

A multibody dynamics-based investigation of wheel-rail wear and rolling contact fatigue for high-speed railway applications in Vietnam

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Received 18 March 2026

Revised 15 April 2026

Accepted 25 April 2026

Abstract

Purpose – High-speed railway development in Vietnam requires an early assessment of wheel-rail degradation mechanisms. This study investigates wheel wear evolution and rolling contact fatigue (RCF) risk under representative high-speed operating conditions.

Design/methodology/approach – A numerical framework integrating multibody vehicle dynamics and wear prediction modelling is developed. Vehicle-track interaction is simulated in VI-Rail using a standard ERRI passenger bogie model. Key wheel-rail contact parameters, including normal forces, creepages and contact locations, are extracted and used in an Archard-based wear model to predict wheel profile evolution, while RCF risk is assessed using Hertzian contact stress indicators.

Findings – The results show that wheel wear alters tread geometry, with a maximum wear depth of approximately 0.2–0.4 mm, affecting vehicle dynamics. The critical speed varies non-monotonically, increasing by about 3–5% at early stages and decreasing by 10–15% with further wear. Track irregularities raise creepage to the order of 10^{-3} and increase wear rate by 20–30%. These effects also indicate an elevated risk of RCF.

Originality/value – The study provides a multibody dynamics-based framework for analysing wheel wear and RCF in prospective Vietnamese high-speed railway applications. The proposed framework also provides a scientific basis for assessing wheel degradation and supporting infrastructure design, operational planning and predictive maintenance in high-speed railway systems.

Keywords Wheel wear evolution, Rolling contact fatigue, Multibody dynamics, Wheel-rail interaction, High-speed railway, Predictive maintenance

Paper type General review

1. Introduction

High-speed rail (HSR) has become an essential component of modern sustainable transport systems owing to its high passenger capacity, superior energy efficiency and strong safety performance.

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Many countries have invested significantly in high-speed railway infrastructure to enhance long-distance mobility while reducing congestion and environmental impacts. In Vietnam, the proposed North-South high-speed railway corridor has been identified as a major strategic infrastructure project, with an anticipated operating speed of up to 350 km/h (Loan, 2024). The implementation of such a system requires careful consideration of vehicle-track interaction mechanisms in order to ensure operational safety, reliability and long-term economic efficiency.

One of the most critical technical challenges in high-speed railway systems lies in degradation processes occurring at the wheel-rail interface. Under repeated rolling contact conditions, the interaction between the wheel and rail generates complex stress states, micro-slip and frictional energy dissipation within the contact patch. These phenomena lead to progressive wheel and rail degradation, primarily through wear and RCF. As these processes modify the wheel and rail profiles, they may increase dynamic forces, reduce running stability and significantly increase maintenance demand. Consequently, the accurate prediction of wheel-rail degradation mechanisms is essential for ensuring safe and efficient operation of high-speed railway systems (Du *et al.*, 2023; Piao, Zhang, Liang, & Zhao, 2013; Wu *et al.*, 2024; Zhang *et al.*, 2023).

In newly planned railway systems where long-term operational data are not yet available, simulation-based approaches provide an effective means of analysing degradation trends and supporting infrastructure design and maintenance planning. Multibody dynamics modelling has been widely applied to investigate wheel-rail interaction, allowing the dynamic behaviour of railway vehicles and the associated contact forces to be evaluated under realistic operating conditions (Consilvio, Febbraro, & Sacco, 2021; H-Nia, Flodin, Casanueva, Asplund, & Stichel, 2024; Li, Jin, Wen, Cui, & Zhang, 2011; Meghoe, Loendersloot, Bosman, & Tinga, 2018).

Extensive research has been conducted on wheel-rail wear and RCF over several decades. Numerous studies have shown that wear is primarily governed by micro-slip and the associated dissipation of frictional energy within the contact patch, which depend on parameters such as creepage, contact pressure and material properties (Akama, 2025; Archard, 1953; Ciotlaus, Kollo, Marusceac, & Orban, 2019; Krause & Poll, 1986; Lewis & Olofsson, 2009; Piotrowski & Chollet, 2005). In contrast, RCF occurs when repeated contact stresses exceed the elastic limit of the rail material, leading to the initiation and propagation of surface or subsurface cracks.

