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The Influence of Track Irregularity in Front of the Turnout on the Dynamic Performance of Vehicles

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Abstract: While the track irregularity in turnout areas has a significant impact on wheel-rail contact, driving safety, and stability, the impact of track irregularity in front of the incoming turnout on vehicles is often overlooked. This paper fills the gaps in the study. As a result, a rigid-flexible coupled dynamic model of the vehicle and turnout is developed. The effect of various irregularities in front of the turnout on the dynamic performance of a vehicle at high speeds has been investigated based on a random sampling method. The results show that different types of track irregularities in front of the turnout have different effects on the dynamic responses of vehicles. The vehicle dynamic performance is most sensitive to short-wavelength (3 m) irregularities in front of the turnout. The alignment irregularity with long-wavelength (40 m) has a significant effect on the wheelset lateral force and lateral acceleration of the vehicle body, caused by two-point contact between the wheel and rail. The frequency range of the effect of the irregularities on the wheel-rail force, safety indicators, and accelerations of the vehicle is mainly below 200 Hz, 50 Hz, and 20 Hz. In this study, a comprehensive assessment of different irregularities is conducted, as well as a quantitative reflection on the effect of the irregularity on the dynamic indicators, providing a reference for maintenance.

Keywords: turnout; track irregularity; dynamic response; time-frequency analysis; multibody dynamics

1. Introduction

One of the reasons which can cause an increase in vehicle vibration, wheel-rail dynamic interaction, and wheel-rail noise is track irregularity [1,2]. Track irregularity not only affects ride comfort but also reduces the service life of track and vehicle systems, and it even jeopardizes driving safety in severe situations. The effect of track irregularity on the dynamic performance of vehicles tends to be more sensitive under high-speed conditions.

Railway turnouts, one of the most critical components of railway infrastructures, play an important role in diverting vehicles from one track to another. Turnouts are composed of a switch panel and a crossing panel connected by a closure panel [3]. However, turnouts are also considered to be one of the three poor links of high-speed railway infrastructures, owing to their complicated structure and multiple components [4,5]. The turnout itself demonstrates structural irregularities, and when vehicles pass through the turnout at high speed, the wheel-rail dynamic response will increase, the vehicle body acceleration more easily exceeds the limit, and safety accidents such as vehicle derailment are more likely to occur in some serious cases. Additionally, special attention should be given to irregularities in front of the turnout. Before the train passes through the turnout at a high speed, if there are poor track irregularities in front of the turnout, the train will...
produce complex spatial displacements and drive into the turnout in a bad or abnormal state of motion. Under the circumstances of structural irregularities of the turnout itself, the train may shake or even derail.

Numerous studies on track and turnout irregularities have been conducted. Most of these studies focused on the influence of track irregularities on the dynamic responses of vehicles. Karis et al. [6,7] and Choi et al. [8] studied the relationship between track irregularity and the vehicle response based on numerical simulation and measured data. Cantero et al. [9] proposed a method to detect track irregularities caused by infrastructural defects by the vertical acceleration of the vehicle based on wavelet transform. Sadeghi et al. [10] adopted the comprehensive parameter analysis method to study the influence of track irregularities on driving comfort, and the results showed that short-wave track irregularities have a great influence on driving comfort. Liu et al. [11] discussed the comprehensive influence of the vehicle speed and track irregularities on the vehicle vibration, and the results showed that the vertical irregularities had the greatest effect at all speeds on vibration discomfort. Youcef et al. [12] adopted a modal superposition method to compare the dynamic response of vehicles with random and non-random track irregularities, and the results showed that the track irregularities had a certain influence on the vertical acceleration of the vehicle. Hung et al. [13] studied the relationship among track irregularity, vehicle vibration, bridge vibration, and vehicle speed by the 3D finite element transient dynamic analysis method. Ji et al. [14] studied the effects of wind loads, track irregularity, and track elasticity on the dynamic characteristics of high-speed vehicles based on the vehicle-track coupled rigid-flexible model. For the turnout structure, the turnout irregularity has a certain influence on the dynamic responses of vehicles. Cao et al. [15] analyzed the influences on dynamic responses caused by turnout irregularities. Gao et al. [16] studied the influences of different rail weld irregularities on the wheel-rail dynamic interactions. Yin et al. [17] and Sun et al. [18] studied the influence of straight-line optimization in front of the switch on the dynamic characteristics of vehicles traveling in the turnout on the main line. Liao et al. [19] studied the influence of superelevation in the turnout on the passing performance of trains, and the results showed that the application of superelevation can optimize the safety indices. Xu et al. [20] and Zhu et al. [21] analyzed the effect of the stiffness irregularities on the dynamic train-turnout interaction. Chen et al. [22] studied the influence of conversion deviation on the safety and comfort of a train when passing the turnout. Xie et al. [23] analyzed the influence of track-distance roughness on the dynamic wheel-rail geometry contact, and the result showed that appropriate gauge widening can improve the structure irregularity.

