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Experimental study of the effect of viscoelastic damping materials on noise and vibration reduction within railway vehicles

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Abstract

Interior noise and vibration reduction has become one important concern of railway operating environments due to the influence of increased speeds and reduced vehicle weights for energy efficiency. Three types of viscoelastic damping materials, bitumen-based damping material, water-based damping coating and butyl rubber damping material, were developed to reduce the vibration and noise within railway vehicles. Two sleeper carriages were furnished with the new materials in different patterns of constrained-layer and free-layer damping treatment. The measurements of vibration and noise were carried out in three running carriages. It is found that the reduction effect of damping treatments depends on the running speed. The unweighted root-mean-square acceleration is reduced by 0.08–0.79 and 0.06–0.49 m/s^2 for the carriage treated by bitumen-based as well as water-based damping materials and water-based damping material, respectively. The first two materials reduce vibration in a wider frequency range of 63–1000 Hz than the last. It turns out that the damping treatments of the first two reduce the interior noise level by 5–8 dBA within the carriage, and the last damping material by 1–6 dBA. However, the specific loudness analysis of noises shows that the noise components between 125 and 250 Hz are dominant for the overall loudness, although the low-frequency noise is noticeably decreased by the damping materials. The measure of loudness is shown to be more accurate to assess reduction effect of the damping material on the acoustic comfort.

1. Introduction

Modern trains are expected to have a higher level of comfort for the passengers in order to keep or increase its commercial competitiveness. Several investigations have identified noise and vibrations as key factors to high comfort. For lightweight vehicles, interior noise and vibration increase sharply due to the reduction in transmission loss of the car body and the rise in air-dynamic noise and rolling noise. Unfortunately, the subject of noise within railway vehicles has been paid less attention in recent years than external

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"environment" noise such as rolling noise. When the new type of luxury sleeper carriage, traveling at speed of 160 km/h, is designed and manufactured in China, the comfort closely related to internal noise and vibration is one major concern. The question arises: what is the most appropriate and low-cost method to provide the comfortable riding environment with low internal noise?

One of the major problems is to prevent noise and vibrations generated by exterior sources, e.g. the wheel–rail rolling noise and the braking noise. The interior noise inside a railway coach is composed of air-borne at middle and high frequencies and structure-borne sound below 250 Hz. With a trend towards lighter trains the structure-borne sound will increase. There is a conflict between light weight structures and low levels of noise and vibrations. It has been proven difficult to achieve a satisfactory comfort level without adding mass to the structure. Many passive and active methods have been developed to improve the riding comfort. Some of them have been used for many years while others are still quite new or under investigation.

For a good vibration environment it is important that the interaction between different parts of the vehicle is kept as low as possible for all frequencies [1]. Traditionally, this has been done by separation of fundamental frequencies of bogie, car body, floor and passenger chair. This method is only applicable to case where there are distinct resonance frequencies. This method cannot reduce the internal noise level. Floating floor construction investigated in the ship and aircraft industry [2–4] has been used to obtain a vibration isolation of the inner floor so as to reduce the acoustical power radiated by the floor and transmission of air-borne sound. The floating floor constructions have been primarily used to minimize the transmission of noise into the compartment. It is interesting to note that although a floating floor construction can effectively reduce vibrations in the high-frequency region, there is also an increase of vibrations in the low-frequency region near the fundamental frequency of the isolation arrangement. Typically, this is the frequency range considered with regard to comfort-related vibrations. Thus, the use of a floating floor may decrease the vibration-related comfort level. There is a need for modification of current vehicle structure to install the floating floor. Koo et al. [5] described the vibration and noise reduction by application of low-noise wheel for motor and trailer cars in the electric multiple unit system. In spite of its promising potential of noise reduction for railway vehicles, the endurance and safety of the low-noise wheel are yet to be verified. As far as the reduction of internal structure-borne noise in the frequency range of 20–250 Hz is concerned, a few methods mentioned above are neither efficient nor suitable.

By semi-active control of an orifice installed between the air spring and the auxiliary air chamber, the vertical vibration on the car body floor has been effectively reduced [6]. The active method by employing elastically suspended mass is described by Holst [7]. A semi-active control strategy using MR damper is proposed to decrease the vibration acceleration on the floor in the vehicle [8]. However, the advance of active and semi-active control technology in vehicles and commercial airplanes has been slow because of the high costs and complexity of the internal sound field. The practical application of these technologies in railway vehicles is still in the beginning stages.

Passive damping using viscoelastic materials is simpler to implement and more cost-effective than semi-active and active techniques. Passive damping has been dominant in the non-commercial aerospace industry since the early 1960s. Advances in the material technology along with newer and more efficient analytical and experimental tools for modeling the dynamic behavior of materials have led to the surface damping treatments in automotives, commercial airplanes and railways [9–17]. Although the viscoelastic damping materials have been widely used in the fields mentioned above, the comprehensive applications for internal vibration and noise control in the car body of the railway vehicle were rarely reported in public and have been almost confined to company brochures so that these case studies are not readily available. Suzuki [18,19] introduced a method, which used viscoelastic constraint layers pasted partially on the outside sheeting of the car body. Based on the theoretical evaluation, it was found at the choice of the optimal length and appropriate characteristics lead to the maximum damping. These optimum parameters could give birth to the maximum improvement of riding comfort in a lightweight car body of a high-speed train. The full scale experimental results showed that the riding comfort level was improved by about 3 dB at 275 km/h. However, the experiments and analysis were limited to a low-frequency range of the riding comfort. The change in the internal noise by damping treatment has not been studied.

Considering that the properties of viscoelastic materials are significantly dependent on environmental conditions such as temperature, vibration frequency, pre-load, dynamic load, environmental humidity and so on, the proper surface treatment, dimension and appropriate characteristics of the damping material is of vital importance for the success of viscoelastic material in adding damping to the structure system. The viscoelastic materials have been used to enhance the damping in a structure in two most common ways: free (unconstrained) layer damping treatment and constrained-layer damping treatment. The damping material is either sprayed on the body and floor panels or bonded using adhesive to provide damping. When the base structure is deflected in bending, the viscoelastic material deforms primarily in the form of extension and compression. The degree of damping is restricted by thickness and weight of base and damping materials. The system loss factor in a free-layer system increases with the thickness, storage modulus, and loss factor of the viscoelastic layer. Constrained-layer damping treatment consists of a thin layer of damping material combined with a constraining layer of metallic foil. When the base structure undergoes bending vibration, the viscoelastic material is forced to deform in shear because of the upper stiff layer. The constrained-layer damping is more effective than the free-layer treatment since more energy is consumed and dissipated into heat within the viscoelastic layer.

In addition, there is tuned viscoelastic damper (TVD) similar to a dynamic absorber or referred to as tuned mass damper. TVDs are generally applicable to reduce vibration/noise with a single frequency or a narrow band of frequency. Even if they are designed to reduce vibration/noise frequency at a given frequency, several TVDs with different frequency range have a wide band effect. The TVDs are very sensitive to the expected operating temperature range and the glass transition temperature of the viscoelastic material. Any temperature change in the damping material caused by energy dissipation into the internal heating is sufficient to alter the dynamic stiffness. This may lead the TVDs to detune itself. This characteristic of the TVDs makes elastomeric materials for TVDs only used in the rubbery region where slight changes in temperature do not have significant effect on the stiffness.

This paper first investigates new viscoelastic damping materials and appropriate damping treatments in sleeper carriages of electric trains. Further, experimental method on the measurement of vibrations and noise within carriages is provided. The effect of the damping material on vibration and noise reduction within carriages is discussed in term of physical parameters such as accelerations and sound pressure level. Finally, taking into consideration the subjective sound comfort within railway vehicle, loudness evaluated gives more valuable insight into the influence of damping materials on internal noise.

2. New viscoelastic damping materials

The material damping is able to extract mechanical or acoustical energy from a vibrating system and convert it into heat, by taking advantage of the viscoelastic damping capacity around the glass transition region. Taking into account the spectral characteristics in internal noise in railway vehicles, three new types of damping materials, such as bitumen-based damping materials, butyl rubber damping materials, and water-based damping coating, are developed for damping treatment of railway carriages to reduce the dominant components of noise within carriages.

