

Article

Control of Wave Reflection-Induced Rail Corrugation and Associated Rolling Noise on High-Elastic Fastener Tracks

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Featured Application

This work provides a practical solution for mitigating short-pitch rail corrugation and associated rolling noise on metro tracks equipped with high-elastic fasteners. The demonstrated dual benefit of rail vibration absorbers—simultaneously suppressing corrugation growth and reducing rail-borne noise by 8 dB(A)—makes the proposed approach particularly suitable for vibration-sensitive urban rail lines where both track deterioration and acoustic comfort are of concern.

Abstract

Short-pitch rail corrugation is prevalent on metro tracks with high-elastic fasteners and is closely linked to rolling noise. Field measurements show that the in situ pass-by noise reaches its maximum in the 600 Hz–1 kHz range. In the present work, the growth processes of short-pitch rail corrugation are investigated in the time domain by considering wheel–track dynamics, rolling mechanics and wear characteristics on tracks with high-elastic fasteners. The results confirm that wave reflections between wheels cause rapid development of short-pitch rail corrugation on tracks equipped with high-elastic fasteners. It is also found that a promising measure to mitigate this type of rail corrugation is to adjust the wheel intervals. Furthermore, the rail-related component of rolling noise is analyzed, and a rail vibration absorber is introduced into the wheel-track system. Both the simulated results and in situ measured pass-by noise demonstrate that rolling noise and rail corrugation within the 600 Hz–1 kHz range exhibit a strong intrinsic coupling. Thus, rail vibration absorbers offer dual benefits: suppressing corrugation growth and reducing associated rail-borne noise.

Keywords: short-pitch rail corrugation; high-elastic fastener; wear; absorber; rolling noise



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1. Introduction

Rail corrugation is related to vehicle–track dynamics, contact mechanics, wear and operation conditions. Grassie [1] classified corrugation into six types and attributed short-pitch corrugation to the pinned–pinned resonance of the track. However, severe corrugation still occurs on tracks with high-elastic fastener, where the pinned–pinned resonance is weak. Although Grassie [1] attributed short-pitch corrugation to the pinned–pinned resonance of the track, corrugation remains severe on tracks with high-elastic fasteners where this mode is weak. Field investigations have identified a distinct type of short-pitch corrugation on tracks equipped with resilient (including high-elastic) fasteners—such as Vanguard fasteners—across multiple metro lines in China. Li et al. [2] demonstrated through field

measurements and numerical simulations that this corrugation arises from a wavelength-fixing mechanism governed by rail vertical bouncing on the resilient fasteners, and showed that the corrugation wavelength is primarily related to the vertical stiffness of the fasteners. Their findings further indicate that increasing fastener stiffness or damping can suppress this type of corrugation [2]. Wave reflections between wheels and track are associated with the pinned–pinned resonance [3]. In particular, Wang and Wu [4] investigated the effects of wheel–wheel wave reflections on short-pitch corrugation specifically on tracks with high-elastic fasteners, and demonstrated that such reflections can become a dominant mechanism, distinct from the pinned–pinned resonance. This suggests that high-elastic tracks may have a different corrugation mechanism, which motivates the present study.

Most studies believe that corrugation is caused by vehicle–track dynamic resonances. Grassie et al. [5] studied wheel–rail interaction forces with a linear model. Xu et al. [6] analyzed the coupling between wheel polygonization and rail corrugation. Hempelmann [7] related corrugation to dynamic component resonances. However, these studies used single-wheel models and neglected wheel–wheel wave reflections, which are likely significant for tracks with high-elastic fastener. Single-wheel models fundamentally exclude the interference patterns created by waves propagating between multiple wheels, and therefore cannot reproduce the multi-peak dynamic force spectra observed in multi-wheel systems or the resulting distributed wear energy across multiple frequencies.

Some researchers have adopted multi-wheel models. Igeland [8] found that corrugation wavelength is related to bogie wheelbase. Wu and Thompson [9] studied multi-wheel–track interaction and the effect of fastener stiffness in frequency domain. Wu et al. [10] calculated corrugation growth rate and found it related to multi-order resonances. Wu et al. [11] and Fang et al. [12] suggested that wave reflection causes corrugation on resilient tracks but lacked time-domain verification. Frequency-domain models are inherently restricted to linearized contact mechanics and cannot simulate the history-dependent, non-linear time-varying growth of corrugation, which depends on the continuously evolving roughness profile. Therefore, the formation mechanism of corrugation on high-elastic fastener tracks is still unclear, motivating the time-domain multi-wheel approach adopted in the present study. In this paper, the formation process of corrugation is simulated with and without wave reflections at the track with high-elastic fasteners in order to explore the reason for corrugation growth.

Rail vibration dominates rolling noise. Rail vibration absorbers can effectively reduce the rail noise component by 8–9 dB [13] and may suppress corrugation related to pinned–pinned resonance [14]. High-elastic fasteners provide little damping, so absorbers can increase damping to weaken wave reflection and control corrugation. Many rail dampers have been applied in metro systems [15–18]. More recently, Xiao et al. [19] developed an acoustic black hole (ABH) absorber specifically for rail vibration suppression and demonstrated its effectiveness through full-scale experiments, achieving a significant increase in rail decay rate. This type of broadband, lightweight absorber could offer new possibilities for simultaneously controlling corrugation and noise.

The 600 Hz–1 kHz frequency range investigated in this study is of particular concern for human health and comfort. Although whole-body vibration standards such as ISO 2631-1 [20] focus on lower frequencies, the structure-borne noise in this band significantly degrades passenger acoustic comfort in enclosed metro carriage environments [21]. For train operators and track maintenance personnel, prolonged exposure to high-frequency rolling noise in tunnel sections may contribute to auditory fatigue and occupational hearing impairment [22]. Moreover, the human ear exhibits peak sensitivity in this frequency range, as reflected in the A-weighting curve [23], making the 600 Hz–1 kHz band a priority target for metro noise control. These health and comfort considerations further motivate

the present investigation into the coupling between short-pitch corrugation and rolling noise [24], and the development of effective mitigation measures.

The control of rolling noise in metro systems also contributes to the broader environmental sustainability goals of urban rail transport. As noted by Lazaro J, et al. [25], noise reduction is a critical component of sustainable urban rail development, improving both environmental performance and community acceptance in densely populated areas. The present work, by addressing corrugation and noise simultaneously, aligns with these sustainability objectives.

This study is a subsequent work following Fang et al. [12], aiming to validate their frequency-domain predictions for short-pitch corrugation by using time-domain simulations of corrugation growth with and without wave reflections, and to investigate the associated pass-by noise in situ. The results also suggest, qualitatively, that wheel interval configuration could influence wave reflection patterns and hence corrugation growth, although a comprehensive parametric investigation considering vehicle dynamics constraints and practical design limitations is left for future work. It is worth noting that the track is deliberately modeled as continuously supported. Under high-elastic fasteners, the discrete support effect is inherently weak owing to the low fastener stiffness; more importantly, a continuous support model eliminates the pinned–pinned resonance by construction. This allows wave reflections between wheels—rather than the pinned–pinned mode—to be isolated and examined as the sole driving mechanism for corrugation growth. The simulation method follows the approach in [26], but the objective is different. Furthermore, the evolutionary process of short-pitch corrugation and the rail component of rolling noise are explored when a rail vibration absorber is installed. Wu [15] previously demonstrated analytically that a rail vibration absorber can suppress short-pitch corrugation growth related to the pinned–pinned resonance, and also reduce rail noise radiation. However, its effectiveness on high-elastic fastener tracks where wave reflections dominate—and the resulting dual benefits on both corrugation and noise—have not yet been comprehensively verified through time-domain simulation and field measurements, which is the focus of the present work. The results simulated and the passing noise measured in situ show that rolling noise and rail corrugation in the 600 Hz–1 kHz range are intrinsically coupled, and thus double benefits are obtained by installing a rail vibration absorber, namely mitigating corrugation growth and decreasing the corresponding rail radiation.