From the previously published literature, there are many research achievements in track irregularity, turnout geometry, and its dynamic responses. The relationship between track irregularity and vehicle dynamic response on the main line has been extensively studied. However, few researchers have investigated track irregularities in front of the incoming turnout in depth, and the impact of the status of railway lines in front of the turnout is often neglected when vehicle-turnout dynamics analysis is performed. Many factors cause bad preceding track irregularity of the turnout, such as short-wavelength irregularity caused by rail welds [15], track buckling caused by inaccurate locked rail temperature or temperature difference [24], track gauge irregularities caused by partial loosening of fasteners, or partial separation between fasteners and rail, and side wear, uneven wear, and other abnormal wear of rail [25] in front of the turnout et al. In April 2020, a train was derailed in the turnout area of Jincheng Line, China, as shown in Figure 1. Due to the inaccurate locked rail temperature, there were significant lateral irregularities in front of the turnout and in the switch area, as a result, the track expanded, which led to derailment accidents. Therefore, it is of great importance and academic value to determine the law of the influence of the preceding irregularity on the vehicles passing through the turnout area, to guide on-site maintenance. In this paper, based on the theory of multibody dynamics (MBD), a rigid-flexible coupled model of high-speed vehicle-turnout-track was established. The influence of different irregularities in front of the turnout on
the dynamic performance of a vehicle passing through the turnout in the direction from the switch to the crossing panel at a high speed was studied and discussed by a random sampling method. Then, a comprehensive assessment of different irregularities on dynamic indicators is conducted. Finally, conclusions and the next steps are presented.

Figure 1. Vehicle derailment accident in the turnout area.

2. Numerical Approach
2.1. Vehicle Model

According to the structural type and suspension characteristics of the locomotive, the vehicle is regarded as an MBD system composed of a vehicle body, bogie frames, swing bolsters, axle boxes, and wheelsets. Among them, the vehicle body, bogie frame, and wheelset have 6 DOFs (degrees of freedom), while only 1 DOF of rotation about the wheel axle is considered in the axle box. The swing bolster is regarded as a zero-mass component with 0 DOF that moves together with the vehicle body [19]. The whole vehicle has a total of 50 DOFs. The vehicle model is established by the MBD software SIMPACK, and the topological graph of the vehicle model is shown in Figure 2. The differential vibration equation of the vehicle system is as follows [26]:

\[ M \dddot{Z}_c + C \dot{Z}_c + K Z_c = P_c \]  \hspace{1cm} (1)

where \( M \), \( C \), and \( K \) are the mass, damping, and stiffness matrix, respectively; \( \dddot{Z}_c \), \( \dot{Z}_c \), and \( Z_c \) are the acceleration, velocity, and displacement vector, respectively, and \( P_c \) is the dynamic load induced by vehicle gravity and track irregularities.

The rigid bodies are connected and constrained by force elements and hinge joints. The suspension elements are considered as spring-damper elements in SIMPACK. The nonlinear properties of critical parts such as the anti-yaw damper and lateral stop have been considered in this paper to realistically simulate vehicle suspension parameters. The anti-yaw damper keeps bogie movement stable by limiting the hunting motion between the vehicle body and bogie frame, and the function of the lateral stop is to limit the lateral displacement between the vehicle body and bogie frame. The nonlinear stiffness parameters of the anti-yaw damper and lateral stop are shown in Figure 3. The damping parameter for the anti-yaw damper is considered constant and equal to 2450 kN·s/m. The wheel tread is LMA type, which is a worn tread for high-speed vehicles in China, and the radius of the rolling circle is 0.43 m. Detailed vehicle parameters are shown in Table 1 [27,28]. The vehicle speed is 300 km/h on the main route.
Table 1. Vehicle parameters.

