

Evaluation of Silent Properties of Wheel for High-speed Railway Vehicles

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Abstract

In high-speed railway vehicles, noise associated with speed increase is a problem. We have developed a new wheel that can damp the rolling noise radiated from the wheel. By developing a method for evaluating the silent properties of this wheel using the finite element method (FEM), we worked to elucidate the noise damping mechanism of the noise damped wheel. As a result, it was clarified that the noise damped device consisting of rubber and metal attached to the noise damped wheel disperses many natural vibration modes of the wheel, and the damping of the rubber acts on these modes to reduce vibration and contribute to noise reduction. Based on this knowledge, the results of studies to improve performance and to reduce cost are presented.

1. Introduction

Railways are public transport systems used by many passengers. Railway stations and tracks are often located in densely populated residential areas. From the standpoint of residential environment protection, it is desirable to reduce the noise from high-speed railway vehicles such as those of the Shinkansen in order to protect the living environment along the railway lines. In Japan, noise levels along high-speed railway lines are regulated by the Environmental Standards¹⁾ established by the Environment Agency (now the Ministry of the Environment) in 1975. To meet this social demand, railway operators and parts manufacturers are developing ways to reduce noise.

Railway noise is generated from various sources during operation, such as vehicles, pantographs, wheels, motors, rails, and substructures. Noise from vehicles can be broadly divided into (1) aerodynamic noise from above the vehicle, (2) noise from below the vehicle, and (3) noise from the current-collection system. Of these, the noise from above the vehicle and the noise from the current collecting system have been reduced by various measures implemented. The proportion of the noise from under the vehicle in the total noise has been relatively increasing. The noise from under the vehicle consists of bogie aerodynamic noise, underfloor equipment noise, and wheel rolling noise. The wheel rolling noise has a large impact on the environment along the railway lines.

Rolling noise is the noise emitted when the wheels and rails vibrate continuously due to minute irregularities on the wheel and rail

surfaces while the train is running. The noise from the wheels is called the wheel rolling noise.²⁾ Nippon Steel Corporation has developed a noise damped wheel that can reduce this rolling noise.^{3,4)} A device called the noise damper and constructed of rubber and steel is installed on the inner diameter side of the wheel rim. The noise damped wheel was designed based on the idea of reducing the vibration caused noise by utilizing the resonance of the noise damper. The noise damped wheel was confirmed to have a certain level of noise reduction by the wheel rolling noise test described later. However, the optimal design technology for noise damped wheels has not been established yet. The noise damped wheel still has room for performance improvement. With the recent development of computer-aided engineering (CAE), in this study we investigated the noise reduction mechanism of the noise damped wheel through the establishment of a quietness evaluation method using the finite element method (FEM) and worked on improving the noise damped wheel by utilizing the knowledge obtained. The study results are reported here. This report is based on our literature.⁵⁾

2. Numerical Calculation

2.1 Model and calculation conditions

Radiation noise occurs when the vibration of a structure creates a pressure change in the surrounding air and propagates through the space. The radiation noise is analyzed by a method that couples the structure with the acoustic space and calculates the airborne noise. This method has the problem of high computational load, however.

