathbf{p}_z(0) = \mathbf{\Lambda}_3\mathbf{p}_z(L_z) + \mathbf{\Lambda}_4\mathbf{q}_c$$

Then, substituting Eq. (36) into the above equation, we have

$$\mathbf{p}_z(0) = \mathbf{H}_1\mathbf{q}_c + \mathbf{H}_2\mathbf{f}, \tag{40}$$

where

$$\mathbf{H}_1 = \Lambda_3(\mathrm{I} + \Lambda_2\,\mathrm{C}\mathbf{B}\mathrm{C}^{\mathrm{T}})^{-1}\Lambda_1 + \Lambda_4, \tag{41}$$

$$\mathbf{H}_2 = \Lambda_3(\mathrm{I} + \Lambda_2\,\mathrm{C}\mathbf{B}\mathrm{C}^{\mathrm{T}})^{-1}\Lambda_2\mathrm{C}\mathbf{B}. \tag{42}$$

As described above, the control law for cancelling the transmitted waves is obtained by nullifying the sound pressure at the rigid wall. Therefore, the control law is derived as

$$\mathbf{q}_c = -\mathbf{H}_1^{-1}\mathbf{H}_2\mathbf{f}. \tag{43}$$

Next, Eq (13) can be rewritten in the vector form as

$$q_{c,lm} = \begin{pmatrix} e_{lm}\cos\frac{l\pi}{Lx}x_{c,1}\cos\frac{m\pi}{Ly}y_{c,1} & e_{lm}\cos\frac{l\pi}{Lx}x_{c,2}\cos\frac{m\pi}{Ly}y_{c,2} & \cdots \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \\ \vdots \end{pmatrix}. \tag{44}$$

$$= \mathbf{d}_{lm}^{\mathrm{T}}\mathbf{q}$$

Furthermore, assuming that the disturbance force f_d is applied at $(x,y) = (x_0,y_0)$, the Eq (43) is rewritten as

$$\begin{pmatrix} \mathbf{d}_{00}^{\mathrm{T}} \\ \mathbf{d}_{01}^{\mathrm{T}} \\ \mathbf{d}_{10}^{\mathrm{T}} \\ \vdots \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \\ q_3 \\ \vdots \end{pmatrix} = -\mathbf{H}_1^{-1}\mathbf{H}_2\begin{pmatrix} \sin\left(\frac{\pi}{Lx}x_0\right)\sin\left(\frac{\pi}{Ly}y_0\right) \\ \sin\left(\frac{\pi}{Lx}x_0\right)\sin\left(\frac{2\pi}{Ly}y_0\right) \\ \sin\left(\frac{l\pi}{Lx}x_0\right)\sin\left(\frac{3\pi}{Ly}y_0\right) \\ \vdots \end{pmatrix} f_d . \tag{45}$$

$$\mathbf{Dq} = -\mathbf{H}_1^{-1}\mathbf{H}_2\mathbf{s}f_d$$

It should be noted that the matrix **D** can be inverted if the number of the target modes is the same as that of the control sources. In this case, the vector of the volume velocity of each control sources can be described in the feedforward control form as

$$\mathbf{q} = -\mathbf{D}^{-1}\mathbf{H}_1^{-1}\mathbf{H}_2\mathbf{s}f_d . \tag{46}$$

As an exception, if the number of the target modes is not equal to that of control sources, the pseudo inverse of the matrix **D** is utilized.

3. NUMERICAL ANALYSIS

In this section, the validity of the active wave control based on the theory described in section 2 is demonstrated by numerical analyses. The specifications of the coupled system are listed in Table 1. The number of mode groups to calculate the space is nine (that is, up to the (2, 2) mode group), the number of modes for calculating the panel is also nine (that is, up to the (3, 3) mode group). The target mode groups are the (0, 0), (0, 1), and (1, 0) mode groups. In order to suppress the waves belonging to those mode groups, three control sound sources are placed in the *x*-*y* plane at z = 0.3 [m]. Their *x*-*y* coordinates in the control plane are $(x, y) = (0, L_z)$, (0, 0) and $(L_x, 0)$. Furthermore, it is assumed that the disturbance force f_d is applied to the position of the plate $(x, y) = (0.01, 0.01)$.

