Improvement of Sound Insulation Performance of Multi-layer Structures in Buildings

RuiLin Mu

Summary

A fundamental measure for improving the sound transmission performance of a homogeneous single partition is to increase the mass and thickness; however, this technique has limitations and is inconvenient. For these reasons, sound transmission through a double-plate system with various compositions has been studied. Modern mass-air-mass multilayer structures are currently used for sound insulation in many fields as a mature technology because they are lightweight and have effective property to prevent dew condensation. However, this kind of structure has issue with sound insulation, such as sound insulation deficit at low and mid-frequencies due to the effect of mass-air-mass resonance.

In this study, three approaches are proposed to improve the sound insulation performance of multi-layer structures in buildings.

One is to investigate improvements to the sound insulation performance of multi-layer structures with a microperforated panel (MPP). MPP can absorb well over a wide frequency range regarding as a next absorption material. MPP is a panel with sub-millimeter holes and an adequate perforation ratio. Although MPP have been investigated over the last several decades, almost all studies have been conducted in terms of sound absorption. Herein the sound transmission loss of multi-layer structures with flexible MPP is theoretically investigated. In this investigation, the calculation is based on the wave equation and the equation of panel vibration including the effect of perforation of the panel. Experiments are conducted using an acoustic tube to validate the calculated results and the reverberation chamber method to verify the actual sound insulation characteristics. Both experiments agree well with the theoretically calculated perforation effects. Consequently, MMPs are confirmed to improve the deterioration of sound insulation performance due to mass-air-mass resonance of multi-layer structures.

To obtain better sound insulation performance and achieve the aim of light weight, another method about the application of damping materials is proposed. One of serious problems of multi-layer structures on sound insulation performance is the sound insulation deficit due to mass-air-mass resonance. However, this problem has not been thoroughly studied. It is considered that this problem can be solved by installing the damping connectors at sound wave crest and trough. Therefore, the improvement of sound insulation performance of double-panel structures by using damping materials is investigated. In this study, an analytical model is proposed for the theoretical study of the improvement in the sound insulation performance achieved by equipping such a structure with damping materials. The damping materials are installed as connectors between a double layer structure. The effect of connectors on the sound insulation performance of double structures is investigated

by considering the relation of the location, numbers, and properties of connectors with the vibrational mode of the panel. Our results indicate that the effectof mass-air-mass resonance on the sound insulation performance can be suppressed by appropriately selecting these connector parameters.

The other one is the reduction on floor impact sound of a slab with panel flooring which is composed of a flooring panel and urethane foam matt. In the history of reduction on floor impact sound, urethane foam are commonly used as matt material when placing panel flooring on the floor slab. An urethane foam matt consists of elastic fibres (solid part) and pores (fluid part) which can strongly affect the insulation performance against floor impact sounds. The method of analysis for floor impact sound of a structure, which consists of urethane foam matt with flooring panel directly placed on the floor slab has not been studied. Furthermore, details of the effects of fibres and pores have not been clarified yet. Hence, the effect of urethane foam matt on floor impact sound of a slab with panel flooring is investigated. In this work, an analytical model for evaluating the insulation performance against floor impact sound is proposed. By comparing to the result of experiment, an empirical formula is obtained. This empirical formula is used to investigate the effect of layer material in the parametric study of Chapter. 4. The results show that the constitution of the foam (either open or closed cells of pores) and the thickness and elastic hardness of the matt layer have a strong effect on the insulation performance of the floor.

These three approaches are considered as effective noise control methods to improve sound insulation performance. The studies of MPP (Chapter. 2) and damping material (Chapter. 3) focus on the reduction of resonance effect. They can be used to glass window and all surrounding walls. While the panel flooring with urethane foam matt can be used to floor and surrounding walls for reducing the structure-borne sound such as floor impact sound. By these approaches, the sound insulation performance of all elements of buildings can be improved. It means that a good acoustical environment can be achieved by improving the total sound insulation performance of buildings with these approaches.

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CHAPTER. I

1 Introduction

1.1 Background

As the development of industry, a lot of new equipments, machinery and institution continuously appeared which directly lead to amounts of new noise sources. Due to people living standard enhancement, human beings start to pay attention to the problems of residential environment. Noise is one of these problems, which have evoked the importance of noise control techniques.

In recent years, the quality of a building is dominated by several elements, such as its design, safety performance, and residential amenities. The sound insulation performance is an important criterion for evaluating residential amenities. Typical problems related to the sound insulation of buildings are sound transmission of windows and walls, floor impact sounds, and general structure-borne sounds due to vibrations induced by equipments and machinery. The main factors for the physical manifestations of these problems are believed to be the vibration of building elements and acoustic coupling with the surrounding air. Based on this fundamental principle, it concludes that the problems of noise can be resolved by reducing the vibrations of building elements and surrounding air. It means that the noise control corresponds to the control of vibrations of building elements and surrounding air.

In the 1930s, noise control became an important subject. Ever since then it has been studied both theoretically and experimentally. Since

Reissner[1] proposed the predictive theory of thin and thick homogeneous single plate, sound transmission loss of single leaf plate of different materials have been extensively investigated. London[2] proposed a theory to simplify the calculation of sound transmission loss of various materials in random incidence field by only considering the effects of mass and frequency, namely the mass law of random incidence. Although this theory is easy to evaluate sound insulation performance of architectural materials, calculated result does not have a good agreement with experimental results, especially at mid- and high frequencies. Cremer[3] investigated the reason of this discrepancy, and pointed out that it caused by the effect of coincidence. In addition, many other researchers, for example, Schoch & Feher[4] and Fescbach[5] also gave some important reports of single plate. After then, single leaf partitions has been extensively studied.

From these previous researches, regarding sound transmission problems for a homogeneous single plate, it is found that a basic measure to improve sound insulation performance is to increase the mass and thickness. However, this approach is limited and outdated. Therefore, it leads to the development of more complicated constitutions, such as multi-layer structures and active noise control techniques[6 - 10]. Although active noise control techniques can reduce the noise level, this method has its limitations. For instance, the improvement of sound insulation performance is limited comparing to the physical approach of noise control. Furthermore, the wide application of active noise control techniques leads to consume vast amounts of electricity. It does not match the demand of low-carbon society. For the reasons mentioned above, various physical noise control approaches of more complicated constitutions have been studied to improve the sound insulation performance of buildings. In the physical noise control approaches, the study of multi-layer structures became a main subject. Based on such background, this study focuses on the improvement of sound insulation performance of multi-layer structures.

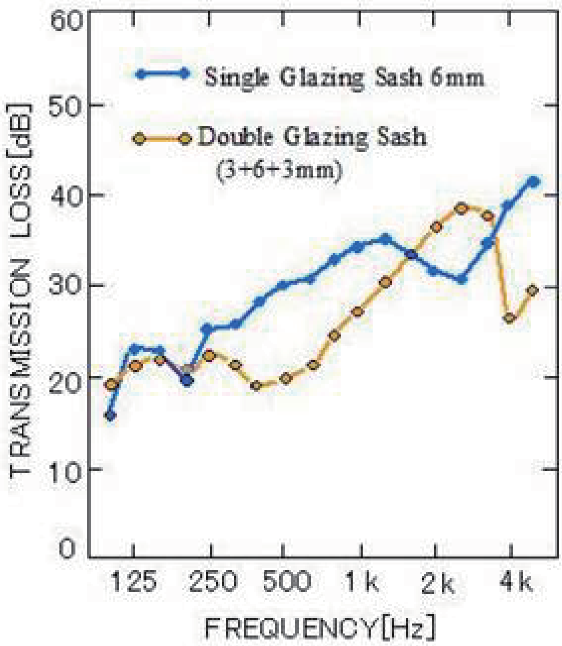

Figure 1.1: Comparison of sound transmission loss double structure and single leaf structure.

Multi-layer structure is the generic name of double layer structure and triple layer structure. The typical structures are double plate and triple plate. Because this kind of structure is developed mainly for the purpose of thermal insulation , sound insulation performance of such a structure has not been studied thoroughly. In the field of multi-layer structures, sound transmission through double plates with air cavity alone is the simplest case. A typical example was presented by London[11]. Nowadays, multi-layer structures have been applied in many fields as a mature technology. However, these structures have issues with sound insulation performance. Especially, sound insulation deficit at low and mid-frequencies due to the effect of mass-air-mass resonance. Figure 1.1 shows the experimental results of single glass and double glass[12].

The blue line is the experimental result of a 6 mm single glazing sash. The yellow line presents the experimental result of a 3 - 6 (thickness of air layer) - 3 mm double glazing sash. Double glazing sash has a significant sound insulation deficit caused by the effect of mass-air-mass resonance which can be seen at around 400 Hz.

To solve these problems and realize high transmission loss, the cavities are often filled with porous materials such as glass wool. However, fibers from porous materials irritate the eyes and skin as well as produce a variety of respiratory ailments. Furthermore, opaque filled material is not appropriate to apply to glass window due to the limitation of transparency. Therefore, inhibiting sound transmission by applying various ideas into multi-layer structures has been extensively studied.

In the present study, three different approaches are proposed to improve the sound insulation performance of multi-layer structures. The first and second approach focus on resolving the sound insulation deficit due to the effect of mass-air-mass resonance which can be applied for glass window and partition walls. The third approach is proposed to reduce the light impact sound of floor. It is considered as an effective resolution for noise insulation problems of floor and partition wall. The details of each study are introduced as follows:

First study is devoted to improvement of the sound insulation characteristics of multi-layer structures by using a microperforated panel (MPP). The effectiveness of this approach is circumstantially investigated in Chapter. 2. MPP appeared in the second half of the 20th century. MPP is a panel with submillimeter holes and an adequate perforation ratio. Figure 1.2 shows a picture of acrylic MPP.

MPPs provide good absorption over a wide frequency range as reported by Maa[13 - 15]. Because MPPs can be fabricated from any sheet materials to meet global environmental demands, they are recognized as next-generation sound absorption materials. Considering the harmlessness to health and diverse applications, such as in a transparent glass window, the sound insulation characteristics of MPPs need to be examined.

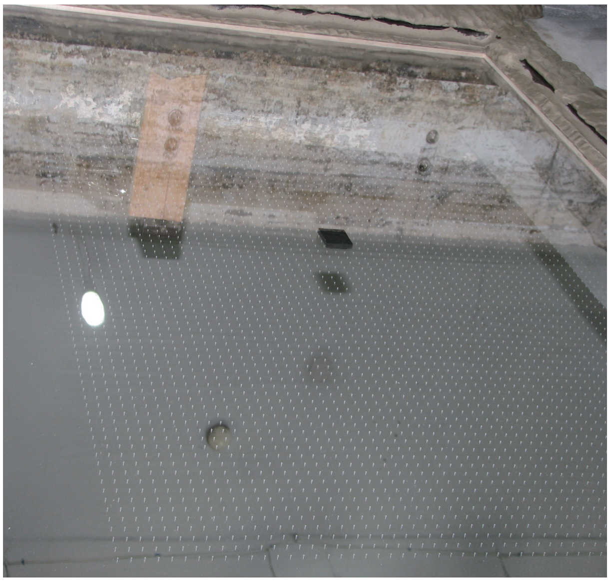

Figure 1.2: The picture of micropeforated panel (MPP).

Investigations of the MPP, which were originally initiated by Maa, have been extensively studied theoretically and experimentally during the past several years. However, almost all recent studies have been conducted in terms of sound absorption[16 - 25]. Some studies on sound insulation performance of multi-layer structures with MPP are also conducted. For example, the effect of MPPs has been discussed using a theoretical model for sound radiation problems of a double-leaf structure excited by a point force[26]. The model assumed the MPP located in the cavity is used as a damping material similar to glass wool inserted into the cavity of a conventional double-leaf structure. However, the function of this type of MPP differs from the model proposed in the present study, which examines sound transmission problems. Moreover, the location of MPP is different from this study. Dupont et al.[27] have investigated the sound insulation performance of a double-leaf structure with an MPP without considering the effect of flexural vibration of MPP. However, MPP used as an absorber layer is installed around the edge of multi-layer structure. In addition, an MPP is used as a sound radiation panel that exhibits a reduction effect to investigate the potential of improving sound insulation performances of multi-layer structures using MPPs[28]. Although reduction of acoustic radiation was studied, MPP is considered as an absorber to resolve the problem of the floor impact sound. We also have previously reported an improved sound insulation performance through the use of an MPP with subdivided air cavities[29]. Therefore, the sound transmission loss of multi-layer structures with flexible MPPs has not been investigated.

In the present study, the application of MPPs to solve sound insulation problems is treated in more fundamental and realistic manner; we investigate the sound insulation performance achieved by perforating conventional window structures (double and triple glass windows) currently available. It means that this study focuses on the sound transmission loss of multi-layer structures with flexible MPP. From the common-sense knowledge of sound insulation techniques, it has been thought that perforating the panel deteriorates the sound insulation performance of the panel structure. The key point of the present study is to propose the " reversal " idea of this common knowledge; developing a new technique for improving the sound insulation performance by perforating the panel.

The second study is the improvement of sound insulation performance of double panel structures by using damping materials. In this study, the damping connectors made of viscoelastic material are installed at mode centers between a double layer structure. An example of damping material is shown in Fig. 1.3.

The damping of acoustic energy can be defined as the process by which materials or structures dissipate acoustic energy and transfer it to heat. These mechanisms have the effect of controlling the amplitude of resonant vibrations and modifying wave attenuation and sound transmission properties.

From the late 20th century, various kinds of damping layers were

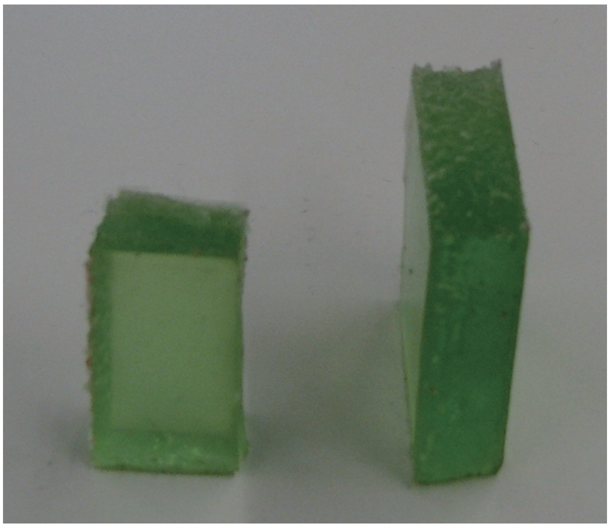

Figure 1.3: The picture of damping material.

extensively used for noise and vibration control of thin walled structures such as automotive body parts, aircraft panels, containers, or casings and considerable research had been devoted[30 - 31]. This approach called passive damping treatment. Compared to the more modern noise control measures – active and semi-active noise control, they are still an attractive alternative or supplement for the reasons of economy and simplicity.

Passive layer damping, usually implemented as constrained layer damping, is the most common form of damping treatment. There are numerous studies on the damping of vibrations in structures via the usage of viscoelastic damping layers such as the works of Nashif et.al.,[32], Sun and Lu[33]. Kerwin is the first researcher, who discussed a three layer beam with a damping layer sandwiched between two face layers [34]. Some authors also proposed to use recycled products to provide alternatives to existing products in a great number of commercial and environmental noise control applications, including building, automotive and business services areas and traffic noise abatement [35 - 38]. However, damping materials are usually installed at whole surface as a layer material in almost all studies. This kind of installation usually involves penalty of weight. Furthermore, opaque filled material is not appropriate to apply to glass window due to the limitation of transparency. In the field of multi-layer structures, sound insulation deficit due to mass-air-mass resonance is the most significant problem. Due to these reasons mentioned above, a new approach by installing damping connectors at sound wave crest and trough is proposed in the present study. From the commonsense of sound insulation technique for multi-layer pane structures, it has been thought that any connector between the panels causes " sound bridge effect ", which deteriorates the sound insulation performance. The key factor of the present study is to propose selection of new materials and how to use the materials of connectors, which improve sound insulation performance of multi-layer structures.

The third study is given to the effect of urethane foam matt on floor impact sound of a slab with panel flooring. Figure 1.4 shows a specimen of urethane foam matt.

