Development of a noise reduction system with piezoelectric material to transmitted noise
(Structure for improvement of the noise reduction effect)

Katsuya YAMAMOTO[1]; Akiyoshi ISHIMORI[2]; Hiroyuki SATO[3]; Mineyuki ASAHINA[4]

Railway Technical Research Institute, Japan

ABSTRACT

We previously developed a noise reduction system consists of piezoelectric material attached on aluminum plate to reduce transmitted noise; the features and principles of this system were presented at the internoise2011. In that paper, each aluminum plate with a piezoelectric material (hereinafter "noise insulation panel: NIP") was assumed to have perfect flat plate and same size and therefore has the identical vibration characteristics. However, when panels were bonded to a frame to form a single structure, their vibration characteristics actually differed, and the noise reduction capability of the system became lower. To solve this problem, we have developed a method in which planar tension is applied to each NIP via a simple frame structure. We have verified that this method improves the noise reduction and allows the target frequency to be tuned. In this paper, we describe the features of this new structure, and present the results of our numerical analysis and experimental tests.

Keywords: Noise Reduction, Piezoelectric material, Transmitted noise, Tension
I-INCE Classification of Subjects Number(s): 38.5.1

1. INTRODUCTION

There is demand for a new method for minimizing noise in our living and working environment; reducing low frequency noise or vibration using lightweight materials or equipment with piezoelectric materials presents a particular challenge(1-7). We have developed a new noise reduction system that employs an array of noise insulation panels (hereinafter "noise reduction system") to counteract transmitted noise(8-10). This system consists of a noise reduction unit—an array of small flat panels to which piezoelectric elements are bonded—and a control circuit in the form of an equivalent inductance circuit designed around operational amplifiers. The capacitance of the piezoelectric elements and the inductance of the control circuit together form a resonance circuit, such that the noise passing through the noise insulation panels (NIPs) is reduced within a given frequency range centered around the circuit's resonant frequency.

This noise reduction system was investigated under conditions where noise waves incident to the NIPs were planar, and all panels vibrated with the same amplitude and phase. Under these conditions, when all piezoelectric elements on the panels are connected in series or parallel, the transmitted noise is expected to be reduced by the control circuit. We verified experimentally that the transmitted noise is reduced when the length of the panel is shorter than the wavelength of the incident noise in a lower frequency range containing a few hundred frequencies. In addition, we established that the optimum control conditions occur when the (1,1) modal vibration frequency of the NIPs and the resonant frequency of the electrical circuit correspond with the target frequency. This means that the NIPs have uniform vibration characteristics.

However, when the NIPs are adhesive-bonded to a frame to construct a prototype noise reduction unit (i.e., a combination of an NIP and the frame), it is difficult to maintain consistent vibration characteristics across multiple NIPs. Therefore, it is more realistic to assume that the vibration

1 katsu@rtri.or.jp
2 ishimori.akiyoshi.09@rtri.or.jp
3 sato.hiroyuki.06@rtri.or.jp
4 asahina@rtri.or.jp

characteristics of the NIPs are not uniform, particularly in a handmade prototype. Although control circuit inductance can be easily adjusted using variable resistance, it is much more difficult to adjust the vibration characteristics of the panels once they are fixed to the frame.

This paper describes a prototype structure in which the (1,1) modal vibration frequency of the NIPs can be adjusted after being attached to a frame. Additionally, the above effects and the improved noise reduction characteristics observed during acoustic excitation testing are detailed.

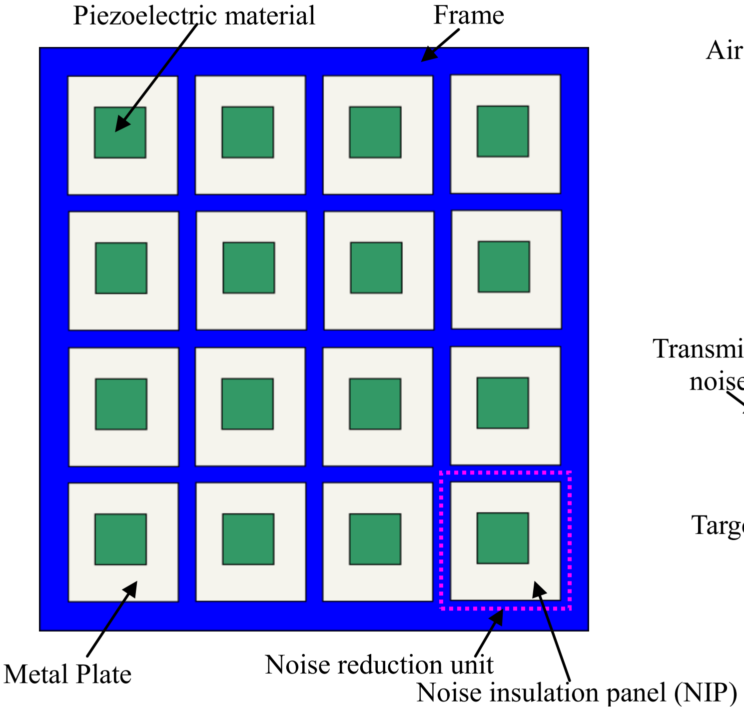
2. OUTLINE OF THE NOISE REDUCTION SYSTEM