| Parameters                                           | Value                  |
|------------------------------------------------------|------------------------|
| Vehicle mass (kg)                                    | $3.376 \times 10^4$    |
| The rolling moment of inertia of the car body (kg·m²) | $1.094 \times 10^5$    |
| The nodding moment of inertia of the car body (kg·m²)| $1.655 \times 10^5$    |
| The yawing moment of inertia of the car body (kg·m²) | $1.561 \times 10^6$    |
| Bogie frame mass (kg)                                | $2.400 \times 10^3$    |
| The rolling moment of inertia of the bogie (kg·m²)   | $1.944 \times 10^3$    |
| The nodding moment of inertia of the bogie (kg·m²)   | $1.314 \times 10^3$    |
| The yawing moment of inertia of the bogie (kg·m²)    | $2.400 \times 10^3$    |
| Wheelset mass (kg)                                   | $1.850 \times 10^3$    |
| The rolling moment of inertia of the wheelset (kg·m²)| $9.670 \times 10^3$    |
| The nodding moment of inertia of the wheelset (kg·m²)| $0.123 \times 10^3$    |
| The yawing moment of inertia of the wheelset (kg·m²) | $9.670 \times 10^3$    |
| Longitudinal stiffness of primary spring (N/m)       | $9.800 \times 10^5$    |
| Lateral stiffness of primary spring (N/m)            | $9.800 \times 10^5$    |
| Vertical stiffness of primary spring (N/m)           | $1.176 \times 10^5$    |
| Vertical damping of primary spring (N·s/m)           | $1.000 \times 10^5$    |
| Longitudinal stiffness of secondary spring (N/m)     | $1.600 \times 10^5$    |
| Lateral stiffness of secondary spring (N/m)          | $1.600 \times 10^5$    |
| Vertical stiffness of secondary spring (N/m)         | $2.400 \times 10^5$    |
| Lateral damping of secondary spring (N·s/m)          | $4.000 \times 10^4$    |
| Vertical damping of secondary spring (N·s/m)         | $2.000 \times 10^4$    |
| Nominal rolling radius of the wheel (m)              | 0.43                   |
| Distance between the mass centers of the bogies (m)  | 17.50                  |
| Distance of bogie fixed axles (m)                    | 2.50                   |

Figure 2. The topological graph of the vehicle model.
2.2. Turnout Model

The turnout model is an elastic track model based on the standard CHN60–1100–1:18 turnout design for high-speed railways (the curve radius is 1100 m, and the turnout angle is 1:18). The total length of the turnout is 69 m, and a straight line of 100 m is established in front of the turnout.

The nominal rail profile changes continuously along the turnout. Based on the critical rail sections of the turnout, as shown in Figure 4a,b, longitudinal interpolation is performed along the rail, which interpolates variable rail cross-sections between the various profiles in the longitudinal direction employing Bézier curves [29]. The numbers of cross-sections in the switch and crossing panels are 72 and 48, respectively, and the variable cross-section lengths of the switch and crossing panels are 21.3 and 14.1 m, respectively [30]. This ensures a smooth transition from one rail profile to the next profile.

The switch and the stock rails are modeled as two separate bodies, obtained by defining separate sets of wheel-rail pairs. The position and angle of the switch and stock rail in the lateral and longitudinal directions can be set independently, so the relative position between the two rails can be simulated. The stock rail and switch rail models are shown in Figure 4c.
2.3. Wheel-Rail Contact Model

The vehicle subsystem and the turnout subsystem are connected by the wheel-rail contact model. The wheel-rail contact includes the identification of the contact patch and the calculation of the normal and tangential contact forces. In this work, a STRIPES-based method [31,32] is used to calculate the number and location of the contact patches.

The semi-Hertz contact theory [33] is adopted to calculate the normal force by calculating the normal wheel-rail equivalent penetration (Figure 5). The tangential contact force adopts the simplified Kalker theory, and the FASTSIM algorithm is used to calculate the nonlinear creep force [34]. The stresses in the semi-Hertzian method are:

\[
\begin{align*}
\sigma_{zi}(x,y) &= \frac{4E(1+\lambda)}{3\pi(n^2(1-v^2))}\frac{h_i}{a_i}\left(1-\frac{x^2}{a_i^2}\right)k_i^2 \\
\sigma_{xi}(x,y) &= \frac{3}{8}G C_{11,i} \nu_{xi} \left(\frac{a_i-x}{a_i}\right) k_i \\
\sigma_{yi}(x,y) &= \frac{3}{8}G C_{22,i} \nu_{yi} + \frac{2}{\pi} m G C_{23,i} \varphi_i \left(\frac{a_i+x}{a_i}\right) \times \left(\frac{a_i-x}{a_i}\right) k_i
\end{align*}
\]

where \(\sigma_{zi}\) are the normal contact stress; \(\sigma_{xi}\) and \(\sigma_{yi}\) are the tangential contact stresses; \(E\) is the Young’s modulus; \(h_i\) is the interpenetrations of the \(i\)th strip; \(n\) and \(m\) are Hertz parameters that depend on two relative curvature constants \(A\) and \(B\); \(v\) is the Poisson’s ratio; \(G\) is the elastic shear modulus; \(C_{11,i}\), \(C_{22,i}\), and \(C_{23,i}\) are Kalker’s creep coefficients of the \(i\)th strip; \(\nu_{xi}\), \(\nu_{yi}\), and \(\varphi_i\) are the longitudinal, lateral, and spin creepage, respectively; \(a_i\) is the half-length of each strip; \(a\) is the semi-axes of the elliptical contact patch in the \(x\)-direction.

