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Mechanism and Mitigation of Carbody Shaking in Chinese High-Speed Trains

X. Dong[1], G. Ren[2], X. Li[3] and Z. Xu[1]

[1]Locomotive and Rolling Stock Research Institute , China Academy of Railway Sciences, Beijing, China
[2]China Railway Group Corporation, China Railway Group Corporation, Beijing, China
[3]Changchun Railway Vehicle Co.Ltd, China Railway Rolling Stock Corporation Limited, Beijing, China

Abstract:

This study investigates the underlying mechanism of car-body shaking, a lateral low-frequency vibration phenomenon frequently observed in in-service high-speed trains. Through extensive field tests on CR400BF EMUs operating on the Beijing–Shanghai High-Speed Railway, lateral accelerations of both bogie frames and car bodies were recorded under real-operation conditions, while wheel and rail profiles were periodically measured to evaluate the evolution of wheel–rail geometric interaction. Results reveal that when the wheel–rail equivalent conicity exceeds 0.35 – typically occurring toward the end of a wheel reprofiling cycle-a pronounced lateral harmonic vibration at approximately 8 Hz emerges in the bogie frame, which is subsequently transmitted to the car body. Complementary modal analyses, including both laboratory-based and operational (working) modal testing, reveal that the car body possesses a diamond-shaped elastic mode with a natural frequency falling within the 7.9~8.1 Hz range across service speeds of 250~380 km/h. This close alignment between the excitation frequency (induced by bogie hunting) and the structural resonance frequency of the car body leads to significant amplification of vibrations, manifesting as perceptible “shaking” for passengers. Based on this resonance mechanism, three targeted mitigation strategies are proposed: (1) optimizing wheel and rail profiles to suppress the growth of equivalent conicity; (2) enhancing car-body structural dynamics to shift critical modal frequencies away from the hunting band; and (3) implementing real-time monitoring algorithms to detect and diagnose

abnormal vibrations during operation. The findings provide a systematic framework for improving ride comfort and structural safety in high-speed rail systems.

Keywords: wheel/rail equivalent conicity, car-body shaking, field test, dynamic performance, lab test, resonance mechanism

1 Introduction

China's high-speed railway network has achieved unprecedented advancements in both operational speed and ride comfort, with commercial services routinely exceeding 300 km/h. Nevertheless, anomalous vibrations-manifesting as lateral car-body 'shaking', bogie swaying, or even perceptible seat trembling-continue to pose significant challenges to passenger comfort and long-term structural integrity. From the perspective of vehicle dynamics, two dominant categories of abnormal vibration have been identified in service: (i) low-frequency vibrations (typically below 10 Hz), associated with bogie hunting instability and resonant car-body responses; and (ii) high-frequency vibrations (~580 Hz), primarily excited by wheel polygonization or rail corrugation [1].

This paper focuses on the 8 Hz lateral car-body vibration commonly reported by passengers as 'shaking' or 'jitter'. This frequency lies within the range most sensitive to human perception and can severely degrade ride quality. Moreover, sustained excitation near structural eigenfrequencies may induce fatigue damage over time, raising concerns for operational safety. While prior work has attributed such phenomena to dynamic stiffness variations in rubber bushings of primary and secondary suspensions due to temperature and frequency dependence [2], effective mitigation strategies remain underexplored.

Internationally, several studies have noted that the natural hunting frequency of high-speed bogies often falls near 8 Hz, and track inspections have occasionally revealed localized periodic irregularities with wavelengths around 2 m, corresponding to excitation frequencies in the same band at typical operating speeds [3–5]. However, these reports seldom establish a clear causal link between track-induced excitations, bogie dynamics, and car-body resonance. In contrast, the Japanese Shinkansen program has successfully mitigated 8~10 Hz car-body resonances through coordinated optimization of the wheel–rail geometric interface and implementation of active suspension systems. Similarly, Deutsche Bahn's research on ICE trains led to the formulation of a coupled stability criterion involving critical speed, hunting frequency, and car-body modal characteristics. The European standard EN 14363 further institutionalizes ride stability assessment by defining the root-mean-square (RMS) of lateral car-body acceleration as a key performance metric.

In China, extensive field tests and simulations have been conducted on CRH-series electric multiple units (EMUs) [6–9]. The pioneering work of Academician Wanming Zhai established the theoretical foundation of "vehicle–track coupled dynamics," elucidating how track irregularities can trigger vehicle-level instabilities. Zhang demonstrated through full-scale testing that certain track segments at 250–300 km/h excite car-body resonances in the 7-9 Hz band, strongly correlated with bogie hunting motion. Despite these advances, most existing studies emphasize rigid-body dynamics and vehicle–track interaction, offering limited insight into elastic modal behaviour, particularly the role of structural resonance in amplifying perceived vibrations.