Recent investigations have increasingly focused on coupling vehicle dynamic models with wear and fatigue prediction approaches in order to simulate long-term wheel-rail degradation under realistic operating conditions. These studies have demonstrated that the integration of multibody dynamics simulations with wear evolution models can provide valuable insight into wheel profile evolution and its influence on vehicle dynamic performance (Jiang *et al.*, 2019; Luo, Shi, Teng, & Song, 2017; Pradhan, Samantaray, & Bhattacharyya, 2019; Yan, Jin, Ma, Zhang, & Zhou, 2023; Zhang & Qi, 2025; Yusuf, 2024).

Despite the substantial international research on wheel-rail wear and RCF, most existing studies have been developed and validated using operational data from mature high-speed railway systems in Europe and Asia (Pradhan *et al.*, 2019; Wu *et al.*, 2024; Zhang *et al.*, 2023; Yusuf, 2024). In contrast, the Vietnamese railway system is currently in a pre-operational stage with no available high-speed service data, which limits the applicability of conventional validation approaches based on field measurements.

This gap highlights the need for predictive modelling frameworks capable of supporting early-stage assessment of wheel-rail degradation under prospective operating conditions. While multibody dynamics and Archard-based wear modelling are well-established methods in railway engineering (Akama, 2025; Gao, Tao, Ren, & Wen, 2025; Wu *et al.*, 2024; Yan, Jin, Ma, Zhang, & Zhou, 2023), their application to data-scarce, emerging high-speed railway systems remains limited.

In this context, the contribution of this study lies in the systematic integration of multibody vehicle dynamics, wheel-rail contact mechanics and wear evolution modelling into a unified simulation framework tailored to the Vietnamese high-speed railway scenario. The framework enables the evaluation of wheel wear evolution, RCF risk and their interaction with vehicle dynamic behaviour in the absence of in-service data.

Specifically, the study provides: (1) a predictive simulation-based methodology for pre-operational assessment of wheel-rail degradation; (2) an application to representative high-speed conditions in Vietnam; and (3) a quantitative analysis of the relationship between wear evolution, contact mechanics and critical speed variation. These contributions aim to support early-stage design evaluation and predictive maintenance planning for emerging high-speed railway systems.

2. Theoretical framework and evaluation criteria for wheel-rail degradation

In this study, wheel-rail wear and RCF are treated as degradation mechanisms arising directly from the contact conditions between the wheel and the rail during operation. The theoretical framework is therefore formulated to establish a consistent link between the contact quantities obtained from multibody vehicle dynamics simulations and the evaluation indices used to characterise wear and RCF.

This section outlines the theoretical models and assessment criteria adopted as the basis for analysing and comparing different high-speed railway operating scenarios in the Vietnamese context.

2.1 Wheel-rail contact modelling for degradation assessment

To evaluate wear and RCF, the first step is to determine the contact state between the wheel and the rail, including the normal contact force and the various forms of micro-slip arising during rolling. In this study, the wheel-rail interaction is modelled as contact between 2 elastic bodies subjected to substantial normal loading, consistent with the typical rolling contact conditions encountered in railway wheel operation.

According to Hertzian contact theory, under conditions of linear elastic contact, the contact between the wheel and the rail assumes an approximately elliptical shape, as illustrated in [Figure 1](#), where the geometry of the contact patch and the corresponding stress distribution

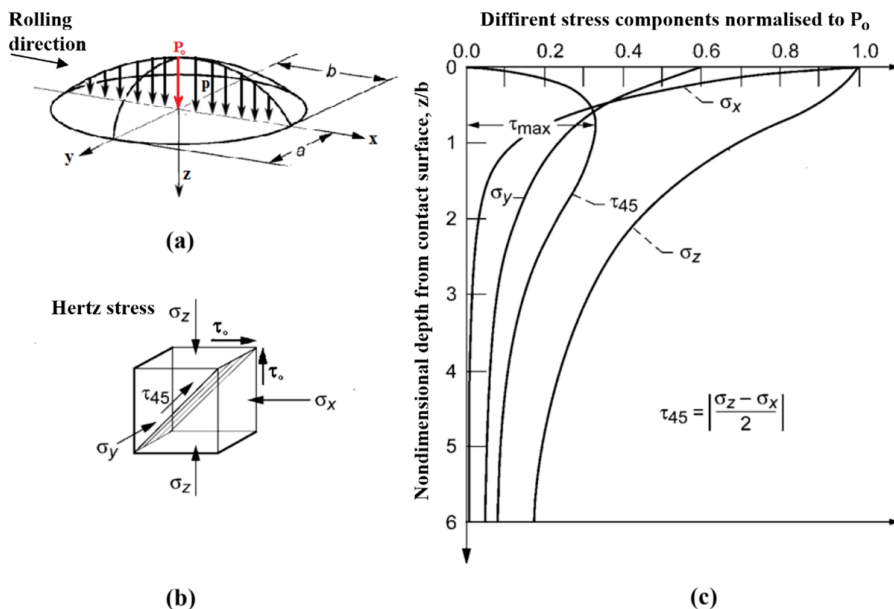


Figure 1. Schematic representation of the wheel-rail contacts and stress distribution according to Hertzian contact theory: (a) elliptical contact patch and local coordinate system; (b) stress state within the contact region; (c) distribution of stress components and shear stress with dimensionless depth. **Source(s):** Oswald, Zaretsky, and Poplawski (2011)

