

Article

Research on the Mechanism of Different Resilient Wheel Structures Affecting Noise

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Abstract

In response to the vibration and noise issues of metro vehicles, this article uses a resilient wheel. The finite element method (FEM) and the boundary element method (BEM) were employed to establish the vehicle-track coupled rolling contact model and the acoustic boundary element model. The reliability of the acoustic boundary element model was verified by comparing simulation results with field measurement data. The effects of different structures on wheel-rail forces and noise were investigated. Compared with a standard wheel, the resilient wheel significantly reduces vibration and noise. The influence of rubber with or without clearances on resilient wheel vibration and noise was further analyzed. Results show that in the 2000–5000 Hz frequency band, the resilient wheel reduces wheel noise by 6–10 dB(A), rail noise by approximately 5 dB(A), and total wheel-rail rolling noise by 5–8 dB(A). The physical mechanisms underlying the vibration and noise reduction performance of the resilient wheel are elucidated, and high-frequency operating noise is examined.

Keywords: resilient wheel; boundary element method; wheel-rail force; noise

1. Introduction

Metro transport is one of the most efficient, safe and environmentally friendly means of transport. In the past few years, urban rail transit has seen significant growth, and the vibration and noise impact caused by vehicles and tracks has intensified significantly [1]. To minimize vibration and noise, several strategies have been implemented on both vehicle and track, such as using a resilient wheel and a ring damper wheel [2], incorporating sandwich damping into the wheel [3], and applying damping solutions for the metro [4] and the track [5]. At present, metro wheel noise is increasingly attracting people's attention.

A resilient wheel has a rubber layer added between the wheel rim and the core to change vibration and noise radiation from the wheel-rail system, thereby reducing wheel and rail noise. Koo et al. [6] conducted tests for minimizing vibration and noise with resilient wheel on the metro and found that resilient wheels reduce interior noise by 4–5 dB(A) on straight tracks. Cigada et al. [7] investigated the connection between the resonance mode of resilient wheels and sound emission using experimental techniques, confirming that mode localization at the wheel web dominates high-frequency radiation. Claus et al. [8,9] created a vertical dynamic model of ICE rail vehicles and studied the potential for minimizing low-frequency noise inside the vehicle with a resilient wheel. Yang et al. [10] examined the longitudinal vibration of resilient wheels and identified distinct vibration decay patterns at low and high speeds, with the rubber layer exhibiting



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notable damping above 500 Hz. Zhou et al. [11] and Cheng et al. [12] reported that novel wheel designs significantly influence radiated noise levels, with up to 10 dB(A) reduction achievable in the 500–2000 Hz range. He et al. [13,14] confirmed that resilient wheels reduce both wheel vibration about 15% and radiation noise compared to standard wheels, mainly due to the rubber interlayer. Wei et al. [15] investigated carriage interior noise and attributed its reduction primarily to suppressed wheel–rail high-frequency excitation. Cao et al. [16,17] analyzed the dynamic response of resilient wheels to external excitation and quantified that the rubber layer cuts noise by 3–10 dB(A) in the 2500–5000 Hz frequency band. Bai et al. [18] and Hamiche et al. [19] proposed efficient boundary element method (BEM) solvers for acoustic problems, enabling high-frequency wheel radiation prediction up to 5000 Hz with controlled cost. Thompson et al. [20] found that reflections from the car body and ground can increase the overall sound pressure level by 2–4 dB(A) and alter directivity. Zhang et al. [21] characterized the directivity of noise radiated by railway wheels, showing a dipole-like pattern dominated by radial modes. Feng et al. [22] revealed that outer-wheel squeal on large-radius metro curves is triggered by lateral creep-induced stick–slip of the tread. Chiello et al. [23] investigated curve squeal noise and identified wheel circumferential modes and rail vertical modes as the main contributors. However, to date, there have been few proposals for rubber with or without clearances on vibration and noise and researches on high-frequency noise. Furthermore, the physical mechanisms underlying the vibration-damping and noise-reducing properties of resilient wheels remain unclear.

Distinct from previous studies on resilient wheel vibration and noise, this work innovatively investigates the influences of rubber installation clearance on wheel dynamic performance. The vehicle-track coupled rolling contact model and acoustic boundary element model are constructed and validated against field test data to guarantee simulation accuracy. Further quantitative analysis reveals the inherent physical mechanism of vibration and noise reduction for resilient wheels under varied wheel structures, with extra exploration focusing on high-frequency operating noise characteristics. This study delivers targeted methodological advancements for resilient wheel analysis, providing practical guidance for wheel structural design and valuable references for metro high-frequency wheel-rail noise research.

2. Vehicle-Track Coupled Rolling Contact Model and Methodology

2.1. Wheel Model

Metro resilient wheels must be designed with sufficient radial and axial stiffness to prevent excessive distortion and creep damage of the rubber block. In the cross-section of a resilient wheel, the rubber at the bottom primarily undergoes compressive deformation, while the arc-shaped portions on either side experience both shear and compressive deformation. Figure 1 illustrates the wheel model: (a) standard wheel, (b) resilient wheel, and (c) Resilient wheel section. Figure 2 shows two different rubber structures: (a) Segmented rubber (b) whole rubber. Both wheel types feature a straight web and have a diameter of 840 mm, but they differ in mass. The standard wheel weighs 325 kg, while the segmented rubber resilient wheel has a mass of 431 kg and the whole rubber resilient wheel has a mass of 432 kg. Table 1 displays the material parameters for wheel and rail.

Table 1. Material parameters for wheel and rail.

Item	Value
Elastic modulus of rubber (MPa)	50
Density of rubber (kg/m ³)	1200
Poisson's ratio of rubber	0.475

Table 1. Cont.