Concurrently, another line of research has focused on bogie-frame hunting instability, where the critical speed is highly sensitive to suspension parameters and wheel–rail contact geometry. Research groups at Southwest Jiaotong University, Beijing Jiaotong University, and others have developed high-fidelity models incorporating flexible bogie frames and suspended traction motor masses[10–12]. Li identified that mismatched stiffness and damping in anti-hunting dampers can induce "quasi-hunting", a sustained low-amplitude oscillation in the 6~10 Hz range that closely resembles real-world shaking events.

Notwithstanding these contributions, a critical gap remains: the lack of an integrated framework that connects evolving wheel-rail geometry, bogie hunting dynamics, and car-body elastic modes, especially the diamond-shaped bending mode known to appear near 8 Hz. Furthermore, while some studies employ bench tests or numerical simulations, few incorporate long-term, in-service monitoring data across diverse track conditions and speed regimes. Crucially, modal analyses have predominantly focused on rigid-body motions, neglecting the coupling between primary structures (e.g., car body) and secondary subsystems (e.g., interior panels, seats, flooring), which may significantly influence vibration transmission and perception.

To address these limitations, this study presents a comprehensive investigation based on long-term field monitoring of in-service CR400BF EMUs on the Beijing-Shanghai High-Speed Railway. We track the co-evolution of wheel/rail profiles, bogie-frame accelerations, car-body responses, and other subsystem dynamics under real operating conditions. By combining operational modal analysis (OMA), also known as working mode testing-with laboratory modal validation and multi-body simulation, we identify a resonance mechanism wherein elevated wheel-rail equivalent conicity (>0.35) triggers 8 Hz bogie hunting, which subsequently excites the car body's diamond-shaped elastic mode. Building on this mechanistic understanding, we propose three practical mitigation pathways: (1) proactive wheel/rail profile management to limit conicity growth; (2) structural enhancement of the car body to detune critical modes; and (3) deployment of real-time vibration monitoring for early diagnosis. This work bridges the gap between theoretical dynamics and operational practice, offering actionable insights for improving ride comfort and structural reliability in high-speed rail systems.

2 Methodology

To systematically elucidate the root cause of the lateral "shaking" phenomenon observed in in-service high-speed trains, characterized by a dominant frequency near 8 Hz, a Multidimensional investigation framework was established. The approach integrates field monitoring, laboratory testing, and numerical modeling, structured around three fundamental aspects of vehicle dynamics: (i) excitation sources, (ii) transmission pathways, and (iii) structural response characteristics. Special attention is devoted to identifying resonant interactions within the vehicle system at approximately 8 Hz, which lies within the human-perception-sensitive band and coincides with typical bogie hunting frequencies.

2.1 Investigation Framework

The investigation is organized along three hierarchical dimensions:

- Vibration Sources: Differentiated into forced (wheel out-of-roundness, bearing defects, motor/gearbox imbalances, track irregularities) and self-excited mechanisms (notably bogie and car-body hunting instabilities).
- Transmission Paths: Encompassing both primary and secondary suspension subsystems—including traction arms, coil springs, air springs, anti-hunting dampers, lateral/vertical dampers, torsion bars, and traction devices-as well as the suspension arrangements of the traction motor and gearbox. The dynamic behaviour of the bogie frame itself (rigid-body vs. elastic modes) is also examined as a potential pathway for energy transfer.
- Structural Response: Evaluated at both global and local levels. Global modes include lateral bending, torsional, and diamond-shaped deformations of the carbody-with and without onboard equipment such as transformers. Local responses focus on interior components (e.g., seats, floor panels) that may amplify passenger-perceived discomfort through secondary resonance.

The core diagnostic objectives are:

- Identify self-excited 8 Hz sources, particularly whether bogie hunting occurs under service conditions when wheel–rail equivalent conicity exceeds critical thresholds.
- Determine if the car-body exhibits natural frequencies near 8 Hz, especially elastic modes such as the diamond-shaped lateral bending mode.
- Assess the integrity of vibration isolation in the suspension system, specifically whether insufficient damping at 8 Hz permits amplification of transmitted vibrations.
- Evaluate seat-level modal characteristics to ascertain their contribution to localized discomfort.

2.2 Field Test Design

A two-year longitudinal field monitoring program was conducted on the leading car of a CR400BF trainset operating in regular commercial service. The campaign featured synchronized measurements of vehicle dynamics and wheel–rail geometry evolution:

- Vibration Monitoring: Lateral accelerations were recorded daily at key locations—including bogie frame ends and carbody center-using calibrated sensors during revenue runs at nominal speeds of 340 km/h ± 5%.

