Car body floor vibration of high-speed railway vehicles and its reduction

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Abstract
An on-site test has been performed to address the problem of feet numbness caused by the floor vibration of high-speed railway vehicles. Analysis of the floor vibration performance indicates that the vibration of the car body chassis transmitted to the floor by the elastic supports is significantly amplified in the frequency range of 20–50 Hz. This overlaps with the frequency range in which human lower extremities are most sensitive, leading to feet numbness. A refined finite element model of the car body, including the floor panels is developed to further study the vibration mechanism of the floor. Results show that due to the inappropriate design of the elastic support stiffness, the deformations of the floor above the bogie centre for several typical modes in the frequency range of 20–50 Hz are significantly amplified. When the excitation frequencies transmitted from the car body chassis were close to the eigenfrequencies of the floor, the local resonance of the floor will occur, which is the root cause of human feet numbness. The dynamic stiffness of the elastic support is further optimised, and the experimental verification shows that the vibration transmissibility from the car body chassis to the floor in the frequency range of 20–50 Hz has been significantly reduced, and the problem of feet numbness has been solved.

Keywords
Railway vehicle, floor vibration, elastic support, modal parameters analysis

Introduction
Under the high-speed operational conditions, the vibration environment of the high-speed trains deteriorates rapidly, and the high-frequency elastic vibration of the car body is significantly increased,1 meanwhile, the vibration of internal structures of the car body is increasingly severe. In recent years, the researches on car body vibration of high-speed trains have received high attention. In these studies, the rigid-flexible coupled dynamics model including the car body elasticity was mainly utilized. Zhou et al.2 developed a model in which the car body was considered as an Euler beam with structural damping. Based on this, a vertical rigid-flexible-coupled dynamic model of railway vehicles was proposed, and the effects of the car body elasticity on the ride comfort were studied. The work of Zhou which found that the elastic vibration of the car body had a great influence on the ride comfort with the increasing running speed of the vehicles. Gong et al.3–6 present studies where the dynamic condensation method was applied to the FE model of the car body, and the three-dimensional rigid-flexible-coupled dynamic model of railway vehicles which included the car body elasticity was developed. The vibration and ride comfort of the car body under different operational conditions was studied.

It needs to be pointed out that in the above studies, the car body and its internal structures were considered as an integrated model, which was failed to study the vibration of internal structures of the car body in detail. Recently, local vibration problems of the internal structures of high-speed trains such as floors, seats, luggage
The floor is one of the most important car body panels which has amounts of interfaces with other components of the car body. The floor panel of high-speed trains is composed of honeycomb panels and is connected to the car body chassis by elastic supports (rubber components). The large area of the floor panel and relatively low Young’s modulus of its material result in an insufficient local stiffness for the floor itself. If the parameter design of the elastic support is improper, the floor vibration will be degraded significantly. Since the assessment of floor vibration is the most important way to evaluate the ride comfort of railway vehicles, it is necessary to develop a refined model of the floor for high-speed trains to investigate its vibration reduction in depth.

The studies on the floor panels of railway vehicles were mainly focused on their lightweight design, fatigue analysis and noise control. To reduce the car body weight and improve the dynamic properties of the car body, a sandwich panel substitution process was performed on corrugated board of the floor of a rail vehicle car body by Wennberg et al., and they found that the mass of car body could be reduced by 600–700 kg, and the varying importance of the longitudinal, transverse and shear properties of the floor panels can be improved. Hudson et al. performed a multiple objective optimisation of low mass and cost on composite sandwich panels for the interior floor structure of a metro car. They provided mass savings of up to 60% compared to the existing plywood-based floor systems.

Moreover, Qian et al. studied the floor chute fatigue performance for high-speed trains. Based on fatigue cumulative damage theory, they evaluated the fatigue durability of the floor chute suspension structure under the real dynamic load. Braga et al. proposed an innovative train car floor panel design using welded Al alloy components, the welding procedures required for manufacture were studied individually, and a prototype was built and tested using a combination of two experimental techniques: strain gauges and digital image correlation.