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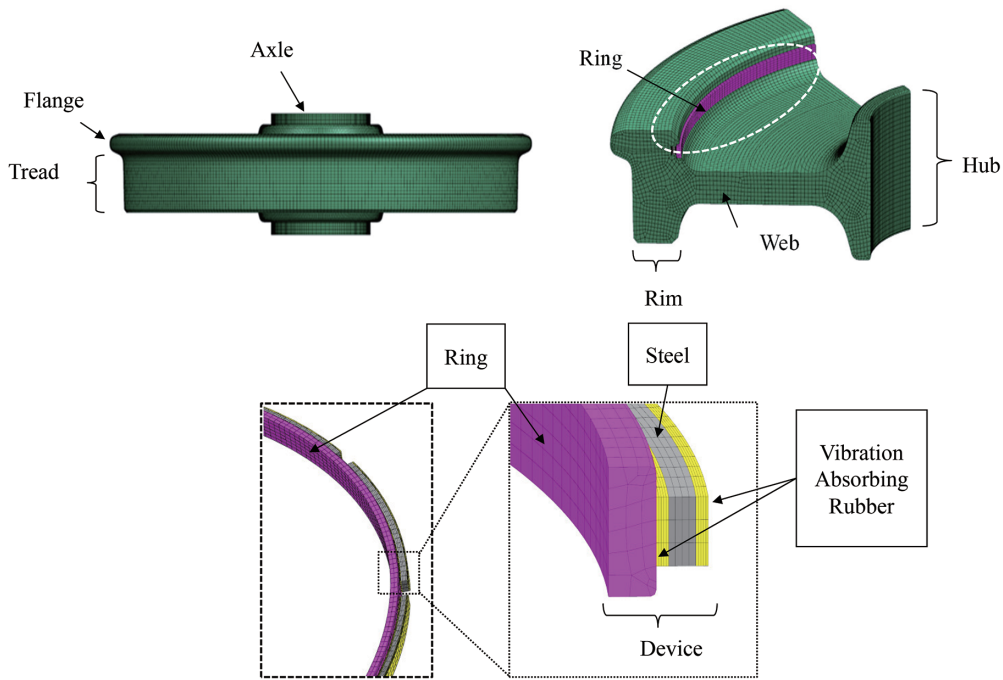


Fig. 1 The wheel model used for calculations⁵⁾

We therefore decided to use the equivalent radiated power (ERP) method that simply calculates the contribution to noise from the results of FEM frequency response analysis. This ERP is the energy released per unit time, i.e., the energy efficiency of vibration, and indicates the acoustic radiation capacity.

The finite element model used in our analysis is schematically illustrated in Fig. 1. As shown in Fig. 1, the noise damped wheel is constructed by installing 12 small noise damper parts, which are composed of three layers of rubber, metal and rubber, in the groove on the inner diameter of the wheel rim and securing them to the wheel with a metal ring. In this case, the metal is steel. Using this model, we calculated the ERP by applying an excitation force simulating the rail reaction force.

2.2 Calculation results

Figure 2 shows the ERP of a grooved wheel without the noise damper (or a non-noise damped wheel) and a noise damped wheel. For the non-noise damped wheel, the ERP is large in the range of 2300 Hz to 5000 Hz and the absolute ERP value decreases as the frequency increases. A similar trend is observed for the noise damped wheel, but its ERP decreases in almost all ranges compared to the non-noise damped wheel. The above results confirmed that the installation of the noise damper provides a certain vibration damping effect.

2.3 Accuracy verification

2.3.1 Wheel rolling noise test

To verify the accuracy of the calculation results, the noise emitted during rolling was evaluated using a wheel rolling noise testing machine shown in Fig. 3. The machine hydraulically presses a roller simulating a rail against the test wheel and drives it with a motor located behind the test wheel to simulate the rolling state.⁶⁾ The specifications of the machine are shown in Table 1. The maximum pressing load is 100 kN and the test can be conducted at a maximum

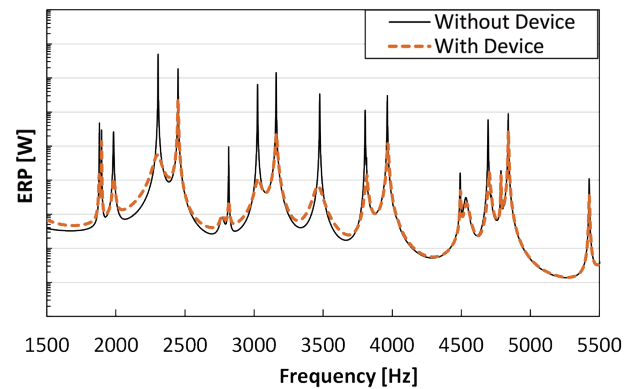


Fig. 2 Effect of damping device on ERP⁵⁾

speed of 400 km/h. A roller with a tread shape equivalent to that of a JIS E 1101 60 kg rail was used. The noise level at 360 km/h was evaluated with a precision noise level meter.