- Wheel–Rail Geometry Tracking: Wheel tread and rail head profiles were measured every 10,000 km of operation using the Greenwood MiniProf wheel profile gauge (accuracy: ±9 μm). Equivalent conicity was computed according to Chinese railway standard TB/T 3332 [15].
- Data Filtering and Evaluation: All acceleration data were processed in accordance with GB/T 5599-2019 [16], the Chinese national standard for railway vehicle dynamic performance assessment. A band-pass filter (0.5~10 Hz) was applied to isolate low-frequency lateral stability indicators. Although GB/T 5599 specifies units in m/s², all reported accelerations in this study are expressed in gravitational units (*g*) to maintain consistency with historical datasets and facilitate cross-study comparison.

This long-term dataset provides a unique empirical foundation for correlating the evolution of wheel-rail contact geometry (particularly equivalent conicity) with the onset and severity of 8 Hz lateral vibrations, thereby enabling data-driven insights into the shaking mechanism and its operational management.

3 Field test result

Longitudinal field monitoring was conducted on a CR400BF high-speed train operating on the Beijing-Shanghai High-Speed Railway, with comprehensive measurements of wheel tread profiles and vehicle dynamic responses performed throughout an entire wheel reprofiling cycle. The equivalent conicity, a key parameter governing lateral wheel-rail interaction dynamics, was computed by pairing measured wheel profiles with the standard Chinese 60 kg/m rail head geometry, in accordance with TB/T 3332.

The results reveal that equivalent conicity increases approximately linearly with accumulated mileage at a rate of 0.007 per 10,000 km. By the end of the reprofiling interval (approximately 244,000 km), the average equivalent conicity across all axles reaches 0.29—still below the design threshold of 0.4 specified for stable vehicle operation. Wheel wear is predominantly localized within ±25 mm of the nominal rolling circle, exhibiting a consistent material loss rate of approximately 0.03 mm per 10,000 km, as illustrated in Figures 1 and 2.

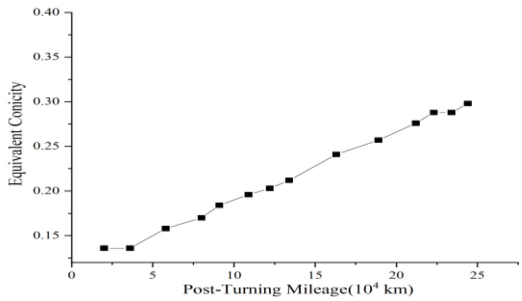


Figure 1: The Variation of Equivalent Conicity with Mileage After Reprofiling

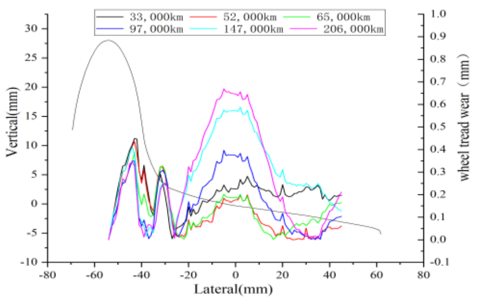


Figure 2: The Variation of wheel wear with Mileage After Reprofiling

Concurrent with geometric assessments, lateral accelerations of the bogie frame were continuously recorded during revenue service at a nominal speed of 350 km/h

(±5%). In alignment with the Chinese national standard GB/T 5599-2019, all acceleration signals were processed using a 0.5~10 Hz band-pass filter to isolate low-frequency components relevant to lateral stability. Statistical analysis of the filtered data demonstrates a progressive increase in peak lateral acceleration as mileage accumulates post-reprofiling. At 244,000 km, the maximum recorded lateral acceleration reaches 0.39 g-observed predominantly during high-speed passage through turnout zones near stations. This value remains well below the safety limit of 0.8 g stipulated in GB/T 5599, confirming adequate operational margins. In contrast, on regular tangent or curved track sections away from station throats, peak lateral accelerations remain around 0.2 g.

Notably, spectral analysis indicates a shift in dominant excitation frequency depending on track context: in station approach areas, the primary oscillation occurs near 7 Hz, whereas on mainline segments, the dominant frequency is approximately 6 Hz. This distinction suggests that local track geometry-particularly the presence of turnouts and transition zones-plays a non-negligible role in modulating the dynamic response.

