Reduction of Carbody Flexural Vibration by the High-damping Elastic Support of Under-floor Equipment

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This paper presents a method for reducing flexural vibrations in a railway vehicle carbody by supporting under-floor equipment using high-damping elastic mounts. This is a kind of dynamic vibration absorber utilizing under-floor equipment as a mass element which has been introduced based on the inspiration of the damping effect of passengers. A series of excitation tests were conducted in the rolling stock testing plant using a Shinkansen type test vehicle by applying the proposed method. As a result of the tests, good vibration reduction performance, including multi-modal vibration reduction and vibration isolation from the equipment, was observed.

Keywords: ride comfort, flexural vibration, underfloor equipment, dynamic vibration absorber, vibration reduction

1. Introduction

Weight reduction and structural simplification of recent railway vehicles has tended to increase the flexural vibrations of railway vehicle carbodies. Since the natural frequency of major modes of flexural vibrations in the vertical direction are usually around 10 Hz and it is commonly considered that human beings are sensitive to the vibrations at these frequencies, suppression of the flexural vibrations is important to improve ride comfort. Flexural vibrations of a carbody are usually supposed to be the first bending mode in a simple free-supported elastic beam. However, according to previous studies [1-3], actual railway vehicle carbodies have a number of other flexural modes such as the different deformation of the roof and floor or cross sectional deformation of the carbody (diagonal distortion), and those complex flexural modes have a more than negligible impact on ride quality. This fact suggests that multi-modal flexural vibration reduction is effective to improve ride quality.

The authors devised a multi-modal vibration control method using active mass dampers to reduce flexural vibrations of railway vehicle carbody and confirmed that the proposed method was able to reduce multiple flexural vibration modes simultaneously [4]. However, simpler and more cost-effective methods which do not have control technology are also needed.

Meanwhile, the authors studied the effect of passengers on the flexural vibration properties of a railway vehicle carbody [5]. These studies confirmed that a large vibration reduction or damping effect can be achieved, even with a small number of passengers and that this damping is also effective against plural flexural modes of the carbody (multi-mode vibration reduction). Therefore, if the damping effect can be mimicked or simulated flexural vibrations of the carbody can be effectively counteracted, thereby improving ride quality. Focusing on this suggestion, a new vibration reduction method was developed which simulates the passenger damping effect by supporting under-floor equipment with a high-damping elastic material. This is a kind of dynamic vibration absorber (DVA) utilizing under-floor equipment as a mass element.

Using under-floor equipment as a DVA is not new e.g. [6]. However the natural frequencies in ordinary DVAs need to be tuned according to the target vibration frequency and ordinary DVAs can be effective in only one vibration mode. By contrast, the high-damping elastic support of under-floor equipment has the following features: no need for cumbersome procedures to adjust the natural frequency and damping characteristics, and a multi-mode vibration reduction effect.

This report first describes the elastic support of under-floor equipment, and then reports on vibration measurement test results which confirm its vibration reduction performance, using an actual railway vehicle.

2. Outline of the vibration measurement test with an actual railway vehicle

In order to verify the vibration reduction performance of high-damping elastic support in under-floor equipment, a supporting member was developed and excitation tests were conducted on an actual railway vehicle at the Railway Technical Research Institute’s rolling stock testing plant. A Shinkansen type test vehicle was chosen for this study as shown in Fig.1. Its carbody shell is made of hollow aluminum alloy, which is a widely used carbody structure for Shinkansen vehicles. It is equipped with interior panels and passenger seats but being a test vehicle, lacks electrical service equipment such as air conditioners and under-floor equipments. This equipment was replaced by a dummy weight under the carbody (hereafter “dummy under-floor equipment”).
Underfloor equipment on an actual Shinkansen vehicle, such as main transformers, main traction converters, etc. weigh more than 3000 kg. Such large masses with elastic support can be expected to greatly reduce carbody flexural vibrations. Therefore, the effect of an elastic support for the equipment was investigated using masses over 3000 kg.

The following two loading configurations were examined: (a) Distributed mass with set of four dummy under-floor equipments (about 245 kg/piece); (b) Concentrated mass of approximately 3400 kg. This report shows the vibration measurement test results which verify the vibration reduction performance of the elastic support under the two loading conditions.