Figure 5. The semi-Hertzian method.

The detailed derivation process of the normal and tangential contacts in the semi-Hertzian method can be found in Reference [35]. The calculated wheel-rail contact force can be laterally and vertically distributed in the vehicle-turnout system through the wheel-rail contact angle, thereby reflecting the coupling interaction between the vehicle and turnout [36].
2.4. Vehicle-Turnout-Track Coupling

The traditional moving track model has an advantage in computation efficiency; however, its effective frequency range is within 20 Hz [37]. For the study in this paper, some short-wavelength irregularities are considered, and the traditional moving track model is, obviously, not applicable to the analysis of short-wavelength irregularities at high frequencies. Therefore, to fully reflect the flexibility of the track structure, a rigid-flexible coupled model is simulated in the model. The rails and sleepers are considered as flexible bodies, and modeled as beam elements and solid elements, respectively, by ABAQUS finite element software. The rails and sleepers are connected by a fastener system. The stiffness and damping of the fastener system are $6 \times 10^7 \text{ N/m}$ and $7.5 \times 10^8 \text{ N-s/m}$, respectively [38]. The wheelsets, bogie frames, and the vehicle body are considered as rigid bodies in SIMPACK, connected by force elements.

The deformation of flexible bodies in multi-body dynamics can be represented by modal coordinates [36]. Therefore, the Craig–Bampton modal synthesis method is conducted to describe the elastic deformation of the track system. The modal synthesis method is then combined with a hybrid coordinate method in the multi-body dynamics system to achieve the coupling of rigid and flexible body motions. The coupling process of vehicle and track models is shown in Figure 6. Concerned with the calculation cost, there is no need to consider the whole length as a flexible track. Therefore, the flexible track length is set as 120 m, which takes into account the whole length of the turnout and the length for setting the irregularities in front of the turnout, and avoids the effects of junction position between the rigid and flexible track. The calculation frequency is 500 Hz.

Figure 6. Coupling of vehicle and track models.

3. Random Sampling Method

Under long-term operating conditions, the vehicle and track dynamics parameters are random due to manufacturing errors, structural aging, coupled loads of temperature, and vehicles, in accordance with normal distribution. To better reflect the wheel-rail contact when a vehicle passes a turnout and obtain the robust dynamic responses that perform well for different operating conditions, a random sampling method for dynamics analysis is applied.
3.1. Latin Hypercube Sampling method

The Latin Hypercube Sampling (LHS) method is adopted to generate random samples with multiple parameters. LHS method is a stratified sampling method, which is the k-dimensional extension of Latin square sampling [39]. In the LHS method, it assumes that there are k random input variables. The sampling area for each random variable is divided into n non-overlapping intervals with a probability of 1/nr, and then a sample is randomly drawn from each interval, thus forming a k-ary sample [40]:

Define a nr×k dimensional matrix P, where each column in P consists of a random permutation of {1,2, ..., nr}. Then define a matrix U of the same dimension as P, consisting of independent random numbers in the uniform distribution [0, 1]. The sampling matrix can be generated based on the inverse transformation method:

\[ x_{ij} = F_{\xi_j}^{-1}\left( \frac{P_{ij} - U_{ij}}{nr} \right), \quad i = 1,2,\ldots,n_r, \quad j = 1,2,\ldots,k \]  

(3)

where \( F_{\xi_j}^{-1} \) is the reciprocal of the target cumulative distribution function of the random variable \( \xi_j \).

3.2. Calculation Process

In this paper, four key parameters, namely vehicle speed, wheel-rail friction coefficient, axle load, and wheel tread profile, are selected as stochastic parameters based on the analysis in Reference [41]. When comparing the degree of variation of multiple random parameters, the relative values of the standard deviation and the mean are used for comparison. \( \sigma \) is the standard deviation, \( \mu \) represents the mean, and the coefficient of variation \( C_v \) can be expressed as:

\[ C_v = \frac{\sigma}{\mu} \]  

(4)

The mean values of the vehicle speed, wheel-rail friction coefficient, and axle load are 280 km/h, 0.4, and 16 t. The coefficient of variation is 0.1. The wheel tread profile is taken from 20 measured worn profiles, as shown in Figure 7.