2.3.2 Comparison of calculation results and test results

Figure 4 shows the 1/3 octave band frequency characteristics determined from the wheel rolling noise test results and from the ERP calculation results. The first axis (left) shows the ERP level (10 times the logarithmic notation of ERP [W]) and the second axis (right) shows the noise level. That is, the calculation results and experimental results of the noise damped wheel are comparatively shown. Consequently, the noise level was highest in the band centered at 3150 Hz for both sets of the results. The difference in the overall (O.A.) noise level, which is the sum of noise in all frequency ranges, was also relatively close between the two sets of results. Although omitted in this report, similar results were obtained from testing with the non-noise damped wheel.⁵⁾ From the above results, we confirmed that experimental results can be sufficiently predicted

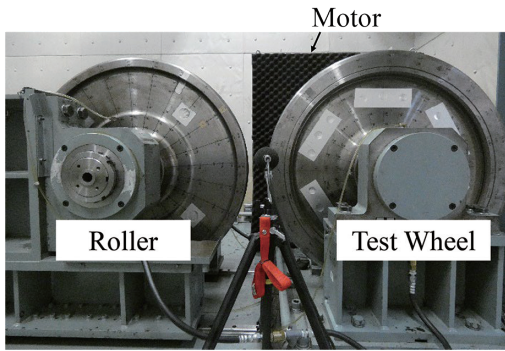


Fig. 3 Rolling noise test machine⁵⁾

Table 1 Capacity of rolling noise test machine⁵⁾

Maximum Load	100 kN
Maximum Speed	400 km/h (2400 rpm)
Roller Diameter	φ 915 mm
Tread Surface Shape of Roller	JIS 60kg Rail
Test Wheel Diameter	φ 660–φ 910 mm

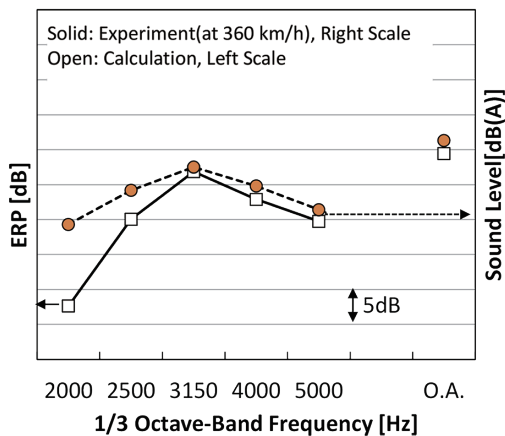


Fig. 4 The comparison of experiment and calculation⁵⁾

using this model.

3. Discussion

3.1 Study of mechanism

3.1.1 Verification of dynamic vibration absorbers

According to the original design concept of the noise damped wheel, the noise damper resonates with the wheel as an assembly of dynamic vibration absorbers.⁷⁾ In other words, the noise damper achieves the vibration reduction effect by absorbing and dissipating the vibration energy of the wheel. In addition, the vibration reduction effect is also achieved by the damping of rubber itself, which is a component of the noise damper. The vibration reduction effect of the noise damped wheel is thus considered to have resulted from the superposition of the two mechanisms noted here.

One configuration of the noise damper is shown in Fig. 5(a). Four types of small parts are sequentially arranged in the circumferential direction with a small space between them. This configuration allows the small parts to vibrate independently. These four types of small parts are the same in the overall thickness but different in the