Bitumen-based damping material is a highly viscoelastic material made from bitumen fillers with added mineral fillers and synthetic rubber. Temperature, excitation amplitude and vibration frequency have great effect on the dynamical mechanical performance of normal bitumen-based damping materials. Normal bitumen materials are brisk at low temperature and have a narrow temperature band. The storage modulus, mechanical loss factor and sensitivity to high temperature were increased by adding SBS to bitumen-based damping materials. A certain amount of clay, petroleum resin and $CaCO_3$, individually, was added to SBS-modified bitumen-based damping material further to improve dynamical mechanical and acoustic properties by us. The 40 samples of different blend ratios of SBS-modified bitumen materials and these three fillers were obtained.

Butyl rubber is the copolymer of isobutylene and a small amount of isoprene. Low levels of unsaturation between long polyisobutylene segments result in the excellent impermeability, air retention and good flex properties of butyl rubber. Research on butyl rubber damping material was involved in the improvement of its property in lower temperature and the extension of its functional zone to higher temperature. The lower temperature property of butyl rubber can be enhanced by blending with other elastomer and co-curing. The functional zone was extended to higher temperature by adding resin composition in butyl rubber. Petroleum resins were added in order to improve the adhesive strength on the surface. Our chlorinated butyl rubber damping materials were binary chlorinated butyl rubber/AO-80 blend and chlorinated butyl rubber/petroleum resin blend. In addition, AO-80 was added to CIIR/petroleum resin blend to regulate the damping property of the binary blend. Sixteen compounds of butyl rubber were obtained as candidates for the acoustic experiments.

Furthermore, 10 among these bitumen samples and four samples selected from butyl rubbers were adhered to 1.2 mm thick steel sheets, which were tested in the acoustic experiment. The dynamic mechanical measurements of samples for bitumen- and butyl-based damping material were conducted.

Water-based damping coating is one kind of spray coating, which is made of water-based resin, dispersant, and a large amount of mica and $CaCO_3$. The ability of water-based coatings to suppress the vibration and to insult the noise depends on material loss factor. We investigated the influence of blended resin system and mica with different diameter and amount on properties of damping coatings. With the increase of the size of mica, the temperature range of high damping became wider and damping peak value decreases. The increase in amount of mica decreased the sensitivity to high temperature. Two kinds of resin blending systems with wider temperature region and higher loss factor were chosen as experiment subjects. Damping coatings of this type were specifically intended to reduce internal vibration and noise in rolling stocks. Ten different thick laminates of water-based damping materials and stick steels were used in the measurement campaign.

The sound absorption and transmission loss of about 80 samples of three types of damping materials were measured by the method of standing wave separation [20]. The sound transmission loss of the least efficient and the most efficient damping materials among three kinds is illustrated in Fig. 1.The three types of damping materials of optimal transmission loss and higher loss factor as shown in Fig. 1(a), (c) and (e), were selected for the survey study. The bitumen-based damping materials in Fig. 1(a) have higher transmission loss at low frequency than the other two damping materials shown in Fig. 1(c) and (e). These three types of viscoelastic damping materials using damping treatment method mentioned above have been applied to the luxury sleeper carriage to investigate the optimal reduction effect of damping materials on noise and vibration. The dynamic mechanical and other sound properties of these three materials are referred to in the literature [21–24]. The damping materials and spraying coatings were developed by us.

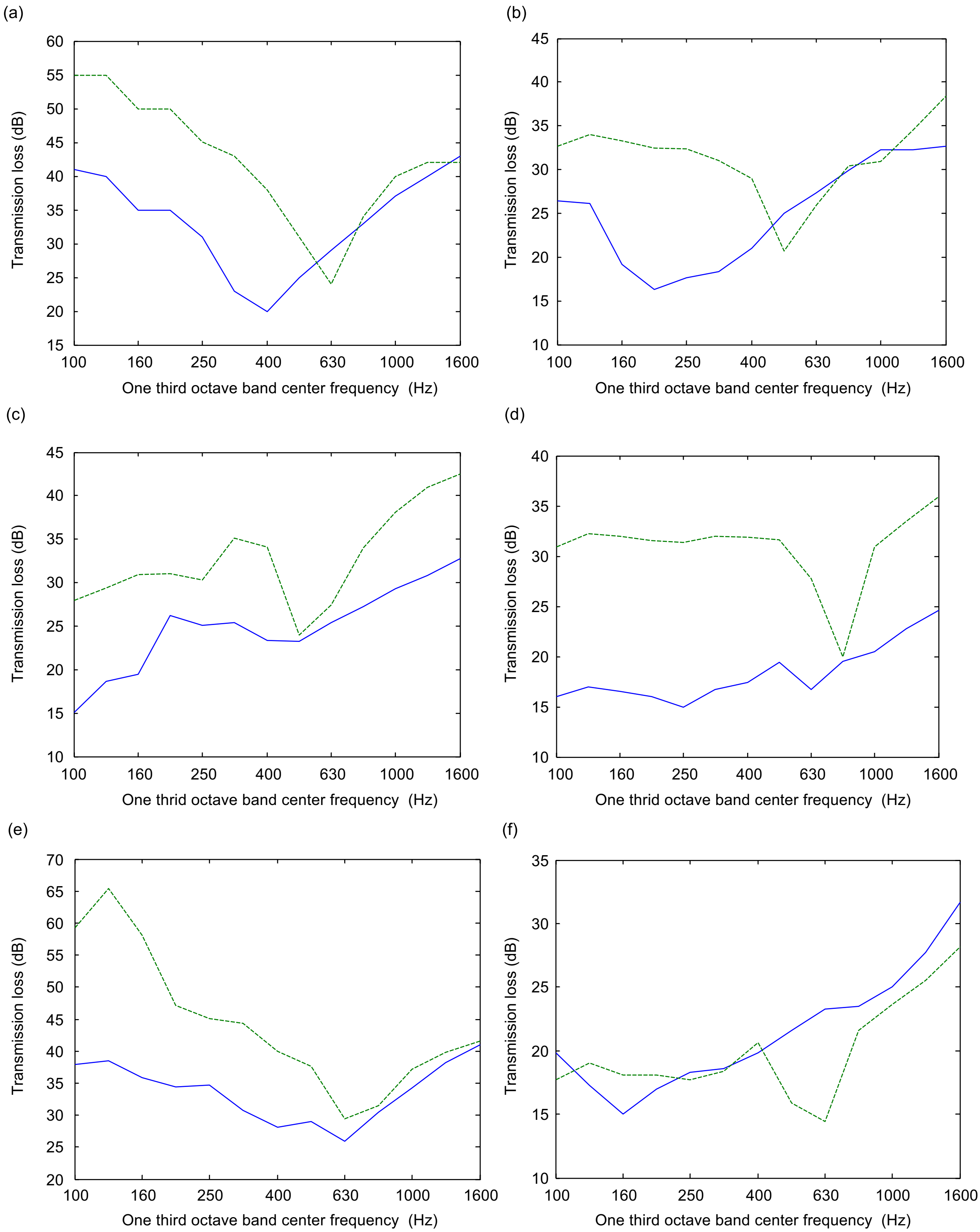
Bitumen based and butyl rubber damping sheet were designed to isolate the transmission of vibration from the bogie frame to the car floor and attenuate the vibration of the wall panel of car body. Water-based damping compound of synthetic resin and fillers is suitable to spray onto the whole internal surfaces of the car body to prevent the transmission of rolling noise through car body.

These three types of damping materials were installed on two carriages C1 and C2. The entire installation of damping materials on the carriage C1 is shown in Fig. 2. The two sleeper carriages C1 and C2 were, respectively, equipped with 3.0 mm thickness of bitumen-based damping sheet and butyl rubber damping sheet on the inner surface of corrugated steel panel under the car floor, the upper surface of the floor panel and the side wall 484 mm high above the floor surface, as shown in Figs. 2 and 3(a). Furthermore, as shown in Fig. 3(b), the water-based damping compound was sprayed onto the whole wall surface of the carriage C1 to replace the sprayed common damping material on the normal sleeper carriage C3.

3. Test methods and setup

To compare between the reduction effect of damping materials on internal noise and vibration, the measurement of vibration and noise on two damping treatment carriages C1 and C2 and the normal carriage

Fig. 1. Sound transmission loss in one-third octave bands: (a) the best efficient bitumen-based damping material, (b) the least efficient bitumen-based damping material; (–) bitumen-based damping sheet with thickness of 2.5 mm; (- - -) the laminate consisting of 1.2 mm thick steel sheet and 2.5 mm thick bitumen-based damping sheet. (c) the best efficient butyl rubber damping material; (d) the least efficient butyl rubber damping material; (–) 3 mm thick butyl rubber damping sheet; (- - -) the laminate consisting of 1.2 mm thick steel sheet and 3 mm thick butyl rubber damping sheet. (e) The most efficient water-based damping coatings; (f) the least efficient water-based damping coatings; (–) 2.4 mm thick water-based damping coating; (- - -) the laminate consisting of 1.2 mm thick steel sheet and 2.4 mm thick water-based damping coating.