The main contributions of this paper are threefold: (i) time-domain verification that wheel–wheel wave reflections, rather than pinned–pinned resonance, are the dominant driver of short-pitch corrugation on high-elastic fastener tracks; (ii) demonstration through both simulation and field measurements that tuned rail vibration absorbers achieve dual benefits—suppressing corrugation growth and reducing rolling noise by 8 dB(A); and (iii) identification of the 600–1000 Hz frequency band as the spectral region where corrugation and noise are intrinsically coupled, providing a clear target for mitigation measures.

Beyond the direct vibro-acoustic analysis, the continuous monitoring of track irregularities and fastener conditions is a critical aspect of modern railway asset management. Recent advances include 3D computer-vision systems for wayside wagon monitoring [27] and deep convolutional neural network methods for automated detection of rail surface defects and fastener damage [28]. The present time-domain corrugation model provides physics-based wear predictions that can complement such data-driven diagnostic tools, offering a potential avenue for integrating numerical wear forecasts with real-time infrastructure inspection data. This connection between mechanistic modeling and intelligent monitoring represents a promising direction for future railway health management systems.

The remainder of this paper is organized as follows. Section 2 develops the time-domain wheels–track model with continuous support. Section 3 describes the rolling

contact mechanics and frictional work-based wear model. Section 4 presents the corrugation development simulation results with and without wave reflections. Section 5 investigates the attenuation of corrugation using rail vibration absorbers. Section 6 analyzes the sound radiation and presents field measurement results from Shanghai Metro Line 11. Section 7 concludes the paper with key findings, practical implications, and future research directions.

2. Wheels–Track Model

A time-domain model for a continuously supported slab track coupled with multiple wheels is presented to simulate the growth in rail corrugation in this section.

2.1. Modeling of Wheels–Track System

Figure 1 schematically shows the vertical vibration model of the continuously supported slab track coupled with multiple wheels and rail vibration absorbers. Due to the symmetric track structure, only 1/2 track is considered in this model. The slab track bed is treated as rigid because the vertical stiffness of high-elastic rail fasteners is far lower than that of the bed. The rail is modeled as an Euler–Bernoulli beam. The continuously supported assumption is introduced not only for computational efficiency, but also for a physical reason: it removes the pinned–pinned resonance from the system. Consequently, any corrugation predicted by the model can be attributed exclusively to wave reflections between wheels, which is the central hypothesis of this study.

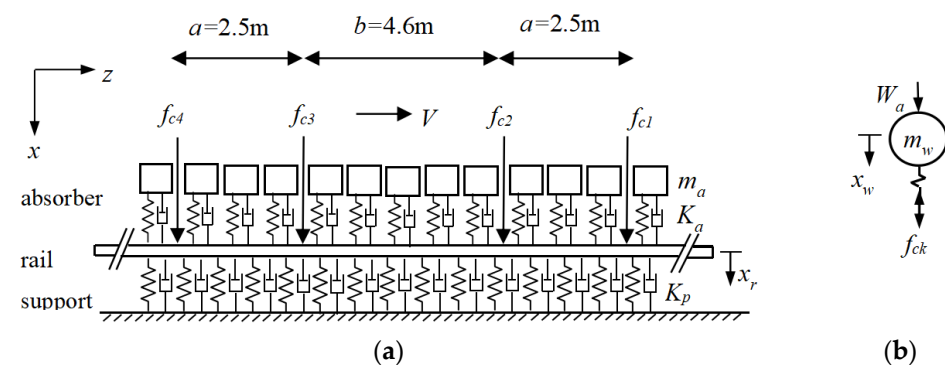


Figure 1. Continuously supported slab track–wheels model: (a) track model; (b) wheel–rail interaction model.

It should be noted that the continuous support idealization represents a deliberate simplification to isolate wave reflections as the corrugation driving mechanism. In reality, the discrete sleeper support introduces spatial periodicity and pinned–pinned resonance, which influences both vibration propagation and noise radiation. Discrete support models, such as those based on the discrete element method or periodic structure theory, can capture the stop-band behavior and spatial decay characteristics that affect broadband noise limits [29]. For noise assessment on discretely supported tracks, instantaneous noise level determination can be enhanced by discrete time-stepping approaches that resolve the pin-to-pin spatial variations in rail vibration. The implications of discrete support for corrugation growth and noise radiation in the context of wave reflection mechanisms represent an important direction for future investigation.

The parameters used for CHN 60 rail are given by

$$E = 2.1 \times 10^{11} \text{ N/m}^2, \rho = 7850 \text{ kg/m}^3, A = 7.745 \times 10^{-3} \text{ m}^2, \\ I = 32.17 \times 10^{-6} \text{ m}^4, \eta_r = 0.01$$

where E is the Young's modulus, ρ is the material density of rail, η_r is the loss factor for the rail, A is the cross-sectional area, and I is the area moment of inertia. The high-elastic fasteners are modeled by the continuous spring and damping. In practice, the discrete supports beneath the rail can be equivalently represented as continuous spring-damper elements to improve computational efficiency. For the high-elastic fasteners, the vertical stiffness of a discrete support is denoted by k_p . The equivalent continuous support stiffness K_p is obtained by averaging k_p over the sleeper spacing of 0.6 m, expressed as a complex stiffness to account for damping: $K_p = k_p(1 + i\eta_p)/0.6$, where the loss factor is $\eta_p = 0.2$. These values are consistent with experimentally reported loss factors for resilient rail fasteners [23], where softer pads typically exhibit higher material damping than stiffer alternatives. The constant stiffness k_p is adopted in accordance with experimental evidence that high-elastic rail pads show limited stiffness variation (below 20%) in the 600–1000 Hz range [23], and the frequency-dependent damping is captured by the loss factor η_p . A viscoelastic constitutive model, while capable of reproducing the exact frequency dependence, would not change the identified resonance patterns and is therefore left for future refinement.

Four wheels on two adjacent bogies are considered in this model. Since the vibration wave reflections between these four wheels and track is the strongest, the effects of other wheels far away from these four wheels may be ignored due to decay of wave propagation [4]. The wheel mass is 750 kg. The vehicle is simplified to be static load W_a to the wheels, $W_a = 80$ kN. These four wheels are spaced at intervals of $a = 2.5$ m and $b = 4.6$ m and the train velocity $V = 20$ m/s. f_{c1}, f_{c2}, f_{c3} and f_{c4} are the contact forces at wheel 1, 2, 3 and 4, respectively. The wheels are coupled with the rail via a Hertzian spring.

To approximate an infinitely long track, the rail is made sufficiently long such that the interaction of reflected waves from the two ends can be neglected. Furthermore, if the beam is long enough, the wheel–track interaction in the middle region is unaffected by the end conditions, where it is simply supported or fixed. It can be observed from Figure 2 that the time histories of vertical contact forces for wheels 1 and 2 tend to stabilize after 8.4 m under simply supported boundary conditions. Therefore, the rail length is selected as 84 m in this study, and its middle section is used to investigate the growth of corrugation on tracks equipped with high-elastic rail fasteners. The rail vibration absorber is incorporated into the system and modeled as a continuous mass-spring-damper system attached to the rail, as shown in Figure 1.

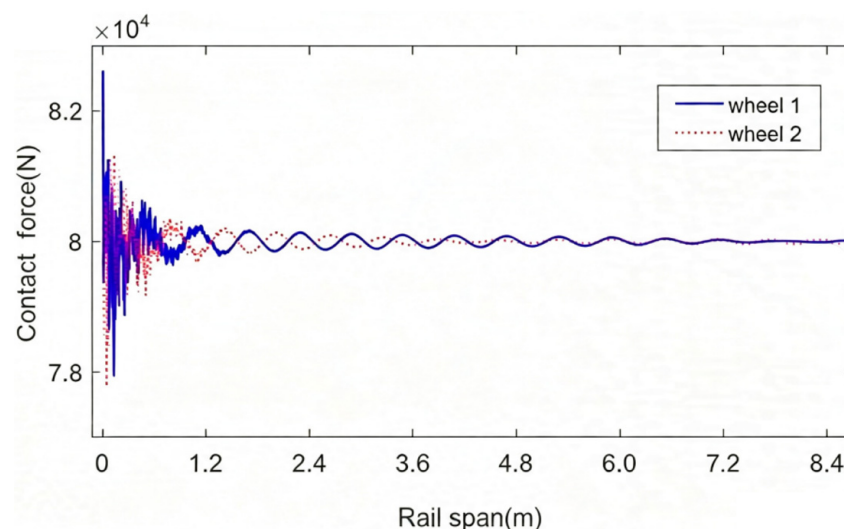


Figure 2. Vertical wheel/rail interaction force, $k_p = 10$ MN/m, $V = 20$ m/s.