provide a fundamental basis for evaluating contact pressure, micro-slip behaviour and the resulting wear and RCF mechanisms. The contact pressure reaches its maximum at the centre of the contact patch (Hertz, 1882; Kalker & Johnson, 1993). The maximum contact pressure is determined as follows Equation (1):

$$p_{\max} = \frac{3F_n}{2\pi ab} \quad (1)$$

where F_n is the normal contact force, and a and b denote the semi-axes of the elliptical contact patch, determined by the contact geometry and the elastic properties of the wheel and rail materials.

In addition to the normal loading, the contact region is characterised by longitudinal, lateral and spin micro-slip components. These arise from differences in relative velocity between the wheel and the rail under dynamic loading, during curving, or when operating with superelevation deficiency. The resulting micro-slip generates tangential forces and plays a governing role in the initiation and progression of wear and RCF (Kalker & Johnson, 1993; VI-grade GmbH, 2025; Xu, Yan, & Sun, 2020).

The contact quantities defined in this section—namely the normal and tangential contact forces together with the micro-slip state—are subsequently employed as input parameters for the wear and RCF indices introduced in the following sections.

2.2 Wear evaluation criteria based on rolling contact

In this study, wheel-rail wear is evaluated based on the relationship between contact loading and micro-slip within the contact patch. A widely adopted and well-established approach in railway engineering is Archard's wear law, which provides a direct link between the contact state and the volume of material removed (Archard, 1953).

According to Archard's law, the wear volume is expressed as Equation (2):

$$\Delta V = k \frac{F_n \cdot s}{H} \quad (2)$$

where k is the wear coefficient, dependent on the material pair and contact conditions; F_n is the normal contact force; s denotes the sliding distance; and H represents the material hardness.

In numerical simulation studies, Archard's law is typically employed not for the absolute prediction of wear magnitude but rather for the formulation of relative wear indices that enable comparison between different operating conditions. Accordingly, the contact patch is discretised into a series of narrow strips, within which the local wear depth is determined from the corresponding local contact pressure and micro-slip distribution (Jiang *et al.*, 2019; Lewis & Olofsson, 2009; Pradhan, Samantaray, & Bhattacharyya, 2018; Wang *et al.*, 2022).

In this study, the wear index is employed to assess the influence of operating speed, track geometry and superelevation deficiency on wheel-rail degradation trends, rather than to provide an exact long-term prediction of the worn profile geometry.

The variation of the wear coefficient with sliding distance is illustrated in Figure 2. The figure shows that the instantaneous wear coefficient K_{act} evolves nonlinearly over the contact cycle, increasing from zero to a peak value before gradually decreasing, while the average wear coefficient K_{ave} represents the equivalent constant value over the same sliding distance. This variation reflects the transient nature of contact conditions, where changes in contact pressure, micro-slip and material response influence the wear process. The use of an averaged wear coefficient enables the practical implementation of Archard-based wear modelling while still accounting for the underlying non-uniform wear behaviour.

2.3 RCF risk index and wear-fatigue competition mechanism

RCF is a damage mechanism arising from repeated contact stresses at the wheel-rail interface that exceeds the elastic load-carrying capacity of the rail material. When this condition is met,

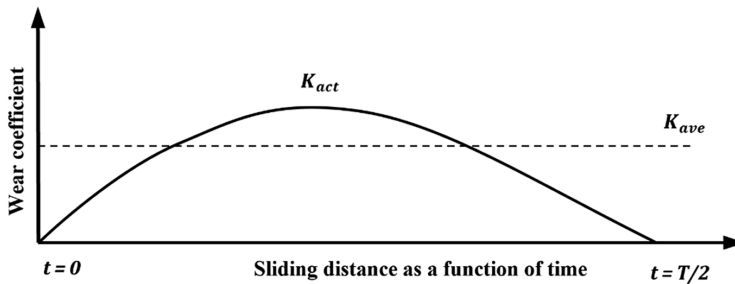


Figure 2. Variation of the wear coefficient with sliding distance and time. **Source(s):** Hanief and Charoo (2021)

accumulated plastic deformation leads to the initiation and subsequent propagation of surface cracks, thereby significantly impairing the service performance of the rail (Akama, 2025; Spangenberg, Fröhling, & Els, 2018; Szablewski & Jahagirdar, 2018; Tawfik *et al.*, 2023).