(a)
Transmission loss (dB)
60 55 50 45 40 35 30 25 20 15
100 160 250 400 630 1000 1600
One third octave band center frequency (Hz)
(b)
Transmission loss (dB)
45 40 35 30 25 20 15 10
100 160 250 400 630 1000 1600
One third octave band center frequency (Hz)
(c)
Transmission loss (dB)
45 40 35 30 25 20 15 10
100 160 250 400 630 1000 1600
One thrid octave band center frequency (Hz)
(d)
Transmission loss (dB)
40 35 30 25 20 15 10
100 160 250 400 630 1000 1600
One thrid octave band center frequency (Hz)
(e)
Transmission loss (dB)
70 65 60 55 50 45 40 35 30 25 20
100 160 250 400 630 1000 1600
One third octave band center frequency (Hz)
(f)
Transmission loss (dB)
35 30 25 20 15 10
100 160 250 400 630 1000 1600
One third octave band center frequency (Hz)

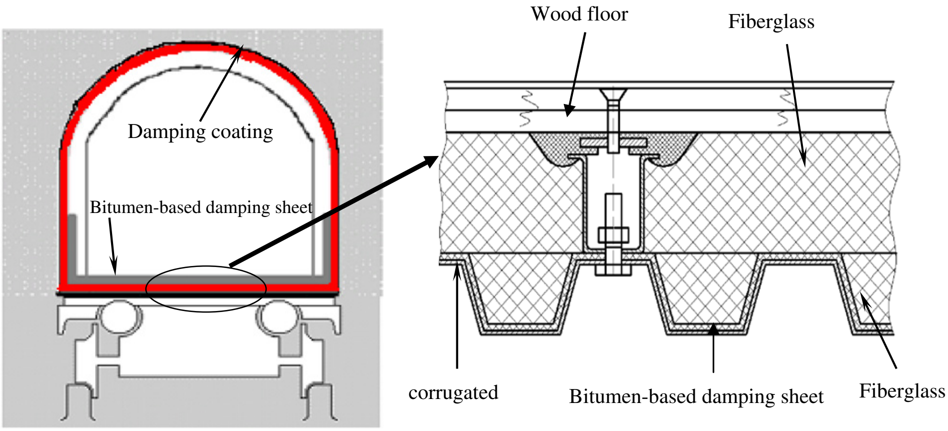


Fig. 2. Schematic diagram of installation of water-based coatings on the whole internal car body and bitumen-based damping sheet on the sidewall and floor panel in the carriage C1.

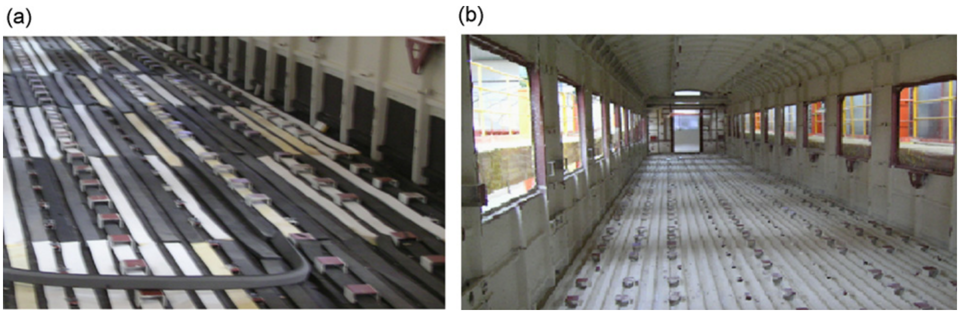


Fig. 3. The damping treatment of the car body: (a) the equipment of damping sheets on the corrugated steel panel and side wall of the car body and (b) the car body sprayed with water-based damping coating.

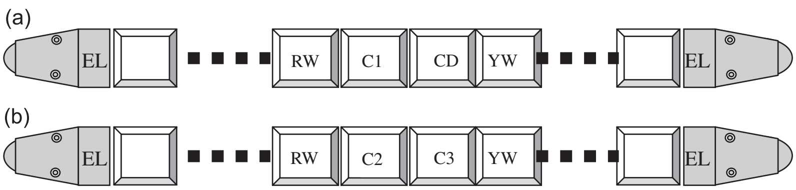


Fig. 4. Two train-sets configuration: (a) DC600v train and (b) intercity train. EL, electric locomotive; RW, top grade cushioned sleeper carriage; C1, test cushioned sleeper carriage; CD, dining car; YW, cushioned sleeper carriage; C2, the tested carriage; C3, the tested normal carriage; EL, electric locomotive.

C3 was carried out. Limited by the carriage manufacturer and train line management, these three carriages were deployed in two intercity train-sets of 12 cars, each composed of two electric locomotives plus trailer cars. The DC600v train-set configuration with the spare carriage C1 is shown in Fig. 4(a). The other two carriages C2 and C3 were parts of the other intercity train-set whose configuration is demonstrated in Fig. 4(b). Both trains transited on the main truck line between West Beijing and Xi'an. The test series fell into two major parts. One was the static test for background noise and the other was the running test in this case where both vibration and noise measurement were taken. The test site was on the main line between Xi'an and Yan'an, where the ballast was traditional. The testing train-sets were traveling on straight level track free of rail joints, dipped welds, and crossings at speeds from 90 to 160 km/h. Air conditioning system operated at maximum load. The compartment doors, the gangway doors between carriages and entry doors as well as intermediate doors were closed. Taking into consideration the spatial complication of internal noises, there were two different setups of noise measurement locations within the entire carriage and one compartment, respectively.

The internal configuration of three carriages that were tested is shown in Fig. 5. Each carriage has nine four-sleeping berth compartments. Since the vertical accelerations on the car body floor and the sleeping berth were measured as a ride comfort criterion, seven locations in three compartments 1, 4 and 9 were chosen to measure the vertical vibration during operation. The center of the 1st compartment next to a bathroom is 3 m away from the center of the nearest bogie. The 4th compartment is in the middle of both bogies. Compartment 9 is above the bogie at the other end of the car body. Accelerometers were installed on vibration measurement locations 1, 3 and 5 on the car body floor and measurement locations 2, 4 and 6 on the sleeping berth. Accelerometer 7 was used for the vibration under the floor above the center of the bogie, as shown in Fig. 6. Additionally, six microphone positions denoted by six consecutive characters from A to E, referred to the standard GB/T 3449-94 [25], were set up in the same three compartments as the vibration measuring points 1–6 within the carriage. The layout of two microphones in one compartment is seen in Fig. 7. The three measuring noise locations A–C for seated persons are at a height of 1.2 m above the floor of the car body in the center of a closed compartment. The other three microphones D–F are 0.2 m above the pillow on the lower sleeping berth.

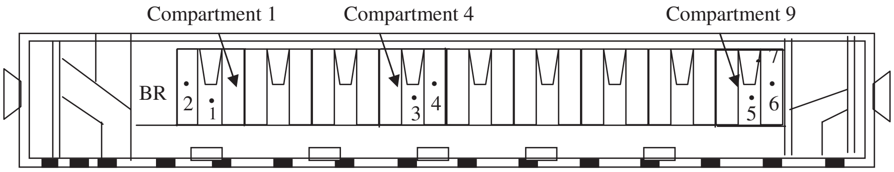


Fig. 5. Setup of the accelerometers on the three carriages: the measurement locations 1, 3 and 5 on the car body floor; the measurement locations 2, 4 and 6 on the lower berths measurement location 7 under the car body floor frame; BR, bathroom.

Fig. 6. Schematic diagram of the accelerometer location 7 under the vehicle floor.

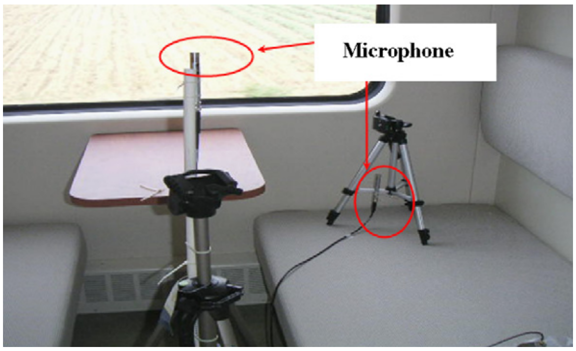


Fig. 7. The arrangement of two microphones in compartment 9.