Sound transmission problems for double plates with an absorptive layer have also been the subject of a number of papers; the absorptive layer is treated as a medium of wave propagation with resistance dissipation. Kropp and Rebillard[39] examined a sandwich construction composed of double plates with an elastic solid core where the core is modelled to have locally reactive elasticity (Winkler-type material). Ford et al.,[40] developed a more rigorous model of the core from the general elastic theory. A structure of double plates with studs was designed as a lightweight construction material. The studs have an effect of short-circuit transmission as well as cavity boundaries. Studies on this problem have been devoted to the effects of studs with an air layer[41 - 43] and studs with an absorptive layer[44 - 46]. Hongisto[47] reviewed these works to compare prediction models. In the construction treated here, two plates were bonded to the elastic core (urethane foam). In this case, the core has a role of wave transmission as a solid

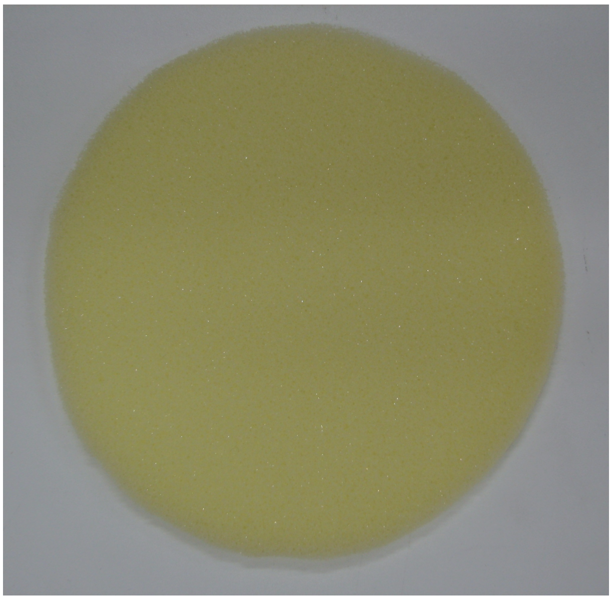

Figure 1.4: The picture of urethane foam matt.

elasticity of the frame as well as a wave propagation medium of pores. The theory of wave propagation in such a porous elastic material was established by Biot [48] and discussed by Allard [49]. Examples of such an application to the core of a double plate system for sound transmission problems have been presented by Bolton et al.[50] and Sgard et al. [51]. In the history of reduction on floor impact sound, urethane foam are commonly used as matt material when placing panel flooring on the floor slab. An urethane foam matt consists of elastic fibres (solid part) and pores (fluid part). Both these two elements could strongly affect the insulation performance against floor impact sounds. In this construction method, the urethane foam matt attached to flooring panel is directly placed on the floor slab. It called direct attachment method. However, the effect of this kind of structure on the reduction of floor impact sound has not been studied. Furthermore, details of the effects of fibres and pores have not been clarified yet, and the effects may change if the material properties and constitution of fibres and pores in the matt are changed. It is considered that two points mentioned above are the novelty of this work.

1.2 Aims of thesis

The aims of this thesis are expressed as follows:

1) Considering both practical use and the demand of light weight, three different approaches are proposed to improve the sound insulation performance of multi-layer structures in buildings. Two approaches (Chapter. 2 and 3) focus on improving the effect of mass-air-mass resonance and the other one is to reduce the floor impact sound (Chapter. 4).

2) Analytical models of the above-mentioned three approaches are developed that will allow the prediction of advantageous effects of each approach.

3) Comparing the calculated results to the measured data, the prediction of the first study (Chapter. 2) is validated and an empirical formula of the third study (Chapter. 4) is obtained. The measurement of the second study (Chapter. 3) will be conducted in the future.

4) With the aid of parametric studies, the effects of some parameters which strongly influence the sound insulation performance of multi-layer structures are investigated.

1.3 Contents and thesis structure

This thesis is divided into five chapters. A short introduction of this thesis is shown in Chapter. 1. The main achievements are concluded in Chapter. 5. Three different approaches which can achieve better sound insulation performance of multi-layer structures are proposed in Chapter. 2 - 4, respectively. In order to provide a general overview of the contents, a short description of the remaining chapters is given below.

In Chapter. 2, the prediction method of sound insulation performance of multi-layer structures with a microperforated panel is proposed. This theory is validated by using the acoustic tube and actual sound insulation performance is confirmed by using a reverberation chamber. Furthermore, the effect of pitch and diameter of the hole are discussed in the parametric study.

In Chapter. 3, an analytical model is proposed for the theoretical study of the improvement on sound insulation performance achieved by equipping such a structure with damping materials. Moreover, the effect of connectors on the sound insulation performance of double structures is investigated by considering the relation of the location, numbers, and properties of connectors with the vibrational mode of the panel.

In Chapter. 4, an analytical model which considers the contribution ratio of pores in the urethane foam matt is used to investigate the effect of a cushion on the sound insulation performance of the floor. To compare the reductions in floor impact sound of experiment and theory, an empirical formula is obtained. In addition, the effects of varying the material properties of urethane foam matt are investigated by this empirical formula in parametric study of Chapter. 4. Finally, in Chapter. 5, the main conclusions drawn from this work and the recommendations for future work are summarized. A simple overview of thesis structure is shown in Figure 1.5.

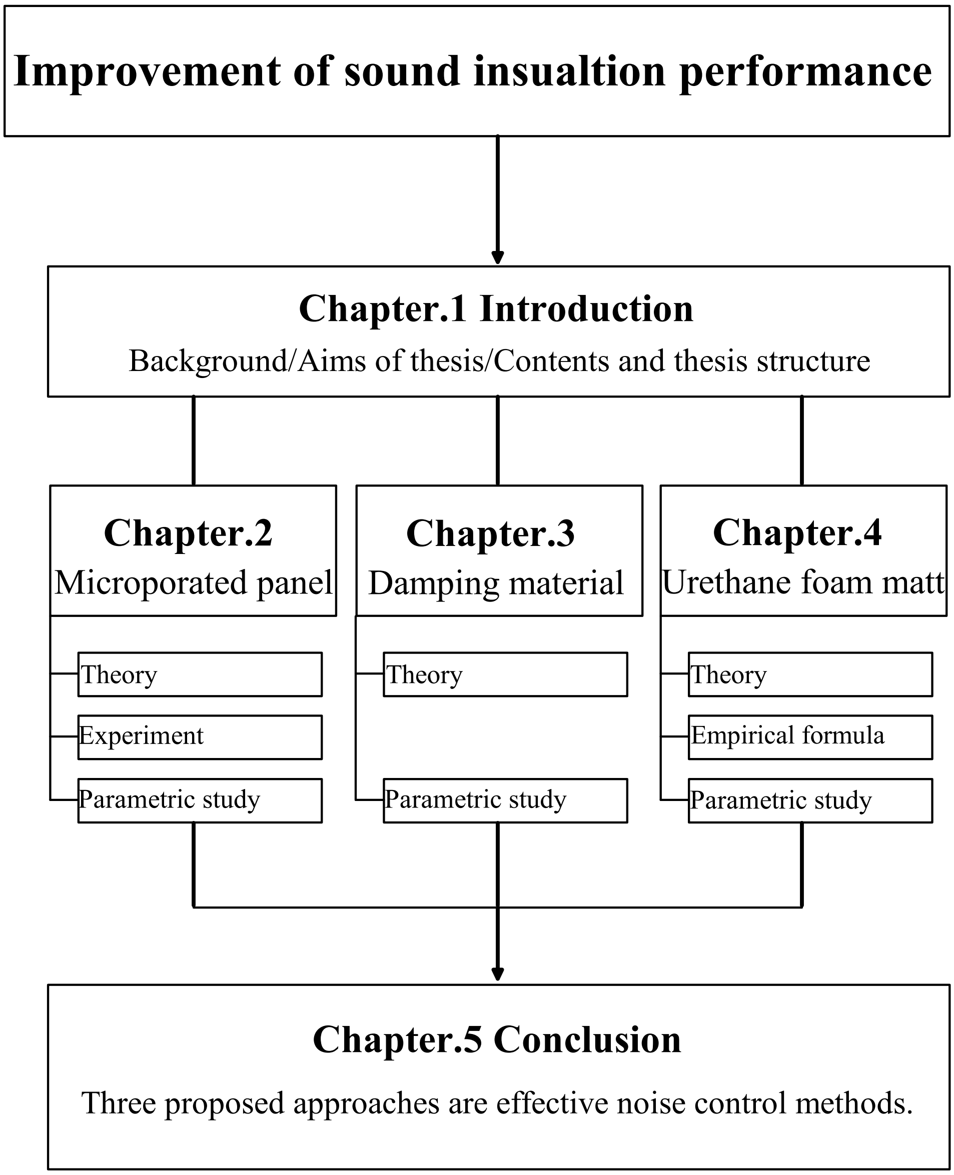

Figure 1.5: The constitution of thesis.

CHAPTER. II

2 Sound insulation characteristics of multi-layer structures with a microperforated panel

2.1 Introduction

Multi-layer structures have been used in many fields as a mature sound insulation technology. However, these structures have issues with sound insulation at low and mid-frequencies due to mass-air-mass resonance. To solve this problem and to realize a high transmission loss, the cavities are often filled with porous materials such as glass wool. However, fibers from porous materials irritate the eyes and skin as well as produce a variety of respiratory ailments. Microperforated panels (MPPs) appeared in the second half of the 20th century. An MPP is a panel with submillimeter holes and an adequate perforation ratio. MPPs provide good absorption over a wide frequency range as reported by Maa [13 - 15]. Because MPPs can be fabricated from any sheet materials to meet global environmental demands, they are recognized as next-generation sound absorption materials. Considering the health risks and diverse applications, such as in a transparent glass window, the sound insulation characteristics of MPPs must be examined.

The investigations of the MPP, which were originally initiated by Maa, have been extended theoretically and experimentally during the past several years. However, almost all recent studies have been conducted in terms of sound absorption[16 - 25]. Some studies on sound insulation performance of multi-layer structure with MPP are also conducted. For example, the effect of MPPs has been discussed using a theoretical model for sound radiation problems of a double-leaf structure excited by a point force[26]. The model assumed the MPP located in the cavity is used as a damping material similar to glass wool inserted into the cavity of a conventional double-leaf structure. However, the function of this type of MPP differs from the model proposed in the present study, which examines sound transmission problems. Moreover, the location of MPP is different from this study. Dupont et al.[27] have investigated the sound insulation performance of a double-leaf structure with an MPP without considering the effect of flexural vibration of MPP. However, MPP used as an absorber layer is installed around the edge of multi-layer structure. In addition, an MPP is used as a sound radiation panel that exhibits a reduction effect to investigate the potential of improving sound insulation performances of multi-layer structures using MPPs[28]. Although reduction of acoustic radiation was studied, MPP is considered as an absorber to resolve the problem of the floor impact sound. We also have previously reported an improved sound insulation performance through the use of an MPP with subdivided air cavities[29]. Therefore, the sound transmission loss of multi-layer structures with flexible MPPs has not been investigated.

In this study, the application of MPPs to solve sound insulation problems is treated in a more fundamental and realistic manner; we investigate the sound insulation performance achieved by perforating conventional window structures (double and triple glass windows) currently available. It means that this study focus on the sound transmission loss of multi-layer structures with flexible MPP. From the common-sense knowledge of sound insulation techniques, it has been thought that perforating the panel deteriorates the sound insulation performance of the panel structure. The key point of the present study is to propose the " reversal " idea of this common knowledge; developing a new technique for improving the sound insulation performance by perforating the panel.

Sound transmission loss is calculated based on the wave equation and the equation for panel vibration. The coupled analysis considers the effects of microperforations at the boundary of the panel and air [52]. Additionally, the average sound transmission loss of each multi-layer structure is determined by considering the directional distribution of the incident energy in a reverberation chamber [53]. To validate the prediction method, an experiment was conducted using an acoustic tube, and the actual sound insulation characteristics are confirmed by the reverberation chamber method. Then the potential of employing MPPs to improve sound insulation performance of multi-layer structures as well as the influence of pitch and hole diameter are discussed.

2.2 Theory

2.2.1 Infinite analytical model

To investigate the ability of MPPs to improve the sound insulation performance of multi-layer structures, an analytical model to calculate sound transmission loss through infinite multi-layer structures with an MPP is introduced. Figures 2.1 and 2.2 show the analytical models for a double panel with an MPP and a single panel with an MPP, respectively. If the MPP is not perforated, then it is regarded as a typical plate. Due to the similarity of the two models, below is theoretical formulation of the double panel with an MPP.

In this model, a plane wave with an incident angle θ is considered as the sound incident upon multi-layer panels that extend infinitely. In each panel with a multi-layer structure and an MPP, both sides are assumed to be filled with the same fluid (air). Additionally, the

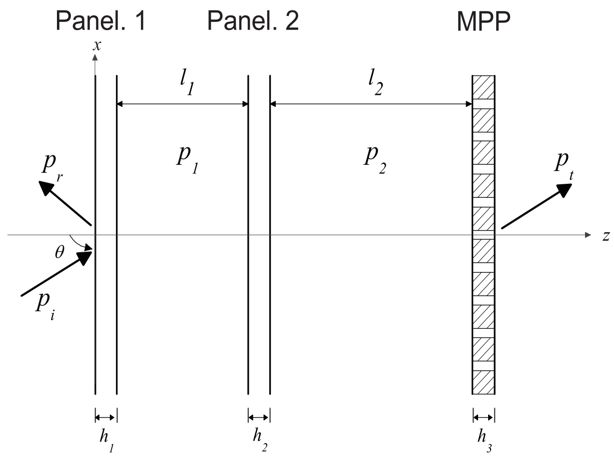

Figure 2.1: Analytical model for double panels with an MPP of infinite extent with an incident plane wave p_i.

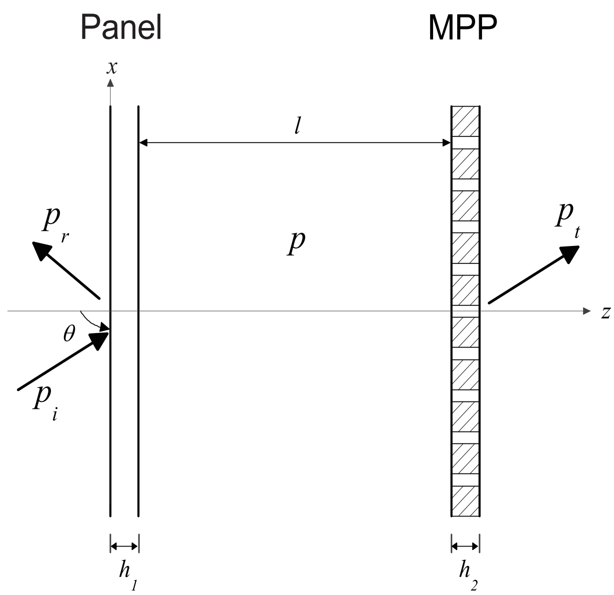

Figure 2.2: Analytical model for a single panel with an MPP of infinite extent with the incident plane wave p_i.

structural homogeneity of this system is assumed to be in the x direction. Moreover, steady state problems assume that the time factor $exp(-i\omega t)$ is suppressed. Here ω is the angular frequency. Therefore, the incident sound pressure p_i can be written as

$$p_i(x,z) = P_i \cdot exp(ikx\sin\theta + ikz\cos\theta). \tag{2.1}$$

Here, k represents the wave number of air, and i is an imaginary unit. P_i is the amplitude of sound pressure, which is considered to be unity in the calculation. Similar to the incident sound pressure, the reflected sound pressure p_r, transmitted sound pressure p_t, first air-layer sound pressure p_1, and second air-layer sound pressure p_2 can be expressed as

$$p_r(x,z) = P_r \cdot exp(ikx\sin\theta - ikz\cos\theta)\ , \tag{2.2}$$

$$p_t(x,z) = P_t \cdot exp(ikx\sin\theta + ikz\cos\theta)\ , \tag{2.3}$$

$$\begin{aligned} p_1(x,z) = P_1^+ &\cdot exp(ikx\sin\theta + ikz\cos\theta) + \\ P_1^- &\cdot exp(ikx\sin\theta - ikz\cos\theta)\ , \end{aligned} \tag{2.4}$$

$$\begin{aligned} p_2(x,z) = P_2^+ &\cdot exp(ikx\sin\theta + ikz\cos\theta) + \\ P_2^- &\cdot exp(ikx\sin\theta - ikz\cos\theta)\ . \end{aligned} \tag{2.5}$$

P_r, P_t, $P_1^{\pm}$, $P_2^{\pm}$ are unknown coefficients. From the incident side, the first panel, second panel and MPP are located at z=0, z=ℓ_1 and z=$\ell_1 + \ell_2$ with their respective Young's modulus E, Poisson ratio ν, and loss factor η shown in Fig. 2.1. The equation of motion for the displacement W by each panel vibration can be expressed as

$$D_1\nabla^4 W_1 - \rho_1 h_1 \omega^2 W_1 = [p_i + p_r - p_1]_{z=0} \ , \tag{2.6}$$

$$D_2\nabla^4 W_2 - \rho_2 h_2 \omega^2 W_2 = [p_1 - p_2]_{z=\ell_1} \ , \tag{2.7}$$

$$D_3\nabla^4 W_3 - \rho_3 h_3 \omega^2 W_3 = [p_2 - p_t]_{z=\ell_1+\ell_2} \ , \tag{2.8}$$

$$D_j = \frac{E_j h_j^3 (1 - i\eta_j)}{12(1-\nu_j^2)} \ . \tag{2.9}$$

where $\nabla^4 = (\partial^2/\partial x^2 + \partial^2/\partial y^2)^2$. D, ρ and h are the flexural rigidity, density, and thickness of the panel, respectively. The panel and MPP are identified by subscripts 1, 2, 3 or j (j=1, 2, 3). The velocity in the z direction, which corresponds to the sound pressure, is given by

$$v_i = \frac{\cos\theta}{\rho_o c} p_i \ , \tag{2.10}$$

$$v_r = \frac{-\cos\theta}{\rho_o c} p_r \ , \tag{2.11}$$

$$v_t = \frac{\cos\theta}{\rho_o c} p_t \ , \tag{2.12}$$

$$v_1 = \frac{\cos\theta}{\rho_o c} p_1^+ - \frac{\cos\theta}{\rho_o c} p_1^- \ , \tag{2.13}$$

$$v_2 = \frac{\cos\theta}{\rho_o c} p_2^+ - \frac{\cos\theta}{\rho_o c} p_2^- \ . \tag{2.14}$$