The mechanism of RCF crack initiation and propagation is illustrated in Figure 3. Under repeated wheel-rail contact, the combined effects of normal contact pressure and tangential forces generate subsurface shear stresses, leading to the initiation of micro-cracks beneath the contact surface. As the wheel continues to roll in the traffic direction, these cracks propagate at shallow angles towards the surface, eventually forming surface-breaking defects. This process reflects the cumulative effect of cyclic loading and highlights the critical role of contact stress distribution and rolling-sliding interaction in the development of RCF damage.

In this study, the risk of RCF is assessed using an index defined as the ratio between the maximum contact pressure and the yield strength of the rail material, expressed as Equation (3) (Spangenberg *et al.*, 2018; Tawfik *et al.*, 2023):

$$R_{mtl} = \frac{p_{max}}{\sigma_y} \quad (3)$$

where p_{max} is the maximum contact pressure defined in Equation (1), and σ_y denotes the yield strength of the rail material.

A higher value of R_{mtl} indicates an increased likelihood of RCF initiation. However, numerous studies have demonstrated that wear and RCF do not evolve independently; rather, they interact in a mutually influential manner. Moderate levels of wear may mitigate fatigue risk by removing surface-initiated cracks, whereas low wear combined with high contact stresses can promote the development and propagation of RCF damage (Akama, 2025; Hutchings & Shipway, 2017; Spangenberg *et al.*, 2018; Szablewski & Jahagirdar, 2018).

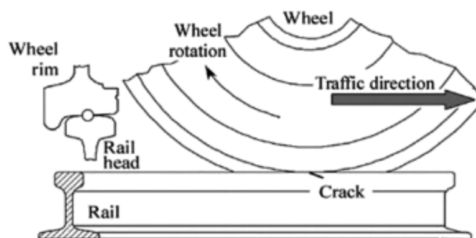


Figure 3. Mechanism of crack initiation and propagation due to RCF at the rail head under rolling-sliding wheel loading in the direction of train movement. **Source(s):** Tawfik *et al.* (2023)

Accordingly, wear and RCF are evaluated concurrently in this study in order to identify operating conditions that may pose a heightened risk of severe degradation at the wheel-rail interface, particularly under high-speed railway operation.

3. Model development

3.1 Scope and study scenario

This study considers a high-speed railway operating scenario in Vietnam with a nominal service speed of 350 km/h, selected to represent the upper-bound operating conditions envisaged in current design proposals. The adoption of a fixed operating speed enables the influence of track geometry and superelevation deficiency on wheel-rail contact conditions, wear and RCF to be isolated and examined in a controlled manner.

The principal modelling parameters are defined with reference to established international high-speed railway standards and previous research and subsequently adapted to reflect the anticipated Vietnamese operating context, as summarised in Table 1.

A curve radius of 5,000 m is adopted to represent typical high-speed railway alignment conditions, under which centrifugal forces and asymmetric load distribution between the inner and outer wheels become significant. Under such conditions, lateral contact forces and micro-slip at the wheel-rail interface—particularly on the outer rail—tend to increase, thereby promoting non-uniform wear development.

The selected curve radius of 5,000 m represents a typical design value for high-speed railway alignments operating at speeds of up to 350 km/h, as recommended in international design guidelines. The adoption of a single representative radius in this study is intended to provide a controlled reference scenario that enables the isolation of key wheel-rail interaction mechanisms. It is acknowledged that, in practical operation, a range of curve radii and operating speeds would be encountered. The extension of the present framework to multiple track geometries and variable operating conditions will be considered in future work to enhance applicability to the full Vietnamese railway network.

A superelevation deficiency of 100 mm is assumed to simulate practical operating conditions in which the train speed exceeds the geometrically balanced speed of the curve. Superelevation deficiency increases dynamic loading and lateral contact forces acting on the outer rail, directly influencing the micro-slip state, frictional energy dissipation and the rate of wheel-rail wear.