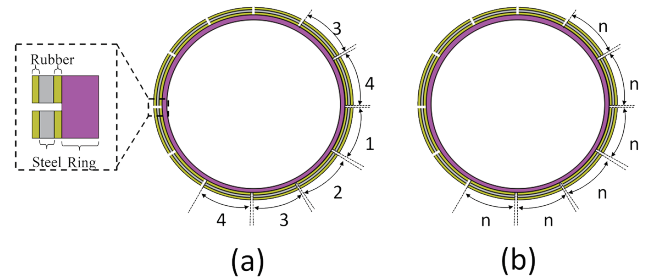


Fig. 5 The image of the configuration of the device⁵⁾

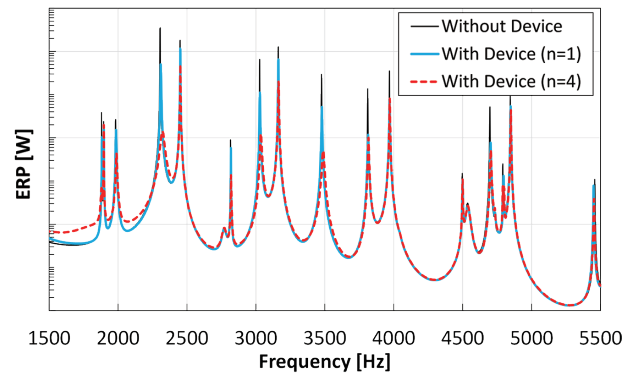


Fig. 6 Effect of noise damper configuration on ERP⁵⁾

rubber/steel thickness ratio. There are four natural vibration modes in the range of 2000 Hz to 5000 Hz. These four modes are thought to have a particularly large effect on the wheel rolling noise. The rubber/steel thickness ratio is changed for tuning to each of the four modes.

First, to verify whether these small parts function as dynamic vibration absorbers, their configuration in the analysis model was changed to either type 1 or type 4 as shown in Fig. 5(b). A noise damped wheel with a type 1 configuration noise damper and a noise damped wheel with a type 4 configuration noise damper were compared with a non-noise damped wheel in terms of the ERP. The results are shown in Fig. 6. The noise damped wheel with the type 4 configuration noise damper has a superior ERP reduction effect in all frequency bands.

For a more detailed study, we focused on a frequency near 2300 Hz where the ERP reduction effect is large and evaluated the degree of natural vibration dispersion by changing the structural damping of rubber to the same value as that of steel. The results are shown in Fig. 7. Figure 7(a) shows the ERP spectra of a noise damped wheel with a type 1 part configuration noise damper (structural damping of rubber changed to the same value as that of steel) and a noise damped wheel with a type 1 part configuration noise damper (structural damping of rubber set different from that of steel) in comparison with a non-noise damped wheel. Figure 7(b) shows the ERP spectra of a noise damped wheel with a type 4 part configuration noise damper (structural damping of rubber changed to the same value as that of steel), a noise damped wheel with a type 4 part configuration noise damper (structural damping of rubber set different from that of steel) in comparison with a non-noise damped wheel. For the type 1 part configuration noise dampers shown in Fig. 7(a), the number of peak dispersions does not change the structural damping of rubber. For the noise damped wheel with the type 4 noise

