Vibration Control of a Plate Structure by an Active Control Device for Reducing Noise*

Kazuto SETO** and Katsumi SAWATARI***

This paper proposes a method of noise reduction by controlling the vibration of a wall between two rooms by an active control device. This is done by using an additional sensor to measure the wall vibration and feed it back to the controller. The board speaker is used primarily to generate sound by exciting the board. However, in this study, it is used as an active vibration control device. The performance of the device is tested on a panel made from a veneer board $1200 \times 900 \times 5$ mm in size which is exposed to sound vibration from the other speaker. The evaluation of this method is performed experimentally by comparison of the dynamic deflection of the panel with and without the device. The experimental results show a significant reduction level to 1/10 of the panel dynamic deflection at frequencies below 100 Hz. The effectiveness of the method is confirmed theoretically by simulation studies. Also, the stability of the control system using the active control device is examined through the root locus method.

Key Words: Vibration control, Noise reduction, Active device, Panel, Board speaker

1. Introduction

Noise reduction between two rooms is obtained by using a noise-reducing wall between a noise source room and the other room to reduce sound noise by the desired amount. A typical method of noise reduction in the past was to use a heavy-weight wall such as a loaded wall and a masonry block⁽¹⁾. However, since the sound noise is transmitted by exciting the wall common to the two rooms, the active method of controlling the vibration of the wall is to use an active vibration controller.

This paper proposes a method of noise reduction by way of controlling the vibration of the wall between two rooms by an active vibration controller. This is done by using an additional sensor to measure

the wall vibration and feed it back to the controller. The board speaker is developed primarily to generate sound by exciting the board; however, in this study it is used as the actuator in the controller.

The performance of the controller is tested on a veneer board $1200 \times 900 \times 5$ mm in size which is exposed to sound vibration by the other speaker. The evaluation of this method is performed experimentally by comparing the dynamic deflection of the board with and without the controller. The experimental results show a significant reduction level to 1/10 of the board dynamic deflection in the frequency region from 10 to 100 Hz. The effectiveness of the method is confirmed theoretically by analogue simulation studies. Also, the stability of the controller is examined by using the root locus method.

2. Experiment

2.1 Construction of the active vibration controller

A typical noise control problem is illustrated in Fig. 1. When an acoustical noise is generated in the room to the left, the radiated sound energy builds up a reverberant sound field in the source room which, in

* Received 7th March, 1990

** Dept. of Mechanical Engineering, National Defense Academy 1-10-20 Hashirimizu, Yokosuka, Kanagawa, 239, Japan

*** Dept. of Mechanical Engineering, National Defense Academy 1-10-20 Hashirimizu, Yokosuka, Kanagawa, 239, Japan

turn, excites the wall. The vibrations in the wall common to the two rooms radiate sound directly into the room to the right to reduce the radiated sound.

To reduce the noise level transmitted to the room to the right, the vibration amplitude of the wall must be reduced. As a result, the noise reduction problem is replaced with the problem of how to suppress the vibration amplitude of the wall by the active vibration controller.

Various properties required in the active vibration controller for reducing noise are to be incorporated optionally into the wall to generate enough power for driving of the wall, and of small and light-weight body. A board speaker is suitable as the active device of the active vibration controller since it is a kind of electromagnetic exciter devised to generate sound by exciting the board. The schematic view of the board speaker used in this study is shown in Fig. 2. The board speaker is of 85 mm diameter and 33 mm thickness on the outside and consists of a ring-type permanent magnet that is 0.78 kg in mass, a supporting spring with 34 000 N/m in spring constant and 34 Ns/m in damping constant, and a voice coil with 10 W in power capacity. The board speaker actuates the

Fig. 1 Sound noise transmitted from source room to receiver room through the wall

Fig. 2 Schematic view of the board speaker

wall using the reactive inertia force of the ring-type magnet; therefore, it is the same kind of actuator principally used in active vibration control for tall buildings in the field of civil engineering.