A track gauge of 1,435 mm is employed in accordance with the international standard gauge, ensuring compatibility with widely used wheel-rail profiles and material datasets reported in previous studies. The principal study parameters are summarised in Table 1 and are applied consistently throughout the simulation procedure to ensure coherence and comparability of the results (Lewis & Olofsson, 2009; Zhou, 2023).

3.2 Vehicle multibody dynamics model for wheel-rail interaction analysis

In this study, a multibody dynamics model is developed for a representative passenger vehicle of a high-speed railway system to investigate wheel-rail interaction, wear and RCF. The vehicle model comprises the carbody, 2 bogie frames and 4 wheelsets, which together constitute the primary components governing load transfer and wheel-rail contact conditions during operation.

Table 1. Study parameters

Parameters	Value
Operating speed [km/h]	350
Curve radius [m]	5,000
Superelevation deficiency [mm]	100
Track gauge [mm]	1,435
Source(s): Authors' own work	

These components are interconnected through primary and secondary suspension systems, allowing the relative motions between the wheelsets, bogies and carbody to be captured under the combined influence of dynamic loading and track geometry. The mass, inertia and suspension stiffness parameters are selected to reflect typical values for modern high-speed rolling stock, ensuring that the model provides a realistic representation of the dynamic behaviour of the wheel-rail system.

The adoption of a single-vehicle model, rather than a full trainset, is intended to isolate and clarify the influence of vehicle dynamic parameters and track geometry on wheel-rail contact conditions. In wear and RCF studies, the contact state at each wheelset is governed primarily by axle load, suspension characteristics and track conditions, whereas longitudinal interactions between adjacent vehicles are of secondary importance for local contact analysis. Consequently, a single-vehicle model is considered sufficient to evaluate wear and RCF trends while significantly reducing computational complexity and simulation time.

3.2.1 Dynamic simulation framework and wheel wear evolution prediction. To evaluate the evolution of wheel wear during service, a coupled simulation framework is established in which the vehicle multibody dynamics model is integrated with a wear prediction model. This approach enables the two-way interaction between the vehicle's dynamic response and the progressive geometric modification of the wheel profile to be captured over the simulated operating period. The overall structure of the simulation framework is illustrated in [Figure 4](#).

The vehicle dynamics model is implemented in VI-Rail within the ADAMS environment to simulate the dynamic response of the vehicle under the combined influence of track geometry, structural characteristics of the vehicle and wheel-rail contact conditions. The principal input parameters include track design characteristics, bogie and suspension properties, wheel and rail profiles, and the prescribed operating speed.

From the multibody simulation results, the global wheel-rail contact quantities are extracted through post-processing, including contact patch dimensions, contact forces at the wheel-rail interface, and the generalised longitudinal, lateral and spin creepages. These quantities serve as input data for the wear model, enabling the estimation of material removal at the wheel tread surface ([Gao et al., 2025](#); [Pradhan et al., 2019](#); [Xu et al., 2020](#)).

In this study, wheel wear evolution is predicted by combining Archard's wear law with a simplified rolling contact theory. Based on the contact state obtained from the dynamic simulation, incremental wear is calculated at each update step. After each computational cycle, the wheel profile is updated and smoothed to ensure geometric continuity before being reintroduced into the multibody dynamics model for the subsequent simulation step.

The entire wear computation and wheel profile updating procedure is implemented in the MATLAB environment, enabling iterative data exchange between the multibody dynamics

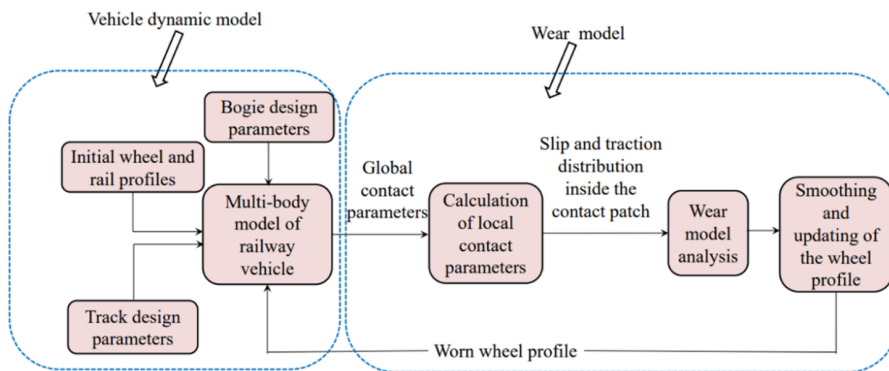


Figure 4. Schematic representation of the wear prediction model. **Source(s):** [Pradhan et al. \(2019\)](#)

model and the wear model. Through this iterative scheme, the influence of wheel wear on the vehicle's dynamic characteristics is assessed progressively over the assumed service period.