Here, ρ_o is the density of air and c is the speed of sound. The boundary conditions for panels 1 and 2 in the coupled analysis of the wave motion and the panel vibration can be expressed as

$$[v_i + v_r]_{z=0} = -i\omega W_1 \ , \tag{2.15}$$

$$[v_i + v_r]_{z=0} = v_1 \mid_{z=0} \ , \tag{2.16}$$

$$v_1 \mid_{z=\ell_1} = -i\omega W_2 \ , \tag{2.17}$$

$$v_1 \mid_{z=\ell_1} = v_2 \mid_{z=\ell_1} \ . \tag{2.18}$$

A detailed theoretical study has examined the flexural vibration of a perforated panel [52]. This study considered the effect of perforation on the flexural vibration of the panel. Let Z_0 be the acoustic impedance of each hole in an MPP. Z_0 is composed of the resistance term Z_{res} and reactance term Z_{rea}, and is expressed as $Z_0 = Z_{res} + Z_{rea}$, where

$$Z_{res} = \frac{8\mu h}{(0.5d)^2}\left(\sqrt{1 + \frac{X^2}{32}} + \frac{\sqrt{2}dX}{8h}\right) \ , \tag{2.19}$$

$$Z_{rea} = -i\rho_0\omega h\left(1 + \frac{1}{\sqrt{9 + \frac{X^2}{2}}} + \frac{0.85d}{h}\right) \ , \tag{2.20}$$

and

$$X = (0.5d)\sqrt{\frac{\rho_0\omega}{\mu}} \ . \tag{2.21}$$

This function, which was proposed by Maa, has been widely applied to calculate the MPP. Here, d and μ are the perforation diameter and air viscosity (=1.8 $\times 10^{-5}$ Pa $\cdot$ s), respectively. Let σ be the perforations ratio, then the boundary condition for the MPP can be expressed as [8]

$$v_2 \mid_{z=\ell_1+\ell_2} = -i\omega\zeta_c W_3 + \frac{[p_2 - p_t]_{z=\ell_1+\ell_2}}{Z_0}\sigma \ , \tag{2.22}$$

$$v_2 \mid_{z=\ell_1+\ell_2} = v_t \mid_{z=\ell_1+\ell_2} \ . \tag{2.23}$$

By coupled analysis of Eqs. (2.6) - (2.8), (2.15) - (2.18), (2.22), and (2.23), the transmitted sound pressure can be determined. According to the effect of the directional distribution of the incident energy in a reverberation chamber, the averaged sound transmission coefficient $\bar{\tau}$ can be calculated by an additional term of the weighting function [53] $w(\theta) = exp(-\beta\theta^2)$, which can be expressed as

$$\bar{\tau}(\omega) = \frac{\int_0^{\pi/2} \tau(\omega,\theta) w(\theta) \sin 2\theta d\theta}{\int_0^{\pi/2} w(\theta) \sin 2\theta d\theta} \ . \tag{2.24}$$

where $\tau(\omega,\theta) = |P_t|^2/|P_i|^2$, and β is constant number (=1.0). By Eq. (2.24), the averaged sound transmission loss can be acquired by

$$\overline{TL} = 10 \log_{10} \frac{1}{\bar{\tau}(\omega)} \ . \tag{2.25}$$

Using Eq. (2.25), the sound insulation characteristics of a double panel with an MPP can be calculated.

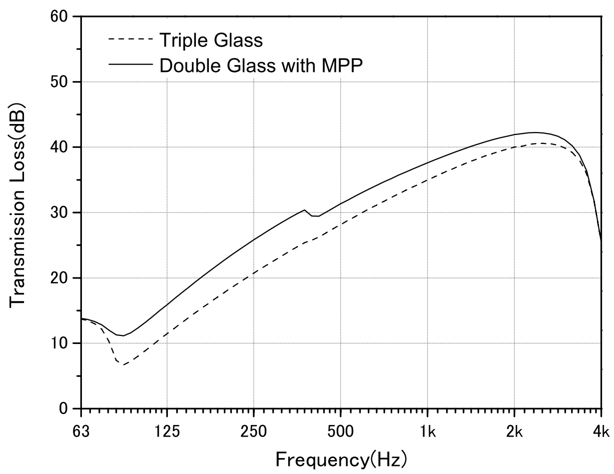

Figure 2.3: Sound transmission loss of a triple glass window (broken line, h_1=h_2=h_3=3 mm, l_1=6 mm, l_2=100 mm) and double glass window with an acrylic MPP (solid line, h_1=h_2=h_3=3 mm, l_1=6 mm, l_2=100 mm). Perforation diameter (d) is 0.3 mm and the perforation pitch (a) is 6.2 mm.

To discuss the potential improvement in the sound insulation performance in a multi-layer structure using an MPP, this section shows select numerical results. The calculation assumes that the glass has Young's modulus of 7×10^{10} N/m^2, density of 2500 kg/m^3, Poisson ratio of 0.22, and loss factor of 2×10^{-3}.

Figure 2.3 compares the sound transmission loss between a triple glass window (the width of glass h_1=h_2=h_3=3 mm, the width of air layer l_1=6 mm, l_2=100 mm) and a double glass window with an MPP (h_1=h_2=h_3=3 mm, l_1=6 mm, l_2=100 mm). Here, the perforation diameter (d) is 0.3 mm, and the perforation pitch (a) is 6.2 mm. Compared to triple glass, double glass with an MPP has improved sound insulation performance at low frequencies to about 5 dB in the range of 100 - 400 Hz. However, the presence of an MPP has a negligible effect at other frequencies. Therefore, changing the transmitted side glass to an MPP can improve the sound insulation performance, and may effectively prevent resonance effects.

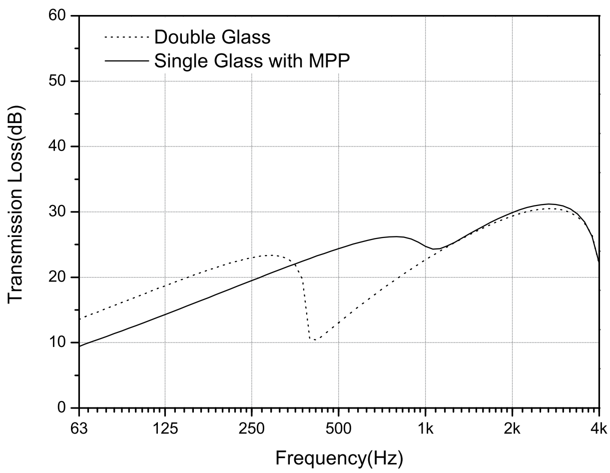

Figure 2.4: Sound transmission loss of a double glass window (dotted line) and a single glass with an acrylic MPP (solid line). Perforation diameter (d) is 1 mm and the perforation pitch (a) is 10 mm.

Figure 2.4 shows the results of sound transmission loss calculated for a double glass window (h_1=h_2=3 mm, l=6 mm) and a single glass window with an MPP (d=1 mm, a=10 mm). Perforating the glass on the transmitted side improves the sound insulation performance in the mid-frequency range and prevents resonance effects in the range of 400 - 1 kHz. However, significant changes are not observed at high frequencies. Although deterioration appears at low frequencies, the expected effect is achieved.

2.2.2 Finite analytical model

In this section, the sound transmission loss of a single panel with an MPP in an acoustic tube is calculated using an analytical model with cylindrical coordinates (Fig. 2.5). This model is used to validate the vibration theory of MPP and to confirm the resonance suppression effect of an MPP compared to the experiment.

The model assumes the panels are clamped at the edge of an acoustic tube with radius b and a normal incidence of plane wave. The sound pressure, particle velocity, and displacement of the panel mentioned in the previous section are changed into cylindrical coordinates,

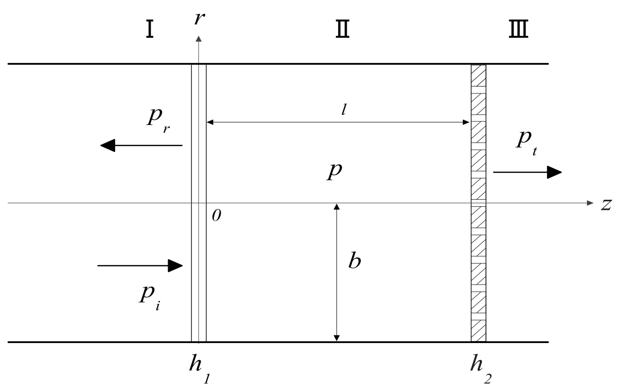

Figure 2.5: Analytical model of a double-leaf structure using cylindrical coordinates with an incident plane wave p_i.

and are expanded using the eigenvalue. Assuming the amplitude of the incident sound pressure is unity, the sound pressure P and particle velocity V in the three regions can be expressed as

region I

$$\begin{gathered} P_I = P_i + P_r = exp(ikz)+ \\ \sum_{m=1}^{\infty} A_m J_0(\bar{\beta}_m r) exp(-ik\varsigma_m z) \ , \end{gathered} \tag{2.26}$$

$$\begin{aligned} V_I = \frac{1}{\rho_0 c}[exp(ikz) - \sum_{m=1}^{\infty} A_m \varsigma_m \\ J_0(\bar{\beta}_m r) exp(-ik\varsigma_m z)] \ , \end{aligned} \tag{2.27}$$

regionII

$$\begin{aligned} P_{II} = P = \sum_{m=1}^{\infty} B_m J_0(\bar{\beta}_m r) exp(ik\varsigma_m z)+ \\ \sum_{m=1}^{\infty} C_m J_0(\bar{\beta}_m r) exp(-ik\varsigma_m z) \ , \end{aligned} \tag{2.28}$$

$$\begin{aligned} V_{II} = \frac{1}{\rho_0 c}[\sum_{m=1}^{\infty} B_m \varsigma_m J_0(\bar{\beta}_m r) exp(ik\varsigma_m z)- \\ \sum_{m=1}^{\infty} C_m \varsigma_m J_0(\bar{\beta}_m r) exp(-ik\varsigma_m z)] \ , \end{aligned} \tag{2.29}$$

region III

$$P_{III} = P_t = \sum_{m=1}^{\infty} D_m J_0(\bar{\beta}_m r) exp(ik\varsigma_m z) \ , \tag{2.30}$$

$$V_{III} = \frac{1}{\rho_0 c} \sum_{m=1}^{\infty} D_m \varsigma_m J_0(\bar{\beta}_m r) exp(ik\varsigma_m z) \ . \tag{2.31}$$

where J_0 and J_1 are the zero- and first-order Bessel functions, respectively. The eigenvalue β_m is the constant from the function $J_1(\beta_m) = 0$. Here, $\bar{\beta}_m$=β_m/b. The quantity ς_m is given by

$$\varsigma_m = \begin{cases} \sqrt{1-(\bar{\beta}_m/k)^2} \,, & (\bar{\beta}_m/k)^2 \leq 1 \\ i\sqrt{(\bar{\beta}_m/k)^2 - 1} & (\bar{\beta}_m/k)^2 > 1 \end{cases} \ . \tag{2.32}$$

The displacement of panel w, which is expanded by the eigenfunction $\varphi_n(r)$, can be expressed as

$$w(r) = \sum_{n=1}^{\infty} W_n \varphi_n(r) \ . \tag{2.33}$$

Here, the eigenfunction $\varphi_n(r)$ is given by

$$\varphi_n(r) = J_0(\frac{\gamma_n r}{b}) - \frac{J_0(\gamma_n)}{I_0(\gamma_n)} I_0(\frac{\gamma_n r}{b}) \ , \tag{2.34}$$

where the eigenvalues γ_n are obtained by the following equation

$$\frac{J_1(\gamma_n)}{J_0(\gamma_n)} + \frac{I_1(\gamma_n)}{I_0(\gamma_n)} = 0 \ , \tag{2.35}$$

This eigenfunction has an orthogonal relation, which is given by

$$\int_0^b \varphi_m(r)\varphi_n(r) r dr = \begin{cases} 0, & m \neq n \\ b^2 J_0^2(\gamma_n), & m = n \end{cases} \ . \tag{2.36}$$

where I_0 and I_1 are the zero- and first-order modified Bessel functions, respectively. By substituting the series expansion for the sound pressure, particle velocity, and panel displacement into Eqs. (2.6), (2.8), (2.15), (2.16), (2.22), and (2.23), and changing the subscript for the double structure, a matrix for the simultaneous equation for unknown quantities A_m, B_m, C_m, and D_m can be acquired. Solving this matrix can calculate the sound transmission loss.

To compare the double acrylic panel (h_1=h_2=1 mm, l=9 mm) to the single acrylic panel with an acrylic MPP (d=0.5 mm, a=10 mm), Figure 2.6 shows numerical examples of the calculated sound transmission loss. The calculation assumes that the acrylic panel has Young's modulus of 3.2×10^{10} N/m^2, density of 1190 kg/m^2, Poisson ratio of 0.3, and loss factor of 3×10^{-2}.

The results indicate that the resonance of the double panel structure appears around 630 Hz, but employing an MPP improves the sound insulation performance around the resonance frequency of the double panel structure by about 13 dB. This improvement is attributed to the shift in resonance frequency. Although the sound insulation performance is weakened at low and high frequencies, the improved performance at the resonance frequency is confirmed. Therefore, changing the panel to an MPP may effectively suppress the resonance effect.

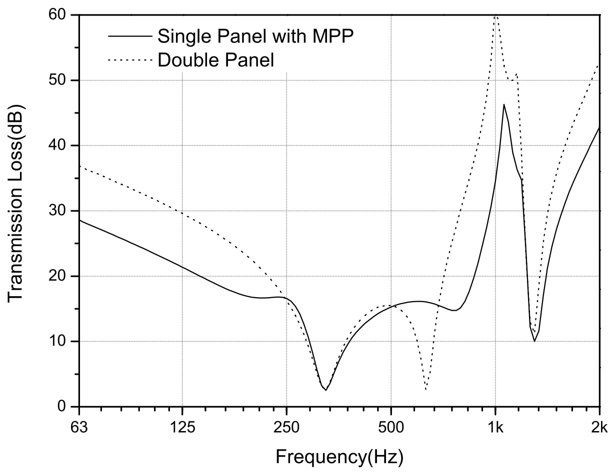

Figure 2.6: Theoretical results of the double acrylic panel (dotted line, h_1=h_2=1 mm, l=9 mm) and a single acrylic panel with an acrylic MPP (solid line, perforation diameter (d) =0.5 mm, pitch (a)=10 mm).

2.3 Experimental studies

2.3.1 Measurement using acoustic tube

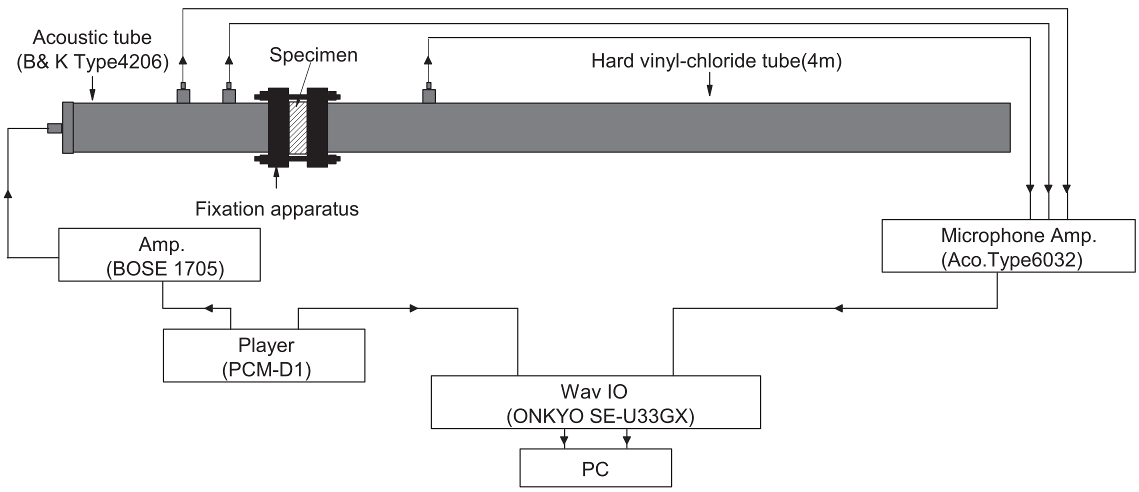

Figure 2.7: Experimental system using an acoustic tube.