The concept of the experimental setup constructed for examining the effect of vibration reduction by the active vibration controller is shown in Fig. 3. The active vibration controller is composed of the board speaker, its driving power amplifier with adjustable gain, and a sensor to measure the vibration of the wall. A panel selected as the control object in place of the wall is made of a veneer board which is $1\,200 \times 900 \times 5$ mm in size. The board speaker is mounted at the center of the panel. The sensor, also, is set near the center of the panel. The voltage corresponding to the vibration signal which appears at the sensor is amplified and then applied to cause the board speaker to drive in such a direction that the voltage is eventually reduced to zero. For excitation of the panel by an airborne sound wave, a driving speaker is set against the panel.

2.2 Experimental results

2.2.1 Vibration characteristics of the panel

Although the effect of noise reduction should be evaluated by the sound-insulating property of the panel, this paper adopts a method of evaluating the degree of the vibration amplitudes of the panel with and without the active vibration controller under the constant experimental condition. Since noise reduction depends on the vibration amplitude, an input voltage to drive the driving speaker keeps constant during the experiment, and the resultant vibration amplitude and phase angle of the panel are measured by a transfer function analyzer.

Figure 4 indicates frequency responses where a dotted line and solid line show the responses of the panel with and without an uncontrolled board speaker. It is found that there are many resonance

Fig. 3 Experimental setup constructed for examining the effect of vibration reduction by the active vibration controller

peaks and antiresonance peaks within 100 Hz (i.e., a fundamental resonance peak at 11 Hz, a small resonance peak at 30 Hz, a large antiresonant peak at 40 Hz, and three resonance peaks in a row between 50 and 100 Hz), and that the fundamental resonance peak moves toward lower frequencies while three resonance peaks move toward higher frequency slightly when the uncontrolled board speaker is mounted. This figure also shows that the dynamics of the panel are scarcely influenced by the dynamics of the board speaker regardless of the natural frequency of the board speaker located at 33 Hz and, for preference, it is influenced by a mass effect of the board speaker.

In order to effectively use the active vibration controller, it is necessary to know the vibration mode shapes at resonance peaks. Figure 5 indicates the mode shapes of three resonance peaks appearing from

Fig. 4 Frequencies of the panel with and without the uncontrolled board speaker

Fig. 5 Vibration mode shapes of the panel

50 to 100 Hz, where solid lines indicated in the inside of the panel represent the vibration nodal line. The panel, while vibrating at 55 Hz, is subdivided into three parts by two vertical nodal lines, as shown in Fig. 5(a). This type of mode shape is called the (3,1) mode. Also, the mode shapes shown in Figs. 5(b) and 5(c) are called the (1,3) and (3,3) modes, respectively. Thus, the mode shape at the fundamental resonance peak is the (1,1) mode.

The mode shapes that involve even-numbered modes such as (2,1), (1,2), and (2,2) take the nodal line on the center of the panel; resonance peaks about these modes are not apparent in the frequency response shown in Fig. 4. This means that the even-numbered modes are uncontrolled as long as the board speaker is mounted at the center of the panel. In support of the concept that even-numbered modes are of little concern in sound noise generation, this paper proposes that odd-numbered modes are merely controlled by the board speaker set at the center of the panel.

2.2.2 Effectiveness of the active vibration controller Comparison of the frequency response of the panel with and without the active vibration controller is shown in Fig. 6. In the case of the frequency response with the controlled board speaker, feedback gains are changed to 5, 10, and 20 by the gain-adjustable amplifier. The dotted, chained, and solid lines indicate these frequency responses at 5, 10, and 20 in gain, respectively. Referring to Fig. 6, it is found that as the gain increases, the fundamental resonance frequency moves toward a lower frequency, and the three resonance frequencies in a row move toward a higher frequency, bordering on the antiresonance frequency at 40 Hz. Then, vibration amplitudes of the panel in the region of 10–100 Hz are gradually reduced as the gain increases. However, no effect is recognized above 100 Hz.

Fig. 6 Comparison of the frequency response of the panel with the active vibration controller when the feedback gain is changed

The frequency response, indicated by the solid line of the panel with the active vibration controller driven by full power, is shown in Fig.7 in comparison with the situation without the board speaker. It is found that the vibration amplitude of the panel in the region of 10-100 Hz is remarkably reduced below 1/10, when the active vibration controller is fully activated. Conversely, the dynamic stiffness of the panel is actively improved by ten times or more. This indicates that effective noise reduction can be realized by using such an active method and is not dependent on a heavy-weight wall.