2.1 Features and Principles of the Control (9)

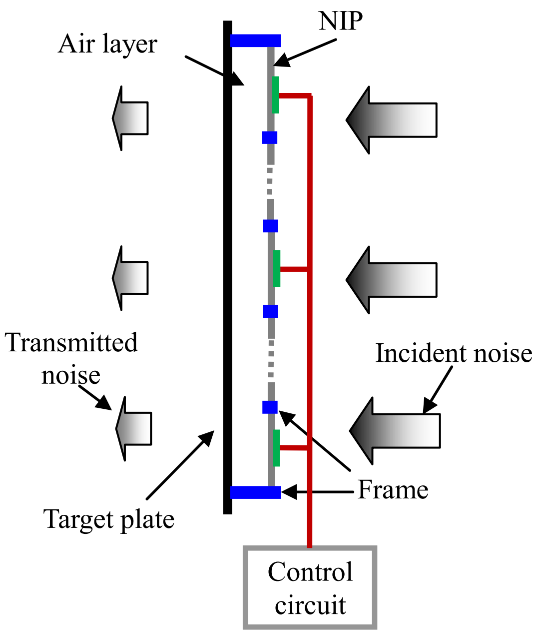
The features of the noise reduction system are shown in Fig.1. The system consists of a control circuit and an array of NIP (metal panels bonded to piezoelectric materials) and attached to a frame. All piezoelectric materials in an NIP are wired to each other in series or parallel and connected to the control circuit. This controller is an equivalent inductance circuit based on a simple analog design using operational amplifiers. Since piezoelectric material is capacitive, a resonance circuit is created between the piezoelectric materials and the control circuit.

The principle of control is as follows: when the incident noise is input to the noise reduction unit, the NIPs are excited at a frequency range centered around the resonance frequency, and the piezoelectric materials generate a voltage. As the generated voltage is input to the control circuit, a voltage anti phase to the generated voltage returns to the piezoelectric materials, causing an anti-phase force to interact with the NIP. The original vibration and the anti-phase force cancel each other out, suppressing the vibration of the NIP. As the vibration is reduced, the noise transmitted through that panel also reduces. This vibration reduction effect, which is produced when an inductive element is connected to a piezoelectric element, is reported to be equivalent to the effects of applying a mechanical dynamic vibration absorber. When several piezoelectric materials are wired to the noise reduction unit, they act as an equivalent capacitance and the vibration of all NIPs is reduced by the uniform anti-phase voltage input to the piezoelectric materials from the control circuit.

When the noise reduction unit is installed on any target panel, a layer of air is formed between the two. As the acoustic power in the air layer is suppressed by the noise reduction system, the noise transmitted through the target plate is also reduced, regardless of its vibration characteristics. Figure 1(a) suggests that the system reduces only the transmitted noise when the panel is placed on the



(a) Noise reduction panel



(b) Cross-sectional view

Fig.1 Outline of the noise reduction system incident noise side of the target plate and structure bone noise when the panel is installed on the transmitted noise side.

This noise reduction system has several advantages:

(1) It is not necessary to identify the vibration characteristics of the target panel or the sound characteristics of the transmitted noise.

(2)Hence, this system can reduce transmitted noise without requiring sensors such as microphones or acceleration pickups.

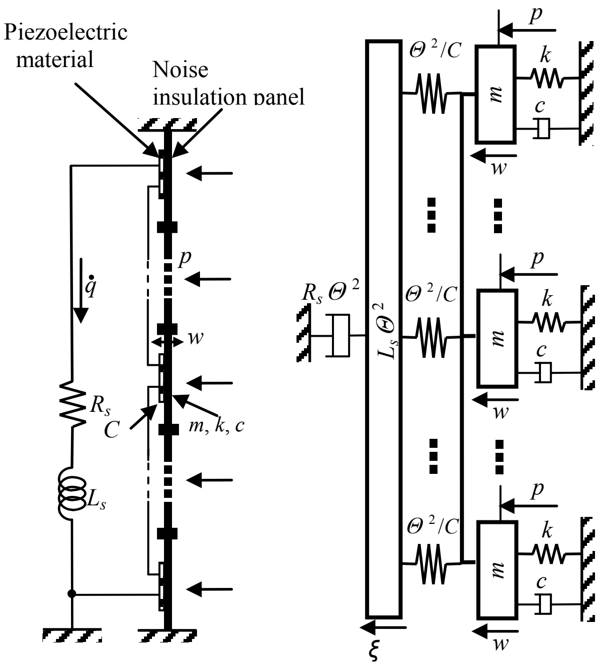
(3)Since the controller is composed of an operational amplifier circuit and the piezoelectric material is driven according to the input voltage, power consumption is low.

(4)As the noise reduction panel and the control circuit can be mass produced, this system can be realized inexpensively.

2.2 Model of the Noise Reduction System (9)

Several studies have dealt with noise reduction systems that use piezoelectric material. Since our noise reduction system implements control by connecting an electrical circuit to piezoelectric materials, the whole system contains both a mechanical subsystem (an NIP) and an electrical subsystem (the piezoelectric materials and control circuit). In previous research, it was demonstrated that these system can be converted to an equivalent mechanical system(11-12).

Figure 2(a) shows the actual circuit formed when all piezoelectric materials are connected in series, and Fig. 2(b) is the equivalent mechanical system. The inductance L_s , resistance R_s , and capacitance C in the actual circuit of Fig. 2(a) can be converted to an equivalent added mass $L_s\Theta^2$, an equivalent damping element $R\Theta^2$, and an equivalent stiffness element Θ^2/C , as shown in Fig. 2(b), using the electromechanical coupling coefficient Θ of the piezoelectric material, where p is the amplitude of the incident noise pressure, w is the displacement of an NIP, m is the modal mass, c is the modal damping coefficient, and k is the modal stiffness of an NIP.