The root-mean-square (RMS) values of lateral bogie acceleration follow a similar trend, peaking at approximately 0.11g at the end of the reprofiling cycle as showed in figure 3 and figure 4. For comparative assessment against international criteria, the lateral stability limit was also evaluated according to EN 14363[17], yielding a permissible RMS threshold of 0.52 g based on the specific bogie parameters of the CR400BF. The measured values thus indicate not only compliance but also a considerable safety buffer under current operating conditions.

Figure 5 provides a detailed time-frequency decomposition of the bogie response at 244,000 km, revealing transient bursts of energy centered near 7~8 Hz during critical track transitions. These observations, combined with the conicity evolution data, establish a clear empirical correlation between increasing wheel-rail geometric mismatch and the emergence of intensified lateral vibrations—laying the groundwork for the resonance-based shaking mechanism analysed in subsequent sections.

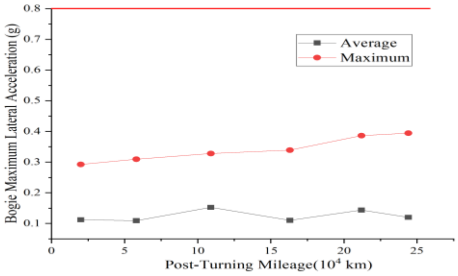


Figure 3: Variation of Lateral Acceleration maximum of the Bogie Frame with Mileage

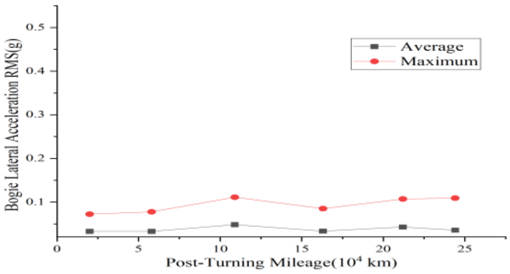
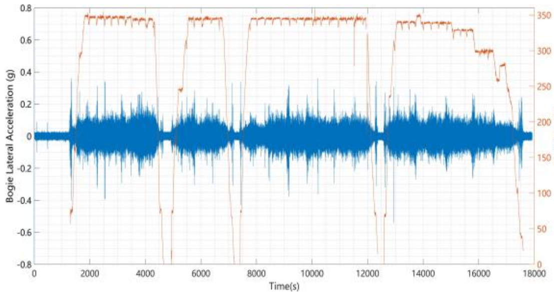
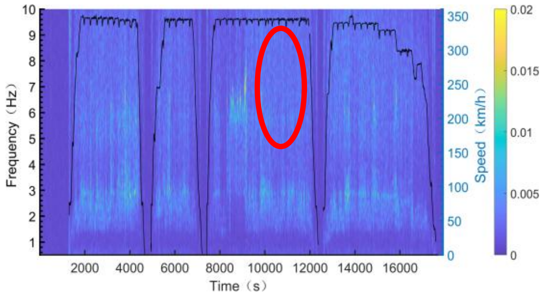


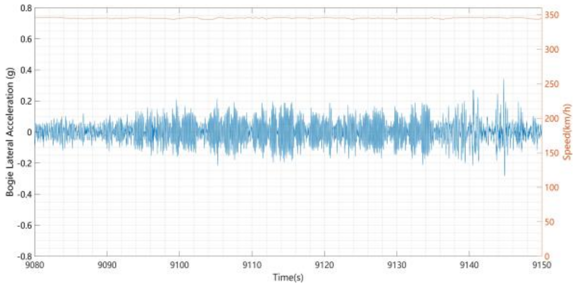
Figure 4: Variation of Lateral Acceleration RMS of the Bogie Frame with Mileage



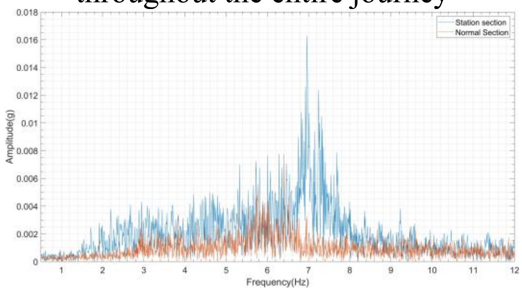
(b) Entire Journey



(b) Variation of frequency over time throughout the entire journey



(c) Local time-domain waveform



(d) Local frequency analysis of the peak values

Figure 5: Analysis of the Bogie Frame Lateral Acceleration at 244,000 km Mileage After Wheel Reprofiling

4 Carbody and Interior Component Modal Characterization

To investigate whether the observed lateral “shaking” phenomenon at approximately 8 Hz is associated with structural resonances of the car-body or interior components, comprehensive modal tests were conducted on both the bare car-body structure and installed seating systems.