To validate the proposed prediction method, a theoretical model with an MPP in cylindrical coordinates is compared to the experimental results using an acoustic tube. Figure 2.7 shows the experimental setup consisting of an acoustic tube with a 10 cm diameter.

The source side is B&K Type4206 and the transmitted side is a 4 m long hard vinyl-chloride tube with an open end. To measure the impulse response at each microphone position, the time-stretched pulse signal [54] is adopted as the source signal. The sound transmission loss is calculated from the measured data obtained at 3 microphone positions based on wavefield decomposition method [55] using the impulse responses at the positions. The transmitted signal is cut off after 24 ms to separate the signals from the reflections. The specimens are sandwiched between wooden boards with holes that have the same diameter as the acoustic tube. Then they are clamped at the edges.

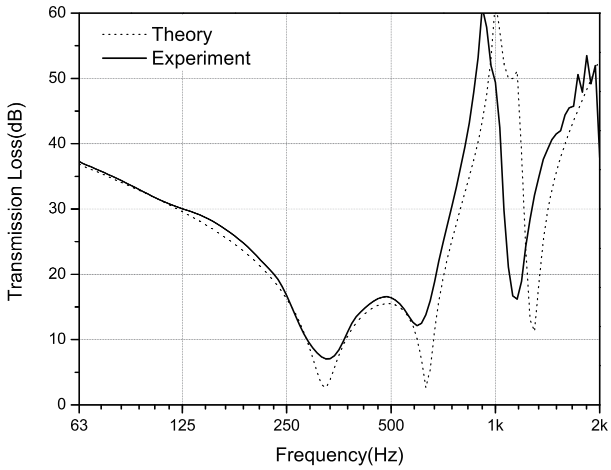

Figure 2.8: Experimental (solid line) and theoretical (dotted line) results of a double acrylic panel in the acoustic tube.

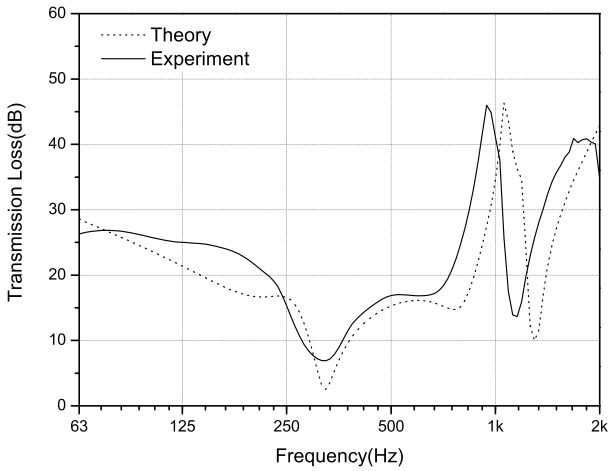

Figure 2.9: Experimental (solid line) and theoretical (dotted line) results of a single acrylic panel with an acrylic MPP in the acoustic tube.

Figures 2.8 and 2.9 compare the calculated and experimental results for a double acrylic panel (h_1=h_2=1 mm, l=9 mm), and a single acrylic panel with an acrylic MPP (holes diameter d=0.5 mm, pitch a=10 mm), respectively. Both the calculated and experimental results agree well, except for the shifts in the peaks and dips. Perforating the panel prevents the sound insulation deficiency due to mass-air-mass resonance of the double acrylic panels near 630 Hz. Hence, the results validate the introduced theory as well as confirm the effect of an MPP. However, there is a slight discrepancy between the theory and experiment at all frequencies, which may be due to the edge condition in the experiments as well as the accuracies of the material constants of the specimen. Moreover, the discrepancy above 1800 Hz may be caused by the limited diameter of the acoustic tube. Furthermore, in the case of a single panel with an MPP, the discrepancy between the theory and experiment at low frequencies may be due to the truncation effect of the measured impulse responses for the transmitted waves.

2.3.2 Measurement using reverberation chambers

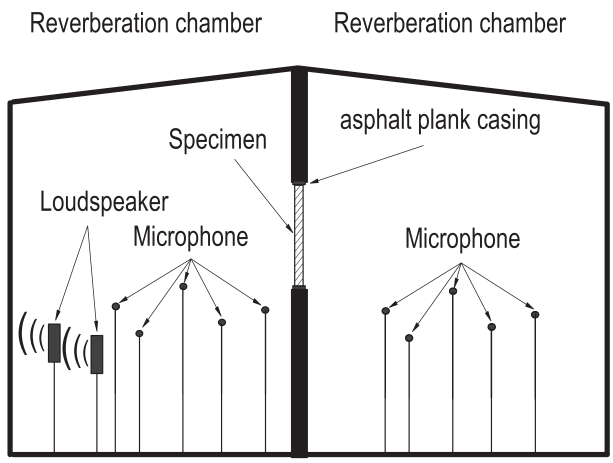

Figure 2.10: Measurement system for multi-layer structures using the reverberation chamber method.

An experiment was conducted using the reverberation chamber method to verify the practicality of improving the sound insulation performance of multi-layer structures via an MPP. Figure 2.10 shows the experimental setup. The volumes of reverberation chambers are 178.5 and 180.0 m^3. The specimen (1.2$\times$ 0.9 m) is installed at the aperture by a casing composed of an asphalt plank. All the gaps between the specimen and aperture are implanted with oil clay. In this case, two panels are attached at the edge by four thin wooden leaves, which are each 6 mm thick and 10 mm wide. A small urethane foam is placed at the center of the two panels to maintain an air cavity with a uniform depth of 6 mm. The sound source is two loudspeakers fed by white noise, and five microphones located in each chamber measure the spatially averaged sound pressure levels over each 1/3-octave band. Here, the parameters of glass and acrylic MPPs are the same as those in Sections 2.2 and 2.3.

The following focuses on the sound transmission loss of glass with an acrylic panel (glass-panel window, h_1=h_2=3 mm, l=6 mm) and glass with an acrylic MPP (glass-MPP window, d=1 mm, a=10 mm).

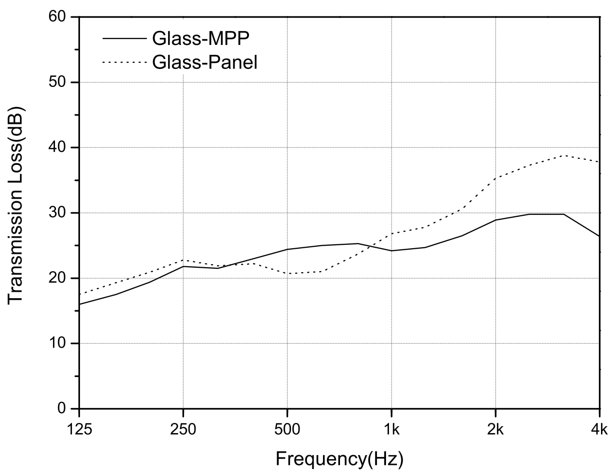

Figure 2.11: Experimental results of glass with an acrylic panel (dotted line) and glass panel with an acrylic MPP (solid line).

From the experimental results shown in Fig. 2.11, the sound insulation deficiency near 500 Hz is suppressed in the glass-panel window using an MPP, and consequently, the improved sound insulation performance using an MPP is confirmed. However, the difference in the sound insulation characteristics at high frequencies between the two structures is seen.

The effect of the directional distribution of incident energy in this reverberation chamber to the sound transmission loss is calculated using a Gaussian function and a modified Hanning window function, which are used as the weighting function in Eq. (2.24). Figures 2.12 and 2.13 show the theoretical results and experimental results of a glass with an acrylic MPP and a glass with an acrylic panel, respectively. The modified Hanning window function is also used:$w(\theta) = 0.5 + 0.5\cos((\theta - 30.0)\pi/55.0)$ ($30^\circ \leq \theta \leq 85^\circ$). When $0^\circ \leq \theta < 30^\circ$ and $85^\circ < \theta \leq 90^\circ$, the normalized energy densities are assumed to be 1 and 0, respectively.

From Figure 2.12, the calculated results of a glass with an acrylic

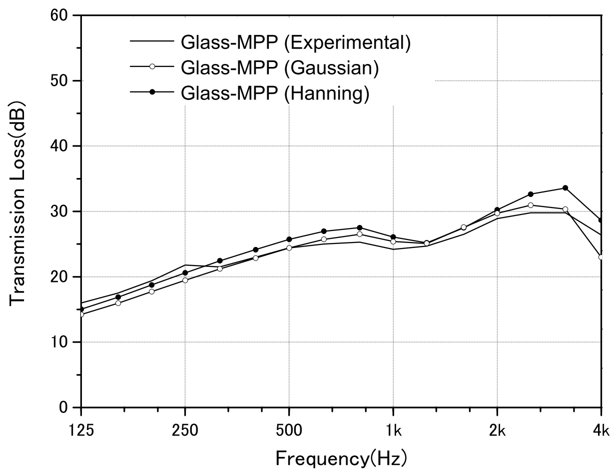

Figure 2.12: Comparison between theoretical results calculated by Gaussian function ($-\circ-$) and modified Hanning window function ($-\bullet-$) and experimental result (—-) of a glass with an acrylic MPP, respectively.

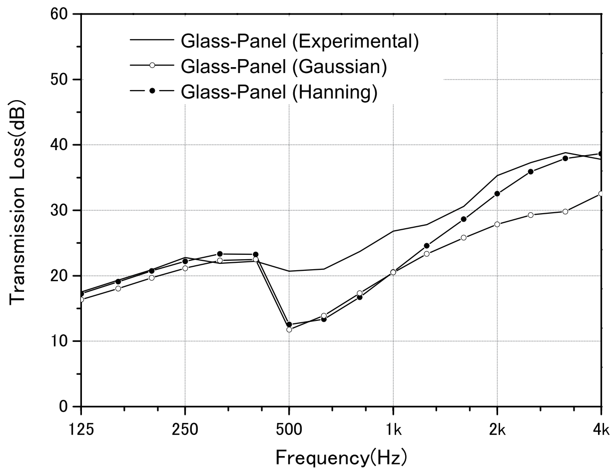

Figure 2.13: Comparison between theoretical results calculated by Gaussian function (– ○ –) and modified Hanning window function (– ● –) and experimental result (—-) of a glass with an acrylic panel, respectively.

MPP have good agreement with the experimental results. The calculated results and experimental results of a glass with an acrylic panel are shown in Fig. 2.13. The calculated result by Gaussian function has a significant discrepancy with the experimental result, especially at high frequencies. The calculated result by the modified Hanning window function has a similar tendency with the experimental result. However, in both cases, the depths of the sound insulation deficiency near 500 Hz do not agree well with experimental result. This may be due to the effect of oil clay and urethane foam. For the glass-panel window, the sound insulation performances, which are calculated by a Gaussian function and the modified Hanning window function, do not differ significantly, except above 1 kHz. The glass-MPP window and glass-panel window exhibit a similar tendency. However, there is a discrepancy above 2 kHz. The results of sound transmission loss of a glass-panel window and a glass-MPP, which are calculated by Gaussian function, are almost the same above 1.25 kHz. However, in the case of the modified Hanning window function, noticeable discrepancies in sound transmission loss between a glass-panel window and a glass-MPP window appear above 1.25 kHz. From the calculated results, the sound transmission loss at high frequencies changes due largely to the effect of the directional distribution of the incident energy. Although there are some discrepancies between theory and experiments, it can be concluded that perforating the panel on the transmitted side of a multi-layer structure is an effective approach to suppress the effect of mass-air-mass resonance.

The improvement effect of MPP may vary depending on the installed condition such as the use of the putty at peripheral part of the glass in practical use of glass window. Moreover, an opposite effect may be occurred by the configuration of perforation. The possibilities of opposite effect are discussed in section 2.4.

2.4 Parametric study

In this section, the effects of the pitch and perforation diameter on the sound transmission loss of a single glass with an acrylic MPP are studied. The parameters of the glass and acrylic MPPs are the same as those in Sections 2.2 and 2.3.

2.4.1 The effect of pitch

Figure 2.14 shows the results of the sound transmission loss calculated for various pitches in 2 mm intervals at a constant diameter of 1 mm. As the pitch size decreases, the resonance shifts to a higher frequency, improving the mass-air-mass resonance. Changing the pitch to 2.5 mm shifts the resonance above 4 kHz without a significant change at low frequencies. Consequently, the effect of resonance is suppressed, and the sound insulation performance at higher frequencies is improved.

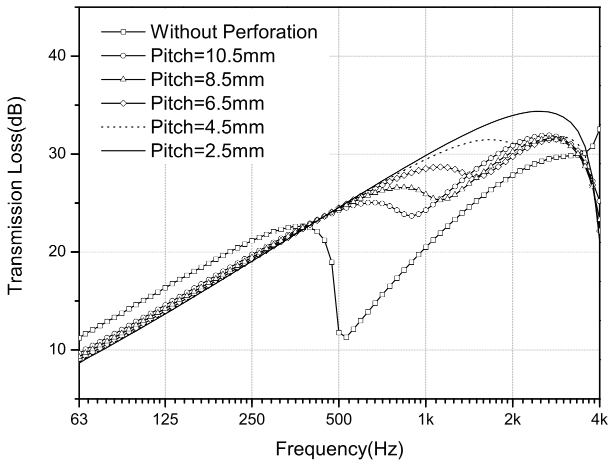

Figure 2.14: Effect of pitch on sound transmission loss in a single glass with an acrylic MPP. Perforation diameter is constant (1 mm). Pitches are 2.5 (—–), 4.5 (···), 6.5 (−◇−), 8.5 (− △ −), and 10.5 (− ∘ −) mm.

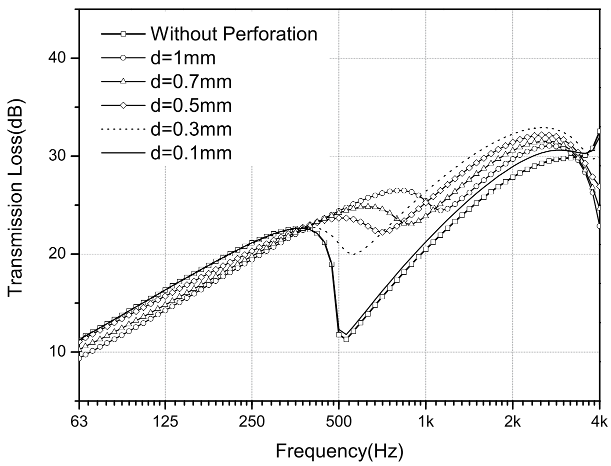

Figure 2.15: Effect of perforation diameter on the sound transmission loss in a single glass with an acrylic MPP. Pitch is constant (10 mm). Perforation diameters are 0.1 (—–), 0.3 (···), 0.5 (−◇−), 0.7 (− △ −), and 1 (− ∘ −) mm.

2.4.2 The effect of perforation diameter

At a constant pitch (10 mm), the sound transmission losses for various diameters are calculated. The perforation diameters are 0.1, 0.3, 0.5, 0.7, and 1 mm. Figure 2.15 shows the results. As the diameter increases, the resonance shifts to higher frequencies. Consequently, the deficiency of resonance is improved. This tendency is similar to the case of pitch. However, the sound insulation performance at higher frequencies deteriorates. Hence, increasing the size of the diameter in the range between 0.1 and 1mm can suppress the effect of mass-air-mass resonance.

2.5 Conclusion

The influence of MPPs on the performance of multi-layer structures is theoretically and experimentally investigated, and the applicability of MPPs to improve the sound insulation performance is discussed. The experimental results using an acoustic tube confirmed both the suppression effect of an MPP on the mass-air-mass resonance and the theory. Moreover, the reverberation chamber method verified the actual sound insulation characteristics. This study demonstrated that perforating a panel on the transmission side of a multi-layer structure effectively prevents mass-air-mass resonance. Moreover, the best sound insulation performance depends on the both the hole diameter and pitch of the perforation.

CHAPTER. Ⅲ

3 Improvement of sound insulation performance of double-panel structures by using damping materials

3.1 Introduction

A fundamental measure for improving the sound transmission performance of a homogeneous single partition is to increase the mass and thickness; however, this technique has limitations and is inconvenient. For these reasons, sound transmission through a double-plate system with various compositions has been studied. A typical example of this research is that of London[11]. Modern mass-air-mass multilayer structures are currently used for sound insulation in many fields because they are lightweight and have effective sound insulation performance at high frequencies. However, resonance of these structures results in problems with sound insulation. To solve this problem and to achieve higher transmission loss, a traditional approach that involves filling the cavities with porous material as a medium of wave propagation with resistance dissipation is adopted. A type of sandwich construction composed of double plates with a solid elastic core was studied by Kropp et al.[39], in which the core was modeled as a locally reactive elastic material (Winkler-type material). In addition, a more rigorous model of the core developed from general elastic theory was studied by Ford et al.[40]. Although porous materials can provide good sound insulation performance, their fibers can cause irritation of both the eye and skin and various respiratory ailments. Moreover, porous materials cannot be used for glass windows because they lack transparency. We have also studied the improvement in the sound insulation performance of multilayer structures by using MPP [56 - 58]. Our results indicated that the effect of resonance on sound insulation performance is suppressed due to a shift of the resonance frequency. However, the sound transmission loss at high frequencies deteriorates owing to the decrease in mass and the effect of holes. Another approach is improving the sound insulation performance of a double structure by means of damping materials.