(a)Actual model (b)Equivalent mechanical model

Fig.2 A model of the noise insulation panel with inductance circuit

In a previous study, the vibration reduction effect Δw of an NIP when the capacitance C of all piezoelectric materials is assumed to be the same has already been obtained against the displacement w_0 without control, as shown below.

$$\Delta w = \frac{|w|}{|w_0|} = \sqrt{\frac{\left(G_{iR}{}^2 + G_{iI}{}^2\right)}{\left(G_{iR} + G_{eR}\right)^2 + \left(G_{iI} + G_{eI}\right)^2}} \ , \tag{1}$$

$$w_0 = \frac{p}{-m\omega^2 + j\omega c + k}, \tag{2}$$

$$G_{iR} = N\left(1/\Omega^2 - 1\right), \tag{3}$$

$$G_{iI} = 2\varsigma N/\Omega,, \tag{4}$$

$$G_{eR} = \mu/\Omega_e^{\,2} + \frac{\mu\left(1-1/\Omega_e^{\,2}\right)/\Omega_e^{\,4}}{\left(1-1/\Omega_e^{\,2}\right)^2 + \left(2\varsigma_e/\Omega_e\right)^2}, \tag{5}$$

$$G_{eI} = \frac{2\mu\varsigma_e/\Omega_e^{\,5}}{\left(1-1/\Omega_e^{\,2}\right)^2 + \left(2\varsigma_e/\Omega_e\right)^2}, \tag{6}$$

$$\mu = m_e/m, \tag{7}$$

$$\Omega = \omega/\omega_1, \tag{8}$$

$$\Omega_e = \omega/\omega_e, \tag{9}$$

$$m_e = L_s\Theta^2 : \text{in series}, \quad L_p\Theta^2 : \text{in parallel}, \tag{10}$$

$$\zeta = \frac{c}{2m\omega_1}, \tag{11}$$

$$\zeta_e = \frac{R\Theta^2}{2m_e\omega_e}, \tag{12}$$

where j is the imaginary unit, ω is the angular frequency, ω_1 is the (1,1) modal natural angular frequency of an NIP, and ξ is displacement of the equivalent added mass. The equivalent electrical mass m_e in Eq. (10) is selected depending on whether the piezoelectric materials are connected in series or parallel. The combined capacitance C_s of the piezoelectric elements is related to inductance L_s because of the formation of a resonance circuit between the combined capacitance C_s and inductance L_s, where ω_{es} denotes the resonant angular frequency (hereinafter electric resonance frequency).

The natural frequency of (1,1) mode is separately derived as follows(13):

$$f_{mn} = \frac{1}{2\pi}\left\{\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right\}\sqrt{\frac{D}{\rho h}}. \tag{13}$$

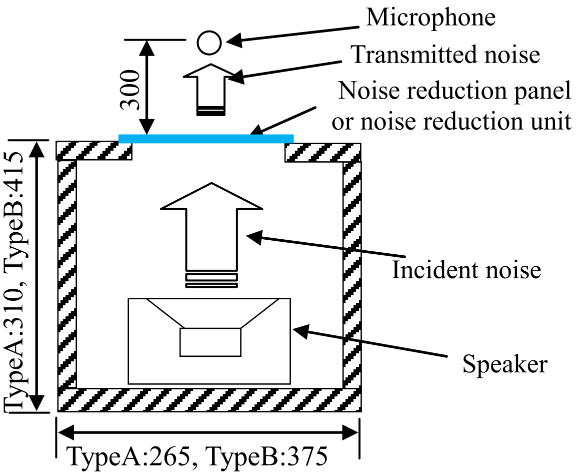


Fig.3 Experimental setup
(TypeA: Smaller Box, TypeB: Larger Box)

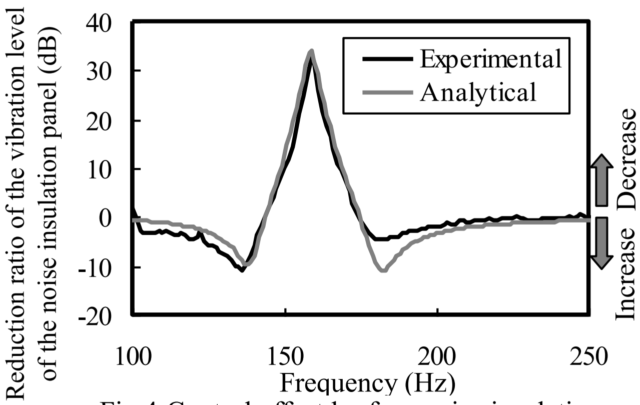


Fig.4 Control effect by four noise insulation panels (the piezoelectric loudspeaker)

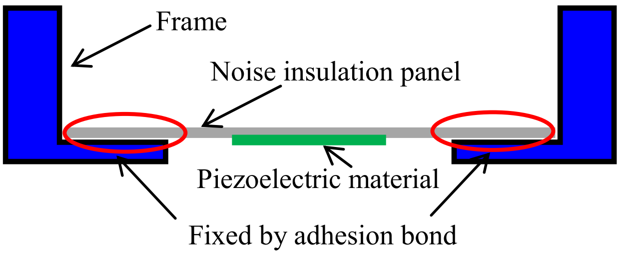


Fig.5 Cross-sectional view of the conventional structure of a noise insulation unit

Here, a and b are the length and breadth of a rectangular plate, ρ is density, D is flexural rigidity, and h is thickness.