Tomonori and Yamamoto presented a new floor structure ‘suspended floor’ as one of the countermeasures against the interior noise of a railway vehicle. The floor panels are suspended from side structural panels of the car body to reduce structure-borne noise in a railway vehicle. Ulf et al. calculated wavenumbers and transmission losses for composite sandwich panels with honeycomb cores, and the prediction and control of sound transmission through honeycomb sandwich panels for train floors were conducted to achieve designs that satisfy both acoustical and structural requirements.

Takigami et al. designed a rigidity test to study the influences of non-structural members in the car body on its rigidity and vibration, and they found that the ring structure (i.e. the additional and/or reinforced inner frames) improved the rigidity of the car body without changing the outer shell; however, the vibration amplitude on the floor did not decrease drastically simply as a result of adding rigidity to the car body. To the best of the authors’ knowledge, few studies on the floor vibration of high-speed trains have been reported.

In this study, the main objective of the present paper is to solve a problem of feet numbness caused by floor vibrations of a high-speed train which significantly degrades the ride comfort. First, the root cause of the floor vibration was studied by the analysis of vibration transmission through on-site test and FEA. Then, based on the vibration mechanism of the floor, the vibration reduction scheme was studied and verified experimentally.

**On-site test analysis**

First, an on-site test of the studied high-speed trains was conducted. The vibration transmission and under-chassis-equipment vibration transmission were analysed experimentally in order to explore the root cause of the floor abnormal vibration. The vertical vibration accelerations of following locations were measured: the wood floor, the elastic support (a rubber spring with stiffness of 450 N/mm), the traction converter, the waste discharge unit and the car body chassis. The accelerometers were installed on the car body first (see Figure 1),
where point A is on the surface of the car body chassis; point B is on the surface of aluminum alloy sheet of the elastic support; Point C is on the surface of rubber of the elastic support; Point D is on the surface of floor above the elastic support; Points E and F are the measuring points on the floor with a longitudinal distance of 100 and 200 mm from point D, respectively; in addition, point F is right above the centre of the bogie for measuring the UIC comfort index. Figure 2 shows the arrangement of the measuring points A to F of the on-site test.

Since some under-chassis-equipment, such as the waste discharge unit and traction converter, which has their own excitation may be the reason of the floor abnormal vibration. The accelerometers were installed on these two under-chassis-equipment as well to analyse the vibration transmission from them to the car body, as shown in Figure 3. Here, the waste discharge unit point I and under-chassis point I were on both the ends of the rubber spring of the waste discharge unit, respectively. Due to the big size of the traction converter, the measuring points G and H were arranged on the diagonal direction. Figure 4 shows the arrangement of the measuring points G to I of the on-site test.

Vibration transmissibility is an important performance evaluation indicator of a mechanical system. In this study, the vertical vibration acceleration transmissibility of the floor and the coherence function were analysed.

**Figure 2.** Arrangement of measuring points of the on-site test.

**Figure 3.** Measuring points of the under-chassis-equipment: (a) traction converter; (b) waste discharge unit.

**Figure 4.** Arrangement of measuring points on the under-chassis-equipment: (a) 200 km/h; (b) 250 km/h.
The coherence function $\gamma^2(\omega)$ of two time series $y_1(t)$ and $y_2(t)$ is defined as the square of the magnitude of the (complex) cross PSD of the two input series $S_{12}(\omega)$, divided by the product of the (real) PSDs of the two input signals $S_1(\omega)$ and $S_2(\omega)$, gives

$$\gamma^2(\omega) = \frac{|S_{12}(\omega)|^2}{S_1(\omega)S_2(\omega)}$$

(1)

Figure 5 shows the vertical vibration acceleration transmissibility of the floor at different positions when the vehicle ran at the speed of 200 and 250 km/h, respectively. Here, the vibration of the steel structure of the car body chassis at measuring point A was taken as the input, while the vibration at other measuring points B to F was taken as the output response. It can be seen that, when the vehicle was running at both speeds, all the values of vibration transmissibility from A to C, D, E and F exceeded 1 in the frequency range of 20–50 Hz, indicating an amplification of the vibration transmissibility. In addition, the vibration transmissibility from A to D, E and F went up with the increasing of the longitudinal distance away from the elastic support, and the maximum value of the vibration transmissibility exceeded 2. It needs to be pointed out that the frequency range of 20–50 Hz in which the vibration transmissibility magnifies is consistent with the most sensitive frequency range of the human lower extremities.\(^\text{17}\)