The damping of acoustic energy can be defined as the process by which materials or structures dissipate acoustic energy and transfer it to connected structures or ambient media. These mechanisms have the effect of controlling the amplitude of resonant vibrations and modifying wave attenuation and sound transmission properties.

From the late 20th century, various kinds of damping layers were extensively used for noise and vibration control of thin walled structures such as automotive body parts, aircraft panels, containers, or casings and considerable research had been devoted[30 - 31]. This approach called passive damping treatment. Compared to the more modern noise control measures – active and semi-active noise control, they are still an attractive alternative or supplement for the reasons of economy and simplicity.

Passive layer damping, usually implemented as constrained layer damping, is the most common form of damping treatment. There are numerous studies on the damping of vibrations in structures via the usage of viscoelastic damping layers such as the works of A.D. Nashif et.al.,[32] and C.T. Sun &Y.P. Lu[33]. Kerwin first discussed a three layer beam with a damping layer sandwiched between two face layers [34]. Some authors also proposed to use recycled products to provide alternatives to existing products in a great number of commercial and environmental noise control applications, including building, automotive and business services areas and traffic noise abatement [35 - 38]. However, damping materials are usually installed at whole surface as a layer material in almost all studies. This kind of installation usually involves penalty of weight and transparency.

To determine effective methods for improving sound insulation performance, a practical technique for improving the sound transmission loss is to use viscoelastic materials as the connectors of double panel structures. Compared to the porous materials and traditional damping materials, the viscoelastic material used as a connector is more appropriate to apply to glass window. Even if an opaque connector is installed, the effect on the transparency is limited due to the limited area of the connector. Furthermore, it is possible to realize a transparency connector. In the field of multi-layer structures, sound insulation deficit due to mass-air-mass resonance is the most significant problem. Due to these reasons mentioned above, a new approach by installing damping connectors at sound wave crest and trough is proposed in the present study. From the commonsense of sound insulation technique for multi-layer pane structures, it has been thought that any connector between the panels causes " sound bridge effect ", which deteriorates the sound insulation performance. The key factor of the present study is to propose selection of new materials and how to use the materials of connectors, which improve sound insulation performance of multi-layer structures.

Many factors such as the location, numbers, and properties of connectors must be considered because inappropriate combinations of these parameters could cause negative effects. Therefore, the effects of various connector parameters on the sound insulation performance of double structures are investigated by considering the effect of normal modes of vibration (vibrational mode) of the panel. In this study, the analytical model of sound transmission through a double-panel system considered in this study consists of simply supported double panels fastened by several connectors, set in an infinite rectangular duct. Sound transmission loss is calculated by a coupled analysis of the panel vibration and the wave motion of air. The trends of improvement of sound transmission loss of double-panel structures are investigated for the case of a single angle of incidence. The effects of viscoelastic materials are discussed for different combinations of location, numbers, and properties.

3.2 Theory

To investigate the possibilities of improving the sound insulation performance of double-panel structures by using viscoelastic materials, an analytical model is introduced as shown in Fig. 3.1.

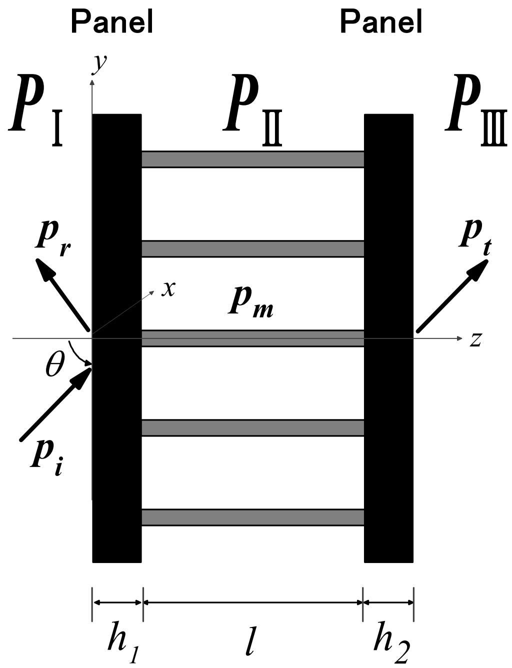

Figure 3.1: A sketch of the analytical model.

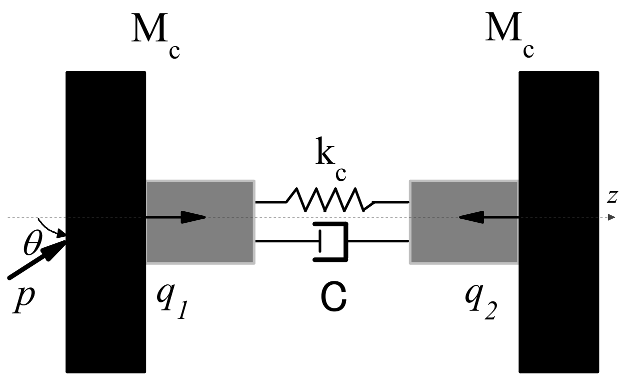

Figure 3.2: The analytical model of the connector with the panel of one vibratory element.

Where p_i and p_r are the incident sound pressure and reflected sound pressure, respectively. p_t is the transmitted sound pressure. Two panels with spacing l were set in the x - y plane of a rectangular duct. The space in the duct was divided into three parts I, II, and III from the incident side by two panels. Both panels were connected by discretely distributed connectors, which are shown in Fig. 3.2. Here, the connector is assumed as a Voigt model.

3.2.1 Formulation

In this model, a plane wave p_i with an incident angle θ is considered as the sound incident upon multilayer panels with simply supported edges set in a rectangular duct of infinite length. The cross-section area of duct is $a \times b$, where a and b represent the x and y coordinates. From the incident side, the first panel and second panel were located at $z = 0$ and $z = l$, respectively, with their corresponding Young's moduli (E), Poisson ratios (ν), and loss factors (η). The inner surface of the duct is assumed to be acoustically rigid, i.e., the particle velocity at the surface is zero. Both sides of each panel were considered to be filled with the same fluid (air). Additionally, the time factor $exp(-i\omega t)$ was suppressed by assuming a steady-state problem. In the time factor, ω and i represent angular frequency and an imaginary unit, respectively. k is wave number of air. The thicknesses of the first and second panels are h_1 and h_2, respectively. From these assumptions, the sound pressure p in each region of the air space can be expressed by using the series expansion of eigenfunction, which satisfy both the wave equation and boundary condition at the duct surface, as follows:

$$p_I = p_i + \sum_{mn} A_{mn}\phi_{mn}(x,y)exp(-ik\beta_{mn}z), \tag{3.1}$$

$$\begin{aligned} p_{II} = &\sum_{mn} B_{mn}\phi_{mn}(x,y)exp(ik\beta_{mn}z)+ \\ &\sum_{mn} C_{mn}\phi_{mn}(x,y)exp(-ik\beta_{mn}z), \end{aligned} \tag{3.2}$$

$$p_{III} = \sum_{mn} D_{mn}\phi_{mn}(x,y)exp(ik\beta_{mn}z), \tag{3.3}$$

$$\beta_{mn} = \begin{cases} \sqrt{1-(\lambda_{mn}/k)^2}\ , & (\lambda_{mn}/k)^2 \leq 1, \\ i\sqrt{(\lambda_{mn}/k)^2-1}\ , & (\lambda_{mn}/k)^2 > 1, \end{cases} \tag{3.4}$$

$$\lambda_{mn}^2 = [(m-1)\pi/a]^2 + [(n-1)\pi/b]^2. \tag{3.5}$$

where, $\phi_{mn}(x,y) = \cos[(m-1)\pi x/a]\cos[(n-1)\pi y/b]$ (m,n = 1,2,3$\cdots$). For the panel vibration, the displacement w is expressed by using eigenfunction of simply supported condition, as follows:

$$w_j = \sum_{MN} W_{jMN}\varphi_{MN}(x,y). \tag{3.6}$$

Here, the panels are identified by subscript j (j = 1,2). The eigenfunctions is $\varphi_{MN}(x,y) = \sin(M\pi x/a)\sin(N\pi y/b)$ (M,N = 1,2,3$\cdots$). The latter function satisfies the following orthogonal condition:

$$\int_0^a \int_0^b \phi_{mn}(x,y)\phi_{m'n'}(x,y)dxdy = \begin{cases} \frac{ab}{4}\epsilon_m\epsilon_n, m=m', n=n', \\ 0, otherwise, \end{cases} \quad (3.7)$$

where

$$\epsilon_m = \begin{cases} 2, m=1, \\ 1, m \neq 1, \end{cases} \quad \epsilon_n = \begin{cases} 2, n=1, \\ 1, n \neq 1. \end{cases} \quad (3.8)$$

The particle velocity v of each region in the z direction, which corresponds to the sound pressure, is given as following:

$$v_I = \frac{cos\theta}{\rho_0 c}p_i - \frac{1}{\rho_0 c}\sum_{mn} A_{mn}\beta_{mn}\phi_{mn}(x,y)exp(-ik\beta_{mn}z), \quad (3.9)$$

$$\begin{aligned} v_{II} = \frac{1}{\rho_0 c}\sum_{mn} B_{mn}\beta_{mn}\phi_{mn}(x,y)exp(ik\beta_{mn}z) - \\ \frac{1}{\rho_0 c}\sum_{mn} C_{mn}\beta_{mn}\phi_{mn}(x,y)exp(-ik\beta_{mn}z), \end{aligned} \quad (3.10)$$

$$v_{III} = \frac{1}{\rho_0 c}\sum_{mn} D_{mn}\beta_{mn}\phi_{mn}(x,y)exp(ik\beta_{mn}z). \quad (3.11)$$

Here, ρ_0 is the density of air, and c, the speed of sound. The boundary conditions for panels 1 and 2 in the coupled analysis of the wave motion and the panel vibration can be expressed as follows:

$$v_I \mid_{z=0} = v_{II} \mid_{z=0}, \quad (3.12)$$

$$v_I \mid_{z=0} = -i\omega w_1, \quad (3.13)$$

$$v_{II} \mid_{z=l} = v_{III} \mid_{z=l}, \quad (3.14)$$

$$v_{III}\mid_{z=l}=-i\omega w_2. \tag{3.15}$$

It should be noted that the eigenfunction of panels in the x-y plane differs from those of the air with rigid boundaries. To solve this problem for simultaneous linear equations,a direct correlation must be made between the two eigenfunctions ϕ_{mn} and φ_{MN}. The relation is written as shown below.

$$\varphi_{MN}(x,y)=\sum_{mn}\alpha_{MN}^{mn}\phi_{mn}(x,y), \tag{3.16}$$

where

$$\alpha_{MN}^{mn}=\frac{(\zeta_{Mm1}+\zeta_{Mm2})(\zeta_{Nn1}+\zeta_{Nn2})}{\epsilon_m\cdot\epsilon_n}, \tag{3.17}$$

$$\zeta_{Mm1}=\begin{cases}0,(M+m-1=0),\\ \frac{1-(-1)^{M+m-1}}{(M+m-1)\pi},(M+m-1\neq 0),\end{cases} \tag{3.18}$$

$$\zeta_{Mm2}=\begin{cases}0,(M-m+1=0),\\ \frac{1-(-1)^{M-m+1}}{(M-m+1)\pi},(M-m+1\neq 0),\end{cases} \tag{3.19}$$

$$\zeta_{Nn1}=\begin{cases}0,(N+n-1=0),\\ \frac{1-(-1)^{N+n-1}}{(N+n-1)\pi},(N+n-1\neq 0),\end{cases} \tag{3.20}$$

$$\zeta_{Nn2}=\begin{cases}0,(N-n+1=0),\\ \frac{1-(-1)^{N-n+1}}{(N-n+1)\pi},(N-n+1\neq 0).\end{cases} \tag{3.21}$$

In the case of a double layer structure, the unknown coefficients A_{mn}, B_{mn}, C_{mn}, D_{mn}, and W_{jMN} can be determined from the matrix that substituted the expressions for the sound pressure, particle velocity, and displacement of the panel into the vibration equations and boundary conditions. The sound transmission loss can be calculated by the formulation 10log10($1/\tau$), where τ is the sound transmission coefficient defined as $\tau=P_t/P_i$, which can be obtained as shown below.

$$P_t = 0.5 \int_0^a \int_0^b Re\{p_{III} \cdot v_{III}^*\}_{z=l} dxdy, \tag{3.22}$$

$$P_i = 0.5 \int_0^a \int_0^b Re\{p_i \cdot v_i^*\}_{z=0} dxdy. \tag{3.23}$$

where * represents the complex conjugate. When the viscoelastic materials are installed, the vibration equations can be expressed as follows:

$$\begin{aligned} D_1\nabla^4 w_1 - \rho_1 h_1 \omega^2 w_1 = p_I \mid_{z=0} - p_{II} \mid_{z=0} + \\ \sum_{j'} q_1 \delta(x - x_{j'}) \delta(y - y_{j'}), \end{aligned} \tag{3.24}$$

$$\begin{aligned} D_2\nabla^4 w_2 - \rho_2 h_2 \omega^2 w_2 = p_{II} \mid_{z=l} - p_{III} \mid_{z=l} + \\ \sum_{j'} q_2 \delta(x - x_{j'}) \delta(y - y_{j'}). \end{aligned} \tag{3.25}$$

In ∇^4=$(\partial^2/\partial x^2$+$\partial^2/\partial y^2)^2$, $D_j = E_j h_j^3(1 - \mathrm{i}\eta_j)/[12(1 - v_j^2)]$. D_j, ρ_j and h_j are the flexural rigidity, density and thickness of the panels, respectively; δ represents the delta function; and j' is the number of the damping material. The forces transmitted from the first panel (q_1) and the second panel (q_2) are given as follows:

$$\sum_{j'} q_1 \delta(x - x_{j'}) \delta(y - y_{j'}) = -(k_c^* - m_c\omega^2) w_1 + k_c^* w_2, \tag{3.26}$$

$$\sum_{j'} q_2 \delta(x - x_{j'}) \delta(y - y_{j'}) = -k_c^* w_1 + (k_c^* - m_c\omega^2) w_2, \tag{3.27}$$

$$k_c^* = k_c - iC\omega. \tag{3.28}$$

Here, m_c is half the mass of the damping material and k_c^* is the complex spring constant. In this case, the spring constant can be written as $k_c = SE_v/h$, which is defined by the cross-sectional area S of viscoelastic material, Young 's modulus E_v, and the length h of the viscoelastic material. The damping coefficient C is related to the damping ratio ζ and the spring constant as $C = \zeta(2M_c k_c)^{1/2}$, where M_c is the equivalent mass, including the vibratory element of the panel. The force transmitted from q_1 and q_2 can be expanded by the eigenfunctions ϕ_{mn} as shown below.

$$\sum_{j'} q_i \delta(x - x_{j'})\delta(y - y_{j'}) = \sum_{mn} \psi_{imn} \phi mn(x, y). \tag{3.29}$$

By a coupled analysis of Eqn. (3.22 - 3.23) and Eqn. (3.25), the unknown coefficients ψ_{imn} can be determined. By substituting the expanded q_1 and q_2 into the vibration equations (3.20 - 3.21) and coupling with the boundary conditions, sound transmission loss can be calculated for cases without damping as well as cases for which the effect of damping is taken into account.

3.2.2 Results and discussions

In this study, the damping material used as a connector is assumed to be installed at the center of each vibratory element of a target vibrational mode. To investigate the trend of improvement by using damping materials, two cases of incident angle (normal and 45° oblique) are calculated in this study, assuming that the glass has a Young 's modulus of 7×10^{10}N/m^2, density of 2500 kg/m^3, Poisson ratio of 0.22, and loss factor of 2 $\times10^{-3}$. Gum was used as the damping material; the gum sample had a Young 's modulus of 1×10^6 N/m^2, density of 1000 kg/m^3, and radius of 5 mm. It was assumed that this material has an arbitrarily varying viscosity. Figure 3.3 shows the installation locations of the 5*5 mode. The width a of the double glass

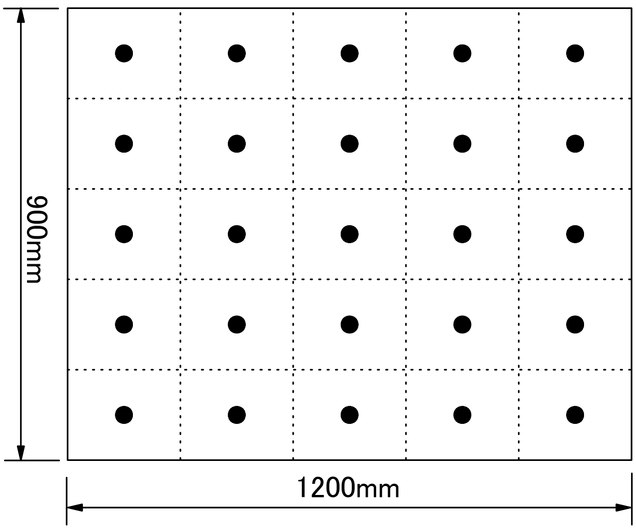

Figure 3.3: Installation locations of the 5*5 mode.

is 1.2 m, and the length b is 0.9 m. The thicknesses of the first glass panel (h_1), second glass panel (h_2) and air layer (l) are 3, 3 and 6 mm, respectively.