In the continuously supported slab track system, the vertical vibration of the rail excited by four moving wheels is governed by

$$\rho A \frac{\partial^2 x_r}{\partial t^2} + EI \frac{\partial^4 x_r}{\partial t^4} + K_p x_r = \sum_{k=1}^{N_k} f_{ck} \delta(z - z_{wk}), \quad (1)$$

where x_r represents the rail displacement, and $N_k = 4$ is the number of wheels. K_p denotes the equivalent continuously supported stiffness. f_{ck} represents the contact force at the k th wheel position z_{wk} . δ is the Dirac delta function. $k = 1, 2, \dots, N_k$.

The equations of vertical motion of the mass wheel are given by

$$m_w \ddot{x}_{wk} = W_a - f_{ck}, \quad (2)$$

where m_w represents the mass of the wheel. x_{wk} represents the displacement of the k th wheel. The wheel–rail contact force f_{ck} is calculated by

$$f_{ck} = \begin{cases} C_H [x_{wk} - x_r(z_{wk}) - r]^{\frac{3}{2}}, & x_{wk} - x_r(z_{wk}) - r > 0 \\ 0, & x_{wk} - x_r(z_{wk}) - r \leq 0 \end{cases}, \quad (3)$$

here C_H is the Hertzian contact constant, taken as $C_H = 93.7 \text{ GN/m}^{3/2}$. r is the roughness on railhead and the sign convention adopted here for r is positive for dip and negative for an asperity.

Equation (1) can be solved for rail vibration using the modal superposition method. For simplicity the beam is modeled to be simply supported at its ends. The displacement response is the summation of each modal, given by

$$x_r(z, t) = \sum_{m=1}^M X_m(z) q_m(t), \quad (4)$$

in which $X_m(z)$ is the mode shape function and is obtained by

$$X_m(z) = \sqrt{\frac{2}{\rho A L}} \sin\left(\frac{m\pi z}{L}\right), \quad (5)$$

The orthogonality works to X_m

$$\int_0^L \rho A X_m(z) X_n(z) dz = \begin{cases} 1, & m = n \\ 0, & m \neq n \end{cases}, \quad (6)$$

and then Equation (1) turns into

$$\ddot{q}_n + \frac{EI}{\rho A} \left(\frac{n\pi}{L}\right)^4 q_n + \frac{1}{\rho A} K_p q_n = \sum_{k=1}^{N_k} f_{ck} X_n(z_{wk}), n = 1, 2, \dots, M, \quad (7)$$

Fourth order Runge–Kutta algorithm is used to solve Equation (7) by combining with Equations (2) and (3). The number of modes for the summation here is chosen to be 500 for a credible solution of the rail response. This model also provides a unified framework for incorporating additional dynamic components—such as rail vibration absorbers—by simply augmenting the support stiffness term, as will be detailed in Section 5.

2.2. Dynamic Wheel–Rail Forces

The dynamic behavior of the multiple wheels/track system are investigated using the model developed in Figure 1. The dynamic wheel/rail forces at wheels 1 and 2 are presented in Figure 3 under a unit force excitation, respectively (for symmetry, the forces

at wheels 1 and 2 are given here). As shown in Figure 3a, many resonance peaks appear between 100 Hz and 1 kHz for both wheels 1 and 2. Since wheel 1 only experiences wave reflections from the front, fewer peaks appear compared to wheel 2. And the highest resonance peak of the dynamic force for wheels 1 and 2 simultaneously appear at around 740 Hz. However, these dramatic fluctuations are absent for the single wheel–rail force, which means that these peaks result from the wave reflections between moving wheels. These are similar to the results in reference [12]. As for the track with normal fasteners, namely $k_p = 60$ MN/m, the force peaks are lower than the case with the high-elastic fastener $k_p = 10$ MN/m, and thus contributes to the difference in dynamic forces for wheels 1 and 2, see in Figure 3b. The main reason for this is that rail pad stiffness is increased and thus the damping is proportional to it, which can improve the suppression of wave reflections among the wheels. The phase of dynamic force is relative to that of roughness; -180° means that the minimum force occurs in the dip, while 0° shows the opposite. In the former case rail corrugation will be developed.

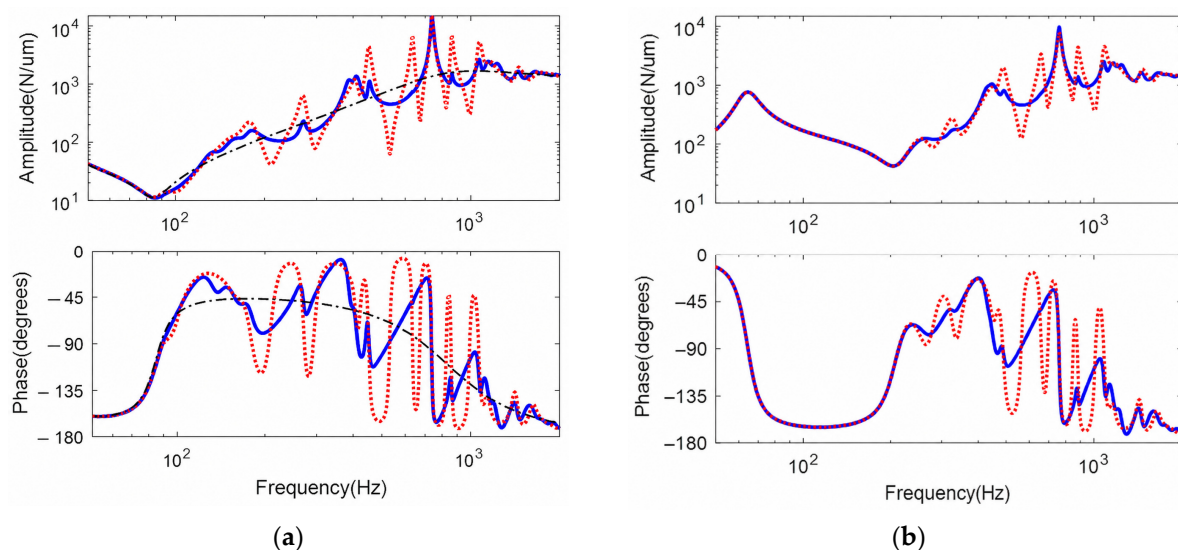


Figure 3. Active forces: (a) for $k_p = 10$ MN/m, (b) for $k_p = 60$ MN/m. (—) excitation at wheel 1. (---) excitation at wheel 2. (- · - ·) single wheel–rail interaction.

To further quantify the effect of fastener stiffness on wave reflections, the peak dynamic force at 740 Hz for $k_p = 10$ MN/m is approximately 2.3 times that for $k_p = 60$ MN/m, indicating that the high-elastic fastener track amplifies the resonance response substantially. The unilateral versus bilateral wave reflection configurations also lead to distinct force spectra: wheel 1, experiencing reflections only from the front, exhibits a single dominant peak, whereas wheel 2 shows multiple secondary peaks (e.g., at 454 Hz and 625 Hz) due to bidirectional wave interference. The phase of the dynamic force relative to roughness is also critical: at frequencies where the phase approaches -180° , the minimum contact force coincides with the roughness dip, promoting wear amplification and corrugation growth.