Vibration transmission from the equipment to the carbody was also expected to fall under the effect of the new method. In some cases, under-floor equipment, such as air compressors or main transformers etc., generates their own constant-frequency vibrations in a frequency range which can be felt by humans. When such vibrations are transmitted to the carbody, they may increase carbody vibrations and lower ride quality. To evaluate the vibration isolation performance of the elastic support, vibration exciters were installed to the dummy under-floor equipments and programmed to generate constant-frequency vibrations. Vibration measurement test results to verify the elastic support’s vibration isolation performance, are also reported.

3. Dummy under-floor equipment and layout

3.1 Dummy under-floor equipment

Figure 2 shows how the dummy under-floor equipment was attached to the carbody. It consisted of a frame mounted with a dummy mass (DM) or stack of thick steel plates, fastened under the carbody. The DM was supported by four supporting members. Two support conditions were applied to the DM; rigid support (fixed normally, using bolts) and elastic support with high-damping material. A simple rubber mount was chosen as the elastic supporting member shown in Fig.3. To exert the high-damping properties, the rubber mount was made of butyl rubber.

3.2 Layout of the dummy under-floor equipment

Four dummy under-floor equipments were built and installed under the test vehicle as shown in Fig.4. The weight of each DM was 238 kg (#1 and #2) and 252 kg (#3 and #4). This test configuration is referred to hereafter as "distributed mass." A second test set up, referred to here-
after as “concentrated mass” configuration was then arranged by connecting the dummy under-floor equipments using steel beams with some additional DMs, as shown in Fig.5. The weight of a large mass was 3380 kg. The loading and supporting conditions are listed in Table 1. According to the report [7], it is better to support a mass elastically with a natural frequency lower than the target frequency. Since the natural frequencies of the major flexural vibrations of the test vehicle (without dummy equipment) were estimated to be in the range of 9 Hz to 11 Hz [4], the rubber mount was designed so that the dummy under-floor equipment had a natural frequency of around 8 Hz to 8.5 Hz.

### 4.2 Acceleration measurement points

The vibration measurement points on the floor are illustrated in Fig.6. Vertical accelerations on the roof were also measured. In addition, vertical accelerations of each DM, and longitudinal and lateral acceleration of DM #2 and #4 were measured.

### 4.3 Mode shapes and natural frequencies of the test vehicle

Figure 7 shows the modal properties (vibration mode shapes and corresponding natural frequencies) identified under simultaneous excitation of all wheels for the distributed mass test (rigid support). Modal analysis was carried out using a multiple-input multiple-output ARX model [8]. Five flexural modes were identified below 20 Hz. The notations “S-11”, “J-1” etc., represent the features related to the vibration shapes. The prefixes S denote the modes in which the roof and the floor deform in the same direction around the longitudinal center of the carbody. Prefix Z is used when the directions of the roof and/or floor vibrations are not clear, or when the vibration amplitude of the roof and that of the floor are extremely different. Prefix J represents the mode shapes with shear deformation in the cross section of the carbody. The numbers following the above prefixes are composed of two parts. The prefix number represents the number of vibration loops observed in the roof structure and the suffix number the number of vibration loops in the floor structure. For J-modes, the suffix number was omitted since in this mode, the roof and the floor always have the same number of vibration loops.

### 4.4 Vibration reduction performance on the floor achieved by applying the proposed method

Figure 8 shows a comparison of the acceleration power spectral densities (PSDs) at f4c, f4r and f6c in Fig.6 when...
Fig. 7 Mode shapes and natural frequencies of the test vehicle
(Distributed mass with rigid support)

(a) J-1, 8.9 Hz  
(b) S-11, 10.9 Hz  
(c) J-3, 14.9 Hz  
(d) Z-10, 15.3 Hz  
(e) Z-20, 18.9 Hz

Roof

Floor

Fig. 8 Acceleration PSDs measured on the floor
(Distributed mass)

(a) At the floor centre (f4c)

(b) At the longitudinal centre near the window (f4r)

(c) Above the rear bogie (f6c)

Fig. 9 Acceleration PSDs measured on the floor
(Concentrated mass)

(a) At the floor centre (f4c)

(b) At the longitudinal centre near the window (f4r)

(c) Above the rear bogie (f6c)
simulated running excitation was applied to the distributed mass configuration. In case of rigid-support (blue line in the figure), the peak around 11 Hz corresponds to the S-11 mode (first bending mode) and the peak around 9 Hz (in Fig.8 (b)) corresponds to the J-1 (diagonal distortion) mode. Both peaks were reduced by applying the high-damping elastic support (red line in the figure), and the multi-modal vibration reduction effect was clearly confirmed. In addition, it was confirmed that vibrations at f6c, which is located away from the under-floor dummy equipment and above the rear bogie, were also reduced.