The results of the comparison between the double glass and the double glass with damping material (damping ratio $\zeta = 0.8$) of 5*5 mode are shown in Figs. 3.4 and 3.5. In addition, the figures show the results obtained from the calculated raw data and that obtained from the moving average. Bandwidth is 1/3-octave, and the shift in averaging is 1/24-octave.

The results indicate that the resonance of double glass appears to occur around 350 Hz. In the case of normal incidence, the sound insulation performance of the double glass with damping material at around resonance frequency is higher than that of only double glass by about 15 dB. At lower frequencies, significant changes are not observed. On the contrary, the sound insulation performance at high

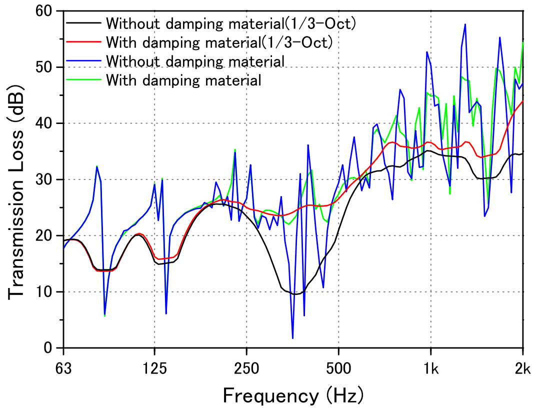

Figure 3.4: Calculated results for the 5*5 mode locations for normal incidence.

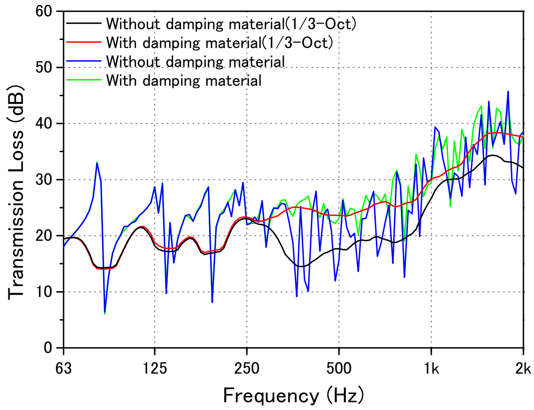

Figure 3.5: Calculated results for the 5*5 mode locations for oblique incidence (45°).

frequencies is slightly improved by the presence of the damping material. In the case of oblique incidence, the trend of improvement is similar to that of normal incidence; the sound insulation performance of the double glass with damping material at around resonance frequency is higher than that of only the double glass by about 10 dB. From the calculated results, it was determined that the effect of mass-air-mass resonance can be suppressed and the sound insulation performance can be improved by using a damping material with the appropriate combination of installation location, numbers, and material properties of the connector. The method of installing the damping material in the air layer of double glass is considered to be an effective sound insulation design approach. The effects of connector parameter combinations are discussed in Sec. 3.3.

3.3 Parametric survey

3.3.1 The effect of the number of installations

To investigate the effect of the number of installation (connectors), calculations were conducted by installing the damping materials with the damping ratio $\zeta = 0.8$ at the center of each vibratory element of a target vibrational mode, as shown in Fig. 3.3. The calculation results of 1*1, 2*2, 3*3, 4*4, 5*5, 6*6, 7*7, 8*8, 15*15 mode locations with normal and oblique incidence are shown in Figs. 3.6 and 3.7, respectively. Here, it is considered that 15*15 mode location is an extreme condition.

The results indicate that an increase in the number of installations decreases the effect of resonance, and the improvement of sound insulation performance is increased over the frequency range of interest. In the case of normal incidence, the improvement effect of the odd-odd mode is slightly better than that of even-even in the range of 750 - 1200 Hz; at other frequencies, the installation has a limited effect if the

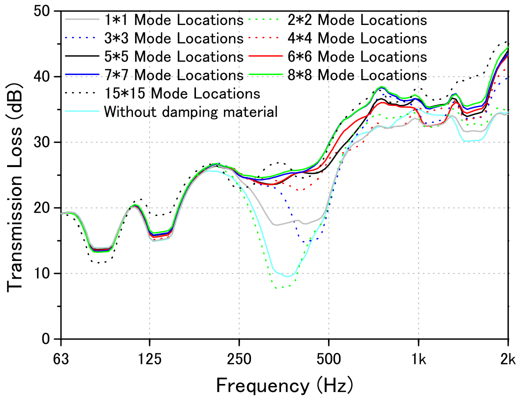

Figure 3.6: Effects of different number of installations for normal incidence.

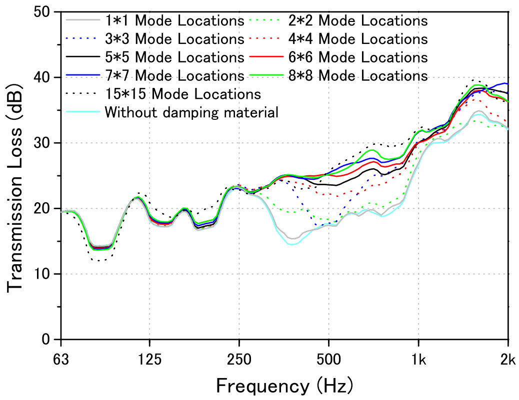

Figure 3.7: Effects of different number of installations for oblique incidence (45°).

numbers of installation are restricted. In the case of oblique incidence, the improvement effect of the odd-odd mode is somewhat better than that of even-even in the range of 700 - 2000 Hz. However, this case also shows that if the numbers of installation are restricted, the improvement effect is limited. Herein, it is concluded that the method of installing the damping material at the center of each vibratory element of odd-odd mode appears to be more efficient than that of even-even. Furthermore, when the number of installations is more than 25, i.e., 5*5 mode, the improvement on sound insulation performance near resonance frequency becomes insignificant. The special example (15*15 mode) shows that the improvement in sound insulation degradation is not significant compared to 5*5 - 8*8 mode locations. Therefore, improvement by increasing the number of installations has an upper limit.

3.3.2 The effect of the installation locations

In Sec. 3.3.1, it was determined that sound transmission loss may change remarkably with different installation locations. From this perspective, the effect of the installation locations should be investigated in detail. The number of 5*5 mode installation locations (25 points) are reduced to 9 (Fig. 3.8). Similarly, the number of 7*7 mode installation locations (49 points) are reduced to 25.

To compare with the effect of the same installation number with different locations, the calculated result of 3*3 mode locations (9 points) is added. The calculated results of normal and oblique (45°) incidence are shown in Figs. 3.9 and 3.10, respectively.

In the case of normal incidence, the improvement effect of 5*5 mode locations (9 points) is more efficient than 3*3 mode locations. Compared with the 7*7 mode locations (25 points), the effect of 5*5 mode locations (25 points) is more efficient. By decreasing the installation locations from 25 to 9 points of 5*5 mode locations, significant changes

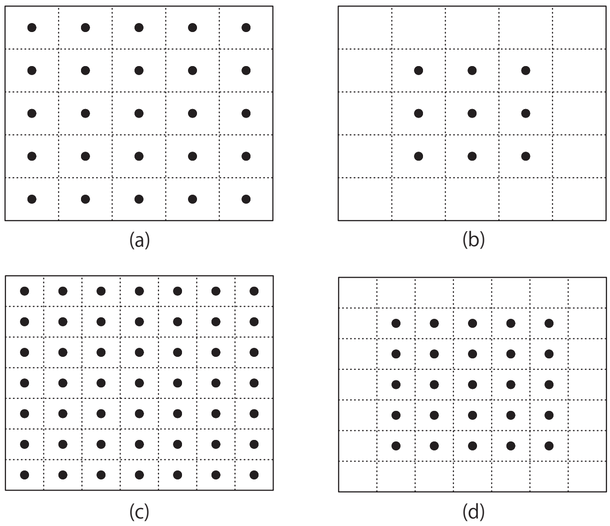

Figure 3.8: Installation locations. (a): 25 points of the 5*5 mode locations, (b): 9 points of the 5*5 mode locations, (c): 49 points of the 7*7 mode locations, and (d): 25 points of the 7*7 mode locations.

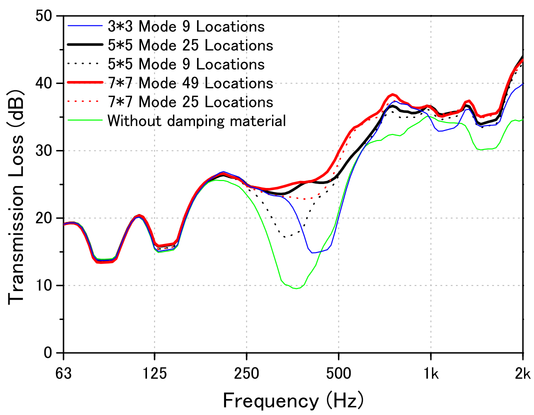

Figure 3.9: Effects of different installation locations for normal incidence.

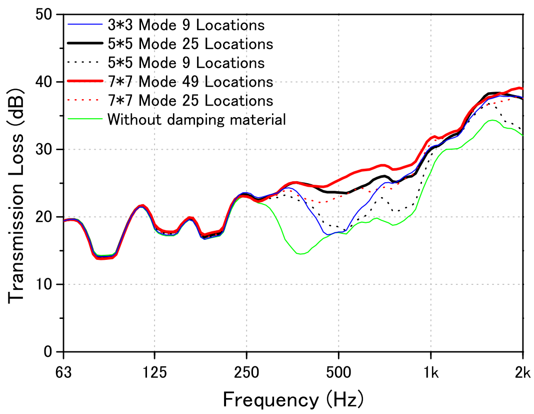

Figure 3.10: Effects of different installation locations for oblique incidence (45°).

are observed near resonance frequency. However, these changes are rarely observed at other frequencies. A similar trend can be seen in the case of 7*7 mode locations. In the case of oblique incidence, the improvement effect of 3*3 mode locations is more efficient than 5*5 mode locations (9 points). The effect of 5*5 mode locations (25 points) appears to be more efficient than that of 7*7 mode locations. From the comparison between 9 and 25 locations of 5*5 mode, a trend similar to that of normal incidence is observed except at high frequencies, which can be seen in the comparison of 49 and 25 locations of 7*7 mode. Therefore, it can be concluded that as long as the number of installation is relatively large, which is 25 in the present case, discrete distribution in an entire area is somewhat more efficient than concentric distribution in a central area. In any case, number appears to be more important than location.

3.3.3 The effect of the damping ratio of the connector

In this section, the effect of damping ratio ζ of the connector is examined. The calculated result of sound transmission loss with various

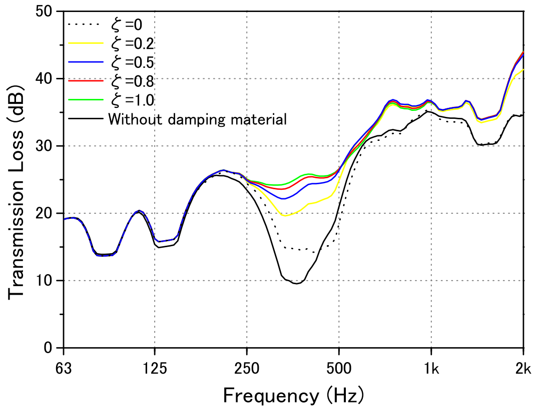

Figure 3.11: Effects of damping ratio for normal incidence.

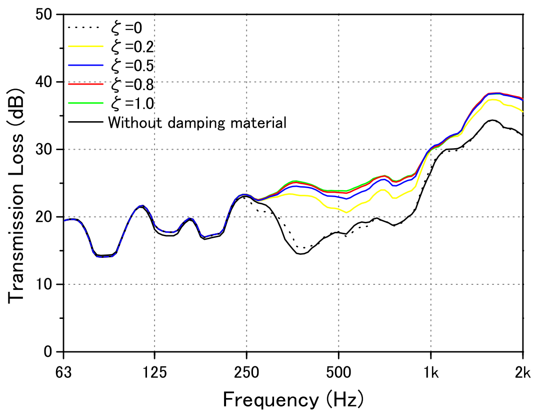

Figure 3.12: Effects of damping ratio for oblique incidence.

damping ratios under normal incidence is shown in Fig. 3.11. Figure 3.12 shows the calculated results of oblique incidence with same damping ratios. Here, the damping ratios are 0, 0.2, 0.5, 0.8, and 1.0. The value 1.0 represents the critical state of vibration.

A decrease in the damping ratio results in an increase in the deficiency of the resonance near 350 Hz. However, significant changes are not observed at other frequencies. In the range of damping ratios of 0.5 - 1.0, significant variation in sound transmission loss is not observed near resonance frequencies; in the range of damping ratios of 0 - 0.5, a significant variation in sound transmission loss is observed. From these results, it is concluded that the damping ratio of 0.5 is a key factor for improving sound insulation performance.

3.4 Conclusions

To suppress the effect of resonance and improve sound insulation performance, a method involving the use of viscoelastic materials as connectors in double-panel structures was proposed. The possibility of improving the sound insulation degradation under normal and oblique incidences was investigated by evaluating the effects of the number, installation locations, and properties of the viscoelastic connectors. Calculated results for different number of connectors and locations revealed that the method of installing the damping material at the center of each vibratory element of odd-odd mode appeared to be more efficient than that in the case of the even-even mode. Installation of the damping material improved the degradation due to mass-air-mass resonance and had a stable improvement effect over the frequency range of interest when the number of installations was relatively large. Results for the effect of damping ratio of the connector revealed that a damping ratio of more than 0.5 provides a stable improving effect over the frequency range of interest and improves degradation due to mass-air-mass resonance. To verify this theory, further investigation including some experiments is needed.

CHAPTER. IV

4 Effect of urethane foam cushion on floor impact sound of a slab with panel flooring

4.1 Introduction

The quality of a building is dominated by several elements, such as its design, safety performance, and residential amenities. The sound insulation performance is an important criterion for evaluating residential amenities. Many studies have been carried out to improve the sound insulation performance. Typical problems related to the sound insulation of buildings are sound transmission of windows and walls, floor impact sound, and general structure-borne sounds due to vibrations induced by equipment and machinery. The main factors for the physical manifestations of these problems are believed to be the vibration of building elements and acoustic coupling with the surrounding air. When there are sound transmission problems for a homogeneous single partition, a basic measure to improve the performance is to increase the mass and thickness. However, this approach is limited and outdated. Therefore, inhibiting sound transmission by using a double plate system with various constitutions has been studied.

Sound transmission through double plates with an air cavity is the simplest case. A typical example was presented by London [11]. Many studies have been reported ever since. Sound transmission problems for double plates with an absorptive layer have also been the subject of a number of papers; the absorptive layer is treated as a medium of wave propagation with resistance dissipation. Kropp and Rebillard [39] examined a sandwich construction composed of double plates with an elastic solid core where the core is modelled to have locally reactive elasticity (Winkler-type material). Ford et al. [40] developed a more rigorous model of the core from the general elastic theory. A structure of double plates with studs was designed as a lightweight construction material. The studs have an effect of short-circuit transmission as well as cavity boundaries. Studies on this problem have been devoted to the effects of studs with an air layer [41 - 43] and studs with an absorptive layer [44 - 46]. Hongisto [47] reviewed these works to compare prediction models. In the construction treated here, two plates were bonded to the elastic core (urethane foam). In this case, the core has a role of wave transmission as a solid elasticity of the frame as well as a wave propagation medium of pores. The theory of wave propagation in such a porous elastic material was established by Biot [48] and discussed by Allard [49]. Examples of such an application to the core of a double plate system for sound transmission problems have been presented by Bolton et al. [50] and Sgard et al. [51].

In this study, the purpose was to clarify the fundamental mechanism of wave transmission through the core, especially the contribution ratio of the frame and pores of the urethane foam cushion to floor impact sounds. The frame was modelled as a Winkler-type frame, and pores were characterized as a wave propagation medium with dissipation. The model was used to compare the reductions in floor impact sounds as obtained by experiments and the contribution ratio of the pores to the elastic fibres for wave transmission from the flooring surface to the slab. The effect of the layer material on the sound insulation performance of the floor was also investigated.