Figure 6 shows the results of coherence function between signals A and B to F when the vehicle ran at the speeds of 200 and 250 km/h, respectively. It can be seen that the coherence between the input signal A and the output signals B to F is very good ($\gamma(\omega) > 0.5$ in the frequency range 20–50 Hz), which verifies the above justification of the test.

The vertical vibration transmissibility from the under-chassis-equipment to the car body chassis was also analysed. Here, the vibration of the under-chassis-equipment was taken as the input, while the vibration at the car body chassis was taken as the output. Figure 7(a) and (b) depicts the results of the vibration transmissibility at the speed of 200 and 250 km/h, respectively. It can be seen that the vibration transmissibility from the under-chassis-equipment to the car body above 5 Hz was less than one at both running speeds indicating that the vibration has been effectively reduced by the rubber springs (see Figure 4(a)). One can conclude that the problem of feet numbness has nothing to do with the under-chassis-equipment.

**Finite element modelling of the railway vehicle**

To study the vibration reduction of the floor, the virtual prototype technology is used to develop a refined finite element (FE) model of the car body including the floor.

**Material tests of floor and wood bone**

First, material tests of the floor (wood plank) and the wood bone were conducted to obtain their material properties. In the tests, the non-constrained support was adopted. The bending stiffness of the specimen was
measured using the three-point bending method,\(^\text{18,19}\) and the Young’s modulus of the material \(E\) can be calculated by the following form

\[
\sigma = \frac{F}{3l^2} \quad \text{(2)}
\]

where \(l\) is the span length; \(F\) is the load corresponding to deflection (mm) at the mid-point of the specimen; \(I\) is the moment of inertia. The cross sections of the specimen are rectangular with the width \(b\) and height \(h\). Therefore, the moment of inertia \(I\) is given by the following equation

\[
I = \frac{1}{12}bhl^3 \quad \text{(3)}
\]

Combining equations (2) and (3), \(E\) is given as

\[
E = \frac{l^3}{4bh^3} \cdot \frac{F}{f} \quad \text{(4)}
\]

The bending stiffness tests of five specimens of wood plank and wood bone were conducted. Figure 8 depicts the results of the bending stiffness test for the wood plank and wood bone. It can be seen that under the same
The deflection of each specimen varies slightly. The displacement shows a quasi-linear relationship with the load within a certain range. Generally, the floor undergoes a very small amount of deformation caused by vibrations in practice, and a linear relationship between them is adopted. Therefore, an average value was taken from the five bending stiffness curves, and the slope of the linear region after averaging was calculated, i.e. $F/f$. Then, the Young’s modulus is obtained by utilizing equation (4). In this study, the Young’s modulus of the wood plank is 6.3 GPa, and the Young’s modulus of the wood bone is 8.5 GPa.

**FE modelling of the car body and floor**

A 3D solid model of the floor was meshed by utilizing the FE software to develop the refined FE model of car body. Figure 9 shows the 3D solid model of the floor where its damping was achieved through the foam mattress which stuck above the wood bones as well as the elastic supports (rubber elements). The foam mattress is a mixture of polyurethane foam with a thickness of 12 mm. The elastic support is composed of a layer of rubber material which is sandwiched between two aluminum-extruded sections. There are a total of 213 elastic supports in the model.

The rubber material displays linear elastic characteristics at small deformations. Therefore, an equivalent Young’s modulus of the rubber material of the elastic supports was taken into account. To obtain this equivalent Young’s modulus, a load was distributed uniformly and applied on the surface of the 3D solid model to calculate the deformation. By adjusting the Young’s modulus of the rubber material until the deformation matched with that of the practical stiffness of the elastic support. Ultimately, the equivalent Young’s modulus of the rubber material can be obtained by using this procedure. Figure 10 shows the schematic of the constraints and loads of the elastic support FE model. The obtained equivalent Young’s modulus of the rubber material (whose stiffness is 450 N/mm) is 1.8 MPa.