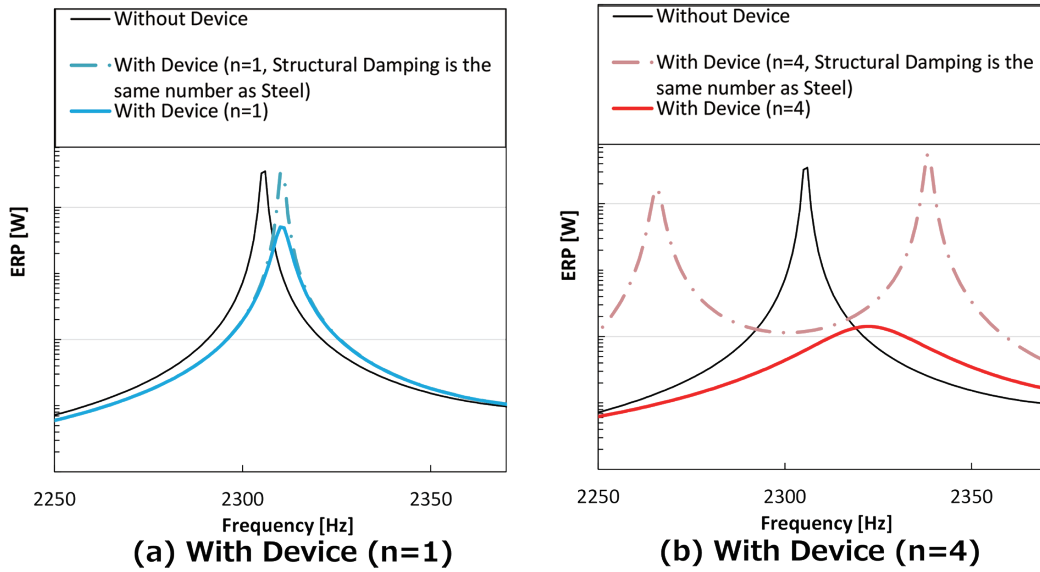


Fig. 7 The ERP spectra around 2300 Hz⁵⁾

damping device configuration (structural damping of rubber changed to the same value as that of steel) shown in Fig. 7(b), the peak is split into two as if straddling the peak of the non-noise damped wheel when the structural damping of rubber is changed to the same as that of steel. This result suggests that the ERP reduction effect is high near the frequency of 2300 Hz because the type 4 small parts function as dynamic vibration absorbers near 2300 Hz. On the other hand, the type 1 small parts exhibit an ERP reduction effect even though no peak split is observed. This effect may be ascribed to the damping of rubber itself. The type 2 and 3 small parts were similarly studied in the range of 2000 Hz to 5000 Hz as shown in Fig. 7. Neither of them functioned as general dynamic vibration absorbers as shown in Fig. 7(b). Therefore, some of the small noise damping parts are considered to function as general dynamic vibration absorbers at least in the vicinity of 2300 Hz. In other frequency ranges, the ERP is considered to be reduced by the damping of rubber itself or by other mechanisms.

Generally, there are many natural vibrations per ERP peak at the high end of the frequency range. The natural vibration of a structure refers to the natural frequency at which the structure vibrates easily. When a structure has many natural frequencies in a narrow frequency range, its natural vibration mode density is said to be high. When the natural vibration mode density is high, it is difficult to identify the dominant natural vibration mode (shape) by frequency-by-frequency evaluation. Thus, we studied the vibration reduction mechanism by narrowing down to a particular frequency and evaluating the natural vibration mode that affects this frequency. **Figure 8** shows the ERP spectra of a noise damped wheel and a non-noise damped wheel in a frequency range of 2100 to 3200 Hz. According to Fig. 8, we focused on the frequencies near 2300 Hz and 3000 Hz where the ERP reduction effect is high.

Next, of the many natural vibrations that affect the ERP peaks in Fig. 8, the natural vibration modes in the nearest frequencies are shown in **Fig. 9** and the natural frequencies that correspond to the natural vibration modes in Fig. 9 are shown in **Table 2**. The color contours are based on blue. The displacement amount increases in the order of green, yellow, and red. In Fig. 9, (α) is a two-diameter node mode in which the rim vibrates in the same phase in the in-

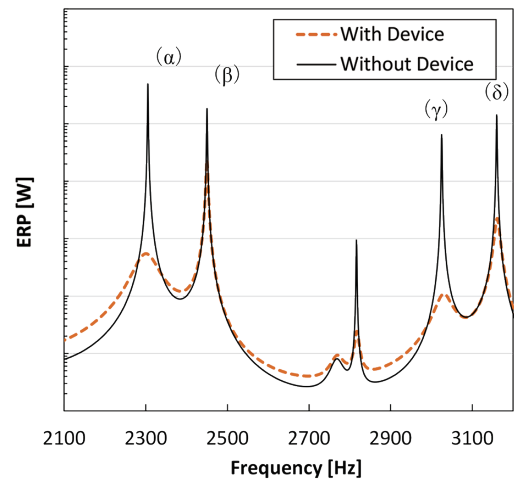


Fig. 8 The ERP spectra of with device and without device from 2100 Hz to 3200 Hz⁵⁾

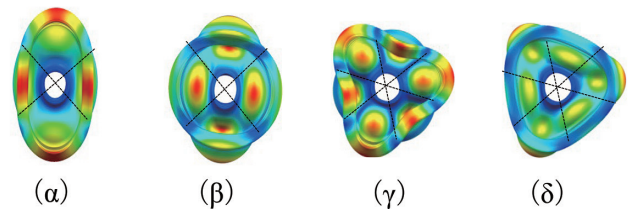


Fig. 9 The wheel vibration mode of without device corresponding to peaks (α) to (δ)⁵⁾

plane direction and (β) is a two-diameter node mode in which the flange side and the anti-flange side vibrate in the anti-phase in the in-plane direction and the plate vibrates in the out-of-plane direction. On the other hand, (γ) and (δ) in Fig. (9) are both three-diameter node modes in which the flange side and the anti-flange side vibrate in the anti-phase in the in-plane direction and the plate vibrates