The tests for noise and vibration were carried out simultaneously. There were two types of data acquisitions. In one case, the noise and vibration measurement data within three carriages were individually collected. For the convenience of high comparability of experimental data within two adjacent carriages C2 and C3, the accelerometers and microphones were sequentially placed in three tested compartments within these two carriages. The sets of vibration and noise data within two carriages were acquired at the same time. The experiment of noise and vibration within the carriage C1 was carried out and the data were alone collected when the train was running under the same conditions of speed and track section as the other train.

4. Results and discussion

4.1. Effect of damping treatment on the physical parameters

4.1.1. Acceleration

For the assessment of the effect of the vibration on the comfort of the passenger and crew in normal health, frequency-weighted root-mean-square (rms) values of the measured accelerations in vertical direction are evaluated according to ISO 2631-1:1997 [26]. Table 1 gives unweighted rms accelerations for the vertical vibration at every measurement point on each carriage, respectively, for running speeds of 100 and 150 km/h. It is clear that vibration magnitudes of the floor at either end of the car body are higher than that in the middle of the car body at both speeds, due to the local deformation of the car body. As the speed increases, the vibration levels at every point in three carriages go up.

For example, for the carriage C2, the unweighted rms acceleration at location 1 is 0.62 m/s^2 in contrast with vibration of 0.32 m/s^2 at location 3 in the case of running speed of 150 km/h. This difference results from the effect of transmission of the bogie frame vibration to the floor frame.

The vibration levels on the floor and the berth pad are much lower within both damping-treated carriages C1 and C2 than within the normal carriage C3. By subtracting vibrations at each location of the carriages C1 and C2 from the vibrations corresponding to locations in the carriage C3, the reduction amount of vibration within two carriages treated by damping materials can be seen more clearly as presented in Fig. 8. Although there is a wide variation in the vibration reduction at different measurement points in each of the carriages C1 and C2 in comparison with the carriage C3, the effect of the vibration decrease is more appreciable for the C1 carriage. From Table 1 it is seen that the overall reduction of 65% and 17% for vibrations at location 5 in the carriages C1 and C2, respectively, are achieved at the speed of 150 km/h. Both the floor structural complexity and different excitation under the floor are attributed to the distinction in vibration reduction between different measurement locations.

Since there exist the high-frequency components of vibration at the floor of the car body, only the vertical vibrations at the sleeping berth are weighted by a frequency weighting curve W_k associated with riding comfort suggested in the ISO 2631-1:1997. Vibration in the frequency region of 0.1–0.5 Hz is responsible for the motion sickness. For the frequency region of 0.5–80 Hz considered for comfort, the human body can be approximated as a system of particles with different resonance frequencies. Important resonance frequencies

Table 1
Unweighted rms acceleration levels (m/s^2) in the vertical direction for vibration measurement points within the three carriages at two running speeds

Running speed (km/h)	Carriage	Measurement points						
		1	2	3	4	5	6	7
100	Carriage 1	0.35	0.07	0.18	0.08	0.43	0.11	0.84
	Carriage 2	0.36	0.12	0.20	0.17	0.79	0.19	1.28
	Carriage 3	0.85	0.18	0.26	0.34	1.02	0.34	1.34
150	Carriage 1	0.35	0.10	0.15	0.09	0.42	0.14	0.97
	Carriage 2	0.62	0.14	0.21	0.19	1.01	0.22	1.39
	Carriage 3	1.06	0.20	0.29	0.42	1.21	0.44	1.33

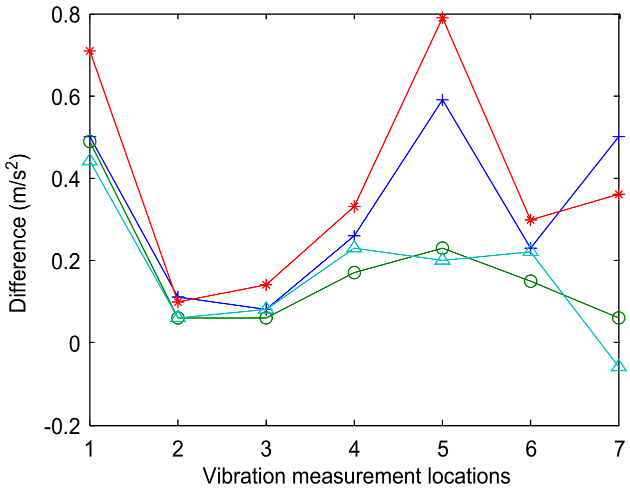


Fig. 8. Vibration acceleration differences between the carriages C1 and C3, C2 and C3: (-+-) carriage C1 at 100 km/h; (-○-) carriage C2 at 100 km/h; (-∗-) carriage C1 at 150 km/h; and (-△-) carriage C2 at 150 km/h.

Table 2
Frequency-weighted rms accelerations (m/s^2) in the vertical direction for vibration measurement points within the three carriages at two running speeds

Running speed (km/h)	Carriage	Measurement points		
		2	4	6
100	Carriage 1	0.06	0.06	0.07
	Carriage 2	0.12	0.10	0.11
	Carriage 3	0.17	0.16	0.17
150	Carriage 1	0.09	0.07	0.08
	Carriage 2	0.14	0.11	0.12
	Carriage 3	0.20	0.17	0.18

are 4–8 Hz for stomach, 20–25 Hz for head/shoulders and 30–80 Hz for eyeloobs. These overall frequency-weighed rms accelerations in the vertical direction for comfort are given in Table 2. The weighted rms accelerations show the reducing effect of the damping materials on the internal vertical vibration.

The comparison of the unweighted and weighted acceleration spectra in one-third octave band center frequency between three carriages is made with respect to vibration related to comfort given in Figs. 9 and 10. The acceleration spectra at the measurement point 2 look different from those at the point 6. It is clear from Figs. 9 and 10 that the frequency contents in the center frequency range of 6.3–10 Hz are dominant within three carriages. The human body is most sensitive to the vibration in the frequency range of 4–10 Hz according to the weighting filter W_k in ISO 2631. The damping materials within the carriage C1 suppress the most influential vibrations to greater extent than butyl rubber does within the carriage C2, as seen in Figs. 9 and 10. Due to the enormous decrement in the vibration in the center frequency range from 6.3 to 10 Hz inside both carriages C1 and C2, the frequency weighting curves for comfort in this frequency span in Fig. 10 are very similar to what the unweighted rms accelerations suggest. Weighting factors below 1.25 Hz and above 20 Hz attenuate the vibration components. The damping materials within both carriages C1 and C2 work less efficiently in these ranges. Therefore, it is found that the comfort weighting evaluation of vibrations is more suitable to indicate the reduction effect of three kinds of the damping materials on the vibration in the frequency range relative to motion comfort. As the overall accelerations and one-third octave band acceleration spectra show, the carriage C1 is the most suitable to travel in.

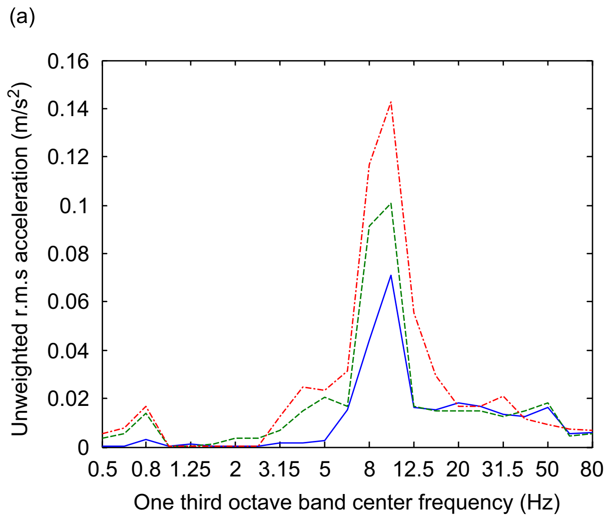


Fig. 9. Comparison of unweighted acceleration spectra at the measurement locations 2 and 6 inside the three carriages at the speed of 150 km/h: (a) location 2 and (b) location 6; (–) carriage C1, (– –) carriage C2 and (– · –) carriage C3.