Initial laboratory-based modal testing of the isolated passenger seat revealed a first-order natural frequency not lower than 20 Hz. However, when the same seat was mounted on the actual car-body floor under in-service boundary conditions, its effective modal frequency dropped significantly to the range of 8~13 Hz. This reduction highlights the strong influence of mounting stiffness and coupling with the primary vehicle structure on local component dynamics.

Separately, the global modal characteristics of the car-body were evaluated using two complementary approaches: (i) Laboratory Modal Testing (LMT) under controlled static conditions, and (ii) Operational (or Working) Modal Analysis (OMA) performed during revenue service across a range of operational speeds (250~380 km/h).

As summarized in Table 1, LMT identified the first six global modes of the empty car-body, with the lowest natural frequency at 9.90 Hz-still notably higher than the 8 Hz shaking frequency reported by passengers. Nevertheless, given that laboratory boundary conditions (support constraints, absence of onboard equipment, and lack of dynamic loading) may not fully replicate real-world operational states, OMA was subsequently employed to capture the true in-service modal behavior.

The OMA results, presented in Table 2, reveal two dominant elastic modes within the 7.9~8.9 Hz band across all tested speeds:

- A diamond-shaped lateral bending mode, with a frequency ranging from 7.9 Hz (at 350 km/h) to 8.1 Hz (at 250 km/h), and
- A first vertical bending mode, varying between 8.6 Hz and 8.9 Hz.
- Both modes exhibit moderate damping ratios (2.6%~4.9%), sufficient to limit uncontrolled amplification but inadequate to suppress resonance when excited near their natural frequencies.

These findings collectively indicate that the perceptible “shaking” is primarily driven by resonant excitation of the car-body’s diamond mode at approximately 8 Hz. This resonance is triggered by external lateral forcing—most notably, bogie hunting motion induced by elevated wheel–rail equivalent conicity during late stages of the wheel reprofiling cycle.

Thus, while the bare car-body appears dynamically safe under idealized conditions, its in-service modal response-coupled with interior subsystems and excited by self-sustained bogie instabilities—creates a resonance pathway that directly manifests as passenger-perceived discomfort.

Figure 6: Modal test in field line

First-order modal natural frequency	9.90Hz
Second-order modal natural frequency	10.12Hz
Fourth-order modal natural frequency	12.90Hz
Fifth-order modal natural frequency	15.73Hz
Sixth-order modal natural frequency	16.03Hz

Table 1: modal natural frequency

Modal	parameter	250km/h	300km/h	350km/h	380km/h
diamond	frequency	8.1Hz	8.0Hz	7.9Hz	8.0Hz
	damping rate	3.5%	2.6%	3.6%	3.5%
vertical bend	frequency	8.9Hz	8.8Hz	8.6Hz	8.8Hz
	damping rate	3.3%	4.3%	4.9%	4.7%

Table 2: modal test result of carbody

5 Shaking mechanisms

The car-body "shaking" observed in high-speed trains can be systematically interpreted through three interrelated aspects: (i) the excitation source, (ii) the vibration transmission path, and (iii) the structural dynamic response of the carbody. The following subsections present a comprehensive analysis of each component based on theoretical modelling, field measurements, and experimental modal data.

5.1 Excitation Source: Bogie Hunting Instability

Bogie hunting is a self-excited lateral oscillation inherent to high-speed railway vehicles, arising from the nonlinear coupling between wheel–rail geometry and creep forces. Classical linear stability theory indicates that the hunting frequency f_h increases with both vehicle speed V and the wheel–rail equivalent conicity λ, as described by the following approximate expression [18,19]:

$$f_h = \frac{V}{2\pi}\sqrt{\frac{2b\lambda}{r_0 {l_1}^2}} \tag{1}$$

where:

- b is the lateral distance from the wheelset center to the axle box,
- $r_0$denotes the nominal rolling radius of the wheel,
- l_1 represents the longitudinal distance from the wheelset center to the lateral guide point,
- V is the operating speed, and
- λ is the equivalent conicity.

Using the CR400BF bogie parameters listed in Table 3 and assuming an equivalent conicity range of λ=0.35~0.45 -typical during the late stage of the wheel reprofiling cycle-theoretical computation yields a hunting frequency of approximately 3.9~8.0 Hz at a service speed of 350 km/h.

Parameter	Value	Unit
λ	0.35~0.45	—
r_0	0.465	m
b	0.12	m
l_1	0.8	m
V	350	km/h

Table 3: CR400BF bogie parameter

Field monitoring data corroborate this prediction: as wheel and rail profiles degrade over operational mileage, the effective conicity gradually increases, thereby lowering the critical speed for hunting onset and shifting its dominant frequency into the 4~9 Hz band. This frequency range aligns closely with the passenger-reported "shaking" sensation, strongly suggesting that bogie hunting serves as the primary external excitation source.