Item	Value
Elastic modulus of wheel steel (MPa)	2.1×10^5
Density of wheel steel (kg/m^3)	7850
Poisson's ratio of wheel steel	0.3
Elastic modulus of rail steel (MPa)	2.1×10^5
Density of rail steel (kg/m^3)	7850
Poisson's ratio of rail steel	0.3

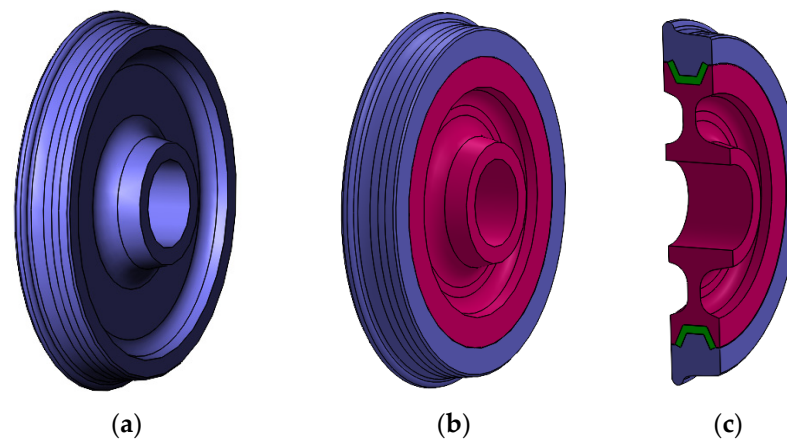


Figure 1. Wheel model: (a) standard wheel; (b) resilient wheel; (c) resilient wheel section.

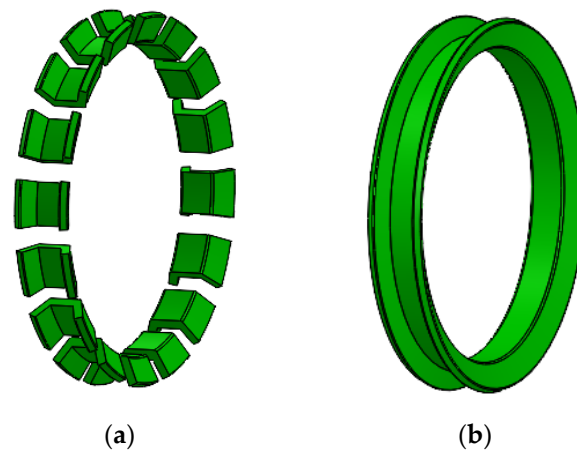


Figure 2. Two different rubber structures: (a) segmented rubber; (b) whole rubber.

2.2. Track Model

Ballastless track is usually used in domestic metro, and the physical object is shown in Figure 3a, which consists of a rail, a fastener system, a prestressed concrete slab with a poured sleeper, and a CA layer. In this paper, the rail is conceptualized as an endlessly long repeating structure, and the infinitely long period orbital system is illustrated in Figure 3b. Based on the existing research [24], the rail vibration

$$M_R \ddot{q}(x, t) + K_0 q(x, t) + K_1 \frac{\partial}{\partial x} q(x, t) - K_2 \frac{\partial^2}{\partial x^2} q(x, t) = f(x, t) \quad (1)$$

among them, M_R , K_0 , K_2 are symmetric matrices, and K_1 is the antisymmetric matrix. These matrices are defined solely by the rail, $q(x, t)$ is the vector representing rail movement, and $f(x, t)$ is the vector that includes the forces exerted on the track, which encompasses both external loads and forces from fasteners.

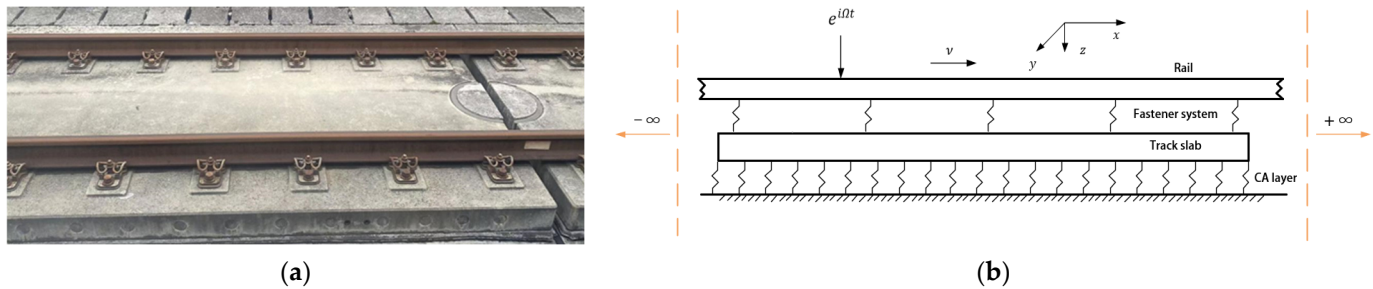


Figure 3. Ballastless track system: (a) ballastless track; (b) infinitely long period orbital system.

2.3. Force Between Wheel and Rail Contaction

Using the wheel and rail models presented above, the acceptance at the contact point between the wheel and rail can be determined. An equation to calculate the wheel-rail contact force can be derived from the forces exerted by the car body at the wheel-rail contact point, and the forces applied by the rail at that same point. Assuming that the wheel and rail are always in contact, the positional relationship between each wheel and the rail can be determined using the following equation [14].

$$w_R(a_l + ct, t) + w_W^l(t) + \delta_0^l + \frac{P^l(t) - W_l}{k_H^l} = z(a_l + ct) \quad (2)$$

in the formula, $w_R(x, t)$ is the displacement of the track in position x at time t ; $w_W^l(t)$ is the displacement of the l -th wheelset; $z(x)$ is the roughness of the wheel-rail at x location; P^l is the wheel-rail force of the l -th wheelset; W_l is half of the axle load of l -th wheelset; a_l is position of first wheelset when t is 0; k_H^l is contact stiffness at contact point of l -th wheelset.

2.4. Methodology

The proposed methodology employs a sequential computational framework integrating multibody dynamics, contact mechanics, and acoustic analysis; the methodology flow chart is shown in Figure 4. Initially, the vehicle-track coupled rolling contact model is established, incorporating inputs such as different wheel structures, wheel-rail roughness excitation, and material parameters, to generate the time-domain wheel-rail force. This dynamic result subsequently undergoes Fast Fourier Transform (FFT) to convert it into the frequency domain. The derived frequency-domain force is then utilized as an excitation source for the acoustic boundary element model, which calculates the radiated noise. Finally, the computed noise results undergo model validation against experimental data, leading to a comprehensive analysis of the noise characteristics.