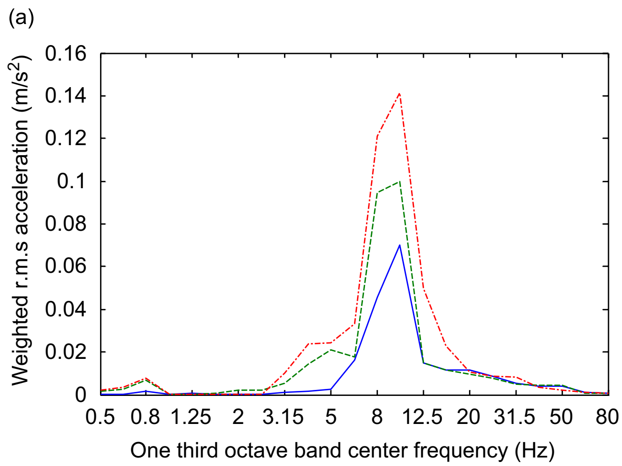


Fig. 10. Comparison of weighted acceleration spectra at the measurement locations 2 and 6 inside the three carriages at the speed of 150 km/h: (a) location 2; (b) location 6; (–) carriage C1; (– –) carriage C2; (– · –) carriage C3.

The distinctive effects of damping materials maybe result from the different damping loss factor of bitumen- and butyl-based rubber damping sheet to attenuate the vibration of the wall and the under-floor corrugated steel. The spraying coatings on the whole wall of the carriage C1 may contribute to the vibration damping to a certain degree.

The vibration in each one-third octave band at the location 1 on the floor in three carriages is illustrated in Fig. 11 for two speeds of 100–150 km/h. The floor vibration is mainly dominated by the frequency range from 0.5 to 80 Hz, as seen in Fig. 11. The car body rigid vibration was assumed to be dominant in low frequency up to 25 Hz and therefore the results for vibration described below at the location 4 on the sleeping berth include only the contribution of frequencies from 0 to 25 Hz, as shown in Fig. 12. From these figures it is found that the components in the whole frequency range have been apparently decreased for the two carriages C1 and C2 for two speeds of 100 and 150 km/h. Comparing the floor vibrations in the carriage C1 with the carriage C2, the carriage C1 has a wider frequency range where dominant frequency components diminish more sharply. The floor vibration at frequencies above 200 Hz in the carriage C1 has been significantly decreased whereas the

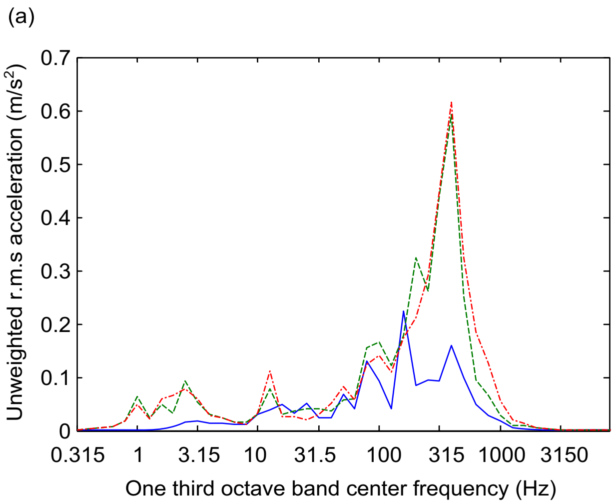


Fig. 11. Comparison of accelerations at the measurement location 1 on the floor within the three carriages for two speeds: (a) the speed of 100 km/h and (b) the speed of 150 km/h; (–) carriage C1, (– –) carriage C2, and (– · –) carriage C3.

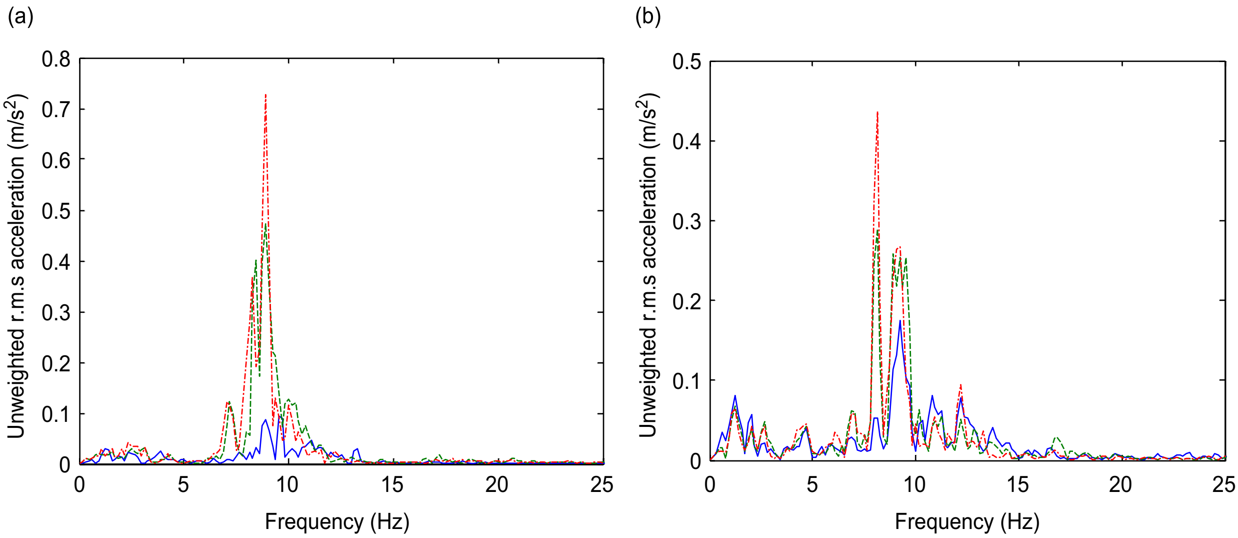


Fig. 12. Comparison of accelerations at the measurement location 4 on the berth pad within the three carriages for two speeds: (a) the speed of 100 km/h, and (b) the speed of 150 km/h; (–) carriage C1, (– –) carriage C2, and (– · –) carriage C3.

floor vibration in one-third octave bands covering the nominal center frequencies of 200–1000 Hz has been damped much less for the carriage C2.

It can be seen in Fig. 12 that a few characteristic frequencies of these three carriages are excited at two speeds of 100 and 150 km/h. The characteristic frequency with maximum amplitude for car body is near 8.75 Hz for each carriage at speed of 100 km/h. Vibrations at the smaller characteristic resonance frequency are found to become higher as the running speed increases. The train speed is mainly responsible for the excitation of different frequencies and amplitude of the carriages. The damping materials installed in the carriages C1 and C2 have reduction effect on the vibration in the resonant frequency regions near 8 and 10 Hz, as shown in Fig. 12. At resonance frequencies, the maximal decrease in vibration from 0.43 to 0.28 m/s^2 and to 0.05 m/s^2 for the carriage C2 and carriage C1 occurs at the speed of 150 km/h, respectively. From these results, it is evident that the higher vibration mitigation is achieved by the damping materials within the carriage C1.