On the basis of the above theory, we have confirmed the noise reduction capability of our design with a system using a piezoelectric loudspeaker (Panasonic, WM-R57A:50*40mm) as an NIP. In this test, the noise reduction array (or the noise reduction unit) was set on a box in which the loudspeaker was installed, as shown in Fig. 3, and a microphone was used to measure the transmitted noise. We verified that the vibration characteristics of the NIPs could be evaluated using the frequency response function of the voltage generated by a piezoelectric material, taking the sound pressure level in the box as the incident noise. In the frequency range centered at the natural frequency of (1,1), evaluation of vibration characteristics was largely based on the transmitted noise. Figure 4 shows a comparison between the noise reduction capability of the experimental system and the analysis based on Eqs. (1)–(12). Since the natural frequency of (1,1) mode was about 160 Hz, the vibration was suppressed by more than 30 dB. This noise reduction system is considered equivalent to a dynamic vibration absorber. Thus, the noise in the side bands increased in accordance with that theory.

Although this piezoelectric loudspeaker is effective as an NIP, its vibration characteristics cannot be modified. As a next step, we tried to develop an NIP using an aluminum plate and piezoelectric materials to achieve noise reduction at arbitrary frequencies. This system comprised ceramic piezoelectric materials attached to aluminum plates adhesive-bonded to an ABS frame. Figure 5 shows a cross-sectional view of that unit. Transmitted noise centered at the natural frequency of (1,1) mode of the NIP was reduced sufficiently.

3. PROBLEMS TO BE SOLVED

For expanding the noise reduction area, we manufactured a larger noise reduction panel composed of four NIPs, as shown in Fig. 6; its dimensions are provided in Table 1. This unit was mounted on a test box, as shown in Fig. 3, and the vibration characteristics of the NIPs were measured. The results are shown in Fig. 7.

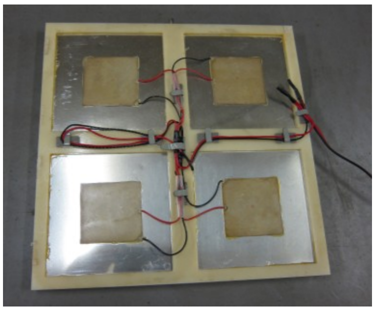


Fig.6 A previous noise reduction panel with four noise insulation panels

Table 1 Dimensions of a previous noise reduction panel

Size of a NIP	100*100 mm
Size of a piezoelectric material	50*50 mm
Thickness of an aluminum plate	0.3 mm
Thickness of a piezoelectric material	0.3 mm
Material of the frame	ABS

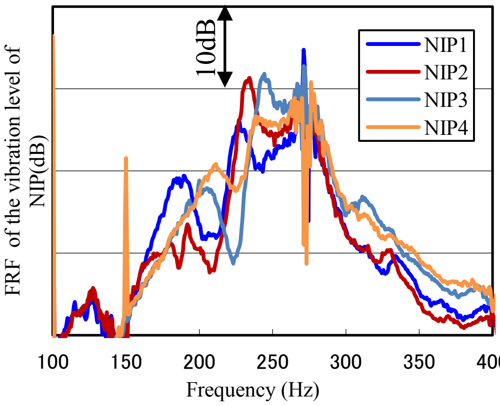


Fig.7 Vibration characteristic of the previous NIP

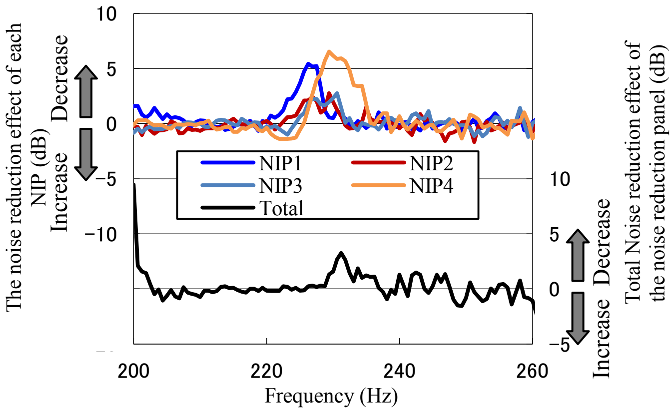


Fig.8 Noise reduction effect (Total: right axis)

It can be seen that the vibration peaks of all NIPs appeared between 220 and 270 Hz, so the vibration around this frequency range was dominated by the (1,1) vibration mode. However, the vibration characteristics were not consistent, suggesting that when the NIPs are fixed to a frame with the same specifications and procedure, their vibration characteristics differ.

Under these conditions, the two piezoelectric materials were connected in series, and two sets were connected in parallel. Finally, this set of piezoelectric elements was connected to the control circuit. By measuring the transmitted noise at a distance of 300 mm from the noise reduction unit and very close to the NIP, we compared the frequency response functions of the transmitted noise and the input noise with and without control. The target frequency was 227.5 Hz, and the inductance of the control circuit was tuned to that frequency. The noise reduction effects are shown in Fig. 8. While the noise transmitted through NIPs 1 and 4 were reduced by more than 5 dB, noise reduction by panels 2 and 3 was almost 3 dB. We consider this difference in noise reduction to be influenced by the vibration characteristics of the NIPs. In particular, when an NIP has slack areas and is not flat, the control force generated by the piezoelectric material might not propagate across the entire area of the plate, such that the effects required in order for the plate to stiffen do not appear.