To ensure computational feasibility while preserving the essential physical characteristics of wheel-rail interaction, a number of modelling assumptions are introduced. These assumptions are consistent with widely adopted practices in multibody dynamics-based wear prediction studies and are justified as follows:

First, the wheel and rail materials are assumed to be identical and to behave as linear elastic bodies. This assumption is commonly adopted in wheel-rail contact modelling to enable the application of Hertzian contact theory and simplified rolling contact formulations, which provide sufficient accuracy for global contact analysis under typical operating conditions (Akama, 2025; Wu *et al.*, 2024).

Second, dry contact conditions with a constant friction coefficient are assumed. Although friction conditions in real railway systems may vary due to environmental factors, previous studies have shown that constant friction assumptions can capture the dominant trends in wear evolution and RCF risk for comparative and parametric analyses (Gao *et al.*, 2025; Yan *et al.*, 2023). The present study therefore focuses on relative degradation behaviour rather than absolute wear magnitude.

Third, rail wear is neglected, and only wheel wear evolution is considered. This assumption is justified by the significantly higher wear rate of wheels compared to rails due to repeated rolling cycles at the same contact location, as widely reported in the literature (Akama, 2025; Wu *et al.*, 2024). The simplification allows the study to focus on wheel profile evolution and its influence on vehicle dynamic behaviour.

Fourth, thermal effects associated with braking and frictional heating are neglected. Under high-speed operation with disc braking systems, heat generation is primarily localised at the brake disc rather than at the wheel-rail interface. Consequently, the thermal influence on wheel material properties is considered secondary for the mechanical wear mechanisms analysed in this study (Yan *et al.*, 2023).

Finally, the simulation is conducted using a representative operating condition with controlled parameters, including constant speed and predefined track geometry. This approach enables the isolation of key variables affecting wheel-rail contact behaviour and degradation mechanisms, which is appropriate for a preliminary assessment in a data-scarce environment. The limitations associated with these assumptions are acknowledged and discussed in the subsequent sections.

Overall, these assumptions provide a balance between model fidelity and computational efficiency, ensuring that the dominant physical mechanisms governing wheel-rail wear and rolling contact fatigue are captured while maintaining the feasibility of iterative simulation.

3.2.2 Vehicle and track modelling for dynamic and wheel-rail contact analysis. In this study, the vehicle multibody dynamics model is developed based on the reference model proposed by the European Rail Research Institute (ERRI), comprising a carbody and 2 bogie frames, and is employed as a benchmark configuration for wheel-rail interaction analysis. The model captures the essential dynamic characteristics required for the assessment of contact conditions, wear and RCF.

The simulation procedure follows a coupled sequential scheme. First, the vehicle dynamics model is used to determine the global contact parameters at the wheel-rail interface, including contact forces and generalised creepages. On this basis, local contact parameters—together with the distributions of creep and tangential traction within the contact patch—are evaluated for subsequent wear calculations (Bysani, Pålsson, & Kabo, 2025; Liu & Bruni, 2022; Piotrowski & Chollet, 2005; Pradhan *et al.*, 2019; Tang, Yuan, Xie, Jiang, & Zhang, 2019).

The wear model utilises these contact parameters to estimate material removal at the wheel surface. After each computational step, the wheel profile is updated and smoothed to ensure geometric continuity before being reintroduced into the dynamics model for the next simulation cycle. The outcome of this iterative process is an evolved wheel profile, which is then used to assess the influence of wear on the dynamic behaviour of the vehicle.

The bogie design parameters, initial wheel and rail profiles, and track geometry characteristics adopted in this study follow the recommendations of the International Union of Railways (UIC) and are implemented consistently within the VI-Rail simulation environment. Adherence to these standards ensures that the simulation results are representative and comparable with those reported in previous investigations (Pradhan *et al.*, 2019).

Within the multibody framework, the carbody is connected to the bogies via the secondary suspension system, while the wheelsets are linked to the bogie frames through the primary suspension. This configuration enables the model to capture the relative motions among the principal vehicle components under dynamic loading and varying track geometry.

The secondary suspension system comprises coil springs, non-linear dampers in the lateral and vertical directions, anti-yaw dampers and bump stops. This system plays a critical role in enhancing curving performance and maintaining passenger ride comfort.