Ride comfort was evaluated using the ride quality level ($L_r$), which is a representative index for evaluating ride comfort on railway vehicles in Japan. $\Delta L_r$ in the legend of the Fig.8 is the difference in ride quality level for each of the different support conditions (a negative value means that the ride quality is improved). Results confirmed an improvement in ride quality at all measurement points in Fig.8 when the high-damping elastic support was used.

Figure 9 shows a comparison of the PSDs at f4c, f4r and f6c in Fig.6 when simulated running excitation was applied to the concentrated mass configuration. The peak values around 9 Hz and 11 Hz were reduced simultaneously at f4r with a high-damping elastic support. In addition, an improvement in ride quality level was up to 5 dB at f4c by applying the high-damping elastic support.

These results confirmed the vibration reduction performance of the new method is effective under both loading conditions (distributed and concentrated mass).

The total reduction effect of the flexural vibrations of the carbody floor, are evaluated using the following equation:

$$ S_{\text{rms}} = \frac{1}{N} \sum_{n=1}^{N} P_{\text{rms}} n $$

where $n$ and $N$ denote the acceleration measurement points and their total number on the floor, respectively ($N=21$ as shown in Fig.6); $P_{\text{rms}} n$ is the band limited RMS (Root mean square) value (BLRMS) of measured acceleration at point $n$; and $S_{\text{rms}}$ is the averaged value of the BLRMS on the floor (called averaged BLRMS, hereafter). Here, the BLRMS value was determined as the root of the integral value of the PSD with a frequency range from 5 Hz to 20 Hz.

Figure 10 illustrates the averaged BLRMS when simulated running excitation was applied to the concentrated mass. About 24% reduction was observed on the floor acceleration in total with a high-damping elastic support. From the result, it was confirmed that vibrations were reduced across the whole of the floor of the vehicle, using the proposed method.

### 4.5 Vibration of the supported mass

Since the proposed method is in fact a kind of DVA utilizing under-floor equipment, the vibration of the supported mass should be estimated. The averaged BLRMS values of the acceleration on the DMs were calculated. As mentioned in Section 4.2, vertical acceleration was measured at the top of each DM, and longitudinal and lateral acceleration were measured on DMs #2 and #4. The averaged BLRMS values of the supported mass are shown in Fig.11 when simulated running excitation was applied to the concentrated mass. The averaged BLRMS values in the vertical direction increased with high-damping elastic support; meanwhile the components of the averaged BLRMS in the longitudinal and lateral direction decreased drastically. In consideration of this situation, the averaged BLRMS with respect to the combined acceleration values in the three directions, was also calculated. As a result, the combined value decreased with high-damping elastic support. Therefore, the proposed method exerts a negligible adverse influence on the vibrations of the supported equipment. However, future studies need to recheck vibrations from the supported equipment, under in actual operational conditions, considering various types of motion, such as passing through curves, acceleration and deceleration.

### 4.6 Performance of equipment vibration isolation

To evaluate the vibration isolation performance of the elastic support, vibration exciters were installed DMs #1 and #2 and used to generate vibrations at a constant frequency of 30 Hz. Figure12 shows the PSD at the longitudinal floor centre near the window (measurement point at f4r) when all wheels were being excited simultaneously and the DMs in distributed mass configuration. The sharp peak at 30 Hz shows the additional vibrations transmitted from the under-floor equipment, but also shows that these vibrations were greatly reduced by the high-damping elastic support.
5. Summary

In this study, a high-damping elastic support for under-floor equipment was developed as a method for reducing carbody flexural vibrations. Dummy under-floor equipment, equipped with elastic mounts were developed and installed to a Shinkansen type test vehicle, and a series of excitation tests were conducted at the Railway Technical Research Institute’s rolling stock testing plant. Good vibration reduction performance was observed when the high-damping elastic support was used, in two loading configurations (distributed and concentrated mass). Using the proposed method, the multi-modal flexural vibrations were reduced simultaneously and ride quality level was improved. Moreover, the vibration isolating performance of the elastic mount was observed and confirmed, and under-floor equipment vibrations did not increase. Measured results confirmed the validity of the proposed method.

The high-damping elastic support for under-floor equipment can be installed on various cars, requires no specific maintenance and is a convenient method for reducing carbody flexural vibration. The authors will continue to develop this method to make it more practical for actual use.

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