5.2 Vibration Transmission Path: Suspension System Dynamics

Vibration energy generated by bogie hunting is transmitted to the car-body via the primary and secondary suspension systems. Static transfer function tests conducted under laboratory conditions show no resonant amplification or significant gain in the 8~13 Hz frequency band, indicating that the suspension components themselves do not introduce problematic dynamics under idealized settings.

However, under real-world operating conditions-characterized by sustained wheel-rail irregularities and nonlinear contact forces-the persistent excitation of bogie hunting enables efficient energy transfer across the suspension interfaces. Although the suspensions are not inherently resonant in this band, their role as a dynamic conduit facilitates the propagation of low-frequency lateral vibrations from the bogie frame to the car-body frame, particularly when the excitation frequency approaches structural resonances of the superstructure.

5.3 Structural Response: Carbody and Component Resonance

Operational modal analysis (OMA) of the in-service car-body reveals two global elastic modes within the critical 8~10 Hz range:

- The first diamond-shaped lateral bending mode (~8.0 Hz),
- The first vertical bending mode (~8.9 Hz).

These frequencies coincide closely with the dominant hunting-induced excitation band (4~9 Hz), creating conditions for resonant amplification. As illustrated in Figure 7, when the bogie hunting frequency overlaps with the car-body's natural frequencies, particularly during the later stages of the wheel maintenance interval, the originally moderate bogie vibrations are significantly amplified into perceptible whole-body shaking.

Moreover, interior components such as seats and luggage racks, when installed on the vibrating floor, exhibit reduced local modal frequencies (8~13 Hz) due to structural coupling and boundary condition changes. These subsystems not only experience intensified local accelerations but also generate secondary effects, including audible "tapping" or "rattling" noises, further degrading ride comfort.

In summary, the car-body shaking phenomenon emerges from a resonance cascade:

- Wheel–rail wear elevates equivalent conicity, triggering bogie hunting near 8 Hz.
- Vibration energy transmits efficiently through the suspension system under dynamic loading.
- The car-body's global diamond mode—and coupled interior components—resonate at matching frequencies, amplifying motion to perceptible levels.

This mechanistic understanding provides a foundation for targeted mitigation strategies, including conicity management, suspension tuning, and structural stiffening of critical car-body regions.

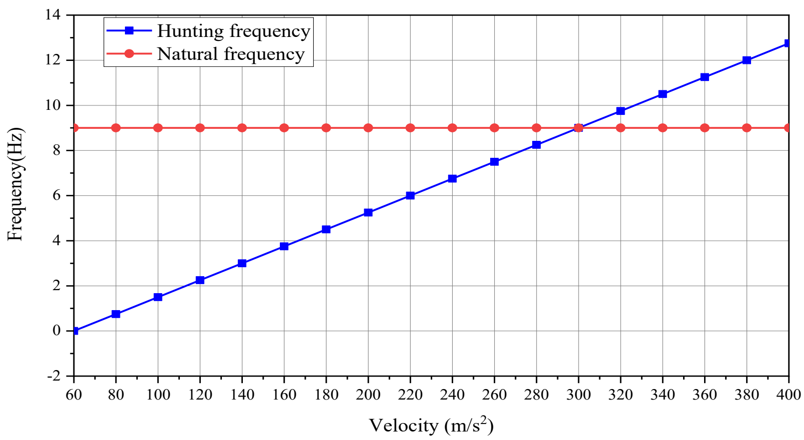


Figure 7: Resonance frequency

6 Integrated Mitigation Strategies for Carbody Shaking

The preceding analysis establishes that the car-body "shaking" phenomenon arises from a resonance cascade triggered by frequency alignment between bogie hunting excitations (4~9 Hz) and low-order structural modes of the car-body and interior components. Consequently, effective mitigation requires breaking this resonance condition through either:
(i) shifting the car-body's modal frequencies away from the excitation band, or

(ii) suppressing the amplitude and frequency content of the excitation source itself—primarily by controlling wheel-rail equivalent conicity.

Field investigations on critical track segments-including K419~K420 on the Beijing–Shanghai High-Speed Railway and K2391+600 on the Lanxin Passenger Dedicated Line—combined with operational modal analysis, wheel-rail contact modeling, and maintenance records, identify four root causes:

- Degraded wheel-rail geometric compatibility leading to elevated equivalent conicity.
- Inconsistent or suboptimal wheel reprofiling practices.
- Frequency coincidence between car-body global modes and bogie hunting.
- Localized track irregularities, particularly in station approach zones, within 1 km downstream of turnouts, and at inter-bureau boundary sections.