The tendency of frequency response shown in Fig. 6 indicates that the more the power capacity of the board speaker is increased, the more the dynamic stiffness of the panel is improved.

3. Theoretical Examination

3.1 Construction of a simulation model

In order to examine the stability of the active vibration controller and the general applicability of the method proposed above, a theoretical consideration is performed using the simulation model.

The frequency response shown in Fig. 4 is characterized by two odd-numbered resonance peaks, which lie at 11 Hz and concentrate at about 70 Hz, and the antiresonance peak lying at 40 Hz. Therefore, dynamic characteristics of the panel are approximated in Fig. 8, where ω_a and ζ_a denote the natural frequency and damping factor at the first vibration mode; ω_b and ζ_b those at the second vibration mode; and ω_r and ζ_r those at the antiresonance peak. The above-mentioned frequency characteristics are realized by a two-degree-of-freedom model shown on the left side of Fig. 9, which shows the board speaker model on the right side. In this figure, m_1 , m_2 , c_1 , c_2 , k_1 , and k_2 are defined as masses, damping coefficients, and spring constants of the 1st and 3rd vibration modes,

Fig. 7 Comparison of the frequency responses of the panel with and without the active vibration controller driven by full power

respectively. Also, m_0 , c_0 , and k_0 are defined as mass, damping coefficient, and spring constant of the board speaker, respectively.

When displacement variables x_1 , x_2 , and x_0 at each mass point are defined, the following equations are introduced based on Fig. 9.

$$m_1 \ddot{x}_1 = f_1 + f_2 - (f_d + f_0 + f_{in}) \quad (1)$$

$$m_2 \ddot{x}_2 = -f_2 \quad (2)$$

$$m_0 \ddot{x}_0 = f_0 + f_{in} \quad (3)$$

$$f_1 = -k_1 x_1 - c_1 \dot{x}_1 \quad (4)$$

$$f_2 = -k_2(x_1 - x_2) - c_2(\dot{x}_1 - \dot{x}_2) \quad (5)$$

$$f_0 = k_0(x_1 - x_0) + c_0(\dot{x}_1 - \dot{x}_0) \quad (6)$$

$$f_{in} = Z \dot{x}_1 \quad (7)$$

where f_d denotes a force excited by the sound noise, f_{in} a control force by the board speaker, and Z a transfer function of the active vibration controller. After Laplace transforming Eqs.(1)~(7), these equations are represented by the signal flow diagram indicated in Fig. 10. By defining the following transfer function which represents the dynamics of the panel, Fig. 10 is reduced in Fig. 11.

$$G_1(S) = \frac{m_2 S^2 + c_2 S + k_2 / m_1 m_2 S^4 + (m_1 c_2 + m_2 c_1 + m_2 c_2) S^3}{(m_1 k_2 + m_2 k_1 + m_2 k_2 + c_1 c_2) S^2 + (k_1 c_2 + c_1 k_2) S + k_1 k_2} \quad (8)$$

Fig. 8 Frequency response of the panel approximated in a two-degree-of-freedom system

Fig. 9 Simulation model

On the other side, the transfer function of the panel approximated in Fig. 8 is expressed as:

$$G_1(S) = \left\{ \frac{1}{k_1} \left(\frac{1}{\omega_\gamma^2} S^2 + \frac{2\zeta_\gamma}{\omega_\gamma} S + 1 \right) \right. \\ \left. / \left(\frac{1}{\omega_\alpha^2} S^2 + \frac{2\zeta_\alpha}{\omega_\alpha} S + 1 \right) \left(\frac{1}{\omega_\beta^2} S^2 + \frac{2\zeta_\beta}{\omega_\beta} S + 1 \right) \right\} \quad (9)$$

where spring constant k_1 is obtained by measurement, and the other parameters are listed in Table 1 based on Fig. 8. By comparing the coefficients of polynomial between Eq. (8) and Eq. (9), we obtain:

$$\frac{1}{\omega_\alpha^2 \omega_\beta^2} = \frac{m_1 m_2}{k_1 k_2} \quad (10)$$