3. Rolling Contact and Wear

Rail corrugation growth is closely correlated with contact mechanics and wear in addition to wheel/track dynamics. Here the two-dimensional method is utilized to depict the tangential contact of the wheel and rail, in which a cylindrical wheel is assumed to roll on a flat rail and the width of the contact patch is assumed to be a constant, and the two- and three-dimensional methods can produce similar results of longitudinal wear [30,31], although the three-dimensional contact model is much more coincident with the real situations. The corrugation wavelengths considered here are longer than the contact patch

length, so the half-space assumption remains valid. The wheel and rail are assumed to deform as elastic half-spaces and thereby Hertz theory works well.

It should be noted that this quasi-static contact treatment is valid provided the corrugation wavelength remains substantially larger than the longitudinal contact patch dimension (~6–8 mm). The dominant corrugation wavelength of 25.6 mm identified in this study satisfies this condition. For significantly shorter wavelengths approaching the contact patch size, non-steady and three-dimensional contact effects would need to be incorporated.

The vertical contact force determines the normal pressure distribution, which can be computed via the Hertz formulation [32], and further determines the slip velocity and tangential stress. According to Carter's theory adopted in previous studies [33,34], the tangential stress distribution is illustrated in Figure 4, where the contact patch is divided into stick and slip zones, and its center moves with the wheel at each time step. In this study, a frictional work-based wear model is employed [10–14]. Wear is assumed to be generated solely by local relative slip and proportional to the frictional work. At each time step, the contact patch dimensions, normal pressure distribution, and tangential stress are updated using the instantaneous vertical force and the local rail curvature derived from the evolving roughness profile. Spin creepage and transient geometric effects on the contact-patch scale are not explicitly included, which is acceptable given that the corrugation wavelengths studied here are substantially larger than the contact patch length. For shorter-wavelength irregularities, three-dimensional non-steady contact models would be necessary. In Figure 4, the local wear in time step Δt can be taken as

$$W_t(x) = Kq(x, t)v(x, t)\Delta t, \quad (8)$$

here K is a material related constant, $K = 2.5 \times 10^{-9}$ kg/Nm [34]. q and v are the tangential stress and slip velocity in contact patch, respectively, and the calculation for them and the longitudinal creepage are referred in [10,14].

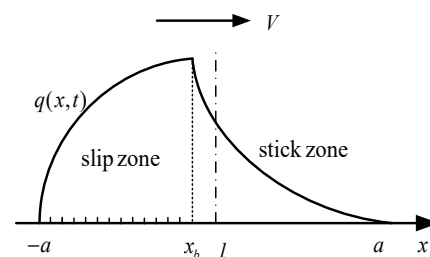


Figure 4. Assumed tangential traction distribution.

The local increment of wear $\Delta r_t(x)$ is obtained by

$$\Delta r_t(x) = \frac{W_t(x)}{\rho}, \quad (9)$$

where ρ is the material density for the rail.

4. Corrugation Development

The simulated development of rail corrugation utilizing the two-dimensional contact model after a large number of passages of wheels is given in this section.

The total wear along the railhead is an accumulation of the local wear. The local increment of wear in slip zone is obtained by Equation (9). After one wheel passage, the global increment of wear $\Delta r(l)$ can be obtained by the accumulation of the local increment of wear $\Delta r_t(x)$ according to the position of the wheel on the rail. To save the calculation time, $\Delta r(l)$ is magnified N times, $N = 1000$. One wheel passage therefore equals to the effects

on corrugation growth of 1000 wheel passages [14,35,36]. Consistent with Refs. [14,31,32], the wear depth per magnified wheel passage is of the order of 10^{-9} m, which is negligible compared with the corrugation amplitude. The acceleration approach therefore provides a linear scaling of the wear process during the growth phase simulated here. A new corrugated profile $r_n(l)$ after one wheel passage can be taken as

$$r_n(l) = r_{n-1}(l) + N\Delta r_n(l), \quad n = 1, 2, \dots, \quad (10)$$

where the old profile $r_{n-1}(l)$ is used as the roughness input for the n th iteration. It should be noted that the wear coefficient K in Equation (8) functions as a linear scaling factor in the frictional work-based wear law. Consequently, K determines the absolute growth rate of corrugation but does not affect the spectral distribution of wear across wavelengths. The dominant wavelength and the relative growth rates across frequencies are therefore insensitive to the specific value of K .

To neglect the effects from the ends, the middle section of the corrugated rail is chosen to observe the corrugation growth. The simulation adopts the parameters of a Chinese A-type metro vehicle. The wheel radius R_w is 420 mm.

To gain a clear understanding of the corrugation growth at the high-elastic track, the initial profile of rail is assumed to be absolutely smooth. Figure 5 shows the simulated corrugation growth induced by wheels 1 and 2 on a track with $k_p = 10$ MN/m. The evolution of the profile and the spectrum of the final profile after 200,000 wheel passages are presented. As observed in Figure 5a,b, wheel 1 generates rail corrugation with a fixed wavelength, which becomes fully developed at 780 Hz, corresponding to a corrugation wavelength of 25.6 mm at a velocity of 20 m/s. Likewise, rail corrugation caused by wheel 2 also develops more rapidly at 780 Hz, with the same 25.6 mm wavelength at the identical velocity, as presented in Figure 5c,d. Nevertheless, the corrugation gradually turns irregular after approximately 100,000 wheel passages, with numerous longer or shorter wavelength components emerging. Furthermore, the growth rate of corrugation at 780 Hz is slower than that induced by wheel 1, whereas components around 300 Hz grow more rapidly. This is attributed to bidirectional wave reflections acting on wheel 2, leading to more intense fluctuations.

A quantitative comparison of the growth rates reveals that the corrugation amplitude at 780 Hz induced by wheel 1 reaches approximately 1.8 times that induced by wheel 2 after 200,000 passages. This difference is attributed to the bidirectional wave reflections acting on wheel 2, which distribute the wear energy across a broader frequency range—primarily the 780 Hz component but also the 300 Hz and 454 Hz bands. Consequently, although the total wear energy is comparable, the spectral concentration differs, leading to a more irregular final profile for wheel 2. After approximately 100,000 passages, the growth rate of the dominant 780 Hz component for wheel 2 begins to decrease, while secondary wavelength components grow more rapidly, indicating a transition from single-wavelength dominance to multi-wavelength irregularity driven by the complex wave interference pattern.

Figure 6 shows the evolution of corrugation for the single wheel model and the corresponding spectrum of final profile after 200,000 wheel passages. Corrugation growth at 780 Hz can be observed to be very slow, compared with Figure 5. The main reason is that there is no wave reflections from other wheels. Consequently, it further indicates that wave reflections between wheels is the dominant reason for the unusual growth in rail corrugation at the track with high-elastic fasteners. In addition, since the single wheel model amounts to the multiple wheel model in which the intervals between wheels are infinite, according to the cause of corrugation growth mentioned above, adjusting the intervals among wheels may reduce wave reflections and thereby rail corrugation, vibration and noise.

in the dynamic force and consequently distributing uneven wear across multiple frequencies (wavelengths). Significantly, the dominant wavelength identified is consistent with the peak amplitude of the dynamic force for both wheels 1 and 2, and matches field measurement results. Because the continuous support idealization intrinsically excludes the pinned–pinned mode, the clear corrugation development observed in the simulation can be unequivocally linked to the wheel–wheel wave reflection mechanism. Therefore, in the absence of the pinned–pinned mode in the continuously supported track, wave reflections between moving wheels are the primary cause of corrugation growth on tracks with highly elastic fasteners.