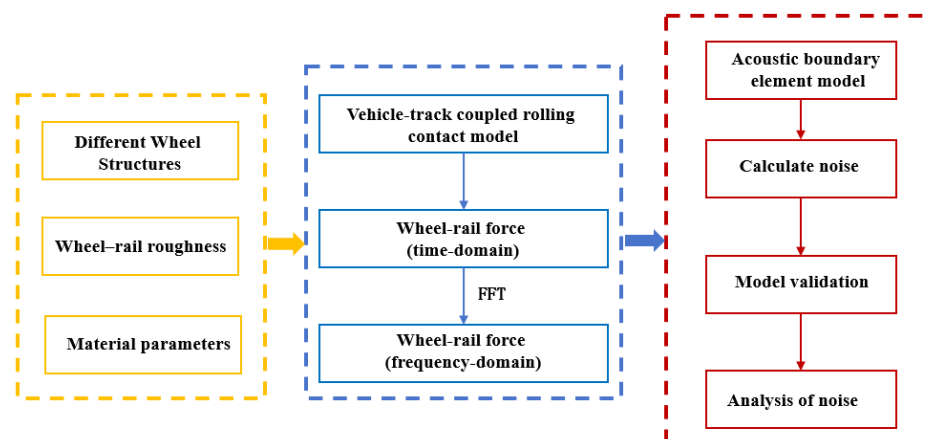


Figure 4. Methodology flow chart.

3. Wheel-Rail Force Analysis

With the purpose of investigating the effects of the segmented rubber resilient wheel and the whole rubber resilient wheel on the wheel-rail vibration and noise, this paper employs the finite element method. In Figure 5, the finite element model of the wheelset is displayed, which was discretized using linear hexahedral elements. A mesh convergence study was performed: three mesh sizes (20 mm, 15 mm, and 10 mm) were compared, and the 20 mm mesh was found to yield a variation in the wheel rim vibration velocity below 5% at frequencies up to 5000 Hz, thus satisfying convergence criteria. Boundary conditions for the structural model included fixed constraints at the axle seat (simulating the bearing support) and free surfaces elsewhere. Material damping assumptions were defined as follows: steel components adopted a structural damping ratio of 0.001–0.002, represented by Rayleigh damping coefficients fitted from the first two radial modes; the rubber layer used a viscoelastic Prony series derived from dynamic mechanical analysis master curves, with a loss factor of 0.3–0.8 over the frequency range 10–5000 Hz. The wheel-rail roughness input was entirely based on field-measured rail irregularity spectra from the tested metro line, reflecting the actual rail surface condition.

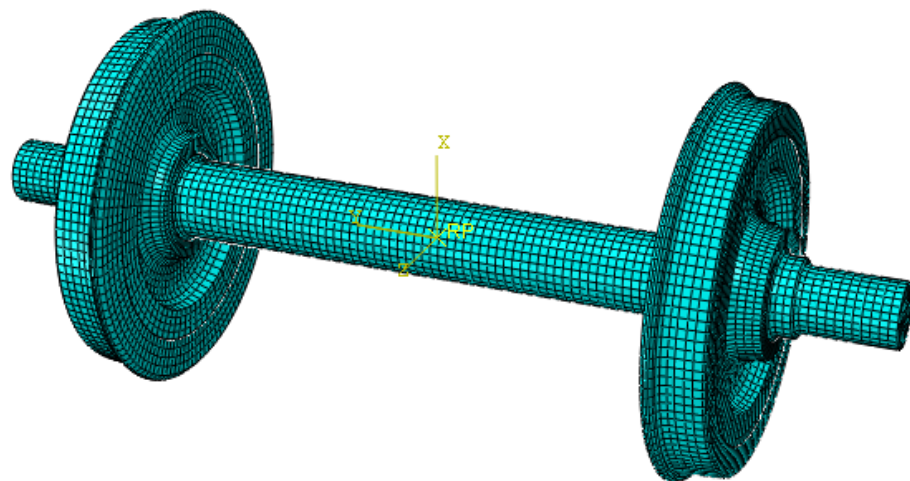


Figure 5. Finite element model of the wheelset.

The flexible wheelset was created using ABAQUS.2021 (finite element analysis software) and SIMPACK.2021 (multibody dynamics software). First, the finite element model of the wheelset was built and meshed in ABAQUS.2021. Static and modal analyses were performed to obtain the natural frequencies and mode shapes. Convergence of the finite element solution was ensured by checking that the nodal displacement changes between successive Newton–Raphson iterations fell below a predefined tolerance. After convergence, the flexible wheelset was reduced using the Craig–Bampton method, retaining a set of dominant modes covering frequencies up to 5000 Hz. The reduced flexible body information—including modal masses, stiffness matrices, and mode shapes—was exported as a standard flexible body file. This file was then imported into SIMPACK.2021. During the time-domain simulation, the wheel–rail contact forces are computed at each time step using a nonlinear Hertzian contact model, and subsequently converted the force into the frequency-domain using the FFT. The data exchange between ABAQUS.2021 and SIMPACK.2021 is unidirectional: ABAQUS.2021 provides the pre-computed flexible body properties, and SIMPACK.2021 utilizes them throughout the dynamic simulation without requiring further iterative feedback. The metro speed is 70 km/h.

The results of the wheel-rail force are presented in Figure 6, where it is evident that:

- (1) Within the range of frequency 1000 Hz to 2250 Hz, wheel-rail bearing forces of the two resilient wheels are smaller than those of the standard wheel. As a result, utilizing a resilient wheel will decrease the interaction force between the wheel and rail within this frequency range, the standard wheel-rail force is the largest followed by the whole rubber resilient wheel, segmented rubber resilient wheel force is the smallest.
- (2) Within the range of frequency 2250 Hz to 5000 Hz, wheel-rail forces of the two types of resilient wheels are very close, and are larger than the standard wheel, primarily due to the fact that the mass of each type of resilient wheel is larger than that of the standard wheel. Therefore, the resilient wheel will enhance the wheel-rail force within this frequency range.