Since such variations in NIP vibration are considered inevitable when a thin NIP is fixed to the frame by adhesive bonding, the importance of creating uniform vibration characteristics is clear.

4. SOLUTIONS TO THE PROBLEM

As discussed above, in this noise reduction system, it is important that NIPs vibrate the same vibration characteristics to produce the noise reduction effect sufficiently. Moreover, the target frequency and the natural frequency of (1,1) mode must coincide. Otherwise, the noise will not be sufficiently reduced. However, preserving these vibration characteristics while fixing the NIPs to the frame is difficult, necessitating a means for adjusting the NIPs' vibration characteristics after fixation.

We propose a method for applying pre-tension to each NIP as a countermeasure against the problem described above. Our method is based on the following two effects. First, the vibration characteristics of the plate should improve if the NIP is fixed loosely to the frame according to the fixation procedure. Second, the natural frequency of (1,1) mode can be adjusted because of the well-known effect of tension on vibration characteristics.

The relationship between vibration and tension in a rectangular plate is known to be as follows. In a rectangular plate with all four edges simply supported, the natural frequency of (1,1) mode with tension is given as follows(14):

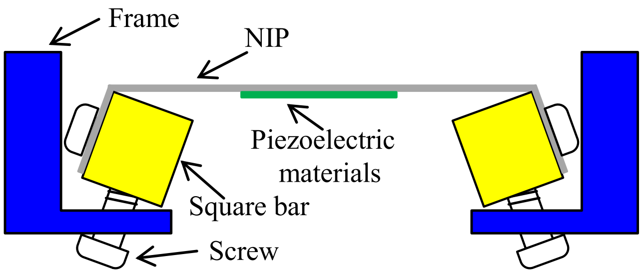
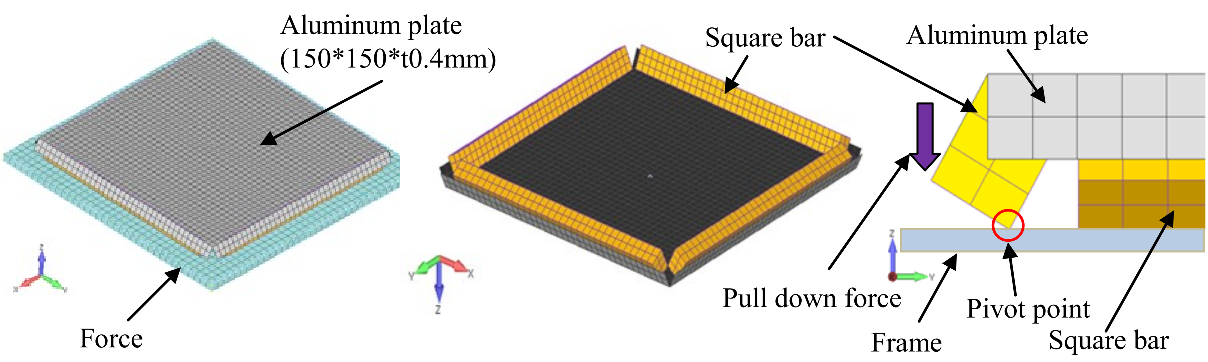
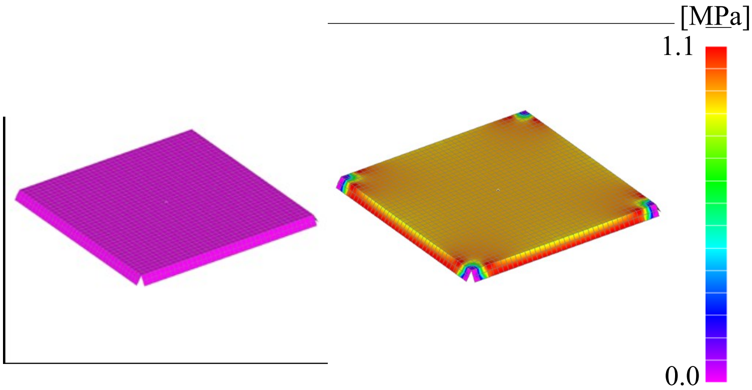


Fig.9 A cross-sectional view of the new structure of a noise reduction panel



(a) Top-down view (b) Bottom-up view (c) Cross-sectional view

Fig.10 A numerical analysis model of a NIP



(a) 0 N
(Initial condition)

(b) 500 N pull

Fig.11 Result of the numerical analysis

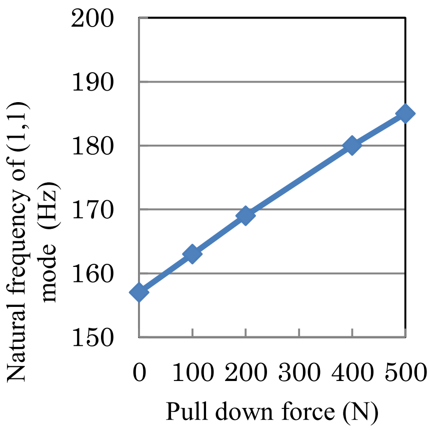


Fig.12 Change in the (1,1) modal vibration frequency

$$f_{mn} = \frac{1}{2\pi}\sqrt{\frac{1}{\rho h}\left[D\left\{\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right\}^2 + N_1\left(\frac{m\pi}{a}\right)^2 + N_2\left(\frac{n\pi}{b}\right)^2\right]}. \tag{14}$$

We devised a structure for easily applying pre-tension to an NIP. A cross-sectional view of the NIP is shown in Fig. 9. The NIP is attached to a frame via four square bars, which are tilted on the frame to be slightly angled, such that the angle of the bar and the pre-tension of the NIP can be adjusted through the tightening force of the screws. In addition, the amount of pre-tension can be controlled by adjusting the angle or torque of the screws.