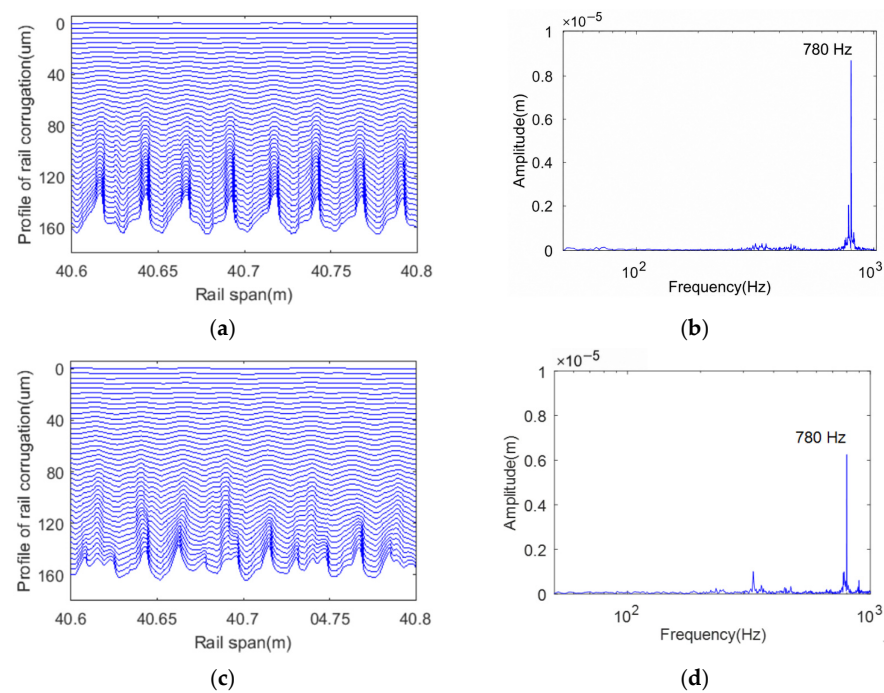


Figure 5. Growth process of corrugation and spectrum of the final profile for (a,c) wheel 1 and (b,d) wheel 2 after 200,000 wheel passages, respectively, $k_p = 10$ MN/m, $V = 20$ m/s. Profiles are shown after every 5000 wheel passages.

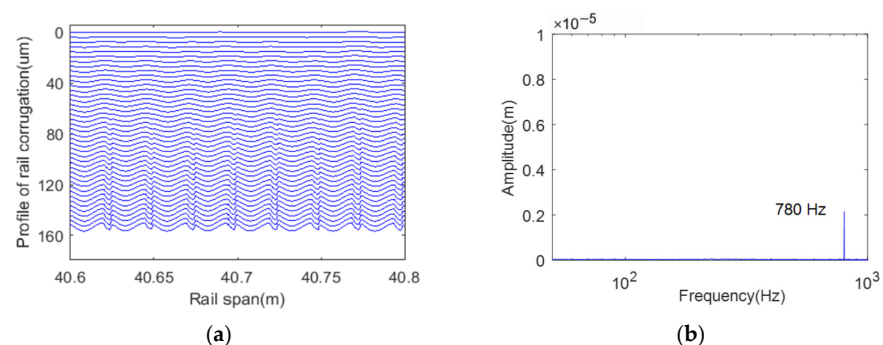


Figure 6. Rail corrugation caused by the single wheel/rail interaction after 200,000 wheel passages: (a) growth process of corrugation, (b) the final profile spectrum, $k_p = 10$ MN/m, $V = 20$ m/s. Profiles are shown after every 5000 wheel passages.

It should be emphasized that the continuous support model eliminates the pinned–pinned mode by construction, ensuring that the predicted corrugation arises exclusively from wave reflections. The strong agreement with field measurements confirms that this mechanism is physically relevant for high-elastic fastener tracks, not an artifact of the modeling simplifications.

Rail corrugation caused by wheels 1 and 2 at the track with normal fasteners, $k_p = 60$ MN/m, are shown respectively in Figure 7 in terms of the evolution of rail corrugation and the spectrum of final profile after 200,000 wheel passages. Rail corrugation caused by wheels 1 and 2 can be observed to be similar to each other, and all of them are scarcely developed in comparison to Figure 5, which accords with the actual situation. This is because more damping contributed by the normal fastener effectively suppresses the wave reflections between moving wheels.

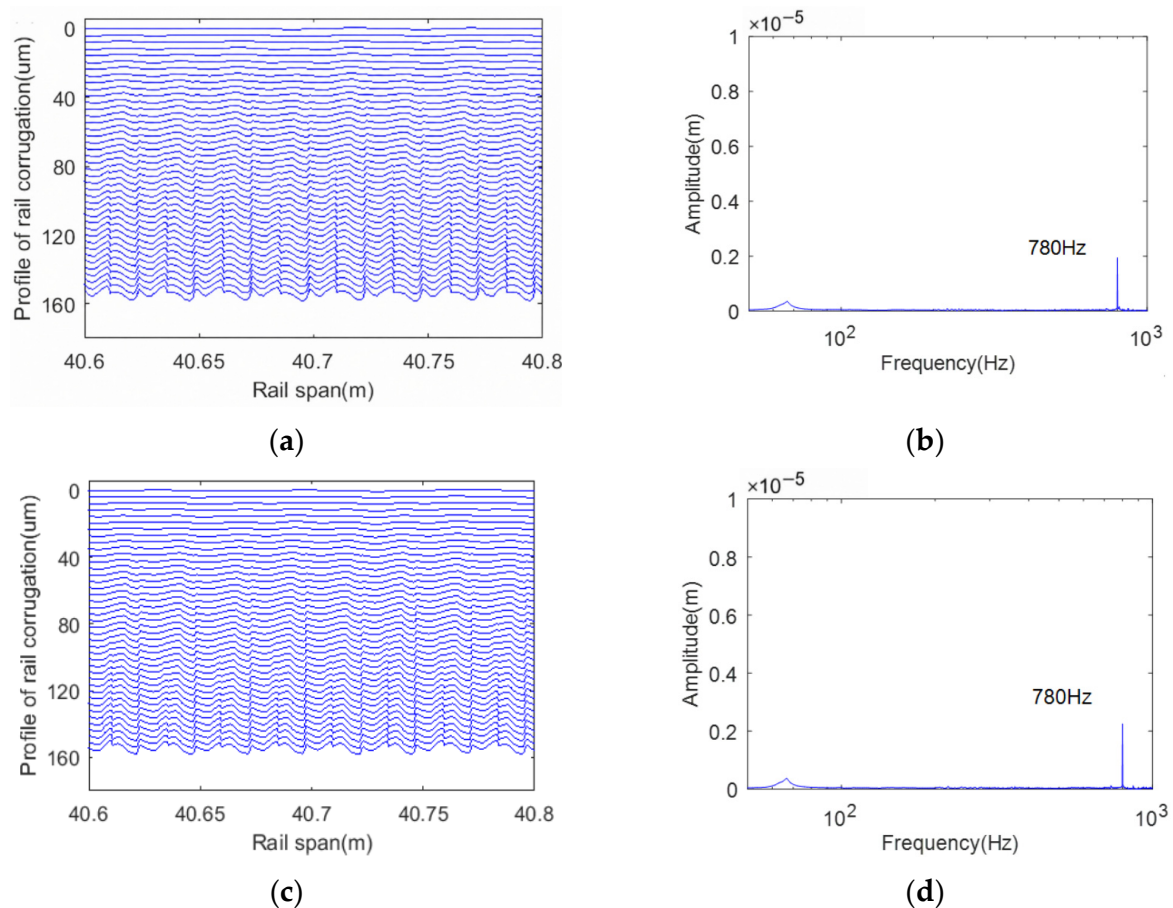


Figure 7. Growth process of corrugation and spectrum of the final profile for (a,c) wheel 1 and (b,d) wheel 2 with rail absorber after 200,000 wheel passages, respectively, $k_p = 60$ MN/m, $V = 20$ m/s. Profiles are shown after every 5000 wheel passages.

5. Corrugation Attenuation

Since a rail vibration absorber can contribute additional damping to the rail and improve the vibration decay rate, short-pitch rail corrugation associated with wave reflections may be weakened by installing them.