Table 2 The wheel vibration mode and corresponding natural frequency⁵⁾

		Mode			
		α	β	γ	δ
Natural Frequency of Wheels [Hz]	Without Device	2306	2450	3028	3162
	With Device	2320	2451	3034	3162

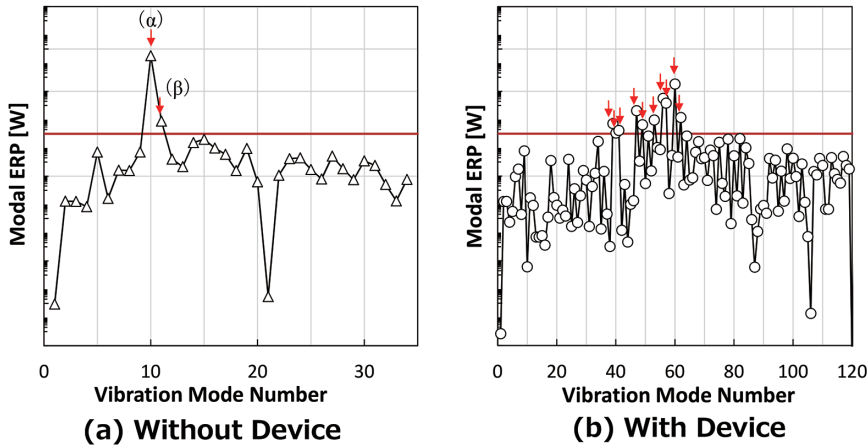


Fig. 10 Modal ERP at 2306 Hz of without device (a) and 2320 Hz of with device (b)⁵⁾

in the out-of-plane direction. The contribution of these natural vibration modes is evaluated in the sections below.

3.1.2 Contribution of natural vibration mode

Figure 10 shows the ERP for each mode (hereafter referred to as modal ERP) at the frequency of the natural vibration (α) near 2300 Hz. The frequency (α) is 2306 Hz for the non-noise damped wheel and 2320 Hz for the noise damped wheel. Figure 10(a) shows the modal ERP for the non-noise damped wheel and Fig. 10(b) shows the modal ERP for the noise damped wheel. The frequencies (α) and (β) are the same as the natural vibrations shown in Fig. 8. For the non-noise damped wheel in Fig. 10, the natural vibrations that exceed the threshold are the two modes (α) and (β) as shown in Fig. 10(a). The mode (α) is particularly dominant.

For the noise damped wheel, on the other hand, there are 10 modes for the natural vibrations exceeding the threshold as shown in Fig. 10(b). This is a significant increase compared to the non-noise damped wheel. Furthermore, Table 3 shows mode deformation diagrams of the ring and steel in addition to the wheel. Of the ten natural vibration modes of the noise damped wheel shown in Table 3, eight (enclosed within the red frames) are thought to be modes with two diameter nodes. The remaining natural vibration modes are considered to have four diameter nodes. Also, the modes shown in Table 3 are the natural vibration modes in which both the ring and steel vibrate in the radial direction. These natural vibration modes are not necessarily symmetrical vibration modes because the steel/rubber thickness ratio is changed depending on the layout.

The above results suggest the possibility that the noise damped wheel increases its vibration reduction effect as the number of natural vibration modes with a high contribution to the vibration reduction effect increases at frequencies with large ERPs, i.e., the noise damper disperses the vibration energy of the wheel.

Near 3000 Hz, the results were similar to those near 2300 Hz. This means that the increase in the natural vibration modes contrib-

utes to the vibration reduction effect. Therefore, the noise damped wheel is considered to display its ERP reduction effect by two actions. One is that the noise damped wheel acts as a general dynamic vibration absorber at some frequencies. The other is that the noise damped wheel disperses the natural vibration mode into many in other frequency ranges. Since the action of the noise damped wheel as a dynamic vibration absorber is limited, it is suggested that the ERP reduction effect mainly comes from the mode dispersion.