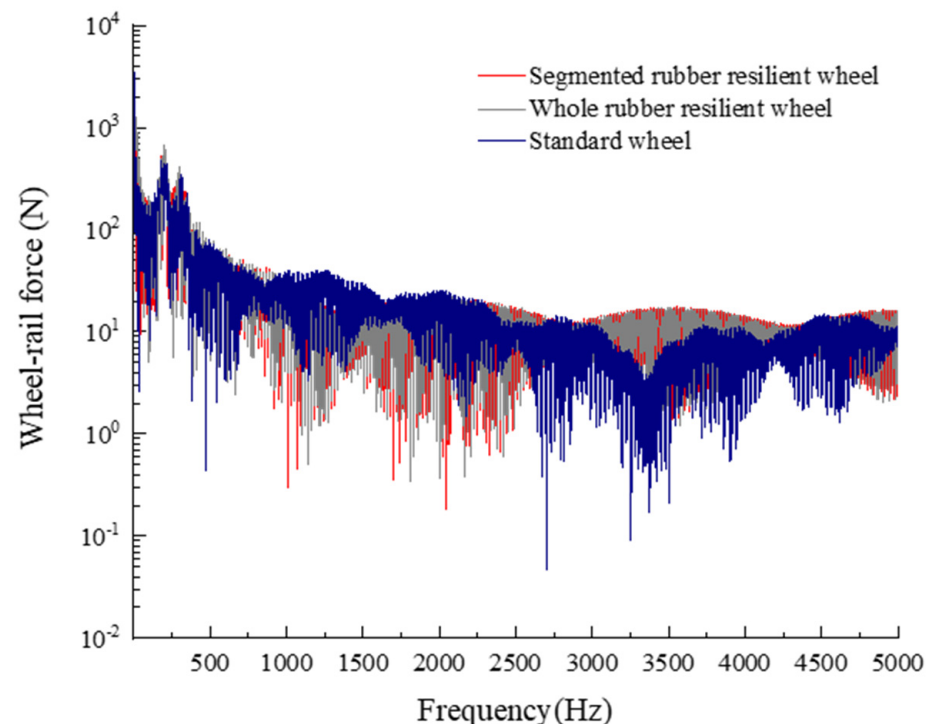


Figure 6. Wheel-rail force.

4. Acoustic Boundary Element Model Validation and Analysis of Noise

4.1. Acoustic Boundary Element Model

This paper utilizes the direct BEM with the complete space acoustic Green's function, along with a quadratic interpolation element. The acoustic boundary element model is employed to determine the sound radiation, LMS.VIRTUAL.LAB.13.6 is used to calculate the noise. The acoustic boundary element model employs tetrahedral elements, with a unit size of 10 mm, with shared nodes established at gaps to prevent sound leakage. The linear step of solution is 80 Hz from 0 Hz to 5000 Hz. A mesh convergence study was performed: three mesh sizes (15 mm, 10 mm, and 7 mm) were compared, and the 10 mm mesh was found to yield a variation in sound pressure level below 5% at frequencies up to 5000 Hz, thus satisfying convergence criteria. The wheel-rail force (from Section 3) is applied as a boundary condition to the acoustic boundary element model to calculate the noise in the frequency domain (no ground reflection). Figure 7 displays the wheel acoustic boundary element model, where (a) is the standard wheel acoustic boundary element model, and (b) is the resilient wheel acoustic boundary element model.

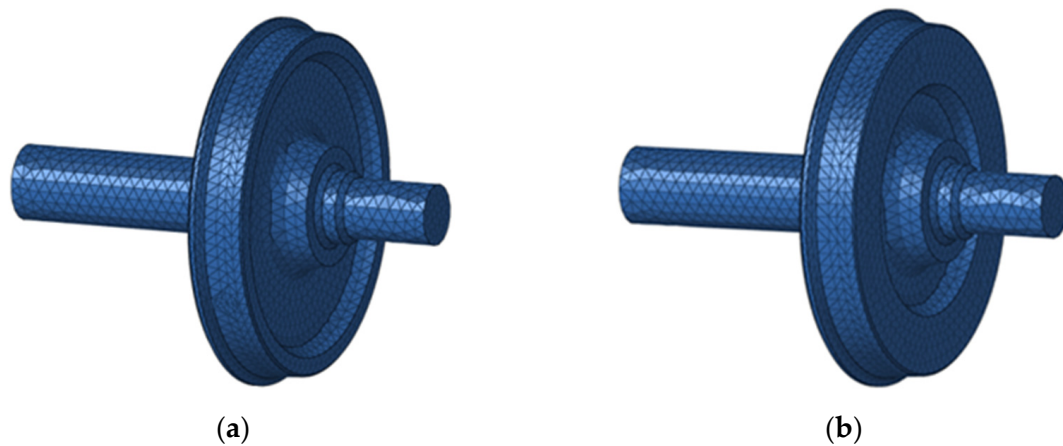
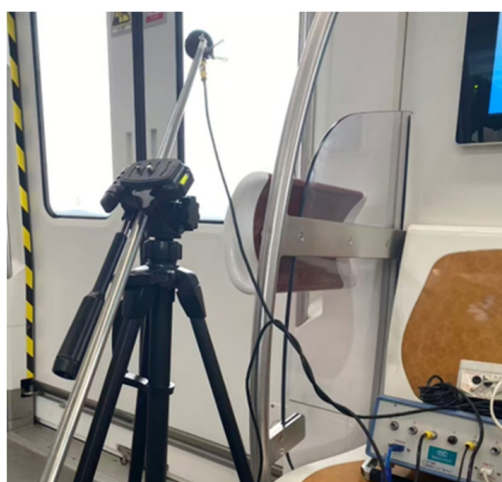


Figure 7. Wheel acoustic boundary element model: (a) standard wheel acoustic boundary element model; (b) resilient wheel acoustic boundary element model.

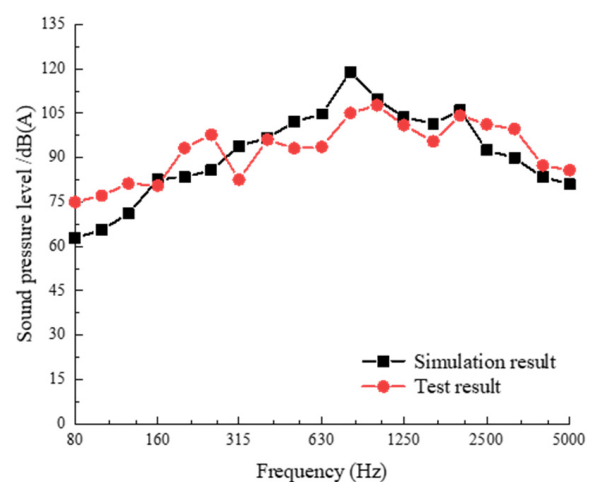
4.2. Noise Measurements and Model Validation

Field measurements were conducted on a particular underground line, and the acoustic boundary element model for calculating noise was verified by comparing simulated A-weighted sound pressure level data with A-weighted sound pressure level data obtained from field measurements.