A random sampling of the three key parameters was carried out using the LHS method. A LHS procedure was established by MATLAB, and 100 sets of random samples were selected. The sampling results of each parameter obey a normal distribution, as shown in Figure 8.

![Figure 7. The wheel tread worn profile.](image-url)
The vehicle-turnout dynamics analysis process based on the LHS method is shown in Figure 9. One hundred groups of random samples are used as input parameters for the dynamic model. The variation patterns of wheel-rail forces, safety, and comfort indicators are calculated under different wavelengths of irregularities in front of the turnout. Finally, the combined effects of irregularities in front of turnout on vehicle dynamic responses are further analyzed in the time domain and frequency domain.

4. Result Analysis

4.1. Simulation of Track Irregularity

Generally, compared with the vehicle passing straight through the turnout, the dynamic behavior and safety of the vehicle passing through the diverging track of the turnout are worse. However, it is challenging to analyze the influence of track irregularity on driving behavior and safety when the curves are considered at the same time [8].

![Figure 8](image.png)

**Figure 8.** Random sample: (a) Vehicle speed; (b) Wheel-rail friction coefficient; (c) Axle load.

![Figure 9](image.png)

**Figure 9.** Calculation process based on the stochastic method.
Therefore, attention is focused on how the preceding track irregularities of the turnout affect the dynamic performance of vehicles in the main direction in this paper.

Track irregularities come in a variety of forms, but they can all be thought of as a single harmonic excitation or a complex wave generated by the superposition of several single harmonic waves. The single harmonic wave can be expressed simply as a cosine function [12,37], which is:

$$Z_p(t) = \frac{1}{2} A \left( 1 - \cos \frac{2\pi vt}{L} \right), 0 \leq t \leq \frac{L}{v}$$  \hspace{1cm} (5)

where $v$ is the vehicle speed; $t$ is the running time of the vehicle; $L$ is the wavelength of track irregularity, and $A$ is the amplitude of track irregularity.

The displacement function is used as the system excitation input for the track irregularity applied in front of the turnout, and its end position is at the position of the starting point of the switch panel, as shown in Figure 10a. The alignment irregularity (AL), the irregularity of the longitudinal level (LL), and the irregularity of the cross level (CL) are considered, as shown in Figure 10b–d. It is worth mentioning that the CL is applied on the switch rail side.

![Figure 10](image-url)  

**Figure 10.** Schematic diagram of irregularity setting: (a) the applied position of the track irregularities; (b) AL; (c) LL; (d) CL.

According to Reference [42], in the case where harmonic irregularities and random irregularities are applied together, it is usually the influence of harmonic irregularities that covers the influence of random irregularities on dynamic responses of vehicles. Therefore, to better reflect the impact of irregularity in front of the turnout, random irregularities are not considered in the model.
4.2. Influence of Different Irregularities

The formation and development of track irregularities are the result of many random factors. Different track irregularities have different impacts on the dynamic performance of high-speed vehicles. In this paper, the influence of the preceding irregularities of the AL, LL, and CL types are studied. The wavelengths of 3 m, 5 m, 10 m, 20 m, and 40 m are considered in the simulations. The amplitudes of AL, LL, and CL are set as 14 mm, 20 mm, and 16 mm, which refer to the management value of the dynamic quality tolerance in the Chinese specification [43].

4.2.1. Analysis of the Maximum in Dynamic Indicators

The effects of different irregularities on the maximum value of dynamic indicators at different wavelengths for 100 groups of random parameter samples are shown in Figure 11. The error bar represents the standard error of the mean maximum value. As a comparison, the black bars in Figure 11 represent the extreme values of the dynamic responses without any irregularities in front of turnout. The AL, LL, and CL in front of the turnout have different degrees of influence on the vehicle dynamics compared to the results without irregularity. The AL mainly affects the wheelset lateral force and lateral acceleration of the vehicle body; the LL has a significant effect on the wheel-rail vertical force, wheel unloading rate, and vertical acceleration of the vehicle body, and the CL has the greatest influence on the wheel unloading rate and lateral acceleration of the vehicle body.
Figure 11. The effects of different irregularities in front of turnout on the maximum value of dynamic indicators at different wavelengths: (a) Wheel-rail vertical force; (b) Wheelset lateral force; (c) Derailment coefficient; (d) Wheel unloading rate; (e) Vertical acceleration of the vehicle body; (f) Lateral acceleration of the vehicle body.