3.1.3 Mode dispersion and rubber attenuation

In the previous section, we clarified that the noise damper reduces the ERP by dispersing the natural vibration modes of the wheel. Here, for a more detailed study, we attempted to separate the ERP reduction effect into that of natural vibration mode dispersion and that of rubber damping.

Figure 11 shows the 2320 Hz modal ERP of a noise damped wheel with the structural damping of rubber set different from that of steel and a noise damped wheel with the structural damping of rubber changed to the same value as that of steel. Note that the number of natural vibrations does not change with the value of the structural damping. As can be seen from Fig. 11, the modal ERP of the noise damped wheel with the structural damping of rubber set different from that of steel does not necessarily match that of the noise damped wheel with the structural damping of rubber changed to the same value as that of steel. Also, in the latter noise damped wheel, there are four natural vibration modes that exceed the threshold as indicated by the red arrows in Fig. 11.

Of these results, the modal ERP in the No. 60 natural vibration mode and enclosed in the red dotted circle in Fig. 11 is dominant. This result indicates that in the noise damped wheel with the structural damping of rubber changed to the same value as that of steel, the contribution of the natural vibration mode No. 60 to the ERP at 2320 Hz is particularly high.

On the other hand, for the noise damped wheel with the structur-

Table 3 List of natural vibrations of with device at 2320 Hz corresponding to (α)⁵⁾

	Wheel	Ring and Steel	Wheel	Ring and Steel	Wheel	Ring and Steel	Wheel	Ring and Steel
Vibration Mode								
Mode Number	39		40		41		47	
Frequency [Hz]	1706		1770		1943		2082	
Vibration Mode								
Mode Number	49		53		56		57	
Frequency [Hz]	2082		2181		2181		2320	
Vibration Mode								
Mode Number	60		62					
Frequency [Hz]	2451		2535					

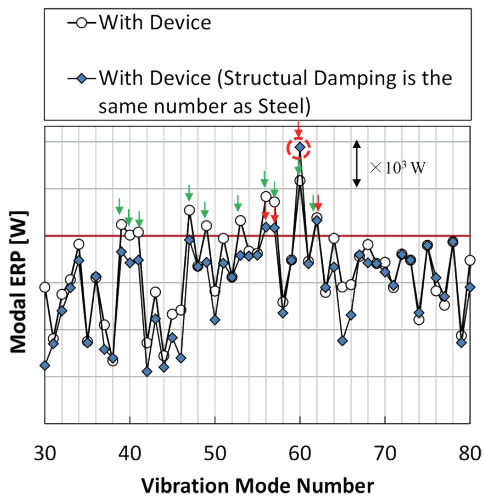


Fig. 11 The modal ERP at 2320 Hz of with device⁵⁾

al damping of rubber set different from that of steel, there are ten natural vibration modes that exceed the threshold, as indicated by the green arrows in Fig. 11. The modal ERP in the No. 60 natural vibration mode is less than 1/100 of that of the noise damped wheel with the structural damping of rubber changed to the same value as that of steel and the modal ERP in many natural vibration modes other than the No. 60 natural vibration mode increases. This result suggests that the contribution of the No. 60 natural vibration mode to the ERP at 2320 Hz has decreased, while the contribution of the other natural vibration modes has increased.

The above results show that the damping of rubber does not contribute to the dispersion of natural vibrations but affects the value of

modal ERP. In other words, the noise damped wheel is considered to exhibit the ERP reduction effect as the noise damper disperses the natural vibrations of the wheel, regardless of the damping of rubber, and as the damping of rubber acts on each of the dispersed natural vibrations.

3.2 Improvement in performance of noise damper

We attempted to improve the performance of the noise damper by effectively utilizing the effect of the vibration mode dispersion using the knowledge described in Section 3.1. Here, we describe the results of examining the effect of the Young's modulus of rubber with the aim of enhancing the mode dispersion effect.