A numerical analysis was carried out to investigate pre-tension characteristics and changes in the natural frequency of (1,1) mode in an NIP. Figure 10 shows a model of the analysis. The NIP was thin aluminum, the square bars were Bakelite specifically, and the frame was assumed to be a rigid body. In this analysis, pre-tension was applied to the NIP by a force that pulls one side of the square bar down to the frame, instead of the tightening force of the screw. Figure 11 shows the stress exerted on the NIP by the force from the square bar. We confirmed that when the square bar applies this force, pre-tension is exerted across the entire area of the NIP. Next, we calculated the natural frequency of (1,1) mode with the applied downward force. Figure 12 shows the relationship between the downward force and the natural frequency of (1,1) mode. Having verified that the natural frequency of (1,1) mode increases in proportion to the downward pulling force, it appears that the natural frequency of (1,1) mode can be adjusted by applying pre-tension to the NIP.

5. VERIFICATION OF THE TUNING METHOD OF NATURAL FREQUENCY OF THE NIP

We manufactured a prototype NIP, shown in Fig. 13. The noise reduction unit was mounted on the test box, as shown in Fig. 3, and the relationship between the natural frequency of (1,1) mode of the NIP and the rotation angle of the screw was evaluated. The result is shown in Fig. 14. Here, “Initial” represents the condition that the head of a screw shown in Fig.9 lightly touched on the frame, i.e. only a little pre-tension was applied to the NIP.

The natural frequency of (1,1) mode at the initial angle was 203Hz, as shown in Fig. 14. The profile of the vibration characteristics changed to become chevron-shaped compared with the vibration characteristics in Fig. 7, and the peak became distinct. As a reason of the improvement to the vibration characteristic without applying per-tension, the NIP might to be a near-flat condition by bending the aluminum

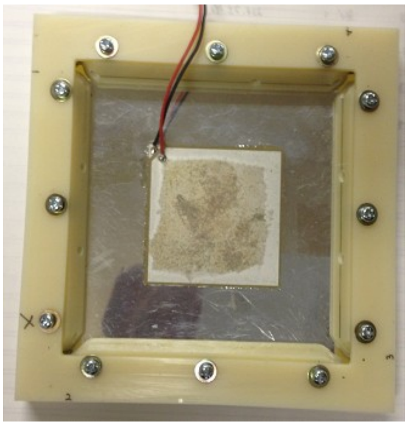


Fig.13 The new type noise reduction unit

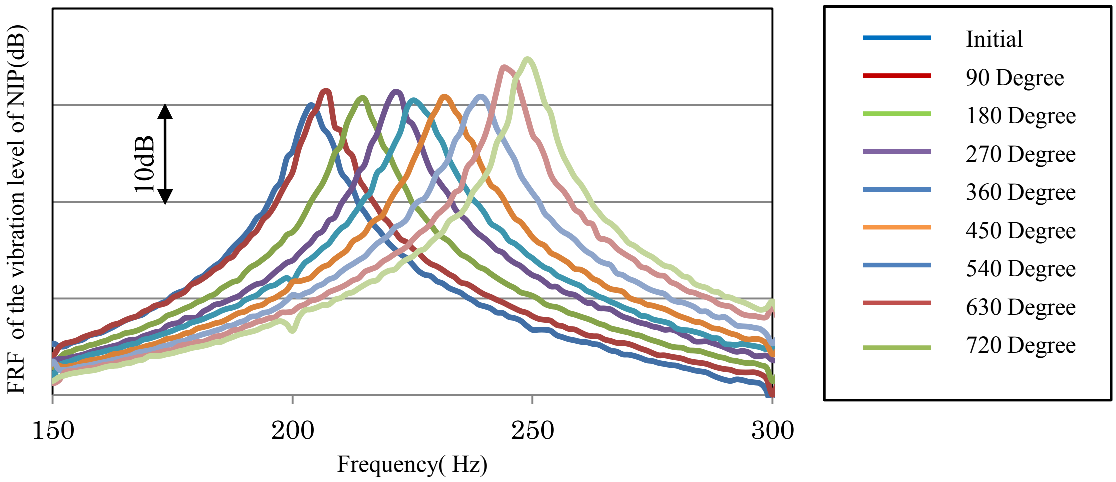


Fig.14 Change of the vibration characteristics of a NIP

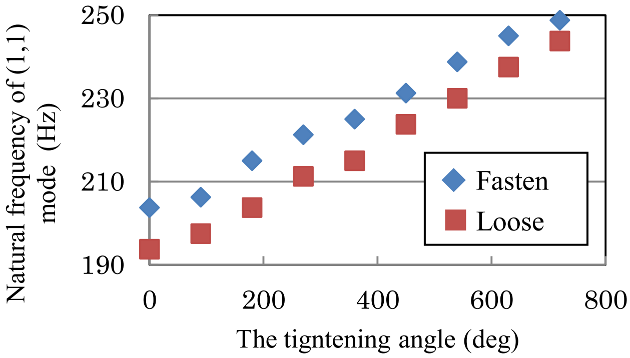


Fig.15 The change of the (1,1) modal vibration frequency

plate at the edge of the square bar.