The rail vibration absorber is introduced into the wheels–track model developed in Section 2, as shown in Figure 1. The absorber is modeled as a continuous mass-spring-damper system attached to the rail, which represents a natural extension of the original continuously supported beam formulation. Accordingly, the vertical motion equation of the rail, originally given by Equation (1), is extended to include the absorber effect, yielding the modified equation of vertical motion for the rail by adding the absorber stiffness K_a to the support term:

$$\rho A \frac{\partial^2 x_r}{\partial t^2} + EI \frac{\partial^4 x_r}{\partial t^4} + (K_p + K_a) x_r = \sum_{k=1}^{N_k} f_{ck} \delta(z - z_{wk}) \quad (11)$$

where K_a denotes the equivalent dynamic stiffness contributed by the continuous absorber layer. The absorber itself is modeled as a continuous mass–spring–damper system, whose vertical motion $x_a(z,t)$ is governed by

$$m_a \frac{\partial^2 x_a}{\partial t^2} + c_a \left(\frac{\partial x_a}{\partial t} - \frac{\partial x_r}{\partial t} \right) + k_a (x_a + x_r) = 0 \quad (12)$$

where m_a , c_a , and k_a are the absorber mass, damping coefficient, and stiffness per unit length, respectively. The damping is characterized by the loss factor η_a , consistent with the hysteretic damping convention used in this paper. Accordingly, under the continuous support assumption and harmonic excitation, the equivalent dynamic stiffness contributed by the absorber is $K_a = k_a - m_a \omega^2 + i \eta_a k_a$, which is fully frequency-dependent and captures the tuned resonance behavior. The modal superposition method used in this study naturally incorporates this complex stiffness in the numerical solution, eliminating the need for an explicit viscous damping coefficient in Equation (11).

The absorber is rigidly clamped to the rail foot via steel clips, as shown in Figure 1, and the continuous mass–spring–damper layer idealization represents the collective dynamic effect of the periodically spaced discrete absorber units. The coupling between the absorber and the rail is achieved through the shared displacement and velocity terms in Equations (11) and (12), which ensures that the absorber responds to rail vibration and, in turn, applies a reaction force proportional to the relative motion.

The rail absorber here is tuned to around 700 Hz, which corresponds to the dominant resonance frequency and rapid corrugation growth. The parameters are as follows:

$$m_a = 12 \text{ kg}, \quad K_a = 232 \text{ MN/m}^2, \quad \eta_a = 0.25$$

The rail response with rail absorbers and corrugation simulation are calculated numerically by using the principle of modal superposition and Runge–Kutta algorithm. The absorber is designed with $\eta_a = 0.25$, providing broadband attenuation across the 600–1000 Hz range rather than a narrow-band notch. This moderate damping level ensures robust performance despite potential deviations from the nominal tuning frequency.

The dynamic properties of the track system after installing the absorber are presented in Figure 8 in terms of the decay rates of rail vertical vibration and the dynamic force for a unit harmonic excitation acting at the rail, respectively. Figure 8 indicates that the decay rate remains unchanged at low frequencies but increases substantially between 600 Hz and 1 kHz after installing the absorbers, thereby reducing rail vibration and the rail component of rolling noise. Because short-pitch rail corrugation is related with wave reflections in this range, the tuned vibration absorber may attenuate this kind of corrugation.

The corrugation growth after installing the vibration absorber is simulated using the developed model in Figure 1. The results are shown in Figure 9 in terms of the growth process and the final profile spectrum after 200,000 wheel passages. It can be observed that, compared with Figure 5, the rail corrugation of 25.6 mm wavelength is attenuated greatly, and rail corrugation formed by wheel 2 with the designed rail dampers and the frequency spectrum of the final corrugation profile are similar to that caused by wheel 1; although, wheel 2 suffers from wave reflections from two sides. Moreover, the growth in rail corrugation becomes quite gentle due to the installation of the designed rail absorber and is moved to a low frequency of 646 Hz. Accordingly, short-pitch corrugation caused by wave reflections among moving wheels can be attenuated by installing the designed rail absorber.

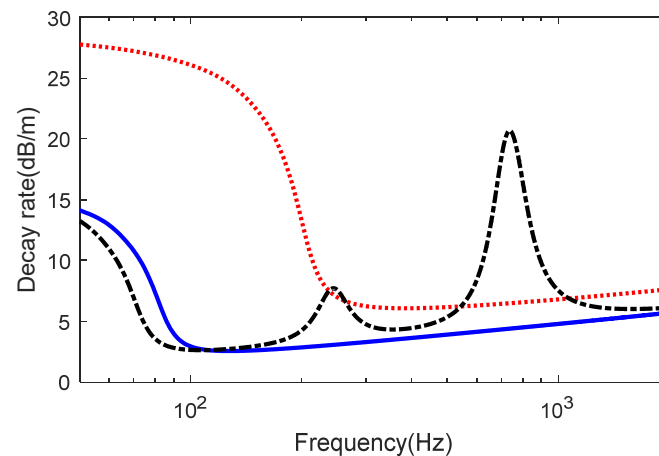


Figure 8. Decay rate of rail vertical vibration. (—) $k_p = 10$ MN/m; (.....) $k_p = 60$ MN/m; (— · — · —) $k_p = 10$ MN/m with rail vibration absorbers.

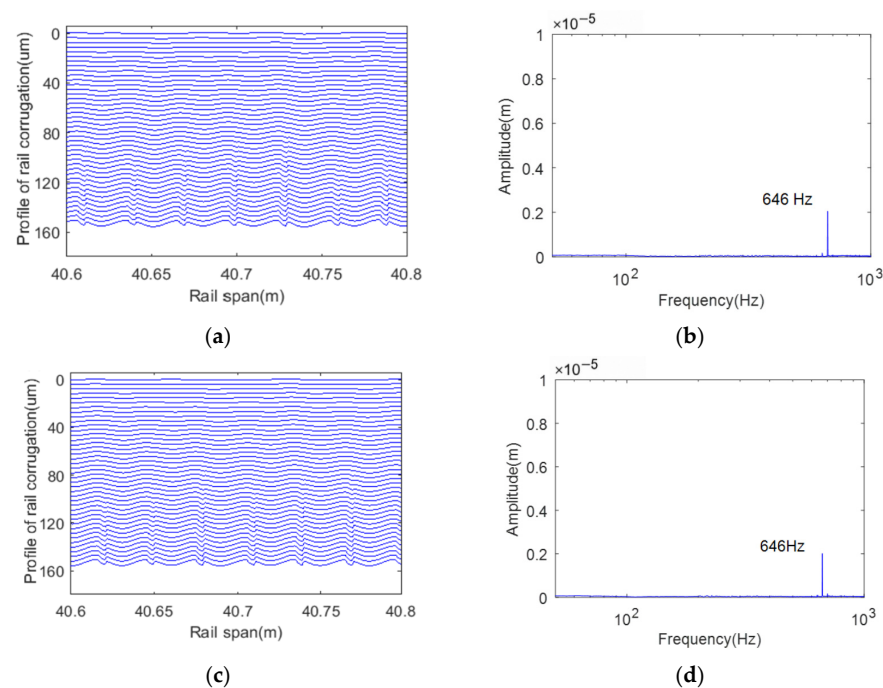


Figure 9. Growth process of corrugation and spectrum of the final profile for (a,c) wheel 1 and (b,d) wheel 2 with rail absorber after 200,000 wheel passages, respectively, $k_p = 10$ MN/m, $V = 20$ m/s. Profiles are shown after every 5000 wheel passages.

Quantitatively, the absorber reduces the corrugation amplitude at 25.6 mm wavelength by approximately 72% compared with the untreated case, as determined from the spectral amplitudes in Figure 9. The shift in the dominant corrugation frequency from 780 Hz to 646 Hz corresponds to a wavelength change from 25.6 mm to 30.9 mm at the same train speed, reflecting the modified track dynamic stiffness due to the added absorber stiffness K_a . The decay rate in the 600–1000 Hz range (Figure 8) increases by a factor of 2–3, which explains both the corrugation suppression and the noise reduction observed in Section 6. The broadband nature of this improvement is consistent with the continuous absorber design, which enhances the vibration decay rate across the entire target frequency range rather than only at a single tuned frequency.