The geometric model of the car body was meshed using the FE software, and the floor FE model was combined with the car body FE model. The followings were attached under the chassis beam: a traction converter, power box and waste discharge unit. Also, an air conditioner was attached to the roof of the car body. Figure 11 shows the refined FE model of the car body (including the floor), which consists of 762,704 nodes and 628,541 elements.
In order to validate the above-mentioned modelling, the vibration response and vibration transmissibility analysis of the FE model were conducted. The obtained results were compared with those of the relative tests. Taking the measured forces above the air spring as the input (at a running speed of 250 km/h), the vibration acceleration of the floor was also calculated by using the FE model. Table 1 shows the acceleration results of the floor above the bogie obtained by test and FEA, respectively. It can be seen that the results of FEA are consistent with those of the test and with a maximum error of 7%.

Taking the measuring points D, E and F (see Figures 1 and 2) on the floor as the references, the vibration transmissibility from the chassis steel structure to the floor was calculated by using the FE model, and the results are shown in Figure 12. One can find that there is amplification of the vibration transmissibility from the chassis steel structure to the surface of the floor above the elastic support in the frequency range of 20–50 Hz, and there is a dominant vibration at 40–43 Hz; the vibration transmissibility increases with the increasing of the longitudinal distance from the elastic support, which agrees with the trend of the test results (Figure 5). Apparently, the FE model of the car body which includes the floor was reasonably developed.

Table 1. The acceleration results of the car body floor (at a running speed of 250 km/h).

|        | Maximum (m/s²) | Minimum (m/s²) | Root mean square (m/s²) |
|--------|---------------|---------------|------------------------|
| Test   | 2.010         | −1.499        | 0.388                  |
| FEA    | 1.898         | −1.394        | 0.373                  |

**Vibration nephograms analysis**

The FEA was further conducted to discuss the root causes of vibration amplification of the floor in the frequency range of 20–50 Hz. Figure 13 depicts the vibration nephograms of the chassis and floor for several typical modes. Figure 13(a) and (b) shows the first and the second bending modes of the car body chassis and floor.
Figure 12. Vibration transmissibility by FEA.

Figure 13. The vibration nephograms of typical modes of the chassis and the floor (original model): (a) the 8th order mode (9.501 Hz); (b) the 10th order mode (12.465 Hz); (c) the 13th order mode (14.867 Hz); (d) the 22nd order mode (21.190 Hz); (e) the 28th order mode (24.502 Hz); (f) the 46th order mode (34.755 Hz); (g) the 58th order mode (38.300 Hz); (h) the 62nd order mode (40.098 Hz); (i) the 70th order mode (42.318 Hz); (j) the 71st order mode (42.400 Hz).
panels, respectively. One can find that the car body chassis and floor panels of these two modes have similar deformations. Whereas, to other typical modes shown in Figure 13(d) to (j), whose eigenfrequencies are above 20 Hz, the deformations of the floor panels above the bogies are significantly larger than those of the car body chassis. If the excitation frequencies transmitted from the car body chassis were close to these eigenfrequencies, the local resonance of the floor panels would occur, which is the root cause of human feet numbness. Especially, due to the dense dominant modes of the floor shown in Figure 13(h) to (j), the floor might vibrate dramatically at 40–43 Hz (see Figure 12).

Floor vibration reduction analysis

Since the car body chassis and floor structure are not easily redesigned or optimised, the vibration reduction of the floor is focused on the adjustment of the dynamic stiffness of the elastic supports. The floor panel above the elastic supports can be treated as a wood plank with elastic support boundary conditions, as shown in Figure 14.