Using the HS5660B-X noise and vibration measurement and analysis system, A-weighted sound pressure level data was collected at the metro site whilst the metro was traveling at a speed of 70 km/h. We recorded 15 s of A-weighted sound pressure level data, converted it to the frequency domain using the FFT, and the 1/3 octave band spectra of A-weighted sound pressure level were extracted. Figure 8 shows the noise measurements and model validation. Figure 8a presents noise (A-weighted sound pressure level) field measurement in the metro. Figure 8b presents the comparison of simulation and field measurement results. The comparison reveals that the simulation results are in close agreement with the measured data; the maximum error occurs at a frequency of 800 Hz, with a deviation of 14 dB(A), thereby validating the reliability of the acoustic boundary element model.



(a)



(b)

Figure 8. Noise measurements and model validation: (a) noise (A-weighted sound pressure level) field measurement in the metro; (b) comparison of simulation and field measurement results.

4.3. Analysis of Noise

To examine the ratio of wheel-rail sound radiation across the entire frequency spectrum, Figure 9a–c, displays the noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for the standard wheel, segmented rubber resilient wheel, and the whole rubber resilient wheel.

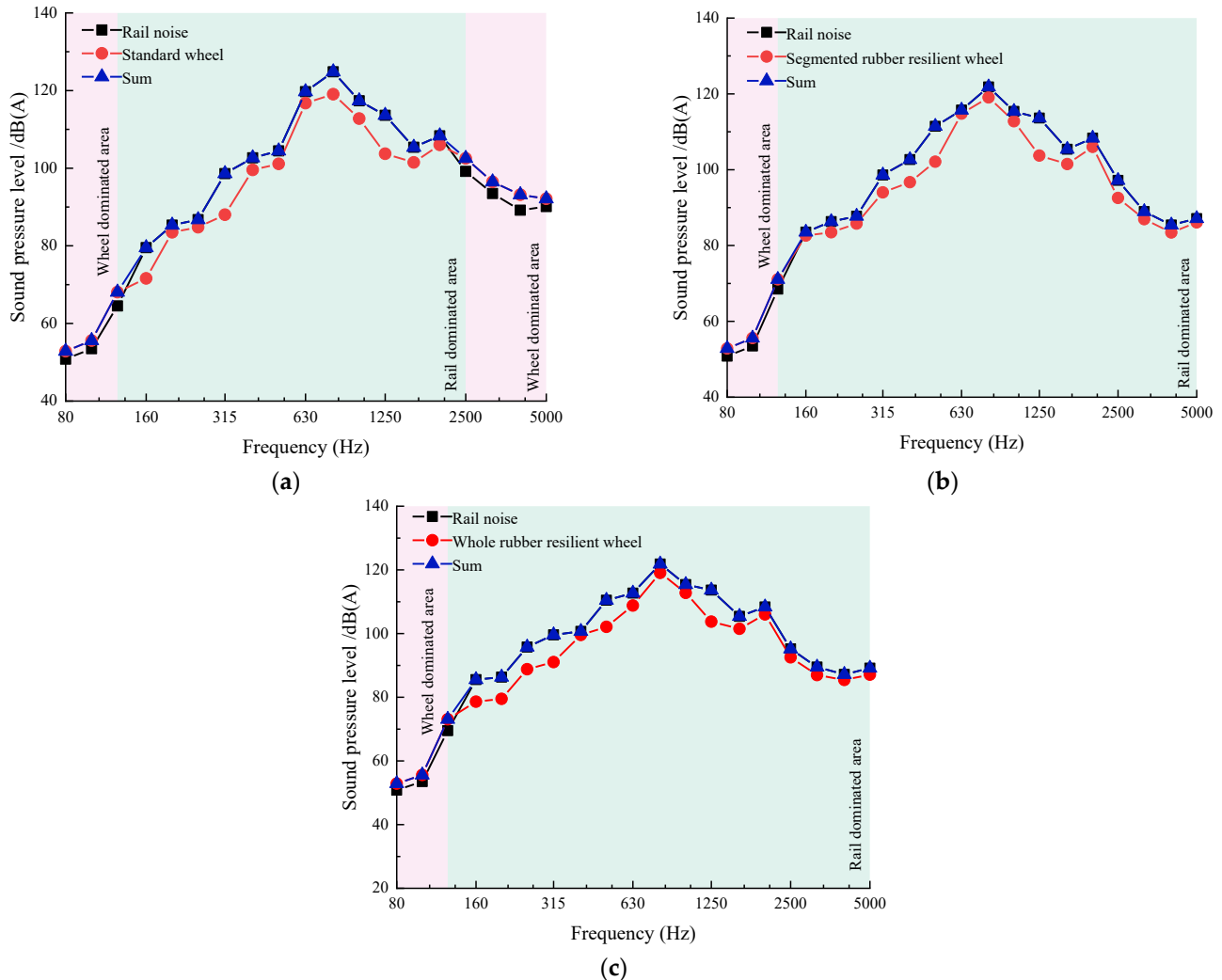


Figure 9. Noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for three wheels: (a) noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for standard wheel; (b) noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for segmented rubber resilient wheel; (c) noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for whole rubber resilient wheel.

As shown in Figure 9, the sound radiation from the wheel-rail interaction is primarily influenced by the sound radiation from the wheel at frequencies below 125 Hz, and wheel-rail sound radiation is dominated by rail sound radiation between 125 Hz and 2500 Hz; over 2500 Hz, the standard wheel-rail system is dominated by the wheel, while the wheel-rail systems of the two types of resilient wheels are still dominated by rail. As a result, the resilient wheel is capable of minimizing the high-frequency noise generated by the wheel-rail system.

To evaluate the airborne noise reduction in each elastic wheel compared to a standard wheel, we used sound pressure level. The A-weighted sound power level L_{WA} can be represented in terms of sound power W as:

$$L_{WA} = 10 \cdot \log_{10} \frac{W}{W_0} \quad (3)$$

L_{WA} is the A-weighted sound power level measured in decibels (dB), W is the sound power measured in watts (W), W_0 is the 0 dB reference level equal to 10^{-12} W. By definition, A-weighted sound pressure level L_{PA} can be articulated in relation to A-weighted sound power level L_{WA} as:

$$L_{PA} = L_{WA} - 10 \cdot \log(4\pi r^2) \quad (4)$$

Taking this relationship into account, the variation in A-weighted sound pressure levels between two distinct wheels, 1 and 2, can be written as:

$$L_{PA1} - L_{PA2} = L_{WA1} - L_{WA2} - 10 \lg(4\pi r_1^2) + 10 \lg(4\pi r_2^2) = 10 \lg \frac{W_1}{W_2} \left(\frac{r_2}{r_1} \right)^2 \quad (5)$$

where W_1 is wheel 1, W_2 is wheel 2, r represents the distance between the sound source and the point where the measurement is taken. This equation is consistent with the experimental findings presented by Efthimeros et al. [25].