To address these interdependent factors, a holistic, multi-layered mitigation framework is proposed, encompassing infrastructure maintenance, vehicle design refinement, and operational adaptation.

6.1 Standardization and Optimization of Rail Grinding Practices

Inadequate or non-uniform rail grinding across railway bureaus often results in excessively wide wheel–rail contact bands (40~50 mm) or severe gauge corner wear, both of which significantly increase equivalent conicity. To mitigate this, rail grinding should be standardized using optimized profiles such as 60N or 60 kg/m asymmetric rails, targeting a controlled contact band width of 20~35 mm. A periodic, data-driven grinding schedule must be enforced, with intensified monitoring and intervention on historically problematic segments (Figure 8). Comparative studies (Figure 9) demonstrate that consistent profile-based grinding reduces conicity variability by over 40% and delays the onset of hunting instability.

Figure 8: Rail absence of grinding

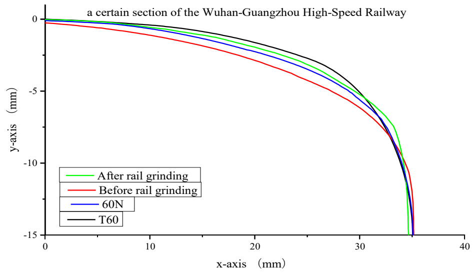


Figure 9: Comparison of grinding

6.2 Adoption of Low-Conicity Wheel Profiles

A newly developed wheel profile,currently being deployed on CRH3-type EMUs (Figure 10) is specifically engineered to reduce initial equivalent conicity while maintaining stable contact mechanics [20]. Key design features include:

- Preservation of the primary tread contact zone to ensure load-carrying capacity.
- Flange thinning and reduced taper at the flange root to minimize unintended

flange–gauge contact, thereby lowering matched conicity under degraded rail conditions.

- Slightly increased taper at the outer tread edge to prevent excessively low conicity (e.g., due to gauge widening), which could otherwise induce vehicle oscillations via loss of self-centering capability.

Field trials confirm that this profile reduces lateral bogie frame accelerations by up to 30% and significantly decreases hunting alarm rates during high-speed operation.

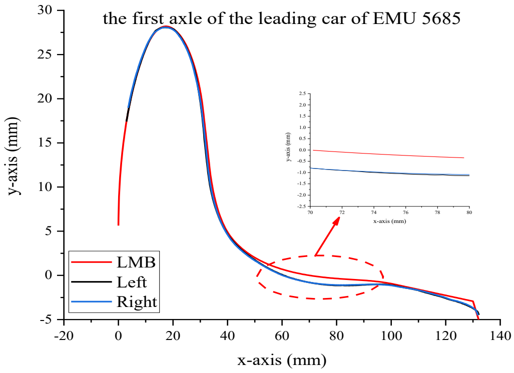


Figure 10: New profile wheel profile

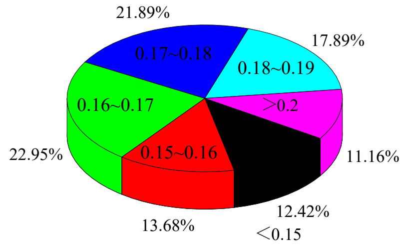


Figure 11: Equivalent conicity statistics

6.3 Enhancement of Wheel Reprofiling Quality and Consistency

Post-reprofiling measurements across 12 EMU depots reveal that more than 10% of wheelsets exhibit equivalent conicity outside the optimal range (i.e., <0.15 or >0.20), directly contributing to instability. This stems from insufficient machining precision, single-pass cutting strategies, and lack of real-time quality feedback. To improve consistency, the following measures are recommended:

- Implementation of a two-pass reprofiling protocol, with a minimum cut depth of 0.5 mm per pass to reduce residual form errors.
- Integration of axle-box lower supports during lathe operations to minimize dynamic deflection and transmission errors.
- Deployment of online equivalent conicity measurement systems to enable closed-loop process control and evidence-based release criteria (Figure 13).

6.4 Elevation of Carbody Modal Frequencies

Operational modal testing confirms that the first diamond mode (~8.0 Hz) and first vertical bending mode (~8.9 Hz) fall squarely within the bogie hunting band. To decouple this resonance, structural modifications should aim to raise the lowest global modal frequency above 10 Hz, ensuring a frequency separation ratio ≥ 1.4 relative to the dominant hunting frequency (Figure 12). Practical interventions include:

- Reinforcement of the underframe longitudinal members and cross-beams.
- Optimization of seat and luggage rack mounting interfaces to reduce local flexibility and avoid secondary resonances in the 8~13 Hz range.