4.2 Theory

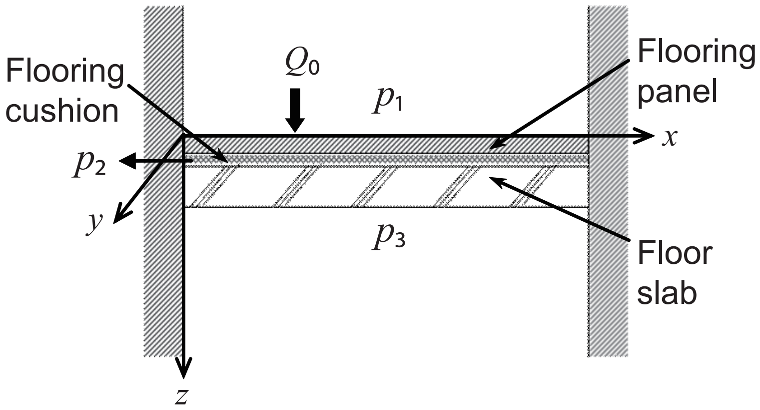

Figure 4.1: Analytical model of a slab with panel flooring excited by a point force Q_0.

Figure 4.1 introduces the analytical model used to investigate the reduction in floor impact sound due to the flooring system. Here, the flooring system, which is composed of the flooring panel and cushion, is directly connected to a floor slab. In this model, the multilayer structure is considered to have simply supported edges set in an infinite rectangular duct of size $a \times b$ m^2 on the x - y coordinate plane. From the source side, the flooring panel located at $z = 0$ is excited by point force $Q_0 exp(-i\omega t)$ at $(x_0,\ y_0)$. The second layer is a urethane foam cushion of thickness L. The inner surface of the duct is assumed to be rigid. Both sides of this structure and the pores of the urethane foam cushion are considered to be filled with the same fluid (air). Additionally, in steady state the time factor $exp(-i\omega t)$ is neglected, where ω is the angular frequency; k represents the wave number of air, and i is an imaginary unit.

The sound pressure p in each region of this structure and the displacement of the flooring panel and floor slab w can be expressed by using series expansions of each eigenfunction with unknown coefficients A_{mn}, B_{mn}, C_{mn}, D_{mn}, and W_{jMN}:

$$p_1 = \sum_{mn} A_{mn}\phi_{mn}(x,y)exp(-ik\beta_{mn}z), \tag{4.1}$$

$$\begin{aligned} p_2 = &\sum_{mn} B_{mn}\phi_{mn}(x,y)exp(-q_{mn}z)+ \\ &\sum_{mn} C_{mn}\phi_{mn}(x,y)exp(q_{mn}z), \end{aligned} \tag{4.2}$$

$$p_3 = \sum_{mn} D_{mn}\phi_{mn}(x,y)exp(ik\beta_{mn}z), \tag{4.3}$$

$$w_j = \sum_{MN} W_{jMN}\varphi_{MN}(x,y), \tag{4.4}$$

$$\beta_{mn} = \left\{ \begin{array}{ll} \sqrt{1-(\lambda_{mn}/k)^2}\,, & (\lambda_{mn}/k)^2 \leq 1, \\ i\sqrt{(\lambda_{mn}/k)^2-1}\,, & (\lambda_{mn}/k)^2 > 1, \end{array} \right. \tag{4.5}$$

$$\lambda_{mn}^2 = (m\pi/a)^2 + (n\pi/b)^2, \tag{4.6}$$

$$q_{mn} = \gamma\sqrt{1+(\lambda_{mn}/\gamma)^2}. \tag{4.7}$$

Here, q_{mn} is a variable limited by Re$\{q_{mn}\} \geq 0$ and Im$\{q_{mn}\} < 0$. γ is the propagation constant of the urethane foam. The flooring panel and floor slab are identified by subscript j (=1, 2). The eigenfunctions are $\phi_{mn} = \cos(m\pi x/a)\cos(n\pi y/b)$ $(m, n = 0,\ 1,\ 2,...)$, $\varphi_{MN} = \sin(M\pi x/a)\sin(N\pi y/b)$ $(M, N = 1,\ 2,\ 3...)$. The function ϕ_{mn} satisfies the following orthogonal condition:

$$\int_a^0 \int_b^0 \phi_{mn}(x,y)\phi_{m'n'}(x,y)dxdy = \left\{ \begin{array}{c} \frac{ab}{4}\epsilon_m\epsilon_n, m = m', n = n', \\ 0, otherwise, \end{array} \right. \tag{4.8}$$

where

$$\epsilon_m = \left\{ \begin{array}{l} 2, m = 1, \\ 1, m \neq 1, \end{array} \right. \quad \epsilon_n = \left\{ \begin{array}{l} 2, n = 1, \\ 1, n \neq 1. \end{array} \right. \tag{4.9}$$

Note that the eigenfunction of air in the x - y plane differs from the eigenfunction of panels. To solve this problem for simultaneous linear equations, a direct correlation must be made between the two eigenfunctions ϕ_{mn} and φ_{MN}. The relation is written below:

$$\varphi_{MN}(x,y) = \sum_{mn} \alpha_{MN}^{mn} \phi_{mn}(x,y), \tag{4.10}$$

where

$$\alpha_{MN}^{mn} = \frac{(\zeta_{Mm1} + \zeta_{Mm2})(\zeta_{Nn1} + \zeta_{Nn2})}{\epsilon_m \cdot \epsilon_n}, \tag{4.11}$$

$$\zeta_{Mm1} = \begin{cases} 0, (M+m-1=0), \\ \frac{1-(-1)^{M+m-1}}{(M+m-1)\pi}, (M+m-1 \neq 0), \end{cases} \tag{4.12}$$

$$\zeta_{Mm2} = \begin{cases} 0, (M-m+1=0), \\ \frac{1-(-1)^{M-m+1}}{(M-m+1)\pi}, (M-m+1 \neq 0), \end{cases} \tag{4.13}$$

$$\zeta_{Nn1} = \begin{cases} 0, (N+n-1=0), \\ \frac{1-(-1)^{N+n-1}}{(N+n-1)\pi}, (N+n-1 \neq 0), \end{cases} \tag{4.14}$$

$$\zeta_{Nn2} = \begin{cases} 0, (N-n+1=0), \\ \frac{1-(-1)^{N-n+1}}{(N-n+1)\pi}, (N-n+1 \neq 0). \end{cases} \tag{4.15}$$

The effect of the flooring cushion is divided into two individual parts in the calculation: the effect of elastic fibres and the effect of pores. w_1 is the displacement of the flooring panel, and w_2 is the displacement of the floor slab. σ is the stress of the flooring cushion. The force relation of the elastic fibres can be expressed as

$$\sigma = E\varepsilon + C\frac{\partial \varepsilon}{\partial t}, \tag{4.16}$$

$$\varepsilon = \frac{E(w_1 - w_2)}{L}. \tag{4.17}$$

Here, Young's modulus E (N/m^2) is directly related to the indentation hardness. C and ε are the damping coefficient and strain, respectively. By the two equations above, the stress can be written as

$$\sigma = \frac{E^*(w_1 - w_2)}{L}, \tag{4.18}$$

where

$$E^* = E(1 - i\eta). \tag{4.19}$$

E^* represents the complex Young's modulus. η is the loss factor. Considering the effects of elastic fibres and pores of the cushion, the equations of motion for the displacements $w_1(x, y)$ and $w_2(x, y)$ of the panels can be expressed as

$$\begin{aligned} D_1\nabla^4 w_1 - \rho_1 h_1 \omega^2 w_1 = p_1 \mid_{z=0} - p_2 \mid_{z=0} - \\ \frac{E(w_1 - w_2)}{L} + q_0\delta(x - x_0)\delta(y - y_0)\ , \end{aligned} \tag{4.20}$$

$$D_2\nabla^4 w_2 - \rho_2 h_2 \omega^2 w_2 = p_2 \mid_{z=l} - p_3 \mid_{z=l} + \frac{E(w_1 - w_2)}{L}. \tag{4.21}$$

where $\nabla^4 = (\partial^2/\partial x^2 + \partial^2/\partial y^2)^2$. D and ρ are the flexural rigidity (Nm) and density (kg/m^3), respectively. h_j (j = 1, 2) is the thickness of the panel (m). δ represents the delta function. The second term of the right-hand side of Eq. (4.20) is the effect of the pores. The third term is the effect of the elastic fibres. The boundary conditions at the interface of each layer can be expressed by particle velocities v_1 and v_2 and the displacement of the panel as

$$v_1 \mid_{z=0} = v_2 \mid_{z=0}, \tag{4.22}$$

$$v_1 \mid_{z=0} = -i\omega w_1, \tag{4.23}$$

$$v_2 \mid_{z=l} = v_3 \mid_{z=l} . \tag{4.24}$$

$$v_3 \mid_{z=l} = -i\omega w_2, \tag{4.25}$$

By substituting the expanded sound pressure, particle velocity, and displacement of the panel into Eqs. (4.20 - 4.21) and coupling with the boundary condition, the unknown coefficients can be determined. The acoustic power P radiated from this structure in the transmitted side can be calculated by

$$P = \int_S \frac{1}{2} Re\{p_3(x, y, L) \cdot v_3^*(x, y, L)\} dS. \tag{4.26}$$

Here, * represents the complex conjugate, and S is taken over the area $a \times b$. The sound insulation performance of the floor structure is evaluated by the difference in the level of power with and without the flooring system.

4.3 Experiment

To investigate the reduction in floor impact sound due to the flooring system, experiments were conducted by means of a simplified method to measure the floor impact sound using a light impact source.

4.3.1 Experiment setup

Figure 4.2 shows the measurement system. The flooring system was installed over an area of 930× 905 mm^2. A 9-mm-thick flooring panel was connected to the 150 mm slab of a reverberation room by a urethane foam cushion. The thickness of the urethane foam was 4 mm. A standard tapping machine was used as an impact source set on the specimen.

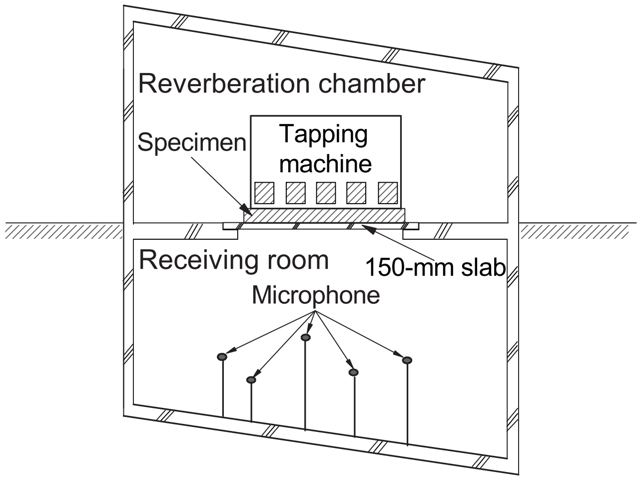

Figure 4.2: Configuration of measurement system (coupled reverberation rooms and a test specimen of the flooring with a tapping machine).

The sound pressure level (SPL) at five positions in the receiving room was measured in the 1/1-octave band and averaged; this value is denoted as L_F (dB). The SPL without flooring was also measured as the reference level L_R (dB). The reduction level in impact sound $\triangle L$ (dB) due to flooring was defined as the difference in level with and without the flooring system. The panel size is known to not have a significant effect on $\triangle L$ for cases of a limp panel with a soft urethane foam cushion. Under these measurement conditions, the corresponding calculations were performed for a double-plate system with dimensions of $3\times\ 4\ \mathrm{m}^2$ ($a = 3$ m, $b = 4$ m). The excitation point in the calculation was taken at an off-centre point ($3a/8$, $3b/8$) so that many modes would be involved. Any size beyond this as well as changes in the excitation point were confirmed to have less effect on $\triangle L$ when averaged in the 1/1-octave band. The properties of the flooring layer materials are presented in Table 4.1.

Table 4.1: Material properties of the layers

Layer	Young's modulus (N/m^2)	Density (kg/m^3)	Poisson ratio	Loss factor
Plywood panel	5.4×10^9	600	0.3	0.01
*Urethane foam	60N(hardness)	-	-	0.1
Concrete slab	2.7×10^{10}	2300	0.2	0.005
*The indentation hardness of the urethane foam is measured following JIS K6400(2004) [59]				

4.3.2 Empirical formula

The results of this experiment were compared with the analytical calculation to examine the roles of the elastic fibres and pores acting as wave transmission elements. In the calculation, five simple cases were considered to determine the effect of the pores. In the first case, only the elastic fibres were considered in the calculation. In other words, the effect of the pores was neglected. Figure 4.3 shows the calculated result for the sound power level (PWL) of the transmitted side caused by a 1 N force. The blue and black lines represent the results of the slab alone, and the green and red lines show the case with the flooring system. The blue and green lines are the raw data, and the black and red lines are the moving average over the 1/1-octave band interval. Figure 4.4 compares $\triangle L$ between the calculated and experimental results.

In another extreme case, only the wave transmission due to pores was considered. In other words, the effect of elastic fibres was neglected, and the effect of pores was considered as a sound wave propagating in air. Figure 4.5 shows the calculated result with and without the flooring system. Figure 4.6 compares $\triangle L$ between theory and the experimental results.

In the third case, both elements were considered equally. The calculated results and comparison of the theoretical and experimental $\triangle L$ values are shown in Figs. 4.7 and 4.8, respectively.

In the fourth case, the wave propagation model of pores was replaced

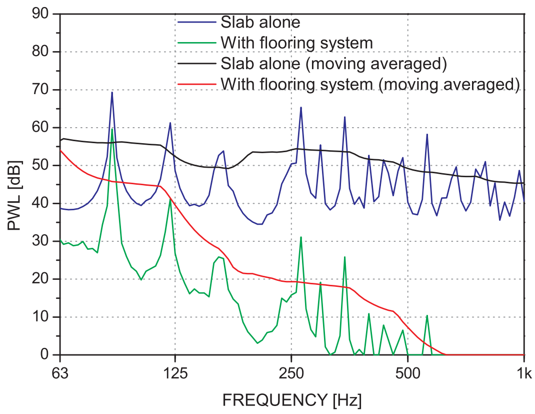

Figure 4.3: Comparison of power level between with and without the flooring system (elastic fibers element alone).

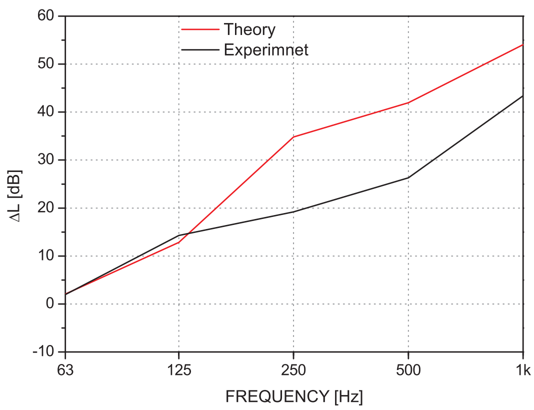

Figure 4.4: Comparison of $\triangle L$ between theory and experiments (elastic fibers element alone).

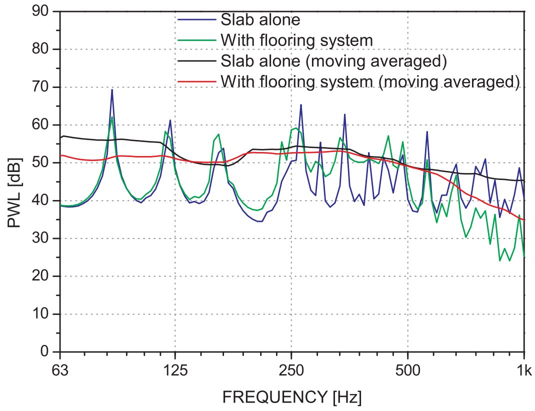

Figure 4.5: Comparison of power level between with and without the flooring system (pores element alone).

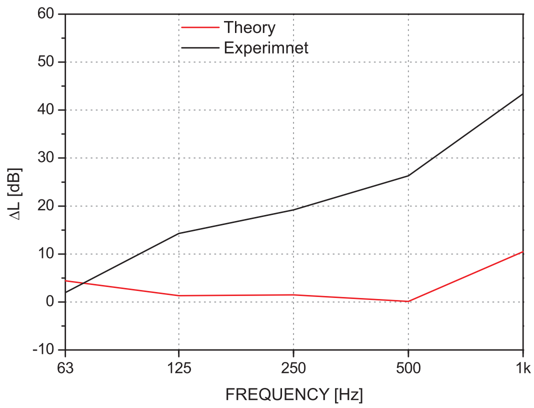

Figure 4.6: Comparison of $\triangle L$ between theory and experiments (pores element alone).