$$\frac{1}{\omega_\alpha^2} + \frac{1}{\omega_\beta^2} = \left(\frac{m_1}{k_1} + \frac{m_2}{k_2} + \frac{m_2}{k_1} \right) \quad (11)$$

$$\frac{1}{\omega_\gamma^2} = \frac{m_2}{k_2} \quad (12)$$

unknown parameters m_1 , m_2 , and k_2 are estimated as follows:

$$m_1 = k_1 \frac{\omega_\gamma^2}{\omega_\alpha^2 \omega_\beta^2} \quad (13)$$

$$m_2 = k_1 \left(\frac{1}{\omega_\alpha^2} + \frac{1}{\omega_\beta^2} - \frac{1}{\omega_\gamma^2} - \frac{\omega_\gamma^2}{\omega_\alpha^2 \omega_\beta^2} \right) \quad (14)$$

$$k_2 = m_2 \omega_\gamma^2 \quad (15)$$

Table 1 List of dynamic characteristics of the panel

ω_α	$6.91 \times 10^3 \text{ rad/s}$	ζ_α	0.05
ω_β	$4.21 \times 10^3 \text{ rad/s}$	ζ_β	0.05
ω_γ	$2.58 \times 10^3 \text{ rad/s}$	ζ_γ	0.01
k_1	$8.50 \times 10^3 \text{ N/m}$

Fig. 10 Signal flow graph representation of the simulation model

Fig. 11 Reduction of the signal flow graph shown in Fig. 10

Also, the unknown parameters c_1 and c_2 are estimated through the above-mentioned procedure as follows:

$$c_1 = k_1 \left[\frac{(2\zeta_\alpha/\omega_\alpha)(1/\omega_\alpha^2 - 1/\omega_\gamma^2) + (2\zeta_\beta/\omega_\beta)(1/\omega_\beta^2 - 1/\omega_\gamma^2)}{1/\omega_\alpha^2 + 1/\omega_\beta^2 - 2/\omega_\gamma^2} \right] \quad (16)$$

$$c_2 = k_2 \left[\frac{2\zeta_\beta/(\omega_\alpha^2 \omega_\beta) + 2\zeta_\alpha/(\omega_\beta^2 \omega_\alpha) - (2\zeta_\alpha/\omega_\alpha + 2\zeta_\beta/\omega_\beta)/\omega_\gamma^2}{1/\omega_\alpha^2 + 1/\omega_\beta^2 - 2/\omega_\gamma^2} \right] \quad (17)$$

Table 2 shows dimensions of the panel estimated by using the above equations.

3.2 Frequency response analysis

To analyze the frequency response of the model, an analogue computer was used, since the signal flow graph is applied for simulation purposes. Figure 12 shows the frequency response of the panel without the board speaker. Comparing Fig. 12 with Fig. 4, it is found that the simulation model expresses the dynamic characteristics of the panel extremely well. Now, the effectiveness of the active vibration controller must be examined using the simulation model shown in Fig. 10.

In comparison to Fig. 8, the frequency response of the panel is computed as shown in Fig. 13 when the gain increases from $Z=0$ to $Z=20$. Also, the theoretical results agree with the experimental results in that the more the gain increases, the more the resonance frequency decreases, and the more the resonance frequency increases to the borderline on the antiresonance, and in that the resultant dynamic stiffness is improved by ten times or more at below 100 Hz.

Table 2 Dimensions of the panel

k_1	$8.50 \times 10^3 \text{ N/m}$	k_2	$6.60 \times 10^4 \text{ N/m}$
m_1	0.67 kg	m_2	1.00 kg
c_1	$14 \text{ N}\cdot\text{s/m}$	c_2	$5 \text{ N}\cdot\text{s/m}$

Fig. 12 Frequency response of the panel only calculated by analogue computer

Fig. 13 Computed frequency responses of the panel with the active vibration controller as the gain increases

Fig. 14 Block diagram representation

3.3 Consideration using root locus technique

The application of the active vibration controller for noise reduction requires an investigation of the control stability due to varying of the feedback loop gain. To examine the control stability when the gain varies, the root locus technique is used and then the following procedure is required.

