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ACOUSTICAL LETTER

Improvement of sound transmission loss of double-layer wall by using vibration absorber

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1. Introduction

A resonance problem exists in a double-layer wall (DLW). When noise is incident to a DLW, the sound energy is completely transmitted through the DLW because of the mass-air-mass resonance. There have already been some researches [1–7] on the use of an absorbent material, resonator or other structures to solve this resonance problem. According to the literatures, applying an absorbent material in a DLW can improve the sound insulation performance at mid- and high-frequency bands [1,2]. Another treatment, the resonator, was reasonably effective at specific frequency bands [6,7]. The application of a vibration absorber in a double-leaf partition was focused on with the aim of improving the sound insulation at the resonant frequency, and the optimal position of the absorber was also investigated [4,5]. In this study, a simple structure vibration absorber composed of a vibration mass layer and a damping material layer was investigated. The vibration absorber obtains the improvements not only at the resonant frequency band but at other frequency bands. Also, from the viewpoint of workability, time is required to analyze the vibration modes of each wall body to decide the optimal positions of vibration absorbers. Therefore, the relationship between the sound insulation performance and the number of vibration absorbers with regular and random distributions was investigated.

2. Analytical model of the sound insulation performance of double-layer wall with vibration absorber

Considering the three-mass MKC system (M: mass, K: elastic constant, C: damping) shown in Fig. 1, the theoretical resonant frequency ω_{rf} is well known to be $\sqrt{k_1(m_1+m_2)/m_1m_2}$ as m_3 is equal to zero.

In the steady state, the equations of motion of the three masses can be written as follows.

$$-\omega^2 m_1 X_1 + k_{\mathrm{a1}}(X_1 - X_2) + jc_{\mathrm{a1}}\omega(X_1 - X_2) = P_{\mathrm{i}} \quad (1)$$

$$-\omega^2 m_2 X_2 + k_{\mathrm{a1}}(X_2 - X_1) + jc_{\mathrm{a1}}\omega(X_2 - X_1) + k_{\mathrm{u}}(X_2 - X_3) + jc_{\mathrm{u}}\omega(X_2 - X_3) = 0 \quad (2)$$

$$-\omega^2 m_3 X_3 + k_{\mathrm{u}}(X_3 - X_2) + jc_{\mathrm{u}}\omega(X_3 - X_2) = 0 \quad (3)$$

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We define ω_{VA} as the natural frequency of the vibration absorber which is equal to $\sqrt{k_{\mathrm{u}}/m_3}$. The vibration energy of m_2 is completely transmitted to m_3 when ω_{VA} and ω_{rf} are equal. The theoretical sound insulation performance of a DLW without and with a vibration absorber obtained using Eqs. (1)–(3) is shown in Fig. 2. In this figure, the mass-air-mass resonance of the DLW is designed to be at the 630 Hz 1/3 octave band; thus, the sound reduction index drops to zero. After applying the vibration absorber, there is a clear improvement at the resonant frequency band.

3. Experiment instruments

3.1. 1/5 Scale model

The sound insulation performance in this study was evaluated by measuring the sound intensity. We used a 1/5 scale model reverberation chamber made of 3-cm-thick acrylic with internal dimensions of 137 cm (W), 87 cm (D), and 156 cm (H). The wall body of the DLW is made of plywood with area density 2.2 kg/m^2 and its air gap is 8 mm; thus, the resonant frequency is 632 Hz. The size of the DLW is 84.5 cm (W) by 62.5 cm (H). The layout of the scale model is shown in Fig. 3.

3.2. Design of vibration absorber

As previously mentioned, the natural frequency of the vibration absorber should be the same as the resonant frequency of the DLW, thus we assembled a vibration absorber using a 2-mm-thick vibration mass with area density 1.72 kg/m^2 and 3-mm-thick urethane foam with area elastic constant 2.73×10^7 N/m^3 and damping coefficient 218 N-s/m^3.

4. Experimental conditions and results

4.1. Cases of applying vibration absorbers

To investigate the effectiveness of the vibration absorbers in the DLW, six conditions were prepared for evaluation by measuring the sound reduction index.

The cross section of Cases 1–4 are shown in Fig. 4(a). Case 1 was a simple model of a DLW without any extra materials or structures. Cases 2 and 3 were variations of Case 1 used to determine the effect on the sound insulation performance of a single material. In Case 4, 48 vibration absorbers were applied. Also, the distribution patterns of Cases 1–4 were the same as shown in Fig. 4(b).

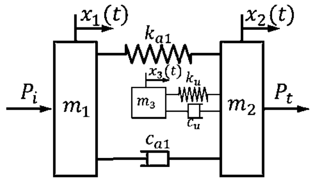


Fig. 1 Three-mass MKC system.

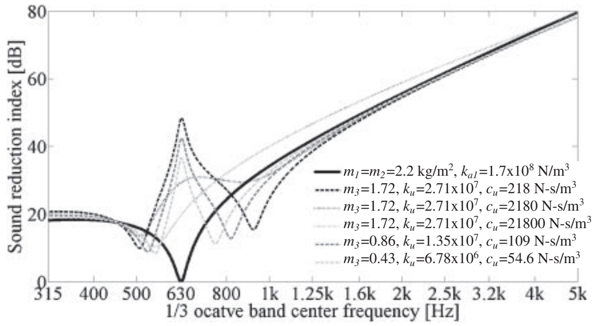


Fig. 2 Sound reduction index with various parameters.