Although the natural frequency of (1,1) mode hardly changed between the initial angle and 90° of tightening, it did gradually increase as the screw was tightened, finally reaching 248 Hz in two turns of tightening (720° of tightening), i.e., it increased by around 50 Hz. The relationship between the screw angle and the natural frequency of (1,1) mode with tightening and loosening is shown in Fig. 15. Although hysteresis was apparent in some ranges, the natural frequency of (1,1) mode changed in proportion to the screw angle, thus demonstrating the feasibility of applying pre-tension to the NIP and adjusting the natural frequency of (1,1) mode according to the tightening angle.

6. IMPROVEMENT OF THE NOISE REDUCTION PERFORMANCE BYTUNING THE NOISE REDUCTION PANELS

In the previous section, it was verified that the natural frequency of (1,1) mode could be adjusted by applying pre-tension to the NIP. To demonstrate the application of this method, we manufactured a noise reduction panel with four intentionally differently sized NIPs, and with the natural frequency of (1,1) mode tunable by means of the tightening force of the screws on the bar. Noise reduction effects were evaluated after connecting an inductance circuit for control.

Figure 16 shows the noise reduction unit, and Table 2 shows its specifications. The vibration characteristics of the NIPs in the initial condition, with the screws completely unfastened, are shown in Fig. 17. The peak frequency appeared at around 140 Hz for NIPs 1 and 2, and at around 160 Hz for NIPs 3 and 4. In theory, if the size of an NIP is larger, the peak frequency will be lower, and vice versa. However, the peak frequencies appeared in two frequency ranges due to subtle differences in conditions.

We tightened the screws of NIP 4 with a small amount of torque and tuned the natural frequency of (1,1) mode to 170 Hz. Pre-tension was incrementally applied to NIPs 1 to 3, and the (1,1) modal frequencies of all NIPs were tuned to 170 Hz, as shown in Fig. 18.

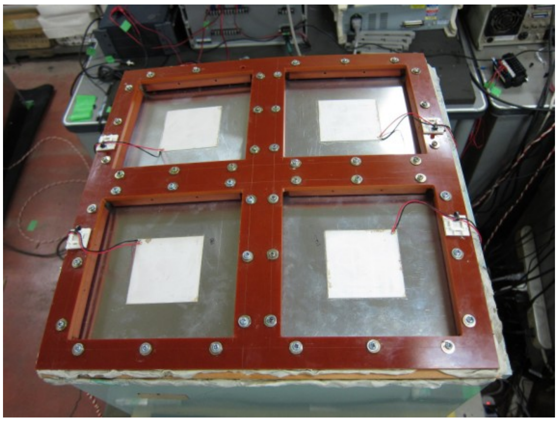


Fig.16 The new type noise reduction panel with four NIP

Table 2 Specification of a new noise reduction panel

Size of a NIP 1	150*150 mm
Size of a NIP 2	148*148 mm
Size of a NIP 3	146*146 mm
Size of a NIP 4	144*144 mm
Size of a piezoelectric material	63*63 mm
Thickness of a aluminum plate	0.4 mm
Thickness of a piezoelectric material	0.4 mm
Material of the frame	Bakelite