- Use of constrained-layer damping treatments in high-strain regions to simultaneously increase stiffness and dissipate vibrational energy.

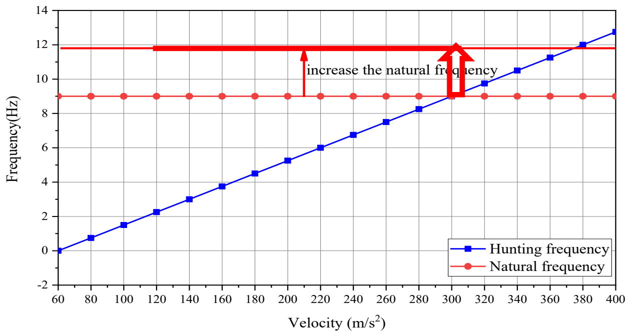


Figure 12: raising modal frequency at service speed

6.5 Dynamic Train Routing Based on Real-Time Risk Assessment

Operational data reveal marked differences in ride smoothness for identical trainsets operating on different lines, for instance, smoother performance on the Hushutong Line compared to the Ningqi Line under similar speed and maintenance conditions. This suggests that line-specific risk profiles can be leveraged for proactive management. For EMUs exhibiting frame instability or passenger-reported shaking, temporary rerouting away from high-conicity or poor-track-condition segments serves as an immediate operational countermeasure. Original routing may be reinstated only after corrective actions—such as wheel reprofiling or targeted rail grinding-are completed, thereby enabling a closed-loop risk mitigation cycle.

7 Conclusions and Contributions

This study presents a systematic investigation into the dynamic interaction between wheel tread profiles and high-speed vehicle–track systems in China's extensive high-speed railway network, with a focus on mitigating the passenger-perceived "car-body shaking" phenomenon. Through an integrated methodology combining field measurements, operational modal analysis, multibody dynamics simulation, and maintenance data mining, the research establishes a clear causal chain: excessive wheel–rail equivalent conicity-driven by suboptimal tread–rail profile matching—triggers bogie hunting in the 4~9 Hz band, which resonantly excites low-order car-body modes (notably the diamond and first vertical bending modes at 8–9 Hz), resulting in amplified whole-body vibrations.

The analysis reveals that conventional European-derived profiles such as LMA and LMB, while effective in their original design contexts, exhibit limitations under China's unique operating conditions. Specifically, the LMA profile, optimized for ultra-high-speed stability on straight alignments, accelerates rail head wear on the frequent small-radius curves characteristic of mountainous corridors. In contrast, thicker-flange LMB-type profiles enhance derailment safety through improved flange clearance but introduce higher contact stresses on tangent tracks, degrading ride comfort and promoting uneven tread wear.

To address these trade-offs, a series of customized wheel tread geometries have been developed using a multi-objective optimization framework that concurrently prioritizes:

(i) lateral stability margin against hunting,
(ii) minimization of wheel and rail wear rates, and
(iii) robust curve negotiation capability.

Designated in internal development cycles as variants such as "LMB-10," these profiles retain compatibility with the national standard GB/T 4608-2019, ensuring fleet-wide interchangeability and maintenance feasibility. Validation through both numerical simulations and in-service trials on CR400AF/BF “Fuxing” EMUs demonstrates significant improvements: up to 25% reduction in tread wear rate and 15% tighter control of equivalent conicity evolution over a full reprofiling interval, all while preserving critical hunting stability thresholds.

Critically, this work underscores that wheel profile design must be co-optimized with track maintenance practices, car-body structural dynamics, and operational strategies. The proposed mitigation framework compassing standardized rail grinding, precision wheel reprofiling, car-body stiffening, and dynamic train routing-represents a paradigm shift from component-level tuning to system-level resilience engineering.

Looking forward, the integration of real-time wheel profile monitoring via onboard sensing and the coupling of tread wear evolution models with predictive maintenance platforms will enable proactive management of wheel-rail interaction states. Such digitalization efforts are essential for sustaining ride quality, extending asset life, and ensuring the long-term safety of high-speed operations.

In summary, rational wheel tread design is not merely a geometric exercise but a strategic lever for system-wide dynamic performance. The methodologies and findings presented herein offer a transferable reference for other high-speed rail networks confronting similar challenges in balancing stability, wear, and passenger comfort under diverse infrastructure and operational constraints.

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