After a reduction of the signal flow graph⁽²⁾ shown in Fig. 11 a block diagram representation of the panel with the active vibration controller is obtained in Fig. 14. Herein, $G_2(S)$ represents a transfer function of the panel with the uncontrolled board speaker as follows :

$$G_2(S) = \frac{(m_2 S^2 + c_2 S + k_2)(m_0 S^2 + c_0 S + k_0)}{A_6 S^6 + A_5 S^5 + A_4 S^4 + A_3 S^3 + A_2 S^2 + A_1 S + A_0} \quad (18)$$

where

$$\begin{aligned} A_6 &= m_0 m_1 m_2 \\ A_5 &= m_0(m_1 c_2 + m_2 c_1 + m_2 c_2) + c_0(m_1 m_2 + m_0 m_2) \\ A_4 &= m_0(m_1 k_2 + m_2 k_1 + m_2 k_2 + c_1 c_2) \\ &\quad + c_0(m_1 c_2 + m_2 c_1 + m_2 c_2 + m_0 c_2) \\ &\quad + k_0(m_1 m_2 + m_0 m_2) \\ A_3 &= m_0(k_1 c_2 + k_2 c_1 + k_2 c_0 + k_0 c_2) \\ &\quad + c_0(m_1 k_2 + m_2 k_1 + m_2 k_2 + c_1 c_2) \\ &\quad + k_0(m_1 c_2 + m_2 c_1 + m_2 c_2) \\ A_2 &= m_0(k_1 k_2 + k_2 k_0) + c_0(k_1 c_2 + k_2 c_1) \\ &\quad + k_0(m_1 k_2 + m_2 k_1 + m_2 k_2 + c_1 c_2) \\ A_1 &= c_0 k_1 k_2 + k_0(k_1 c_2 + k_2 c_1), \quad A_0 = k_0 k_1 k_2 \end{aligned}$$

Therefore, an open-loop transfer function in this closed-loop system is expressed as follows :

Fig. 15 Root locus

$$G(S) = Z \frac{m_0 S^2 (m_2 S^2 + c_2 S + k_2)}{A_6 S^6 + A_5 S^5 + A_4 S^4 + A_3 S^3 + A_2 S^2 + A_1 S + A_0} \quad (19)$$

where if Z is repressed by K , K is the feedback loop gain to vary.

In this equation, there are six poles and four zeros with two zeros at the origin. The locations in the poles and zeros at the S -plane are indicated in Fig. 15 with the marks "x" for poles and "o" for zeros. When K varies, the root loci are sketched in Fig. 15 based on the pole-zero configuration. Figure 15 shows that there are three conjugate root loci, i.e., a closing conjugate root locus to the origin at the S -plane, a middle conjugate locus, and a locus heading away from the origin (the "going-away" locus). The closing conjugate root locus moves toward the origin parallel to the imaginary axis, while the going-away root locus moves toward infinity, parallel to the imaginary axis. Then, the middle root locus approaches the near zero. The conjugate root on the closing root locus corresponds to the fundamental resonance frequency, and the going-away root locus corresponds to the higher-order resonance frequency. Therefore, the root loci clearly explains how the response peaks move toward lower or higher frequencies as the gain

increases, as shown in Figs. 7 and 13.

The most important point indicated in Fig. 15 is that the root locus still keeps to the left side in the S -plane regardless of the gain K , and that the panel with the active vibration controller is stable independent of the gain K . Therefore, if a larger capacity in power of the board speaker is used, the dynamic stiffness of the panel will be improved further.

4. Conclusions

In order to actively reduce sound noise, the active vibration controller using the board speaker was constructed. It was found that the vibration amplitude of the panel in the region of 10-100 Hz was remarkably reduced below 1/10, when the active vibration controller was fully activated. This indicates that

effective noise reduction can be realized using the active method not dependent on a heavy-weight wall. It was shown by means of the root locus technique that the panel with the active vibration controller is stable, independent of the variation of feedback gain. This means that the more the power capacity of the board speaker is increased, the more the dynamic stiffness of the panel is improved.

References

- (1) Beranek, L. L., Noise and Vibration Control, (1971) McGraw-Hill.
- (2) Tominari, N. and Seto, K. Analysis and Design of Servo Control Systems, (1979) Corona Publishing Co..