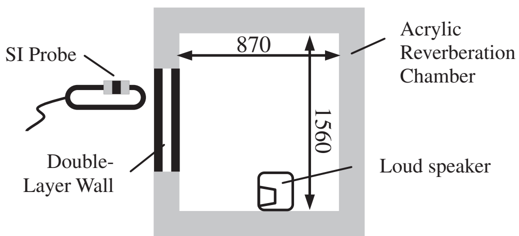


Fig. 3 Layout of 1/5 scale model.

Moreover, we prepared two other patterns with 24 vibration absorbers (Case 5) and 12 vibration absorbers (Case 6) as shown in Fig. 4(b). These cases were used to verify the effect on the sound insulation performance of different numbers of vibration absorbers.

4.2. Measurement results

The results for Cases 1–4 were shown in Fig. 5(a). Case 1 corresponded to the sound insulation performance of a simple DLW in which the resonance occurs at the 630 Hz 1/3 octave band as designed. Case 2 showed a great improvement at high-frequency bands because of the sound absorbance, but resonance still existed. In Case 3, because of the increase in the mass of the wall body, the sound insulation performance was improved at each frequency band. However, there was still a dip at the 630 Hz 1/3 octave band. Case 4 showed about 6.8 dB improvement at the 630 Hz 1/3 octave band because of the vibration absorbers. Also the resonance appeared at the 500 Hz band as theoretically predicted.

Figure 5(b) shows the sound insulation performance of the DLW with different numbers of vibration absorbers. Comparing the results for Cases 4 and 5, the improvement at the 630 Hz band was almost the same. At frequency bands other than the resonance frequency band, the sound insulation performance of 48 vibration absorbers was better than that of 24 absorbers at the 800 Hz band, but worse at the 500 Hz band, and the improvements at the other frequency bands were not different significantly. Case 5 showed a smoother curve than Case 4. On the Basis of these results, we noticed that there was a limit to the improvement at the resonant frequency and that increasing the number of vibration absorbers caused a dip at frequency bands other than the resonant frequency band. Upon reducing the number of vibration absorbers as in Case 6, the sound insulation performance at the 630 Hz band was close to that in Case 1 owing to the reduced number of vibration absorbers.

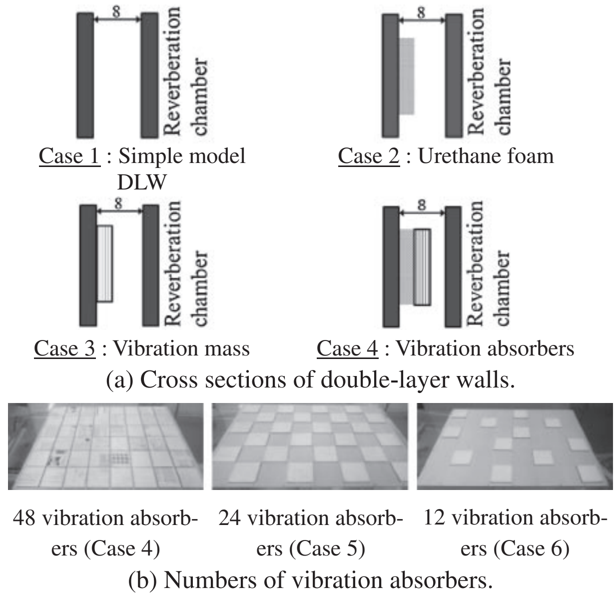
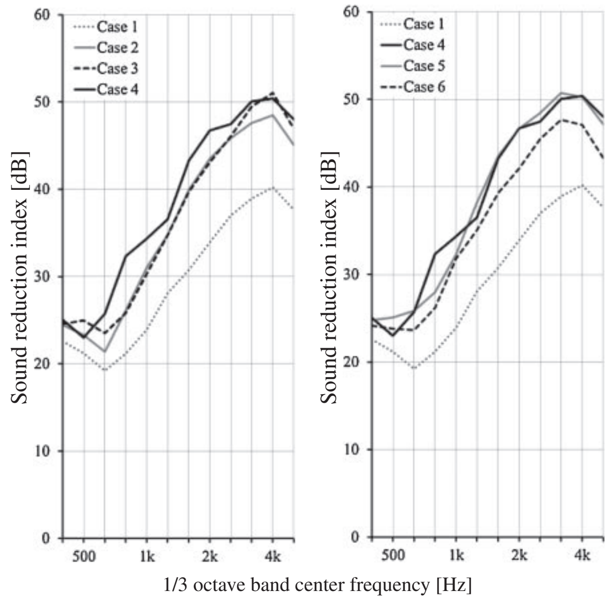


Fig. 4 Cases used for measurement.



(a) Results for different materials. (b) Results for different numbers of vibration absorbers.

Fig. 5 Sound reduction index for different cases.

5. Conclusions

In this study, the effects of applying vibration absorbers to a DLW were investigated theoretically and experimentally. As a result of the investigation, it was confirmed that a vibration absorber whose natural frequency was tuned to be the same as that of the mass-air-mass resonance of the DLW reduced the deterioration of the sound insulation performance at low frequencies. Also, it was found that the effectiveness depended on the number of absorbers set inside the DLW.

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