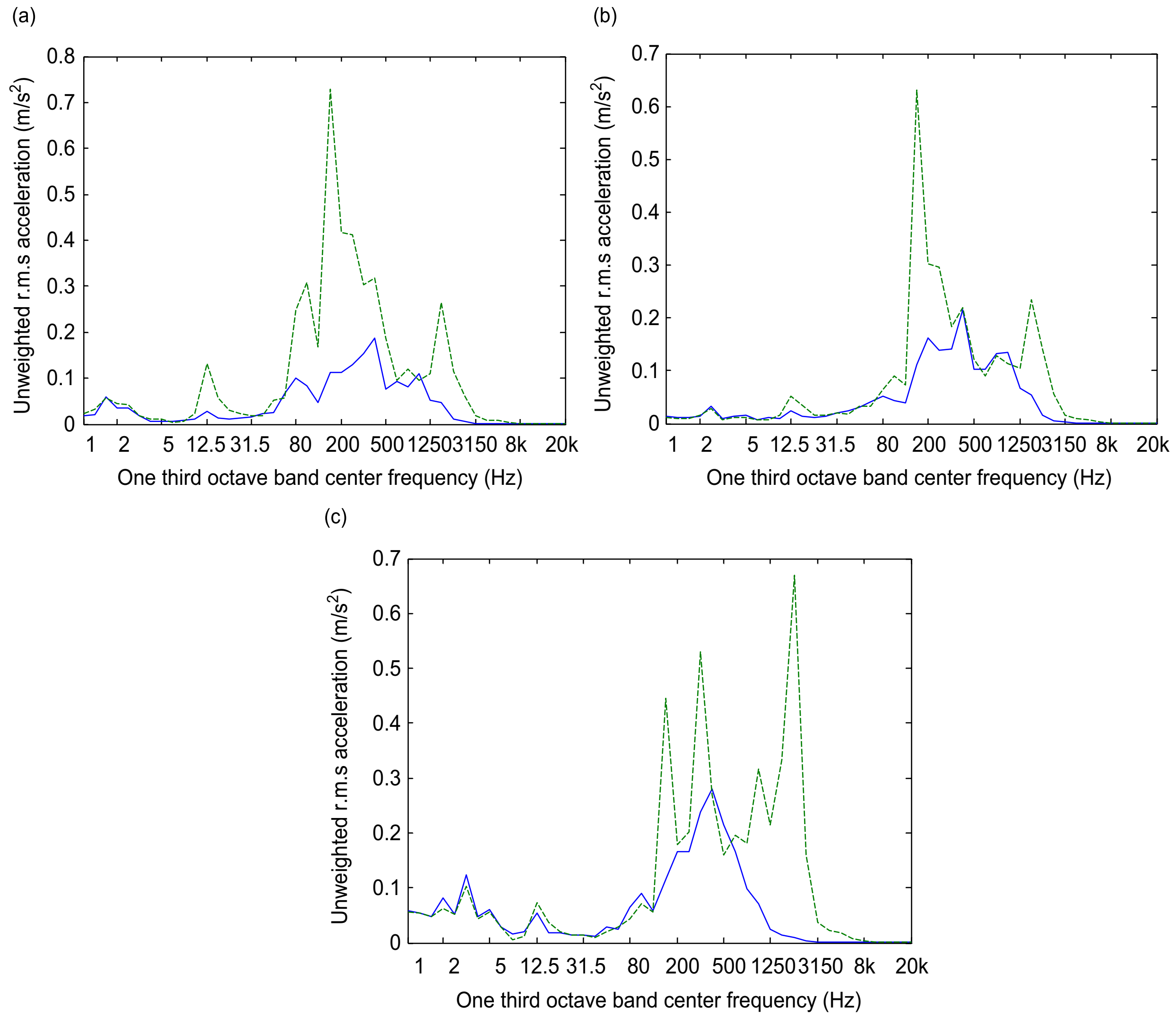


Fig. 13. Comparison of vibration magnitudes at two measurement locations 5 and 7 upper and under the floor panel for running speed of 150 km/h:(a) carriage C1, (b) carriage C2 and (c) carriage C3; (–) measurement location 5 and (––) measurement location 7.

Fig. 13 illustrates vibration magnitudes in one-third octave band above and below the floor panel from three carriages with the running speed of 150 km/h. The difference in vibration rms acceleration between both locations 5 and 7 for each floor is given in Fig. 14. From Fig. 14, it turns out that for the vibration acceleration difference at location 5 the vibration isolation of car body floor of the carriage C1 is the most effective in the major frequency range of 125–600 Hz among three carriages. As can be seen in Fig. 14, there are much higher vibration acceleration difference for the floor of the carriage C1 than for the other two carriage floors although vibration acceleration differences at frequencies from 0 to 500 Hz are almost similarly low for the floor of each carriage.

It is interesting to note that there is actually slight deterioration in vibration isolation at given low frequencies for the carriage C3. The deterioration varies from 0 dB up to approximately 10 dB at frequencies below 500 Hz. The vibration amplification on the top floor is the most serious near center frequency 8 Hz for the carriage C3, as seen in both Figs. 13(c) and 14. The damping treatments of the floor of the carriage C1 almost eliminate this phenomenon of vibration amplification. However, it seems in Fig. 14 that the maximum reduction in vibration in one-third octave bands of center frequencies above 500 Hz is achieved by the carriage C3, which is in contrast with what is previously stated. The acceleration difference curve of the carriage C3

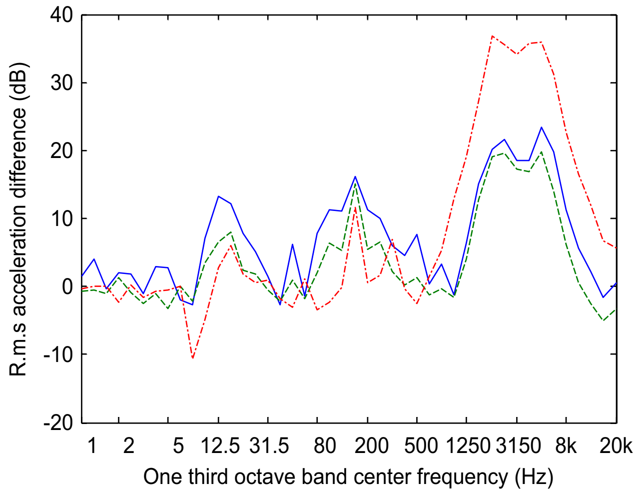


Fig. 14. Comparison of vibration acceleration difference between two measurement locations 5 and 7 upper and under the floor panel within the three carriages: (–) carriage C1, (– –) carriage C2 and (– · –) carriage C3.

Table 3
A-weighted equivalent continuous sound pressure levels (dBA) for six measurement points within the three carriages at two running speeds

Running speed (km/h)	Carriage	Measurement points					
		A	B	C	D	E	F
140	Carriage 1	55	51	60	61	57	64
	Carriage 2	58	54	62	65	58	66
	Carriage 3	62	57	67	66	62	71
150	Carriage 1	58	52	62	62	58	65
	Carriage 2	60	55	64	65	60	67
	Carriage 3	66	59	70	68	64	72

varies from −10 to 38 dB with the change in frequency. It appears that the bitumen-based damping sheet in the carriage C1 and butyl-based rubber damping sheet in the carriage C2 have a negative effect on the vibration reduction of the steel-corrugated floor. In fact, the discrepancy in calculated vibration isolation level among each carriage is due to the unnormalized vibration magnitudes under the car body floor, where the vibration at these frequency range 500–3150 Hz under the floor of the carriage C3 is the largest. The decrease in vibration in one-third octave bands above 500 Hz at measurement location 7 under the floor is believed to come from damping effect using the constrained-layer treatment of different damping materials such as bitumen-based damping and butyl-based rubber material. In addition, water-based coatings are helpful to attenuate the vibration of the floor of the carriage C1. As a result, the vibration acceleration difference insufficiently indicates the capacity of vibration insulation between upper and under car body floor. It is necessary to take the input to the floor vibration into account when evaluating the insulation effect of the damping treatments on the floor vibration.

4.1.2. Sound pressure level

The equivalent continuous A-weighted sound pressure level (L_{Aeq}) for integration time of 5 s, used as the noise indicator, is prescribed by GB/T 12816-91 “Evaluation about noise of railway passenger cars” [27], which establishes limits on indoor noise reception within the passenger cars. The values of equivalent continuous A-weighted sound pressure level L_{Aeq} at six measuring points within three carriages for two speeds of 140 and 150 km/h are given in Table 3. There exist distinguishing spatial distribution characteristics in the interior noise. The distribution of the sound level is dependent on the distance from the floor and stationary in space. The sound pressure levels at places at a height of 1.2 m above the floor is less than that at measurement points above the sleeping berth. The highest sound pressure levels are reached at each extremity of the car body above the bogies. Therefore, the sound pressure levels in the ninth compartment are the highest and the sound pressure levels in the center of the carriage are lower compared to two other compartments.

Since rolling noise is usually the dominant source of environmental noise arising from the operation of trains, the noise inside the carriage consists of structure-borne sound and air-borne and rolling noise transmitted through doors and windows. Just as the vibration under the floor is used as the referential factor of inside vibration reduction, the exact level of rolling noise, which can be measured by a microphone inside the bogie, should have been considered to evaluate the effect of the inside noise reduction. The interior noise is dominant in the low-frequency range of 25–250 Hz while most energy of rolling noise measured in other experiments is in the middle and high range of 200–3000 Hz. Therefore it is assumed that the most part of middle and high-frequency rolling noise is insulated by the car body and interior noise is principally radiated by the structure vibration. Meanwhile, limited to the number of signal channel, the rolling noise was not collected in the measurement of noise. The effect of damping materials on interior structure- and air-borne noise reduction is directly evaluated by comparison of the noise inside the prototype carriage C3 with those inside carriages C1 and C2. Although this method has a little influence on assessment of reduction effect of interior noise, under or overestimation of interior noise reduction presented in the paper can be ignored in this case.