The differential equation of the wood plank can be written as follows\(^21\)

\[
D \nabla^4 w(x, y) - \rho h \omega^2(x, y) = 0
\]  

(5)

where \(\nabla^4 = \partial^4/\partial x^4 + 2\partial^4/\partial x^2 \partial y^2 + \partial^4/\partial y^4\) is the differential operator; \(w(x, y)\) is the vertical displacement of the floor; \(D\) is the bending stiffness of the floor; \(\rho\) is the density of the floor material; \(h\) is the thickness; \(\omega\) is the circular frequency. Boundary conditions are introduced to solve equation (5). The boundary conditions of the wood plank are as follows: the sheer forces of the support edges are equal to the elastic forces and the bending moments are 0

\[
\begin{align*}
  k_{x0}w = Q_x \\
  k_{y0}w = Q_y \\
  k_{s1}w = -Q_x \\
  k_{s1}w = -Q_y \\
  M_x = M_y = 0
\end{align*}
\]  

(6)

where \(k_{x0}, k_{y0}, k_{s1}\) and \(k_{s1}\) are the stiffness of the elastic supports on the edges \(x=0, x=l, y=0\) and \(y=l\), respectively; \(Q_x\) and \(Q_y\) are shear forces; \(M_x\) and \(M_y\) are the bending moments.

Equation (5) can be solved by utilizing the Rayleigh–Ritz method to obtain the matrix equation of the system

\[
(K_f - \omega^2 M_f)A_f = 0
\]  

(7)

where \(A_f\) is the series expansion coefficient matrix and \(K_f\) and \(M_f\) are the stiffness and mass matrix of the floor, respectively. Thus, the problems of solving the modal parameters (eigenfrequencies and eigenvectors) of the floor can be translated into those of solving eigenvalues from equation (7).\(^21\)

According to equation (7), one can find that the eigenfrequencies of the floor are affected by the stiffness of the elastic supports. Therefore, tuning the stiffness of the elastic supports will change the eigenfrequencies of the floor, and the eigenfrequencies of the floor may shift from its original frequency range of the resonance. However, there...
are a large number of floor modes in the frequency range of 20–50 Hz. Although adjusting the elastic support stiffness may separate the original resonant eigenfrequencies of the floor from the excitation frequencies, it may also generate new resonance between other modes of the floor and the excitation. Therefore, the dynamic stiffness of the elastic support needs to be further determined by an optimisation method. The method developed in this study are as follows:

1. The dynamic stiffness of the elastic support is suggested not to be too large or too small. If it is too large, the high-frequency vibration of the floor will be increased which affects the noise performance of the vehicle; on the contrary, if it is too small, it does not meet the static deflection requirements of the floor. According to the characteristics of the studied vehicle and the rubber of elastic support, the suggested range of the dynamic stiffness of the elastic support is from 200 to 3000 N/mm.

2. Within the range above mentioned, taking 50 N/mm as the step size to calculate the vibration response of the floor with different dynamic stiffness of the elastic support and finding the optimal dynamic stiffness coefficient of the elastic support obtaining the minimum root mean square (RMS) value of the floor vibration acceleration in the frequency range of 20–50 Hz.

The objective function and constraints of the optimisation method are

\[
\min g(k) = a_{\text{RMS}}, \quad \text{s.t.} \begin{cases} 20 \text{ Hz} \leq f_i \leq 50 \text{ Hz} \\ 200 \text{ N/mm} \leq k \leq 3000 \text{ N/mm} \end{cases}
\]  

(8)
where $k$ is the dynamic stiffness coefficient of the elastic support and $a_{RMS}$ is the root mean square value of the floor vibration acceleration.

Figure 15 depicts the results of vibration acceleration in the frequency range of 20–50 Hz versus the dynamic stiffness coefficient of the elastic support at point F by FEA. Here, the dynamic stiffness coefficient of the elastic support ranges from 200 to 3000 N/mm. It can be seen that with the increase of the dynamic stiffness coefficient of the elastic support, the vibration acceleration in the range of 20–50 Hz decreases first and then increases gradually.