6. Sound Radiation

6.1. Vibration Energy and Sound Power

The vertical vibration energy of the rail contributing to sound radiation, E_r , is obtained from Equations (1)–(3), and is calculated by

$$E_r = h_p \Delta z \sum_{L_c} v_r^2 \quad (13)$$

where v_r is the vibration amplitude, Δz is the length of a small rail section, and h_p is the total projected width of the rail section in the vertical direction, allowing for the top and bottom of the rail head and foot. The summation is performed within one car length, $L_c = 22.8$ m.

The rail vertical vibration energy for both the tracks with 10 MN/m and 60 MN/m rail fasteners due to the 1 μm roughness excitation are derived and presented in Figure 10 for comparison between the two tracks before and after installing the designed rail absorber. The rail vibration energy is observed to be dramatically reduced to 500–1200 Hz due to the rail absorber. Furthermore, the energy reduction for both tracks is nearly same.

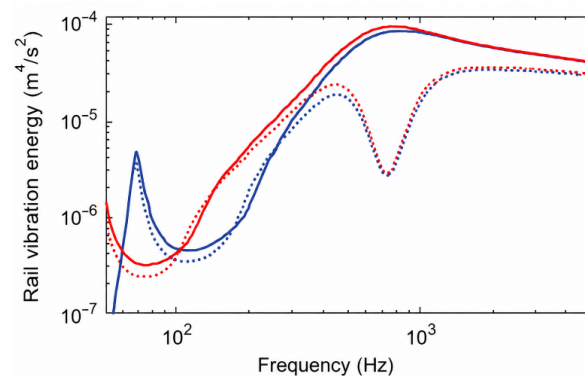


Figure 10. Energy of rail vertical vibration under 1 μm roughness excitation. (—) 60 MN/m no rail absorber, (.....) 60 MN/m with rail absorber, (—) 10 MN/m no rail absorber, (.....) 10 MN/m with rail absorber.

The radiated sound power from the rail, W_r , is estimated as

$$W_r = \rho c \sigma S \langle \bar{v}_r^2 \rangle = \rho c \sigma E_r \quad (14)$$

where ρ is the air density, c is the sound velocity in the air, σ is the radiation ratio for the rail structure, S is the surface area, $\langle \bar{v}_r^2 \rangle$ is the surface-averaged mean-square normal velocity, and E_r is calculated by Equation (13).

Two roughness spectra are used as input in the calculation. One is measured from the corrugated rail, and another is the rail roughness limit from ISO 3095:2005 [37]. They are presented in Figure 11. The radiation ratio of the rail is taken from reference [38] and is shown in Figure 12. The A-weighted sound power spectra radiated by vertical rail vibration in the 1/3 octave frequency band are presented in Figure 13. The peak at 800 Hz is sharply attenuated by installing the rail absorber. Accordingly, since the contribution of the 800 Hz component dominates, the overall A-weighted sound power level decreases greatly by 8.6 dB. For the ISO 3095 limit roughness, the overall A-weighted sound power level is reduced by 4.3 dB. The main reason is that the contribution from other bands above 1 kHz becomes dominant, although the 800 Hz component is also dramatically reduced by the rail absorber. These two roughness cases span the range from a corrugation-dominated spectrum to a broad, smooth-spectrum limit, providing an indication of the expected noise

reduction range in practice. For metro lines with active corrugation, reductions approaching the higher end of this range are anticipated.

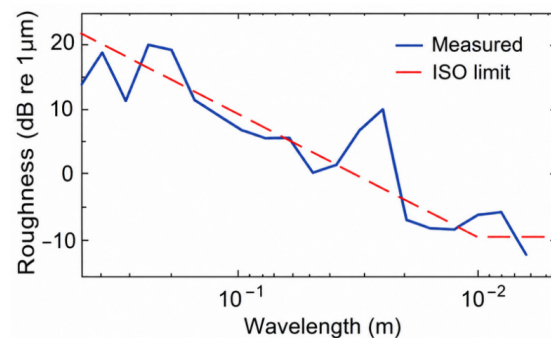


Figure 11. Roughness spectra used as input.

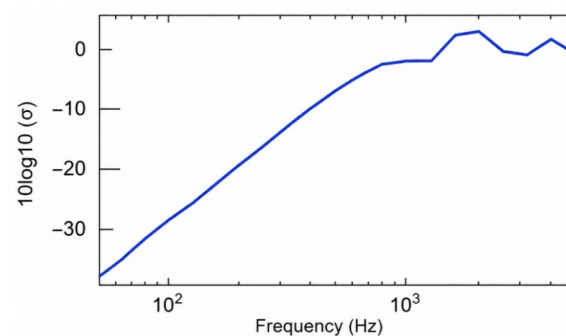


Figure 12. Radiation ratio for the rail vertical vibration.

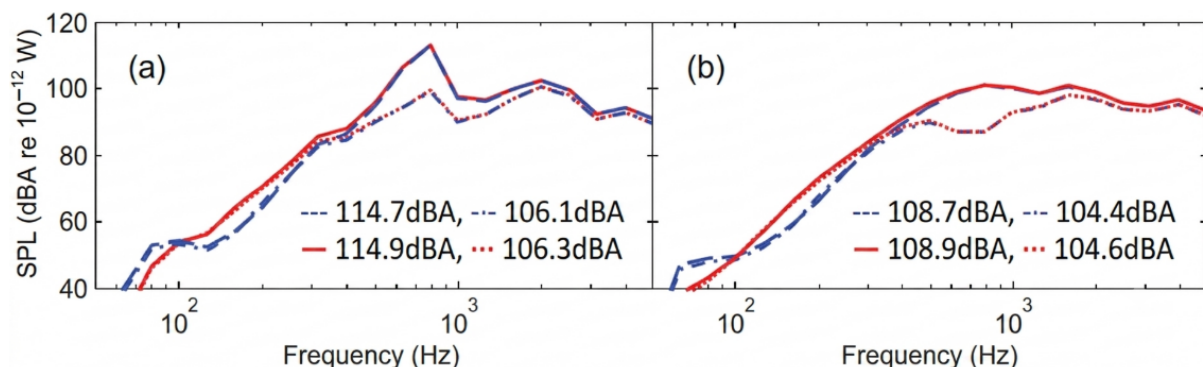


Figure 13. A-weighted sound power level radiated by vertical vibration for (a) corrugated rail and (b) ISO limit: (—) 60 MN/m no rail absorber, (---) 60 MN/m with rail absorber, (—) 10 MN/m no rail absorber, (.....) 10 MN/m with rail absorber.

The difference in A-weighted sound power reduction between the two roughness cases—8.6 dB for the corrugated rail versus 4.3 dB for the ISO limit—is explained by the spectral distribution of the roughness excitation. For the corrugated rail, the roughness spectrum is dominated by the 800 Hz component corresponding to the 25.6 mm corrugation wavelength, which is precisely the frequency band where the absorber is most effective. Consequently, attenuating this single dominant component yields a large overall reduction. For the ISO limit roughness, the excitation is more evenly distributed across multiple frequency bands, including those above 1 kHz where the absorber provides less attenuation. As a result, although the 800 Hz component is still substantially reduced, the overall A-weighted level decrease is smaller because other frequency bands contribute significantly to the total. The discrepancy between the estimated (9.4 dB) and measured (8 dB) noise reduction in the field test is attributable to several factors not captured in the

simulation: ground reflections, the presence of the sound barrier, and contributions from non-vertical rail vibration modes. The 1.4 dB overestimation is within an acceptable range for engineering predictions.

6.2. Measurement Results

Short-pitch corrugation was measured on the high-elastic fastener track of Shanghai Metro Line 11 (terminating at Disneyland, Pudong) and its wavelength spectrum was analyzed in this study. To mitigate this type of short-pitch corrugation, the Angel rail vibration absorber—a commercially available tuned mass—spring-damper device clamped to the rail foot—was installed, as shown in Figure 14a. The section where the absorber was installed is about 300 m long (see Figure 14b). The absorber was only installed on the low rail, because the short-pitch corrugation was produced at the low rail, and it scarcely grew at the high rail. Field observations confirm that short-pitch corrugation is effectively mitigated after absorber installation.