The change in maximum values of dynamic indicators varies for different wavelengths of irregularities. The LL and CL at wavelengths of 3 m and 5 m cause significant increases in wheel-rail vertical forces and wheel unloading rate. The vertical force at 3 m wavelength of the LL is close to the limit of 170 kN, and the wheel unloading rate at 3 m wavelength of the LL and CL is nearly 1.0, which is much higher than the limit of 0.8. The AL at wavelengths of 10 m and above has a significant effect on wheelset lateral force. The LL in each wavelength band has a significant effect on the vertical acceleration of the vehicle body, among which the wavelength of 3 m has the greatest influence. The AL and CL in the whole wavelength section have a significant effect on the lateral acceleration of the vehicle body. The lateral acceleration under the AL is close to or greater than the limit of 0.6 m/s², with the maximum value occurring at 40 m wavelength; the maximum value of the lateral acceleration under the CL occurs at the wavelength of 3 m.

4.2.2. Time-Frequency Analysis

To deeper represent the effect of the irregularities in front of the turnout on the dynamic responses of the vehicle, key dynamic indicators are selected, and time-frequency analysis is carried out. The following are the results of calculations under standard parameters, where the speed is 300 km/h, the wheel-rail friction coefficient is 0.4, the axle load is 16 t, and the wheel tread profile is the standard LMA worn profile.

1. Alignment irregularity

The influence of the AL in front of the turnout on variations in the wheelset lateral force and lateral acceleration of the vehicle body are shown in Figure 12a,b, and the variations along the distance in the wheelset lateral force and wheelset lateral displacement under the AL at 40 m wavelength are shown in Figure 12c. At the position of the irregularities applied in front of the turnout, extremums of the wheelset lateral force exceed 30 kN under short and medium wavelengths, and the maximum value is 49.8 kN at the wavelength of 10 m. When the vehicle drives into the turnout, the influence of the irregularities in the short and medium wavelengths on the wheelset lateral force is small, and the maximum value of the wheelset lateral force in the turnout area is 32.9 kN at the wavelength of 40 m. As can be seen from Figure 12c, the long wavelength of the irregularity in front of the turnout leads to the wheelset moving left and right in a simple harmonic form. When the vehicle drives into the turnout in this attitude, the wheelset lateral displacement is right at the extreme value, which means that the wheelset travels close to the switch rail
side at this time, and the curved stock rail and the straight switch rail are in contact with the wheel tread at the same time, which is a two-point contact, resulting in a surge in the wheelset lateral force. For the lateral acceleration of the vehicle body, the different wavelengths of the irregularities in front of the turnout have a significant effect on the entire turnout area. The extremums of the lateral acceleration occur at 20 m away from the switch rail point under the irregularities at short and medium wavelengths (3 m–10 m), while the maximum of the lateral acceleration occurs at the end of the turnout under the irregularities at long wavelengths (20 m–40 m).

The Short-Time Fourier Transform with a fixed frequency-domain window function (STFT-FD) is applied for the time-frequency analysis [44]. The STFT-FD method fixes the window size in the frequency domain and uses different window sizes for different frequencies. Better frequency resolution can be achieved with the STFT-FD method. Figure 13 shows the time-frequency diagrams of the wheelset lateral force and lateral acceleration of the vehicle body under the AL in front of the turnout at wavelengths of 3 m and 40 m. The frequency range of the effect of the irregularity at 3 m wavelength on the wheelset lateral force is below 100 Hz, and the influence is mainly in the area in front of the turnout, and the influence on the turnout area is small; however, under the irregularity at 40 m wavelength, the wheelset lateral force has a larger response within the 30 m range behind the switch rail point, and the frequency range of the effect is below 20 Hz. For the lateral acceleration of the vehicle body, the main frequency range affected by the irregularity is below 5 Hz, and the irregularity at 40 m wavelength affects the lateral acceleration in the whole range of the turnout area.