The damping materials reduce the noise levels within the whole carriage. The noise reduction of 5–7 and 6–8 dBA, respectively, for the speeds of 140 and 150 km/h, are obtained in the carriage C1. The noise reduction varies at different locations, although the damping treatments do not alter the general trends of sound spatial distribution within three carriages. The increase in the running speed has positive influence on the overall sound pressure level. Both the overall noise level and the decrease in noise have been improved as the speed runs up, and meanwhile the traveling speed makes no change in spatial distribution of noise. The reduction effect of two patterns of damping treatments on interior noise is about 2–7% for the carriage C2 and 8–11% for the carriage C1 at 140 km/h. The noise decrement can be as low as 4–9% for the carriage C2 and 9–12% for the carriage C1 at 150 km/h.

The linear sound pressure levels in one-third octave bands at locations C and F in the ninth compartment, where noise levels are the highest among three points at the same height, are illustrated in Fig. 15. These figures show that the noise levels are dominated by the frequency components in the span of 25–250 Hz. The noise is decreased, to a higher degree, in entire frequency bands within the carriage C1 than within the carriage C2. From these results it is clear that the noise levels in dominant frequency bands from 50 to 250 Hz within these two carriages C1 and C2 are diminished for two speeds. The difference in the sound spectrum levels at two measurement locations between three carriages become smaller in one-third octave bands below 125 Hz with increasing speed, as seen in Fig. 13(b) and (d). This means that the interior noise sources make almost constant contributions to noise levels at frequencies above 125 in the cases of two speeds. The efficiency of damping materials to attenuate low-frequency noise become lower as the speed increases, although the reduction of the overall A-weighted equivalent continuous sound pressure levels at these two measurement locations within the carriage C1 and C2 grows up with higher speed.

4.2. Assessment of the effects of reduction on sound comfort

Due to simplicity and convenience of A-weighted sound pressure level, where A-weighted filter is by inverting the 40 dB equal loudness curves at 1 kHz, the A-weighting has become a popular and often useful frequency weighting for assessing the perceived magnitude of noise. However, dBA is an overall value which may simulate neither the spectral selectivity of human hearing nor its nonlinear relation to sound intensity. As analysis of interior noise, the effects of low-frequency noise need particular concern because of its pervasiveness and efficient propagation. The study of psychoacoustics [28] suggests that A-weighting underestimates the sound pressure level of noise with low-frequency components. The reason is that loudness increases due to bandwidth increase and that spectrum shape is not accounted for to a satisfactory degree by the A-filter. Thus if sounds with different spectral envelops are compared, the dBA value obtained may be an inaccurate indicator of human subjective response.

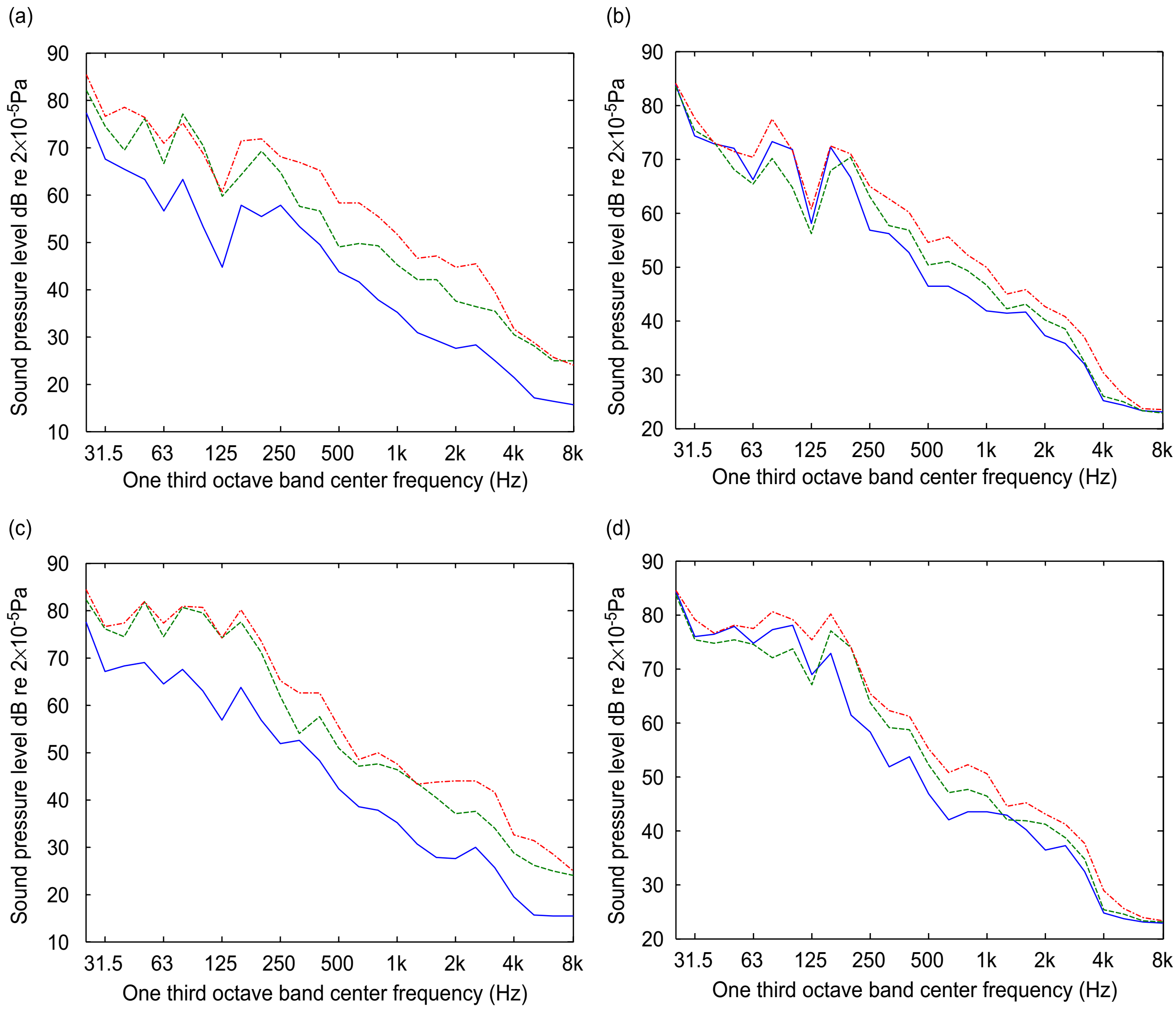


Fig. 15. Comparison of linear sound pressure levels at measurement locations C and F within the three carriages for two speeds of 140 and 150 km/h: (a) location C for 140 km/h, (b) location C for 150 km/h and (c) location F for 140 km/h, and (d) location F for 150 km/h; (–) carriage C1; (– –) carriage C2 and (– · –) carriage C3.

With a growing interest in the concept of "passenger comfort" rather than an objective measurable quantity, the acoustic environment inside a vehicle is judged in complex ways. A variety of criteria may be considered suitable for the quantification of interior train noise. These include the noise criteria, the preferred noise criteria, the noise rating, the room criteria, the related acoustic quality index, the loudness level and simple A–D-weighted sound pressure levels.

Subjective assessment of noise within passenger trains [29–31], indicates that the A-weighted sound pressure level has not been found to correlate well with perceived acoustic comfort and that a decrease in A-weighted sound pressure level does not necessarily result in a corresponding increase in the sound comfort level. As for British Railways of air-conditioned rolling stock, measures taken to reduce the L_A led to increased levels of complaints. It is reported in Ref. [29] that the use of "Room Criterion" can represent sound perception rather than A-weighted SPL. When the preferred speech interface level (PSIL) is below a certain threshold, Hardy said that the noise spectral shape should be the dominant factor. Binaural sound experiments [30] reveal that loudness is the most important parameter. A simple parameter based on the specific loudness calculations is developed to describe the different appreciations of the low-frequency content of sounds. Therefore, to

Table 4
Overall loudness (sones) and overall loudness level (phons) for measurement points C and F within these three carriages at two running speeds

Running speed (km/h)	Carriage	Loudness		Loudness level	
		C	F	C	F
140	Carriage 1	10	10	73	74
	Carriage 2	18	20	82	84
	Carriage 3	24	25	86	86
150	Carriage 1	16	17	80	81
	Carriage 2	18	20	82	83
	Carriage 3	22	24	84	86

identify the effect of noise reduction on the comfort level of carriage, the loudness level of sound is used as an alternative to A-weighted sound pressure level to access the efficiency of damping material.

The loudness model from Moore and Glasberg [32,33] was used in this work. This model differs from Zwicker's one in the following aspects: the assumed sound transmission through outer and middle ears; the calculation of excitation patterns from auditory filters; the critical bandwidth, especially for the frequencies below 500 Hz; the calculation of specific loudness from the excitation patterns. The presentation mode of loudness calculation for internal noises is diffuse field and monaural hearing.