3.1 Frequency Characteristics

When the observation point is $(x, y, z) = (0.01, 0.01, 0.2)$, the frequency characteristics of the sound pressure level before and after control is shown in Fig 2. Comparing the frequency characteristics after control with those before control, it is confirmed that the asymptotic line of the gain curve after control becomes lower than the case before control. Although there are some overlapping peaks, the meaning of this result is considered in the following.

The frequency characteristics of each mode group in the sound pressure level before and after control is shown in Fig 3. For example, consider the peak at 113 [Hz] before and after control. As shown in the figure, this peak mainly belongs to the (0, 1) mode group before control. However, when the control is applied, the (1, 1) and (2, 1) mode groups have the high-level peak at the same frequency, while the (0, 1) mode group is completely suppressed. Thus, this result is due to the spillover effect, that is, when controlling the (0, 0), (0, 1) and (1, 0) mode groups, the control sound sources affect the non-target mode groups through the coupling between sound and vibration field. As shown in Fig. 3 (b), since the sound pressure levels of the (0, 0), (0, 1) and (1, 0) mode groups are low, it is demonstrated that the waves belonging to the target mode groups is reliably suppressed by the proposed method.

Table 1 – Prameters of the coupled rectangular cavity

The overall length of the x direction	0.18[*m*]	Young's modulus	206[*GPa*]
The overall length of the y direction	0.38[*m*]	Panel density	7900[kg/m^3]
The overall length of the z direction	0.466[*m*]	Poisson's ratio	0.29
Speed of sound	340[*m/s*]	Disturbance force	1[*N*]
Air density	1.21[kg/m^3]	Panel thickness	0.0008[*m*]

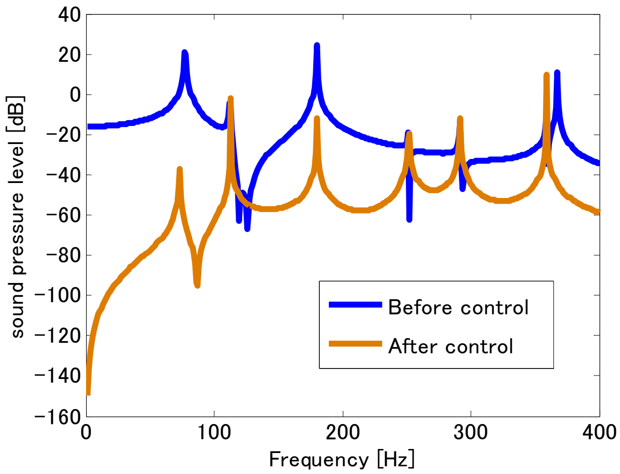

Figure 2 – Frequency of sound pressure level at (x, y, z) = (0.01, 0.01, 0.2) before and after control