We calculated the ERP by changing the Young's modulus of rubber in the industrially manufacturable ranges of E1 to E5 and organized the data by the 1/3 octave band frequency characteristics. The results are shown in Fig. 12. The Young's modulus at the time of basic design corresponds to E1 and the Young's modulus is set to increase from E1 to E5. From Fig. 12, it is evident that the installation of the noise damper reduces the ERP, regardless of the Young's modulus of rubber. As the Young's modulus of rubber increases from E1 to E3, the ERP tends to decrease. At E5, however, the ERP increases in the 1600 Hz band to the 3150 Hz band compared to E4. The O.A. evaluation is smallest for E3 and E4. Accordingly, there is an appropriate range for the Young's modulus of rubber. When the Young's modulus is low outside this range, the energy attenuation amount of the wheel is small. When the Young's modulus is large, the noise damper vibrates together with the wheel, making it impossible for the rubber to fully exert its damping effect.

3.3 Cost reduction of noise damper

The noise damper discussed in this report is composed of 37 parts, including small parts, and presents a high manufacturing cost problem. We thus investigated a cost-saving configuration. As a re-

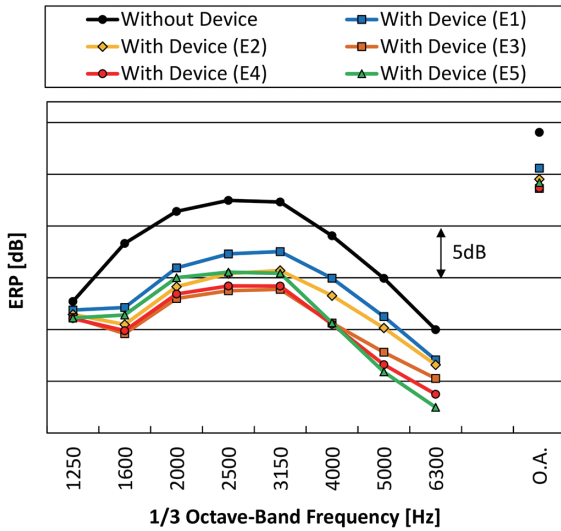


Fig. 12 Effect of Young's modulus on ERP⁵⁾

result, we found that it is possible to control the number of mode dispersions by changing the number of layers in the noise damper. Based on this finding, we developed a ring-shaped configuration in which the noise damper parts are not divided in the radial direction as shown in Fig. 13. When the total number of layers is increased to at least four or more, the noise damper can fully display the mode dispersion effect. This configuration can reduce the number of parts to a minimum of four, significantly reduce the manufacturing cost, and improve maintainability.

4. Conclusions

We investigated the vibration reduction mechanism of noise damped wheels through the establishment of an FEM quietness evaluation method. As a result, we estimated the equivalent radiated power (ERP) of noise damped wheels, evaluated their ERP by mode (modal ERP) at specific frequencies, and clarified natural vibration modes that contribute to the specific frequencies. Accordingly, we

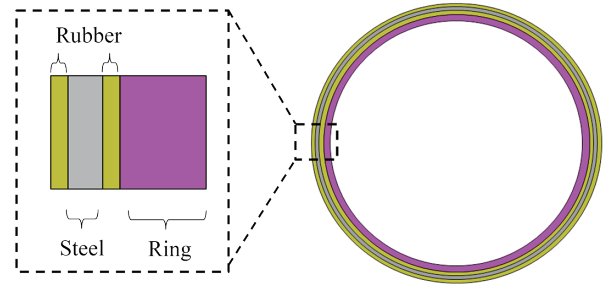


Fig. 13 Configuration of a low-cost device

found that the vibration reduction mechanism of the noise damped wheel is such that the noise damper disperses the natural vibration modes of the wheel into many and that the damping of rubber acts on the noise damper. We also clarified that depending on the configuration, the effect of the dynamic vibration absorbers is also added in some frequency ranges. We also found that the optimization of the Young's modulus of rubber increases the number of mode dispersions and improves the vibration reduction effect. Furthermore, we clarified that the cost can be reduced by controlling the number of layers in the noise damper. We will contribute further to the development of the railway field and the improvement of railway convenience by applying the quietness evaluation method established in this study to the development of railway wheels.

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