To assess the impact of both the segmented rubber resilient wheel and the whole rubber resilient wheel more thoroughly on rail sounds, wheel sounds, and total wheel-rail rolling sounds, Figure 10a–c presents the noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for wheel, rail and wheel-rail. As can be seen:

- (1) At 80–2000 Hz, the variations in A-weighted sound pressure level of the three types of wheels are not significant. However, at 2000–5000 Hz, the graph is in the shape of “Y”, the A-weighted sound pressure level of the standard wheel is greater than those of the two types of resilient wheels. The resilient wheel can reduce the high-frequency noise by 6–10 dB(A) of the wheel, and the impact of noise emission of the segmented rubber resilient wheel is greater than that of the whole rubber resilient wheel.
- (2) The variations in the A-weighted sound pressure level of the rail of the three types of wheels are not significant in most frequency bands. However, at 2000–5000 Hz, the graph is in the shape of “Y”, the A-weighted sound pressure level of the standard wheel on the rail is larger than those of the two types of resilient wheels. The resilient wheel can help minimize high-frequency noise around 5 dB(A) from the rail, and the segmented rubber resilient wheel is more effective at minimizing rail noise compared to the whole rubber resilient wheel.
- (3) Within the frequency range of 80 Hz to 2000 Hz, the differences in A-weighted sound pressure level for the three types of wheels are not significant. However, at 2000–5000 Hz, the graph is in the shape of “Y”, the total A-weighted sound pressure level of the standard wheel-rail system is higher than those of the two types of resilient wheels. The resilient wheel is capable of decreasing the total high-frequency noise generated by the interaction between the wheel and rail by 5–8 dB(A), and the noise reduction effect of the segmented rubber resilient wheel has a greater height compared to the whole rubber resilient wheel.

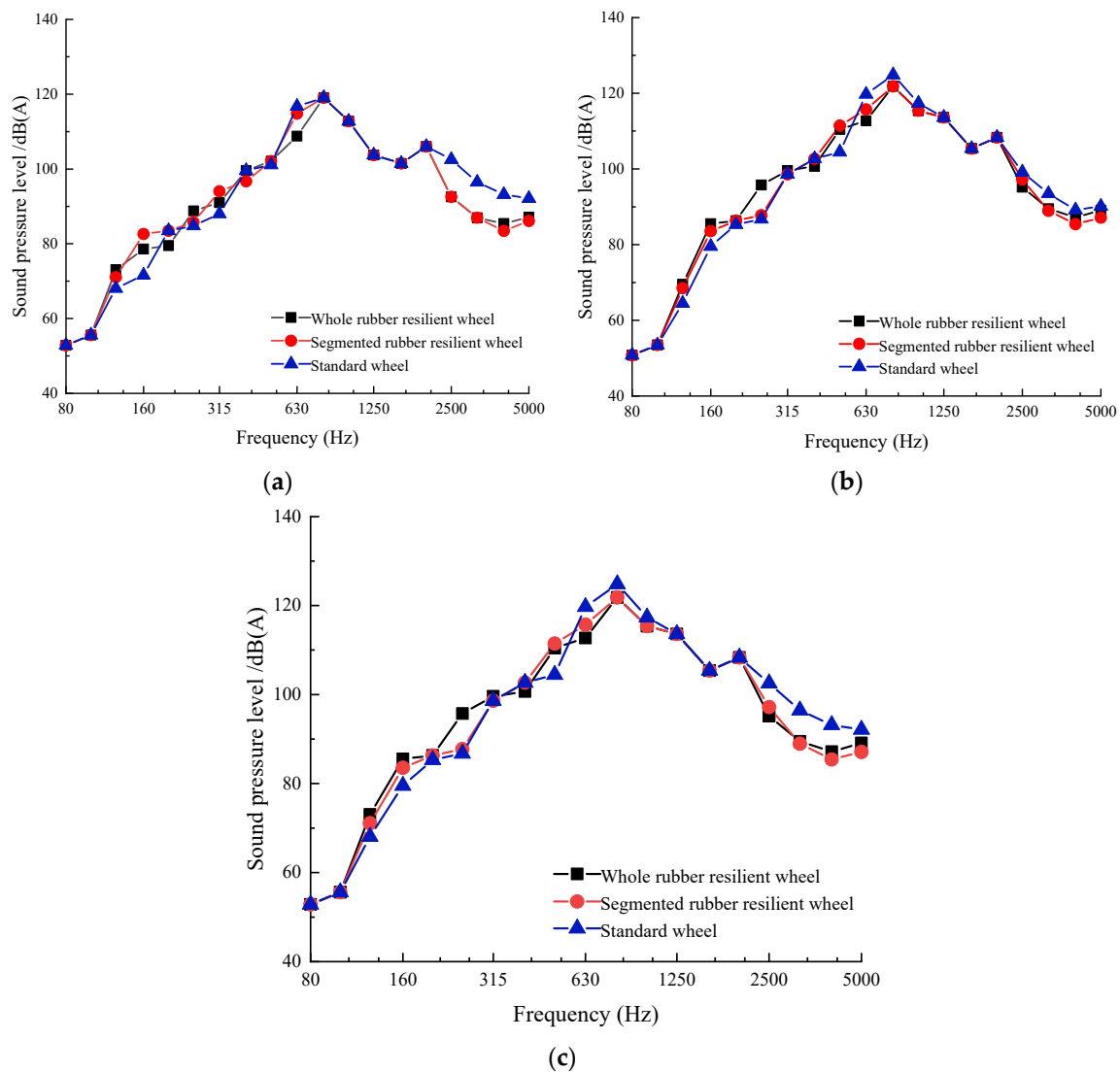


Figure 10. Noise (A-weighted sound pressure level) of spectra in 1/3 octave bands: (a) noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for wheel; (b) noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for rail; (c) total noise (A-weighted sound pressure level) of spectra in 1/3 octave bands for wheel-rail.