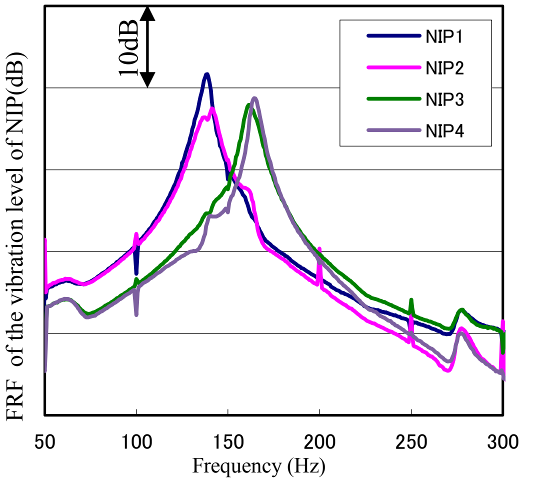


Fig.17 Initial vibration characteristic of each NIP

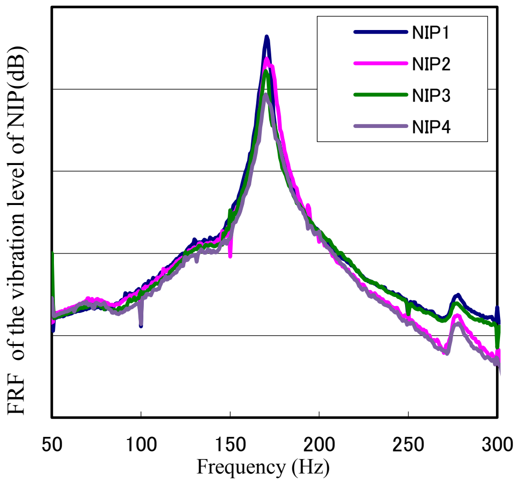


Fig.18 Vibration characteristic of each NIP after tuning

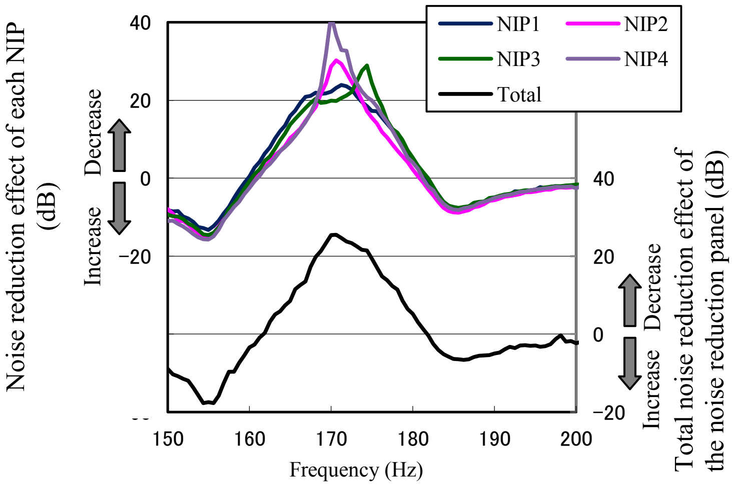


Fig.19 Noise reduction effect of new type NIP (left axis) and the noise reduction panel (right axis)

Finally, four piezoelectric materials were wired to each other (in a 2-2 series–parallel configuration) and connected to an inductance circuit. The inductance was tuned to an electrical resonance frequency of 170 Hz, and the effect upon noise transmission was measured. The result is shown in Fig. 19. The transmitted noise was suppressed by more than 20 dB by each NIP, with a total noise reduction of about 25 dB. The noise increased at 150–160 Hz or 180–200 Hz in agreement with the dynamic vibration absorber theory. The structure was thus demonstrated to offer improved noise reduction characteristics.

7. CONCLUSION

We confirmed that the difference of the natural frequency of the NIP exert influence to the noise reduction performance, and proposed the structure to adjustable the vibration characteristics of the NIP.

1. When an NIP is fixed to a frame with an adhesive bond, it will almost inevitably affect the vibration characteristics of the NIP, and it is impossible to subsequently tune those characteristics.
2. Variation of natural frequency of the (1,1) mode of the NIP causes performance degradation of the noise reduction system.
3. A noise reduction panel with thus altered vibration characteristics cannot greatly reduce the transmitted noise.
4. By applying pre-tension to each NIP, the vibration characteristics improve significantly
5. A structure that can easily apply pre-tension to the NIPs is presented.
6. The vibration characteristics of the noise reduction unit correspond with the different vibration characteristics of the NIPs, and significant improvements in transmitted noise are demonstrated.

REFERENCES

1. Wu SY. Piezoelectric shunts with a parallel R-L circuit for structural damping and vibration control. Proc SPIE 1996; 26 Jan- 2 Feb 1996; San Diego, USA 1996.p.256-269.
2. Fein OM. A model for piezo-resistive damping of two-dimensional structures, Journal of sound and vibration.2008;310:p.865-880.
3. Fleming AJ, Moheimani SO. Synthesis of optimal piezoelectric shunt impedances for structural vibration control. Proc SPIE 2004; 15-18 Mar 2004; San Diego, USA 2004. p.516-526.
4. Fleming AJ, Behrens S, Moheimani SO. An Autonomous Piezoelectric Shunt Damping System, Proc SPIE 2003; 2-6 Mar 2003; San Diego, USA 2003. p.207-216.
5. Kim SM, Wang S, Brennan MJ. Dynamic analysis and optimal design of a passive and an active piezo-electrical dynamic vibration absorber, Journal of sound and vibration.2011;330:p.603-614.
6. Casadei F, Dozio L, Ruzzene M, Cunefare KA. Periodic shunted arrays for the control of noise radiation in an enclosure, Journal of sound and vibration.2010;329:p.3632-3646.
7. Bein T, Elliott S, Ferralli L, Casella M, Meschke J, Saemann E, Nielsen FK, Kropp W. Integrated Solutions for Noise & Vibration Control in Vehicles. Procedia - Social and Behavioral Sciences. 2012; 48:p.919-931.
8. Yamamoto K, Asahina M, Mamada S, Sato D. Development of a Noise Reduction System with Piezoelectric Material to Transmitted Noise (Feature and Control Principle). INTER-NOISE 2011; 4-7 September 2011; Osaka, Japan 2011. CD-ROM.
9. Yamamoto K, Tanaka N, Asahina M. Development of a System to control noise transmitted through internal panels of railway vehicle (2nd report, Proposal for a Noise Control System Arrayed the Noise Insulation Panels). Transactions of the Japan Society of Mechanical Engineers, Series C. 2009; 75(752): p.890-891.
10. Yamamoto K, Asahina M, Mamada S, Sato D. Development of a System using Piezoelectric Materials to Reduce the Interior Noise of a Railway Vehicle. Noise and Vibration Mitigation for Rail Trans. Sys., Springer 2012:p.475-482.
11. Yamada K, Matsuhisa H, Utsuno H, Sawada K. Optimum tuning of series and parallel LR circuits for passive vibration suppression using piezoelectric elements. , Journal of sound and vibration. 2010; 329: p.5036-5057.
12. Yamada K, Matsuhisa H, Utsuno H, Park JG. Hybrid vibration suppression of flexible structures using piezoelectric elements and analog circuits. JSME Journal of Environment and Engineering 3 (2). 2008: p.424-435.
13. Fahy F. Sound and Structural Vibration. Academic press. 1987:p.81-89.
14. Leissa A. Vibration of plates. Acoustical Society of America. 1973:p.276-281.