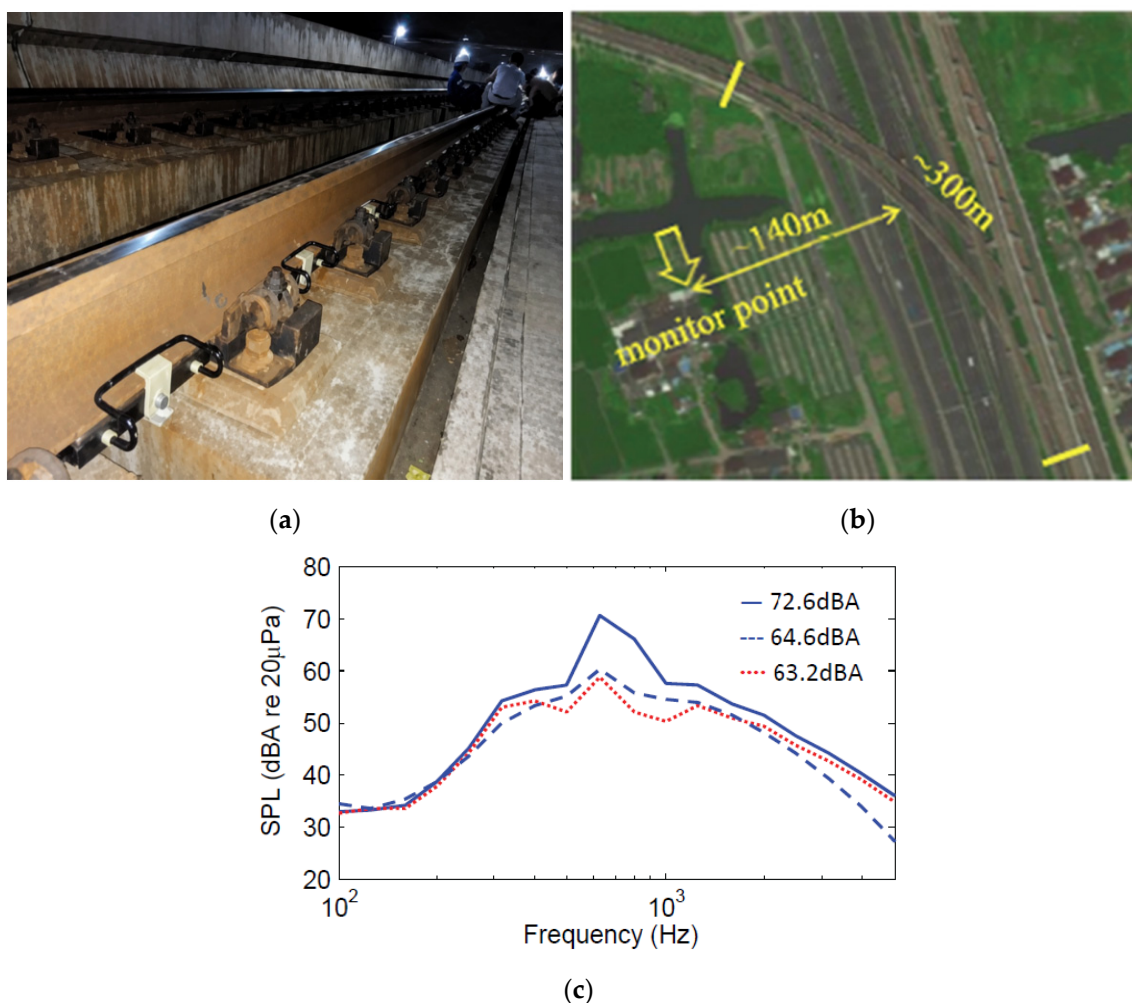


Figure 14. Measurement of passing noise: (a) Angel rail vibration absorber installed to rail foot by clips on slab track. (b) Scene of measurement at curved track with absorber. (c) A-weighted sound pressure level of passing noise in 1/3 octave band at train velocity 20 m/s: (—) measured before vibration absorber installed, (---) measured after rail vibration absorber installed, (.....) estimated with rail vibration absorber applied.

Meanwhile, the passing noise was measured before and after absorber installation. The noise monitoring point was located approximately 140 m from the corrugated rail (see Figure 14b), and the train velocity was about 20 m/s. A sound barrier was installed on

the viaduct wall along the track toward the monitoring side. The measured results are presented in Figure 14c in terms of the A-weighted sound pressure level in the 1/3 octave band. From it the components in the 500–1000 Hz bands are observed to be dominant before installation of the rail vibration absorber, and mainly consist of rolling noise. The components below 250 Hz and above 2500 Hz are secondary and their contribution to the overall A-weighted sound level may be neglected. The noise peak appears around 620–800 Hz, corresponding to the 25 mm wavelength corrugation and the train speed 20 m/s. The peak can be observed to be effectively attenuated by installing the vibration absorber. A significant 8 dB decrease at the monitor point is achieved by the rail vibration absorber via decreasing the rail component of rolling noise. However, if the rail is not corrugated, the rolling noise reduction will not be as effective as in this case.

The estimated noise reduction due to the absorber (dotted line in Figure 14c) is about 9.4 dB, derived by subtracting the simulated sound power reduction (Figure 13) from the pre-installation measured sound pressure level. Although the reduction is over-estimated by about 1.4 dB in comparison to the measured result 8 dB, it is acceptable and can be used as an approximate prediction for noise reduction in the rail absorber. Accordingly, rolling noise and rail corrugation in 600–1 kHz have a symbiotic relationship and thus double benefits could be obtained from using a rail vibration absorber, namely attenuating corrugation growth and decreasing noise radiation from the rail.

7. Conclusions

The main contribution and novelty of this work lie in the time-domain verification that wave reflections between moving wheels—rather than the conventional pinned–pinned resonance—are the dominant driver of short-pitch rail corrugation on high-elastic fastener tracks. The deliberate use of a continuous support model to exclude the pinned–pinned mode by construction enables unambiguous isolation of the wave reflection mechanism, which represents a methodological contribution to corrugation research.

The key findings are as follows. The dominant corrugation wavelength is 25.6 mm (corresponding to 780 Hz at 20 m/s), consistent with in situ measurements on Shanghai Metro Line 11. Corrugation growth without wave reflections is negligible, confirming that wheel–wheel wave reflections are the primary cause. Installing a tuned rail vibration absorber simultaneously suppresses the 25.6 mm corrugation growth and reduces the rail-related rolling noise by 8 dB(A) at the monitoring point. Furthermore, rolling noise and rail corrugation in the 600–1000 Hz range exhibit strong intrinsic coupling; mitigating one intrinsically controls the other.

The validated effectiveness of rail vibration absorbers demonstrates their immediate applicability as a practical countermeasure for metro lines with high-elastic fastener tracks. The identified target frequency range (600–1000 Hz) and dominant wavelength (25.6 mm) provide actionable input for absorber tuning and track maintenance scheduling.

Several directions warrant future investigation. The continuous support idealization, while effective for isolating the wave reflection mechanism, may underestimate broadband noise characteristics influenced by discrete sleeper periodicity; extending the framework to discrete support configurations is a natural next step. The qualitative suggestion that optimizing wheel spacing could mitigate corrugation requires dedicated parametric studies considering vehicle dynamics constraints. Given that wheel spacing is largely fixed in existing bogie designs, this measure is more applicable to new vehicle design rather than retrofit applications. Long-term field monitoring is needed to assess corrugation recurrence after absorber installation. The long-term performance of the vibration absorber under varying environmental conditions—including temperature-induced stiffness shifts in the elastomeric tuning elements and aging of the rubber components—remains an open

question. Dedicated durability testing under controlled temperature cycles and accelerated aging protocols are recommended to verify that the absorber maintains its target tuning frequency over its intended service life. Future studies should also investigate the human health impacts of the 600–1000 Hz noise band on passenger comfort and occupational exposure. Regarding generalizability, while the field results are specific to Shanghai Metro Line 11, the underlying wave reflection mechanism and the absorber tuning methodology are transferable to other metro lines with high-elastic fastener tracks, provided the absorber is tuned to the specific resonance frequency of the target system.

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