From the calculation of wheel vibration velocity response under wheel-rail force excitation in Equation (6), it is evident that the vibration velocity response is lower and the vibration level resulting from the wheel-rail force is also lower. Therefore, the A-weighted sound pressure level of the resilient wheel is relatively low as well. From Equation (7), it is evident that the higher the frequency, the greater the vibration velocity response and the higher the A-weighted sound pressure level, which coincides with the experimental results reported in the literature [26].

$$v_w = \frac{i\omega R Y_w}{Y_r + Y_w + Y_c} \quad (6)$$

The relationship between wheel vibration velocity response and frequency can be written as:

$$v_w = \frac{2\pi f i R Y_w}{Y_r + Y_w + Y_c} \quad (7)$$

where v_w is the wheel vibration velocity response, rail mobility is Y_r , wheel mobility is Y_w , contact mobility is Y_c , amplitude of the combined wheel-rail roughness is R , f is the vibration frequency.

The noise reduction results obtained in this study are compared with existing resilient wheel configurations and conventional metro noise mitigation measures reported in the literature. In the 2000–5000 Hz high-frequency band, the proposed resilient wheel with rubber clearance achieves a wheel noise reduction of 6–10 dB(A), a rail noise reduction of approximately 5 dB(A), and a total wheel–rail rolling noise reduction of 5–8 dB(A). These mitigation effects are superior to those of conventional resilient wheel structures in previous studies, which mainly focus on low-to-medium frequency noise control and exhibit limited high-frequency noise attenuation. Compared with other common metro noise reduction measures [11,12], the presented resilient wheel configuration delivers more prominent and targeted high-frequency noise suppression, compensating for the deficiencies of existing research. Furthermore, incorporating the perspective of assessing such environmental noise [27,28] will provide a more comprehensive evaluation framework.

It is found that the rubber layer of the resilient wheel realizes the vibration energy conversion through the viscoelastic behavior of the molecular chain. When the resilient wheel is impacted, the rubber molecular chain deforms and generates internal friction, which converts mechanical energy into thermal energy and dissipates it. This effect is effective in most frequency bands, and thus realizes the purpose of noise reduction. In addition, the low stiffness characteristics of the rubber enable it to change the intrinsic frequency of the system; reducing the rubber stiffness can reduce the intrinsic frequency, thus making the vibration transmission rate decrease significantly. The middle rubber layer forms a secondary attenuation, effectively isolating the vibration triggered by the impact of the wheel and rail, significantly reducing the noise. Furthermore, the difference in acoustic impedance between the rubber and the air causes sound waves to be reflected at the interface. The unique structure of the resilient wheel can regulate the equivalent acoustic impedance of the rubber between the metal hub and the air, reducing the transmission of sound waves, especially in the high-frequency band, where the noise reduction effect is remarkable.

The segmented rubber resilient wheel effectively divides the vibration propagation path by discretizing the rubber layer, suppressing the continuous resonance phenomenon of high-frequency vibration in the whole rubber resilient wheel. The convoluted structure of the rubber molecular chain creates more localized deformation nodes in the segmented design, so that vibration energy is dispersed in multiple independent zones, reducing the intensity of a single resonance peak. And the multiple interfaces formed by the segmented gap increase the frictional heat conversion efficiency inside the rubber. As the vibration wave travels to the segment boundaries, the viscoelastic nature of the rubber (hysteresis angle effect) allows more mechanical energy to be converted to heat through molecular chain slip. In contrast, the noise reduction effect of the whole rubber resilient wheel has frequency band limitations and energy conversion efficiency bottlenecks due to the vibration coupling effect of the continuous structure and the homogeneous material properties.

5. Conclusions

To address metro vibration and noise, a vehicle-track coupled rolling contact model and an acoustic boundary element model were established using the FEM and BEM to investigate the effects of different wheel structures on wheel-rail forces and noise. The reliability of the acoustic boundary element model was verified by comparing simulation results with field measurement data. Compared with a standard wheel, the resilient wheel significantly reduces vibration and noise. The physical mechanisms underlying the vibration and noise reduction in the resilient wheel are elucidated, and high-frequency operating noise is examined. The results demonstrate that the resilient wheel effectively alleviates carriage noise, thereby improving acoustic comfort in metro environments.

- (1) Within the frequency range of 1000 Hz to 2250 Hz, wheel-rail forces for the two types of resilient wheels are smaller than those for the standard wheel. Therefore, the implementation of a resilient wheel minimizes the forces between the wheel and rail within this frequency range. The standard wheel-rail force is the highest, followed by the whole rubber resilient wheel, and the segmented rubber resilient wheel-rail force is the lowest. Within the range of frequency 2250 Hz to 5000 Hz, the wheel-rail forces of the two resilient wheels are very close to each other and are larger than those for the standard wheel, mainly due to the fact that the mass of the two resilient wheels is larger than that of the standard wheel. As a result, the resilient wheel enhances the wheel-rail force within this frequency range.
- (2) Below 125 Hz, the sound produced by the wheel-rail interaction is primarily influenced by the sound produced by the wheel; the sound radiation from the rail is most prominent in the frequency range of 125 Hz to 2500 Hz; and above 2500 Hz, the standard wheel-rail system is primarily influenced by the sound produced by the wheel, but the wheel-rail systems of the two types of resilient wheels are still primarily characterized by rail sound, and are all relatively low. As a result, the resilient wheel is capable of minimizing the high-frequency noise generated by the wheel-rail system.
- (3) It is found that in the high-frequency band (2000–5000 Hz), the two types of resilient wheels can reduce wheel noise by 6–10 dB(A), rail noise by approximately 5 dB(A), and total wheel-rail rolling noise by 5–8 dB(A), and the noise-reducing effect of the segmented rubber resilient wheel is better than that of the whole rubber resilient wheel.

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Abbreviations

FEM	Finite element method
BEM	Boundary element method
FFT	Fast Fourier Transform

References

1. Han, J.; Xiao, X.; Wu, Y.; Wen, Z.; Zhao, G. Effect of rail corrugation on metro interior noise and its control. *Appl. Acoust.* **2018**, *130*, 63–70. [[CrossRef](#)]
2. Deng, X.; Zhang, X.; Geng, Z.; Chu, L. Analysis of key parameters of metro vehicle amping ring wheel during service period. *J. Railw. Sci. Eng.* **2018**, *15*, 2140–2147. (In Chinese) [[CrossRef](#)]
3. Merideno, I.; Nieto, J.; Gil-Negrete, N.; Ortiz, J.G.; Landaberea, A.; Iartza, J. Theoretical prediction of the damping of a railway wheel with sandwich-type dampers. *J. Sound. Vib.* **2014**, *333*, 4897–4911. [[CrossRef](#)]
4. Asmussen, B.; Stiebel, D.; Kitson, P.; Farrington, D.; Benton, D. *Reducing the Noise Emission by Increasing the Damping of the Rail: Results of a Field Test*; Springer: Berlin/Heidelberg, Germany, 2008.