![Figure 12](image-url)

**Figure 12.** Variations along the distance in the dynamic indicators under the AL in front of the turnout: (a) Wheelset lateral force; (b) Lateral acceleration of the vehicle body; (c) Comparison of the wheelset lateral force and wheelset lateral displacement.
2. The irregularity of longitudinal level

The influence of the LL in front of the turnout on variations in the wheel-rail vertical force and vertical acceleration of the vehicle body are shown in Figure 14. The LL of the short wavelengths (3 m and 5 m) in front of the turnout has a significant effect on the wheel-rail vertical forces. When the vehicle passes through the LL of short wavelengths, the wheel-rail vertical force will greatly increase. Afterward, there is an instantaneous detachment and collision between wheel and rail, and the vertical force decrease to 0 kN. When the separated wheels and rails contact again, there will be a great impact force on the wheel and rail. The range of wheel-rail detachment increases as the wavelength decreases. Therefore, some wheel-rail detachments will continue into the turnout area, and it also causes the overrun of the wheel unloading rate. The vertical acceleration of the vehicle body presents a periodic simple harmonic variation, and the maximum value appears at a short wavelength of 3 m. Interestingly, comparing the changes in the waveform of the wheel-rail vertical force and vertical acceleration of the vehicle body, it is discovered that the peak value of the vertical acceleration of the vehicle body presents a certain delay, which is caused by the time difference between the wheel-rail dynamic interaction and vehicle vibration. In addition, the LL in front of the turnout has longer-lasting effects on the vertical accelerations of the vehicle body. The changes in the wheel-rail vertical force barely lasts 20 m away from the switch rail point, while the changes in the vertical acceleration of the vehicle body are throughout the turnout.

Figure 15 shows the time-frequency diagrams of the wheel-rail vertical force and vertical acceleration of the vehicle body under the LL in front of the turnout at wavelengths of 3 m and 40 m. The LL at the wavelength of 3 m in front of the turnout has a much greater effect on the wheel-rail vertical force and the vertical acceleration of the vehicle body than the irregularity at the wavelength of 40 m. Under the effect of the LL at the wavelength of 3 m, the wheel-rail vertical force presents a high-frequency response at the extremes, while the vertical acceleration of the vehicle body has a large response until 40 m away from the switch rail point, and the frequency effect is concentrated below 20 Hz.
Figure 14. Variations along the distance in the dynamic indicators under the LL in front of the turnout: (a) Wheel-rail vertical force; (b) Vertical acceleration of the vehicle body.

Figure 15. Time-frequency graphs of dynamic indicators at the LL in front of the turnout: (a) Wheel-rail vertical force; (b) Vertical acceleration of the vehicle body.

3. The irregularity of cross level

The influence of the CL in front of the turnout on variations in the wheel-rail vertical force, wheel unloading rate, and vertical acceleration of the vehicle body is shown in Figure 16. The CL of the short wavelengths (3 m and 5 m) in front of the turnout has the same effect as the LL at short wavelengths. It causes a certain range of wheel-rail detachment, the wheel-rail vertical force sharply dropped to 0 kN, and the wheel unloading rate rapidly grows to 1.0. The lateral acceleration of the vehicle body is affected by the CL in front of the turnout at different wavelengths, and the maximum value appears at the wavelength of 40 m.
Figure 16. Variations along the distance in the dynamic indicators under the CL in front of the turnout: (a) Wheel-rail vertical force; (b) wheel unloading rate; (c) Lateral acceleration of the vehicle body.

Figure 17 shows the time-frequency diagrams of the wheel-rail vertical force, wheel unloading rate, and lateral acceleration of the vehicle body under the CL in front of the turnout at wavelengths of 3 m and 40 m. Similar to the influence of the LL, the CL at the wavelength of 3 m has a significant effect on wheel-rail vertical force and wheel unloading rate. Under the CL at the wavelength of 3 m in front of the turnout, the high-frequency response of the wheel-rail vertical force appears at the extreme value. The frequency range of its influence on wheel unloading rate is below 50 Hz. The CL at the wavelength of 40 m has a more obvious influence on the lateral acceleration of the vehicle body, and its main influence frequency range is less than 5 Hz.

Figure 17. Time-frequency graphs of dynamic indicators at the CL in front of the turnout: (a) Wheel-rail vertical force; (b) wheel unloading rate; (c) Lateral acceleration of the vehicle body.
4.3. Comprehensive Assessment of Different Irregularities on Vehicle Dynamics

According to the study of the dynamic responses of different irregularities in the previous section, it can be seen that different types of irregularities in front of the turnout have different effects on different dynamic indicators. To further quantify the influence of different irregularities on different dynamic indicators, this paper adopts normalized Power Spectral Density (PSD) for a comprehensive evaluation by referring to Reference [37]. Firstly, derailment coefficient ($Y/Q$), wheel unloading rate ($\Delta Q/Q$), lateral and vertical acceleration of the vehicle body ($\ddot{y}$ and $\ddot{z}$), and wheelset lateral force ($\Sigma Y$) are selected as evaluation indicators.

The results of the PSD are closely related to the selection of the calculation length and frequency range. Therefore, according to the time-frequency analysis results in Section 4.2.2, the calculated length and frequency of the PSD are determined, as shown in Table 2.