(a) Before control (b) After control

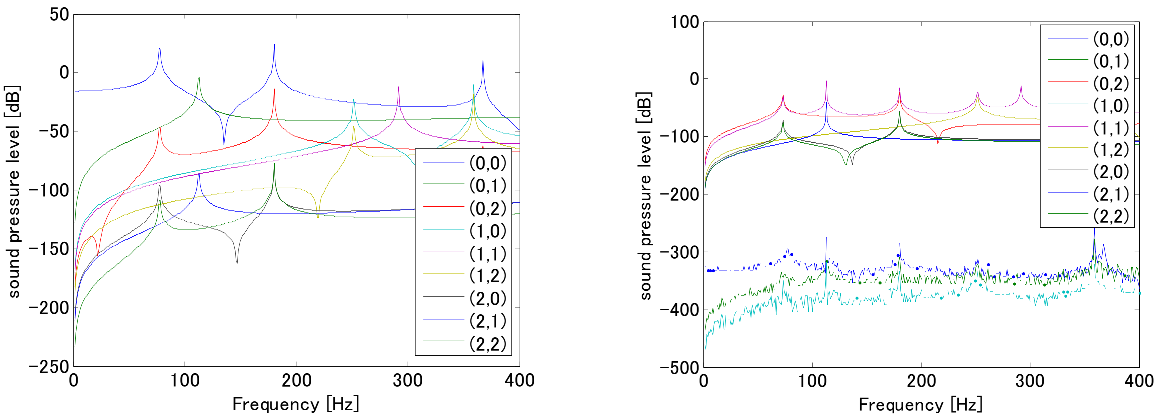

Figure 3 – Frequency response of sound pressure level at (x, y, z) = (0.01, 0.01, 0.2) before and after control

3.2 Sound Pressure Distribution on the Walls

Based on the results of the frequency characteristics shown in the previous subsection, whether a quiet space is generated is confirmed by comparing the sound pressure distribution on the walls before and after control. Sound pressure distributions at 77 [Hz] and 180 [Hz] before and after control are shown in Figs. 4 and 5. (◎: the disturbance force, ●: the control sound source)

(a) Before control (b) After control

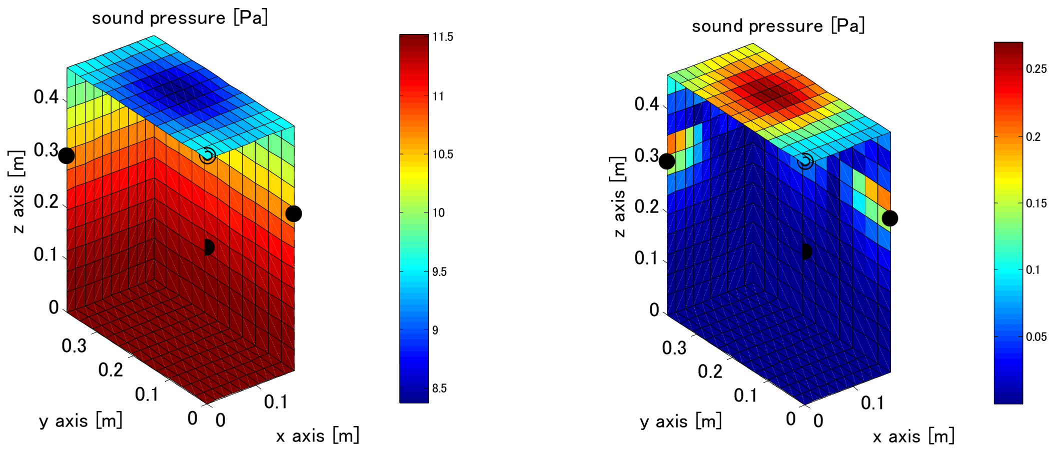

Figure 4 – Sound pressure distribution at 77 [Hz]

(a) Before control (b) After control

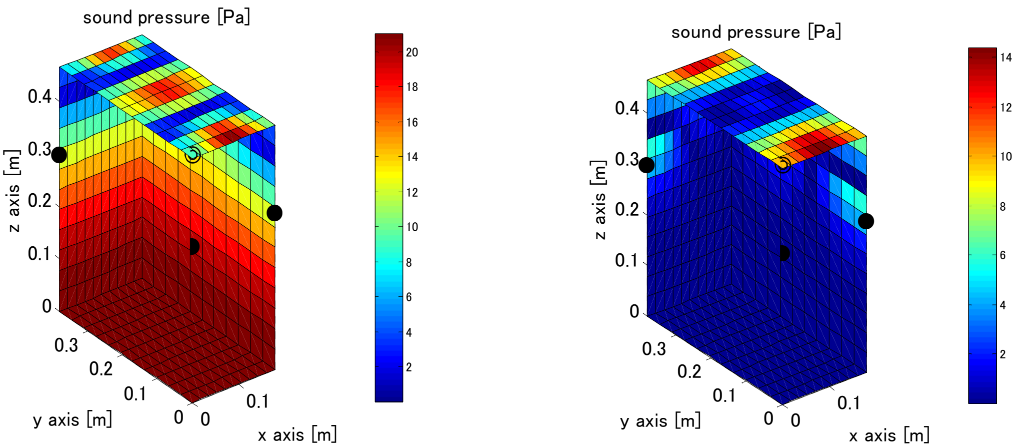

Figure 5 – Sound pressure distribution at 180 [Hz]