The primary suspension includes coil springs, vertical dampers and axle box assemblies, responsible for transmitting loads from the carbody to the wheelsets while controlling relative oscillations between the wheelsets and the bogie frames. To ensure stable running on straight track, the suspension elastic elements are designed with stiffness components in both longitudinal and lateral directions. In addition, non-linear vertical dampers are incorporated to limit excessive relative displacements during operation.

In this study, the suspension stiffness coefficients are considered in the longitudinal, lateral and vertical directions, corresponding to the principal spatial motion components of the vehicle.

Other model components, including the wheelsets, bogie frames and carbody, are represented as rigid bodies with appropriate mass and inertia properties. The same vehicle model is employed consistently for both the dynamic simulations and the determination of global contact parameters at the wheel-rail interface.

The key parameters of the multibody vehicle model are summarised in Table 2, including the mass and inertia properties of the carbody, bogie frame, wheelsets and axle boxes, as well as the stiffness and damping characteristics of the primary and secondary suspension systems. These parameters define the dynamic behaviour of the vehicle, governing its response to track irregularities and wheel-rail contact forces. In particular, the higher stiffness of the primary suspension ensures effective wheelset guidance and stability, while the relatively lower stiffness of the secondary suspension enhances ride comfort and reduces the transmission of vibrations to the carbody. The damping elements further contribute to energy dissipation and vibration control, playing a critical role in stabilising the vehicle dynamics under high-speed operating conditions.

The stiffness values of the primary and secondary suspensions are selected to reflect their distinct functional roles, with the primary suspension providing higher stiffness for wheelset guidance and the secondary suspension offering lower stiffness to improve ride comfort and dynamic stability.

Transition curves are introduced between straight and circular track sections, as well as between curves of differing radii, to ensure a continuous variation of curvature and to minimise abrupt changes in lateral acceleration acting on the vehicle. Simultaneously, the cant (superelevation) is gradually varied along the transition, from zero at the end of the straight section to the design value in the circular curve, thereby promoting a smooth evolution of the vehicle's dynamic response during operation (Pradhan *et al.*, 2019; VI-grade GmbH, 2025).

The principal input parameters of the vehicle dynamics model include the UIC S1002 wheel profile, bogie structural characteristics, mass and inertia properties of the main components, and the prescribed operating speed profile. The suspension stiffness and damping parameters, together with the characteristic inertia properties of the bogie frame, are selected in accordance with the standard ERRI bogie data embedded within the VI-Rail simulation environment.

In addition to the vehicle model, a separate track model is constructed within ADAMS/VI-Rail to represent the influence of track geometry on vehicle dynamic behaviour. The simulated track comprises straight sections, transition curves and circular curves. In curved sections, the

Table 2. Mass, inertia and suspension parameters of the investigated vehicle model according to UIC standards and the VI-Rail reference configuration

Multibody model component	Parameter value (SI units)	Quantity
Carbody mass	$M_{\text{body}} = 32,000$	1
Carbody rotational moment of inertia	$I_{xx} = 5.68 \times 10^4$ $I_{yy} = 1.97 \times 10^6$ $I_{zz} = 1.97 \times 10^6$	
Bogie frame mass	$M_{\text{bogie}} = 2,615$	1
Bogie frame rotational moment of inertia	$I_{xx} = 1,722$ $I_{yy} = 1,476$ $I_{zz} = 3,067$	
Wheelset mass	$M_{\text{wset}} = 1,504$	2
Wheelset rotational moment of inertia	$I_{xx} = 810$ $I_{yy} = 810$ $I_{zz} = 112$	
Axle box mass	$M_{\text{axle}} = 150$	4
Axle box rotational moment of inertia	$I_{xx} = 2.1$ $I_{yy} = 5.6$ $I_{zz} = 5.6$	
Primary suspension stiffness	$K_x = 6.17 \times 10^5$ $K_y = 6.17 \times 10^5$ $K_z = 7.32 \times 10^5$	4
Secondary suspension stiffness	$K_\theta = K_\alpha = 63.5$ $K_x = 1.6 \times 10^5$ $K_y = 1.6 \times 10^5$ $K_z = 4.3 \times 10^5$	4
Primary suspension vertical damper	$K_\theta = K_\alpha = 183.3$ 1×10^6	4
Primary suspension vertical damper	6×10^6	2
Secondary suspension anti-yaw damper	3×10^7	2
Secondary suspension lateral damper	6×10^6	2
Source(s): Authors' own work		

outer rail is elevated relative to the inner rail to balance the centrifugal force generated during vehicle motion; the height difference between the 2 rails is referred to as superelevation (cant).