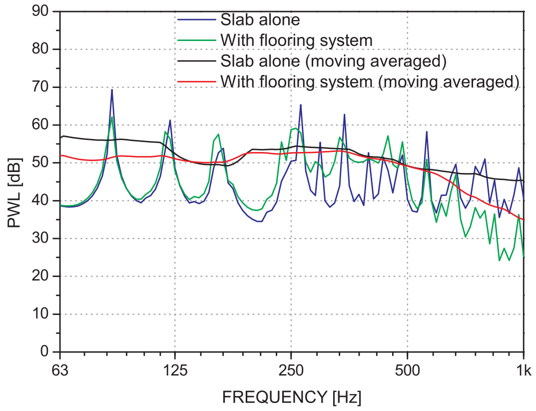

Figure 4.7: Comparison of power level between with and without the flooring system (both elastic fibers and pores elements equivalently).

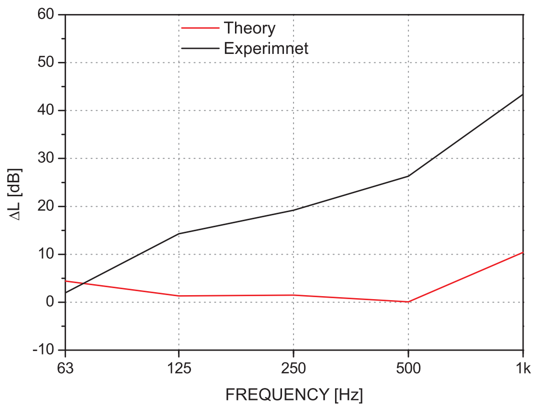

Figure 4.8: Comparison of $\triangle L$ between theory and experiments (both elastic fibers and pores elements equivalently).

with the equivalent spring model. The pores were considered as a spring without any energy loss. Figures 4.9 and 4.10 compare the results.

In the second to fourth cases, the calculated results were similar, but they were very different in the first case (elastic fibres alone). The pores were concluded to play a more important role than the elastic fibres when calculating the effect of porous materials. Therefore, the adverse effects may be caused by inadequate consideration of pores.

The above results showed that the results calculated by the proposed analytical model without any change to the contribution ratio of the wave transmission of pores to the elastic fibres do not agree well with the experimental results. However, changing the contribution ratio of pores to 2% caused the calculated results to align with the experimental results. Herein, it is considered as the empirical formula for evaluating the reduction of floor impact sound by means of a flooring system with porous materials. The contribution ratio of 2% means that the second term of the right-hand side of Eq. (4.20) and the first term of the right-hand side of Eq. (4.21) are multiplied by 0.02. The calculated results and $\triangle L$ between the theory and experimental results of the fifth case are shown in Figs. 4.11 and 4.12, respectively.

From the above results, it is concluded that the disparity between the theoretical and experimental values becomes increases as the contribution ratio of pores is increased when comparing all cases for $\triangle L$. For the equivalent spring model, where the pores are modelled as isolated cells, the disparity between the theoretical and experimental is the highest; however, the wave propagation model also showed similar results. Therefore, considering the pores as the medium of wave propagation is almost the same as considering the pores as the equivalent spring when the thickness of the porous layer is smaller than the wavelength. In the case of considering pores to be the only contributor, theoretical values do not agree with the experimental values. This

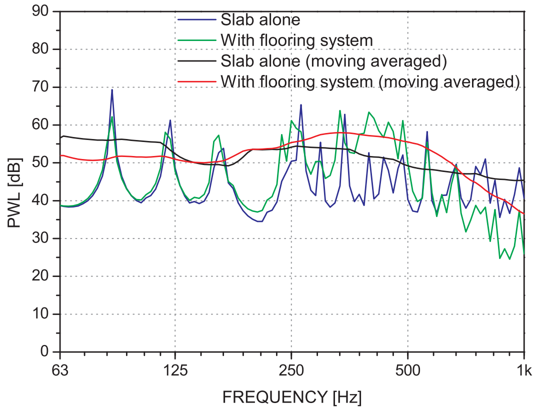

Figure 4.9: Comparison of power level between with and without the flooring system (equivalent spring model instead of wave propagation model of pores).

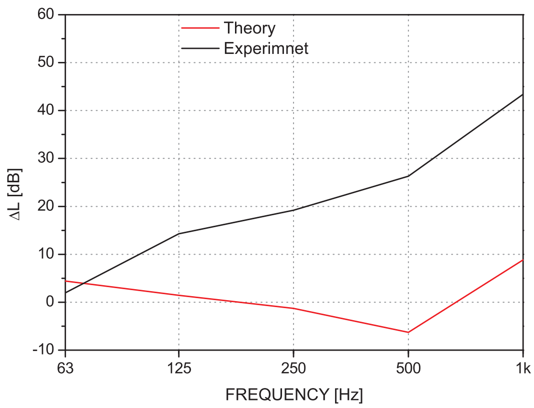

Figure 4.10: Comparison of $\triangle L$ between theory and experiments (equivalent spring model instead of wave propagation model of pores).

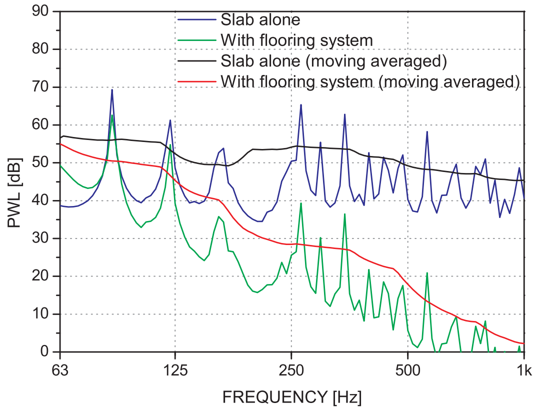

Figure 4.11: Comparison of power level between with and without the flooring system (pores element 2%).

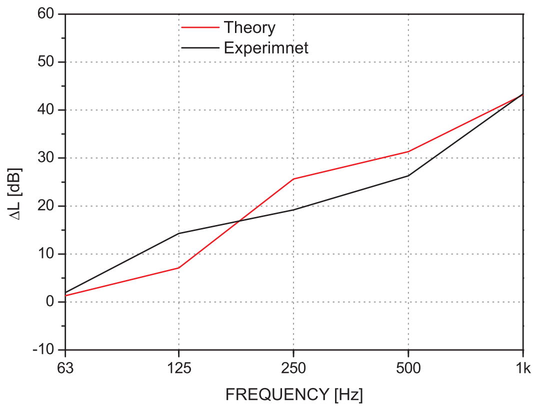

Figure 4.12: Comparison of $\triangle L$ between theory and experiment (pores element 2%).

may imply that some additional effect reduces the wave transmission in such a thin layer of a wave transmission medium. Although a 2% contribution ratio of pores was used in the calculation, this assumption needs to be clarified in order to establish an effective analytical model. This will be examined in future work. In the following section, the effect of the cushion is evaluated using the obtianed empirical formula.

4.4 Parametric study

4.4.1 The effect of thickness

The effects of varying the thickness of the cushion was investigated. Figure 4.13 shows the calculated result. The floor impact sound is significantly reduced by increasing the thickness of the cushion. From this result, it can be concluded that the sound insulation performance can be improved by thickening the urethane foam.

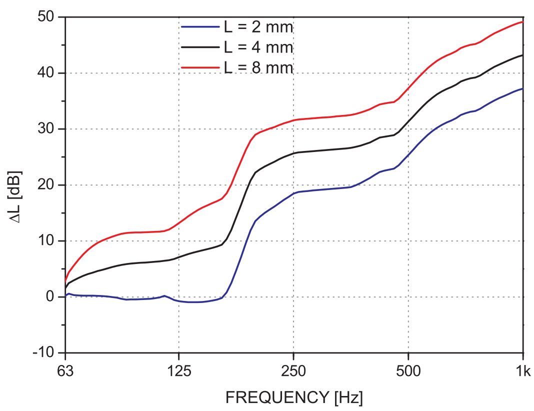

Figure 4.13: Effects of variation in thickness of the urethane foam on $\triangle L$.

4.4.2 The effect of hardness

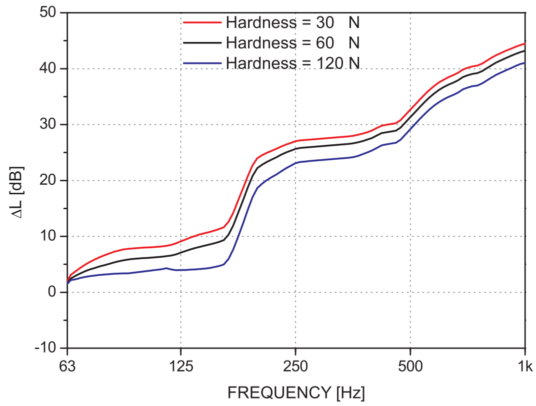

Figure 4.14: Effects of variation in hardness of the urethane foam on $\triangle L$.

Comparing to effect of thickness, a less significant change is observed when the hardness of the cushion is varied. The calculated result of different hardness of urethane foam is shown in Fig. 4.14. From this investigation, it is found that softening the urethane foam can slightly improve the sound insulation performance of flooring.

4.4.3 The effect of varying thickness and hardness with a constant ratio

The effect of varying the thickness and hardness on the feeling of walking over the floor was considered; the reduction in floor impact sound was examined for a constant hardness-to-thickness ratio. Figure 4.15 shows the results, which were similar to the results for thickness. $\triangle L$ was shifted to the safe side when both the thickness and hardness of the urethane foam were increased at a constant ratio.

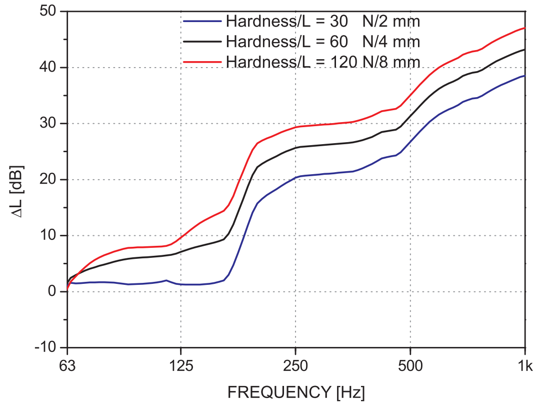

Figure 4.15: Effect of change in both hardness and thickness of the urethane foam on $\triangle L$ with the constant ratio.

4.4.4 The effect of flow resistance

Figure 4.16 shows the effect of flow resistance. With increasing flow resistance, although $\triangle L$ becomes larger in the range of 80 - 1000 Hz, the sound insulation performance is somewhat diminished at low

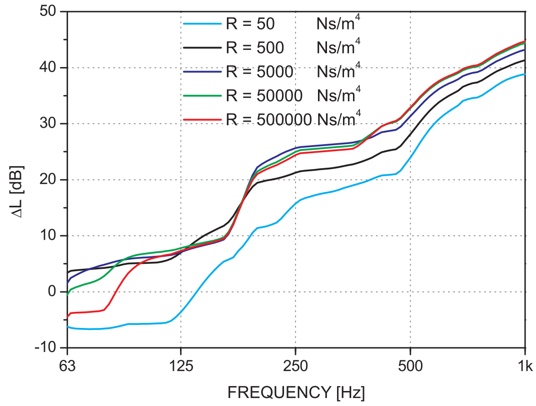

Figure 4.16: Effects of variation in flow resistance of the urethane foam on $\triangle L$.

frequencies. When the flow resistance exceeds a certain value, there is no significant change in $\triangle L$ at 80 - 1000 Hz. In this case, the value was about 5000 Ns/m^4.

4.4.5 The effect of loss factor

In addition, the effect of the loss factor was investigated, as shown in Fig. 4.17. There was no significant change when the loss factor was increased. Therefore, the effect of loss factor can be neglected.

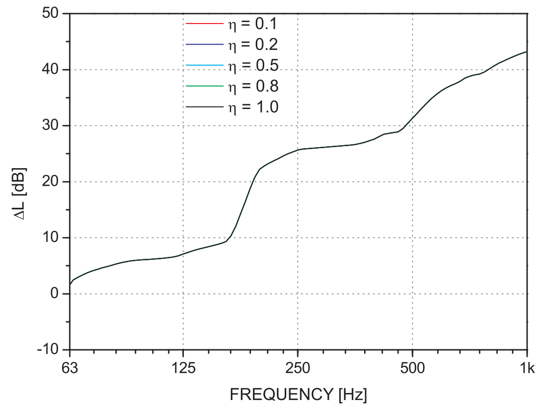

Figure 4.17: Effects of variation in loss factor on $\triangle L$.

4.5 Conclusion

To evaluate the insulation performance against lightweight floor impact sounds, a new analytical model was proposed that considers the effects of both elastic fibres and pores of a urethane foam urethane foam. The contribution ratio of the pores to the elastic fibres for wave transmission from the flooring surface to the slab was investigated with this model to compare the reduction in floor impact sounds between experimental and calculated results. Results calculated with a

2% contribution ratio of pores showed good agreement with the experimental results. In other words, increasing the rate of the pores' contribution had an adverse effect. By this investigation, an empirical formula is obtained.

The effects of the material properties of the porous material such as thickness, elastic hardness, flow resistance and loss factor on the insulation performance of the floor were studied by this empirical formula. A parametric survey on the changes in the properties of the urethane foam urethane foam showed that the sound insulation performance can be improved by softening or thickening the urethane foam. Changing both the hardness and thickness at a constant ratio provides an effective improvement. Although the obtained empirical formula can be applied for all porous materials , the value 2% contribution ratio of pores maybe change. Therefore, it is neccessary to validate this theory by experimental results of various porous material in further work.

CHAPTER. V

5 Conclusions

In this study, three effective approaches are proposed to improve the sound insulation performance of multi-layer structures in buildings. The first and second approach focus on resolving the sound insulation deficit due to the effect of mass-air-mass resonance which can be applied for glass window and partition walls. The third approach is proposed to reduce the light impact sound of floor. It can be used to floor and partition wall. It is concluded that these studies in this thesis included noise control of all component elements of building. In other word, sound insulation performance of the whole building can be improved by these three methods. The main achievements of each study are concluded as follows:

Study One

• The influence of MPPs on the performance of multi-layer structures is theoretically and experimentally investigated, and the applicability of MPPs to improve the sound insulation performance is discussed.

• The experimental results using an acoustic tube confirmed both the suppression effect of an MPP on the mass-air-mass resonance and the theory.

• The reverberation chamber experiments verified the actual sound insulation characteristics.

• By the investigation of parametric study, it is found that the best sound insulation performance depends on both diameter of the hole and pitch of the perforation.

This study demonstrated that perforating a panel on the transmission side of a multi-layer structure effectively prevents mass-air-mass resonance. Therefore, the method using MPPs is an effective approach to improve the sound insulation performance of multi-layer structures.

Study Two

• To suppress the effect of resonance and improve sound insulation performance, a new method using viscoelastic materials as connectors in double-panel structures was proposed.

• The possibility of improving the sound insulation degradation under normal and oblique incidences was investigated by evaluating the effects of the number, installation locations, and properties of the viscoelastic connectors.

• Calculated results for different number of connectors and locations revealed that the method of installing the damping material at the center of each vibratory element of odd-odd mode appeared to be more efficient than that in the case of even-even mode.

• Installation of the damping material improves the degradation due to mass-air-mass resonance and has a stable improvement effect over the whole frequency range of interest when the number of installations is relatively large.

• Results for the effect of damping ratio of the connector revealed that a damping ratio of more than 0.5 provides a stable improving effect over the frequency range of interest and improves degradation due to mass-air-mass resonance.

By theoretical investigation of this approach, the possibility of improving the sound insulation degradation was investigated. Therefore, it can be said that it may be an effective approach to improve the sound insulation performance of multi-layer structures. To verify the proposed theory and confirm the actual improvement of sound insulation performance, further investigation including some experiments is needed.

Study Three

- To evaluate the insulation performance against lightweight floor impact sounds, a new analytical model was proposed that considers the effects of both elastic fibres and pores of a urethane foam.
- The contribution ratio of the pores to the elastic fibres for wave transmission from the flooring surface to the slab was investigated by comparing this model with the experimental results. This leads to an empirical formula, which can be used for evaluating the effects of the material properties of the porous material such as thickness, elastic hardness, flow resistance and loss factor on the insulation performance of the floor.
- A parametric survey on the changes in the properties of the urethane foam urethane foam showed that the sound insulation performance can be improved by softening or thickening the urethane foam.
- Changing both the hardness and thickness at a constant ratio provides an effective improvement.

Although the obtained empirical formula can be applied for all porous materials , the value 2% contribution ratio of pores maybe change. Therefore, it is necessary to validate this theory by experimental results of various porous material in further work.

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Research performance

Research paper (Journal):

1. Masahiro Toyoda, RuiLin Mu, Daiji Takahashi, " Relationship between Helmholtz-resonance absorption and panel-type absorption in finite flexible microperforated-panel absorbers "Applied Acoustics, 71, 315-320, (2010).
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