Overall loudness values of interior noise at measurement locations C and F inside three coaches are presented in Table 4.The loudness values of noises within the carriage C1 keep minimal in these traveling conditions. Loudness reduction between each carriage treated by damping materials and the normal carriage C3 are, respectively, 6–15 sones for the carriage C1 and 4–6 sones for the carriage C2. The corresponding differences obtained in the loudness level are 4–13 phons for the carriage C1 and 2–4 phons for the carriage C2.

These specific loudness curves at two measurement locations C and F with three carriages, just as illustrated in Fig. 15, are shown in Fig. 16. The enormous differences in specific loudness of noises gives clearer indication of the evident effect of the damping materials on the sound comfort. By comparing the linear sound pressure levels in Fig. 15 and specific loudness Fig. 16, the specific loudness curves of noise more directly and accurately show the strong influence of the running speed on the ability of different damping materials to reduce the internal noise in the whole frequency range. At lower speed, the specific loudness of noises within the carriage C1 takes smaller value across the whole ERB scale.

Fig. 16(a) displays that noise specific loudness of each carriage reaches maximum value around 7.1 ERB_N number, and that specific loudness in the ERB_N numbers from 5 to 10, corresponding to 160–444 Hz in a frequency scale, makes most significant contributions to overall loudness of noises. The frequency sensitivity of auditory filter increases with increasing loudness level whereas human hearing system has lower sensitivity to the low-frequency noise. Consequently, since the noises at center frequencies below 160 Hz are at larger levels than at center frequencies above 500 Hz, the low-frequency components of noises obtain quite more intense perceived loudness. However, A-weighted sound pressure levels calculated at frequencies below 160 Hz or above 500 Hz take almost equal values. The influence of damping materials on the low-frequency noises within the carriages would be improperly judged if the A-weighted sound pressure level of noises were used as a measure to assess the effect of damping materials on the internal acoustical comfort.

It can be found that as running speed increases, the overall loudness of noises becomes larger and the reduction effect of damping material on the noise loudness C1 gets smaller within the carriage, as presented in Fig. 16. At higher speed, it can be seen from Fig. 16(c) and (d) that the major contributive specific loudness is in higher bands from 4 to 7 ERB_N numbers (when expressed in Hz, these bands correspond to 125–250 Hz). Compared to the carriage C3, the components at low frequencies below 125 Hz have a lesser influence on loudness of noises within the carriage C2 than that at frequencies above 250 Hz, as shown in Fig. 16(c) and (d). However, although the damping materials lower the peaks of the specific curves for noises for carriage C1, the specific loudness of measurement location F at ERB_N numbers less than 4 excels at higher ERB_N numbers

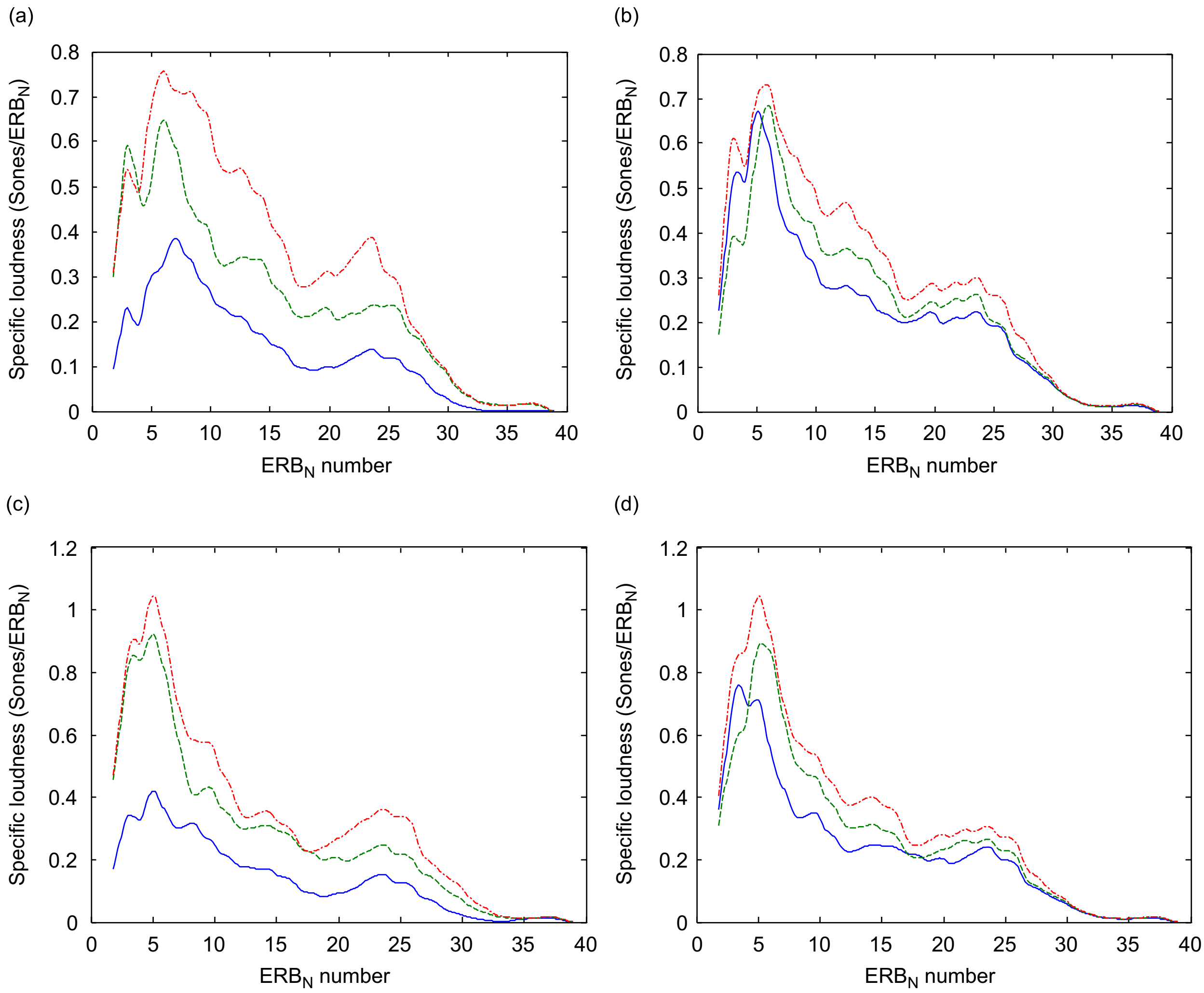


Fig. 16. Comparison of specific loudness of sounds at the measuring points C and F within the three carriages at speeds of 140 and 150 km/h: (a) location C for 140 km/h, (b) location C for 150 km/h, (c) location F for 140 km/h, and (d) location F for 150 km/h; (–) carriage C1, (––) carriage C2 and (– –) carriage C3.

within the carriage C1 at speed of 150 km/h. By taking into consideration vibration isolation of the car body floor of two carriages C1 and C3 shown in Fig. 11, the phenomenon described above can be explained by the decrease in the capability of the bitumen-based damping material and water-based damping coatings to isolate external noises.

5. Conclusions

The method of application of constrained-layer and free-layer damping treatments to luxury sleeper carriages has been developed. Running experiments show that three new damping materials can reduce the internal vibration and noise and provide a more comfortable traveling environment relative to motion and sound for the passengers.

The reduction effect of the internal vibration and noise has been shown to depend strongly on train speed and the measurement location selected. The use of proper sound measure such as loudness rather than A-weighing can evaluate and assess more accurately the efficiency of different damping materials to reduce the internal noise.

In the same measuring case, the damping treatment of both bitumen-based and butyl rubber damping sheets have been found to reduce vibration levels in the whole frequency range, to different degree, at every measurement location within the carriage. The former damping material along with water-based coating decreases the unweighted rms acceleration of the carriage by 0.08–0.79 m/s^2, and the latter by around 0.06–0.49 m/s^2. It is found that the bitumen-based damping material can reduce the vibration in much wider frequency range than the butyl rubber damping material. The damping materials reduce less efficiently the low-frequency vibration of the carriages. The damping treatment of carriages can attenuate the dominant components of noises at frequencies from 25 to 160 Hz. Partly because of the difference in the sound transmission loss for three damping materials, the reduction in A-weighted sound pressure level of noises is 5–8 and 1–6 dBA for bitumen-based damping material as well as water-based coatings and butyl rubber material while a decrease in noise loudness level by 4–13 and 2–4 sones occurs.

Acknowledgments

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