Table 2. The calculation parameters for the PSD.

| The Calculation Parameters | $Y/Q$ | $\Delta Q/Q$ | $\ddot{y}$ | $\ddot{z}$ | $\Sigma Y$ |
|----------------------------|-------|--------------|------------|------------|------------|
| Calculated length range    | 0–30 m| 0–70 m       | 0–30 m     |            |            |
| Calculated frequency range | 0–50 Hz| 0–20 Hz      | 0–200 Hz   |            |            |

The PSD is calculated by performing a Hamming window. The PSDs for different irregularities at different wavelengths are obtained. Then, the Influence Factor (IF) is defined according to Equation (6):

$$ IF_{ij} = \frac{P_{ij}}{P_0} $$

where $P_0$ is the PSD without irregularities in front of the turnout; $P_{ij}$ represents the PSD under an irregularity at a certain wavelength; $i$ and $j$ represent different wavelengths and different irregularities, respectively.

Since the study is to compare the effect of different types of irregularities, the maximum PSD in all cases is selected for normalization. The IF is shown in Equation (7) after normalization by the maximum:

$$ IF_{ij, norm} = \frac{IF_{ij}}{IF_{ij, max}} $$

The comprehensive evaluation of the effect of the irregularities in front of the turnout is shown in Figure 18. The values in the figures represent the IFs, ranging from 0 to 1. The closer the value is to 1, the greater the influence of the irregularity on the indicator. As can be seen from the figures, for the safety indicators: the greatest influence on the derailment coefficient is the AL at the wavelength of 40 m. The AL, LL, and CL at the wavelength of 3 m all have a great influence on the wheel unloading rate, among which the LL has the greatest effect. As for the acceleration indicators, the AL at wavelengths of 3 m, 5 m, 10 m, and 40 m, and the CL at wavelengths of 3 m and 5 m have a significant influence on the lateral acceleration of the vehicle body, and the greatest effect is the CL at the wavelength of 3 m; while the greatest influence on the vertical acceleration of the vehicle body is the LL at the wavelength from 3 m to 10 m. The CL at the wavelength of 3 m has the greatest influence on the wheelset lateral force, and the AL at the wavelength of 40 m also has a significant influence, with an IF of 0.852.
5. Conclusions and Future Works

In this paper, the effects of three irregularities in front of turnout with the alignment, longitudinal level, and cross level on the dynamic performance of vehicles are studied. The results of this paper will be helpful for understanding the relationship between track irregularities in front of the incoming turnout and running performances of high-speed trains. The wavelengths and amplitudes of the irregularities considered in this paper are referenced from the Chinese standards for track irregularities and the observed track irregularity characteristics of high-speed railway lines. Due to regular repairs and maintenance on high-speed railways, the amplitudes actually observed on high-speed railway lines rarely exceed the values considered in the simulation study. The results obtained in the ranges of amplitudes and wavelengths considered in this study can be summarized as follows.
The alignment irregularity in front of the turnout mainly affects the wheelset lateral force and lateral acceleration of the vehicle body; the irregularity of the longitudinal level in front of the turnout mainly affects the wheel-rail vertical force, wheel unloading rate, and vertical acceleration of vehicle body, and the irregularities of the cross level in front of the turnout mainly affect the wheel unloading rate and lateral acceleration of the vehicle body.

2. The dynamic performance of the vehicle is most sensitive to short-wavelength irregularities in front of the turnout. Short wavelength irregularities are more likely to cause a greater wheel-rail force and wheel load reduction, resulting in a greater safety risk. Thus, special attention should be given to short-wavelength irregularities in front of the turnout during maintenance.

3. Another interesting aspect is that the alignment irregularity with a long wavelength (40 m) has a significant effect on the wheelset lateral force and lateral acceleration of the vehicle body, which is caused by the two-point contact between the wheel and rail.

4. The frequency range of the effect of the irregularities in front of the turnout on the wheel-rail force, safety indicators, and accelerations of the vehicle are mainly below 200 Hz, 50 Hz, and 20 Hz, respectively, according to the frequency analysis.

5. The comprehensive assessment of different irregularities in front of the turnout from time-frequency analysis represents a quantitative and intuitive reflection of the effect of the irregularity on the dynamic indicators, which provide a reference for maintenance.

In future research, the influence of track irregularities in front of the turnout on the dynamic performance of a vehicle when passing through the turnout on the sideways and from the crossing panel to the switch panel should be further studied. The analysis can also be carried out in combination with included lines, curve lines, and different positions of the irregularity in front of the turnout.

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