(a) Before control (b) After control

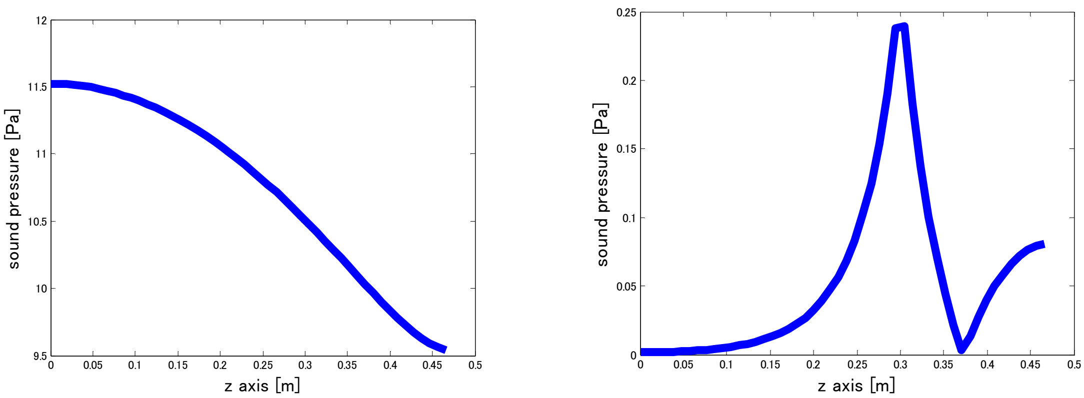

Figure 6 – Sound pressure distribution along the z axis at $(x, y) = (L_x, 0)$ at 77 [Hz]

As can be seen from Figs. 4 and 5, it is confirmed that a quiet space is generated at both frequencies by confining the waves belonging to the target mode groups with the control sources placed in the control plane at $z_c = 0.3$. Focusing on Fig. 4 (a), the space is confirmed to have high sound pressure overall when no control is applied. Furthermore, the sound pressure in the vicinity of the plate is relatively low since the characteristics of the plate as a boundary condition are similar to those of the open-end compared to a rigid wall. In contrast, when the control is applied, the sound pressure inside the cavity is suppressed overall. Although relatively high sound pressure is observed at the location of the plate, it shows that the disturbance energy is confined. In addition, the relatively high sound pressure is confirmed near the control sound sources, and this is considered to be the effects of the non-target mode groups. This result is verified by checking the sound pressure distribution in the z-direction.

3.3 Sound Pressure Distribution in the z-Direction

In this subsection, the sound pressure distribution along the z direction obtained in the previous subsection is focused. When the frequency is 77 [Hz], the sound pressure distribution in the z direction before and after control is shown in Fig. 6. If the control is applied, relatively high sound pressure is observed at $z_c = 0.3$[m] where the control plane is placed. The sound pressure becomes zero at Z = 0.37 [m], and it indicates that this point is a node, that is, there exists the positive and negative traveling waves in the uncontrolled region. In addition, the sound pressure distribution is in the form of a real exponential function in the controlled region. This indicates that the cut-on frequencies of the non-target mode groups are higher than 77 [Hz].

4. CONCLUSIONS

In this paper, we began with the introduction of a transfer matrix method which can treat the wave dynamics in the cavity. This is followed by modeling of the coupled system using fusion of modal coupling and a transfer matrix method In addition, we derived the control law that cancels progressive waves transmitting the control plane consisting of multiple control sources. In numerical simulation, we have demonstrated the validity of the proposed method by analyzing the sound pressure distribution and frequency characteristics. As a result, it was clarified that a quiet space is generated in the coupled system by entrapping the disturbance energy belonging to the (0, 0), (0, 1) and (1, 0) mode groups near the flexible plate.

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