5. Liang, S.Y.; Xiao, A.X.; Geng, C.Z. Analysis of vibration isolation performance of floating slab track structure with damping steel springs. *Noise Vib. Control* **2013**, *1*, 136–139. (In Chinese)
6. Koo, D.H.; Kim, J.C.; Yoo, W.H.; Park, T.W. An experimental study of the effect of low-noise wheels in reducing noise and vibration. *Transp. Res. Part D Transp. Environ.* **2002**, *7*, 429–439. [[CrossRef](#)]
7. Cigada, A.; Manzoni, S.; Vanali, M. Vibro-acoustic characterization of railway wheels. *Appl. Acoust.* **2008**, *69*, 530–545. [[CrossRef](#)]
8. Claus, H.; Schiehlen, W. Stability analysis of railways with radial elastic wheelsets. *Veh. Syst. Dyn.* **2002**, *37*, 453–464. [[CrossRef](#)]
9. Claus, H.; Schiehlen, W. Dynamic stability and random vibrations of rigid and elastic wheelsets. *Nonlinear Dyn.* **2004**, *36*, 299–311. [[CrossRef](#)]
10. Yang, Y.; Ding, J.-J.; Li, F.; Li, J.-c. Longitudinal vibration of a resilient wheel under the adhesion limit. *Proc. Inst. Mech. Eng. Part F J. Rail Rapid Transit* **2019**, *233*, 370–381.
11. Zhou, X.; Liu, Z.; Zhong, S.; An, T.; Sheng, X.; Thompson, D. In-situ assessment of the sound and vibration reduction performance of a new type of resilient wheel installed on a metro train. *Veh. Syst. Dyn.* **2024**, *62*, 2936–2955. [[CrossRef](#)]
12. Cheng, G.; He, Y.P.; Han, J.; Sheng, X.Z.; Thompson, D. An investigation into the effects of modelling assumptions on sound power radiated from a high-speed train wheelset. *J. Sound. Vib.* **2021**, *495*, 115910. [[CrossRef](#)]
13. He, Y.P.; Wang, X.H.; Han, J.; Xiao, X.B.; Sheng, X.Z. Study on the influence of resilient wheels on vibration and acoustic radiation characteristics of suburban railway concrete box girder bridges. *Appl. Acoust.* **2022**, *187*, 108529. [[CrossRef](#)]
14. He, Y.; Zhou, Q.; Xu, F.; Sheng, X.; He, Y.; Han, J. An investigation into the effect of rubber design parameters of a resilient wheel on wheel-rail noise. *Appl. Acoust.* **2023**, *205*, 109259. [[CrossRef](#)]
15. Wei, Z.; Sun, X.; Yang, F.; Ke, Z.; Lu, T.; Zhang, P.; Shen, C. Carriage interior noise-based inspection for rail corrugation on high-speed railway track. *Appl. Acoust.* **2022**, *196*, 108881. [[CrossRef](#)]
16. Cao, Y.; Tian, J. Study on the response of elastic wheel to excitation. *Int. J. Comput. Methods* **2026**, 1–9. [[CrossRef](#)]
17. Cao, Y.; Tian, J. Research on the mechanism of metro wheel high-order polygons affecting sound pressure level. *Mech. Sci.* **2026**, *17*, 61–70. [[CrossRef](#)]
18. Bai, X.; Zhang, X.; Li, Y. Efficient calculating of the vibro-acoustic behaviour of railway wheel based on boundary element method with adaptive order. *J. East China Jiaotong Univ.* **2024**, *41*, 48–55.
19. Hamiche, K.; Li, Y.; Atak, O.; Beriot, H.; Vansant, K. A novel high-order and adaptive boundary element method solver (BEMAO) for small and large acoustics problems. In Proceedings of the DAGA, Hamburg, Germany, 6–9 March 2023.
20. Thompson, D.J.; Zhao, D.; Ntotsios, E.; Squicciarini, G.; Cierco, E.; Jansen, E. The influence of reflections from the train body and the ground on the sound radiation from a railway rail. *Railw. Sci.* **2024**, *3*, 1–17. [[CrossRef](#)]
21. Zhang, X.; He, Y.; Thompson, D.J.; Hu, Z. The directivity of noise radiated by a railway wheel in situ. *Proc. Inst. Mech. Eng. Part F J. Rail Rapid Transit* **2025**, *239*, 669–678. [[CrossRef](#)]
22. Feng, X.; Chen, G.; Dong, B. Study on the mechanism of outer wheel squeal on subway large-radius curves. *Tribol. Trans.* **2025**, *68*, 498–510. [[CrossRef](#)]
23. Chiello, O.; Tufano, R.; Rissmann, M. Estimation of vibration limit cycles from wheel/rail mobilities for the prediction of curve squeal noise. In *Noise and Vibration Mitigation for Rail Transportation Systems*; Springer Nature: Singapore, 2024; pp. 473–482.
24. Sheng, X.; Zhong, T.S.; Li, Y. Vibration and sound radiation of slab high-speed railway tracks subject to a moving harmonic load. *J. Sound. Vib.* **2017**, *395*, 160–186. [[CrossRef](#)]
25. Efthimeros, E.A. Vibration/noise optimization of a FEM railway wheel model. *Eng. Comput.* **2002**, *19*, 922–932. [[CrossRef](#)]
26. Zhou, X.; Han, J.; Zhao, Y.; Wu, B.; Cheng, G.; Xu, S.; Jin, X. Characteristics of vibration and sound radiation of metro resilient wheel. *Chin. J. Mech. Eng.* **2019**, *32*, 67. [[CrossRef](#)]
27. Danilevičius, A.; Karpenko, M.; Křivánek, V. Research on the noise pollution from different vehicle categories in the urban area. *Transport* **2023**, *38*, 1–11. [[CrossRef](#)]
28. Danilevičius, A.; Danilevičienė, I.; Karpenko, M.; Stosiak, M.; Skačkauskas, P. Determination of the instantaneous noise level using a discrete road traffic flow method. *Promet-Traffic&Transportation* **2025**, *37*, 71–85. [[CrossRef](#)]

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