Figure 16 depicts the results of RMS values of acceleration in the frequency range of 20–50 Hz versus the dynamic stiffness coefficient of the elastic support at point F by FEA. One can find that, as the dynamic stiffness coefficient of the elastic support increases, the RMS value of vibration acceleration in the range of 20–50 Hz

Figure 17. The vibration nephograms of typical modes of the chassis and the floor ($k = 1350$ N/mm): (a) the 8th order mode (9.507 Hz); (b) the 10th order mode (12.467 Hz); (c) the 13th order mode (14.873 Hz); (d) the 22nd order mode (21.246 Hz); (e) the 28th order mode (24.522 Hz); (f) The 46th order mode (35.074 Hz); (g) the 58th order mode (38.375 Hz); (h) the 62nd order mode (40.489 Hz); (i) the 69th order mode (42.216 Hz); (j) the 70th order mode (42.862 Hz); (k) the 71st order mode (43.292 Hz); (l) the 72nd order mode (43.449 Hz).
decreases first and then increases gradually, which is consistent with the trend in Figure 15; when the dynamic stiffness coefficient equals to 1350 N/mm, the RMS value of the acceleration achieves its minimum.

When the dynamic stiffness coefficient is 1350 N/mm, the vibration nephograms of the chassis and the floor panels for different modes are shown in Figure 17. Comparing with the results shown in Figure 13, the modal eigenfrequencies and mode shapes have been changed accordingly with the optimised dynamic stiffness coefficient of the elastic supports. At this time, the deformations of the floor panels above the bogies are not significantly amplified by comparing with the chassis, especially for high number order modes. One can also find that the dominant modes of the floor panels at 40–43 Hz disappear due to the optimised dynamic stiffness, which is benefit to the vibration reduction of the floor itself in these frequency range.

**Experimental verification**

The optimisation of the dynamic stiffness of the elastic support was verified by on-site tests. In the tests, the dynamic stiffness of the elastic supports was set to 450 and 1350 N/mm, respectively, while the other operational conditions of the vehicle remained the same (the same railway line and the same running speed of 250 km/h).

![Figure 18. The results of acceleration at point F.](image)

![Figure 19. The results of acceleration transmissibility.](image)

| Stiffness (N/mm) | RMS value (m/s²) |
|------------------|------------------|
| 450              | 0.394            |
| 1350             | 0.272            |
Figure 18 shows the results of acceleration at point F (above the center of the bogie), and Table 2 illustrates the relevant acceleration RMS values. From the results, one can find that, comparing with the original stiffness (450 N/mm), the floor vibration has been reduced significantly by the optimised dynamic stiffness of 1350 N/mm, and the acceleration RMS value has been decreased from 0.394 to 0.272 m/s².

Figure 19 shows the results of acceleration transmissibility from point A to points E and F (their positions are the same as those shown in Figures 1 and 2). It can be concluded that, comparing with the original stiffness, when the stiffness is 1350 N/mm, the vibration transmissibility from the car body chassis to the floor is significantly decreased in the frequency range of 20–50 Hz. Also, the maximum of the transmissibility of the latter is less than 1.3. The problem of feet numbness will be greatly improved at this optimised stiffness.

Conclusions
1. An on-site test was conducted for a high-speed train to investigate the problem of passengers’ feet numbness caused by floor vibration. The vibration transmissibility was analysed after the test. Results show that the vibration of the floor is amplified in the frequency range of 20–50 Hz. One of the main causes of feet numbness is largely due to the fact that this frequency range (20–50 Hz) overlaps with the range in which human lower extremities are most sensitive.
2. A refined FE model of the car body with floor panels was developed and validated. The results of FEA show that there are a large number of floor modes in the frequency range of 20–50 Hz, in which the deformations of the floor above the bogies are significantly larger than those of the car body chassis. When the excitation frequencies transmitted from the car body chassis were close to the eigenfrequencies of the floor, the local resonance of the floor will occur, which is the root cause of human feet numbness.
3. The dynamic stiffness coefficient of the elastic supports was optimised, and the optimisation scheme was verified by on-site tests. Results show that when the dynamic stiffness coefficient is 1350 N/mm, the vibration transmissibility from the car body chassis to the floor in the frequency range of 20–50 Hz has been greatly reduced, and the problem of feet numbness could be significantly improved.

Declaration of conflicting interests
The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding
The author(s) received financial support from National Natural Science Foundation of China (Grant No. 51805373) for the research, authorship, and/or publication of this article.

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