The geometric and structural parameters of the track typically include track gauge, curve radius, rail inclination, superelevation, transition length and track irregularities. However, within the scope of this study, the analysis is restricted to an elastic straight track to isolate the influence of track irregularities on dynamic response and wheel-rail contact behaviour.

The parameters characterising track elasticity, including stiffness, damping coefficients and inertia properties, are summarised in Table 3. For a straight track, wheel-rail contact occurs predominantly at the wheel tread, with an approximately elliptical contact patch consistent with elastic rolling contact assumptions.

Different types of straight-track irregularities are employed for distinct analytical purposes. For the determination of critical speed and ride comfort assessment, an elastic straight track with small-amplitude irregularities is considered, including short-wavelength gradients of limited amplitude and width, as well as measured track irregularity data (Kumar, Rastogi, & Pathak, 2012). In contrast, for wear prediction analysis, an elastic straight track with sinusoidal irregularities is adopted to generate a stable and repeatable excitation state at the wheel-rail interface, facilitating the evaluation of long-term wear trends (Pradhan *et al.*, 2019; VI-grade GmbH, 2025).

The primary contact patch forms at the wheel tread, where the normal load is predominantly transmitted. A secondary contact patch may arise when the wheel profile approaches closer to

Table 3. Elastic properties of the track (according to UIC standards and the VI-Rail reference dataset)

Mass properties					Elastic and damping element properties				
	Mass (kg)	I_{xx} (kg/m ²)	I_{yy} (kg/m ²)	I_{zz} (kg/m ²)			Lateral	Vertical	Roll
Ballast	500	100	100	100	Stiffness	Ballast	3.7×10^7	1×10^9	1×10^7
					(N/m)	Rail	2.4×10^5	1×10^6	1×10^4
Rail	60	10	10	10	Damping	Ballast	4.3×10^7	2.4×10^5	1×10^7
					(Ns/m)	Rail	2.4×10^5	5×10^7	1×10^4

Source(s): Authors' own work

the rail profile due to lateral displacement, reflecting a modification of the contact geometry under asymmetric operating conditions.

The simultaneous presence of multiple elliptical contact patches indicates that the wheel-rail interface may transition from single-point to multi-point contact when the wheelset undergoes lateral shift. This phenomenon alters the distribution of contact pressure and the associated micro-slip components, thereby directly influencing the mechanisms governing wear and RCF at the wheel and rail surfaces.

Figure 5 illustrates the elliptical contact patches formed between the wheel and the rail at the wheel tread when the wheelset is laterally displaced to the left relative to the track centreline. Under such lateral displacement, the wheel-rail contact position becomes asymmetric, leading to the simultaneous formation of 2 distinct contact patches.

In the vehicle dynamics simulations, the wheel and rail profiles are selected according to standard configurations in order to ensure representativeness and facilitate comparison with results reported in previous studies.

Specifically, the rail is modelled using the standard UIC 60 profile with a rail inclination of 1:40, while the wheel adopts the standard UIC S1002 profile with a nominal diameter of 920 mm. The track gauge in the model is set to 1,435 mm, corresponding to the international standard gauge. The adopted wheel and rail profiles are illustrated in Figure 6 below (Bizić, Petrović, & Pančić, 2015).

3.2.3 Simulation approach. Figure 4 presents the schematic workflow adopted to couple the multibody dynamics analysis with the wheel wear simulation. The computational procedure begins with the initial unworn wheel profile. During the dynamic simulation stage, the multibody model employs input parameters including track geometric and structural characteristics, bogie design parameters, wheel and rail profiles, and operational conditions such as vehicle speed and travelled distance. These data are defined in accordance with the specifications outlined in the preceding sections.

The results obtained from the dynamic simulation are subsequently processed to extract the characteristic wheel-rail contact quantities, which serve as general input parameters for the wear model. These include the contact force at the wheel-rail interface, the generalised creepages within the contact patch, the contact location along the wheel profile, the contact patch dimensions and the contact duration. The extracted data are then transferred to the wear model implemented in MATLAB for further computation. The wear evolution prediction

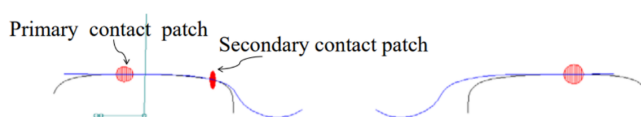


Figure 5. Elliptical contact patches at the wheel tread when the wheelset is laterally displaced to the left relative to the track centreline. **Source(s):** Authors' own work

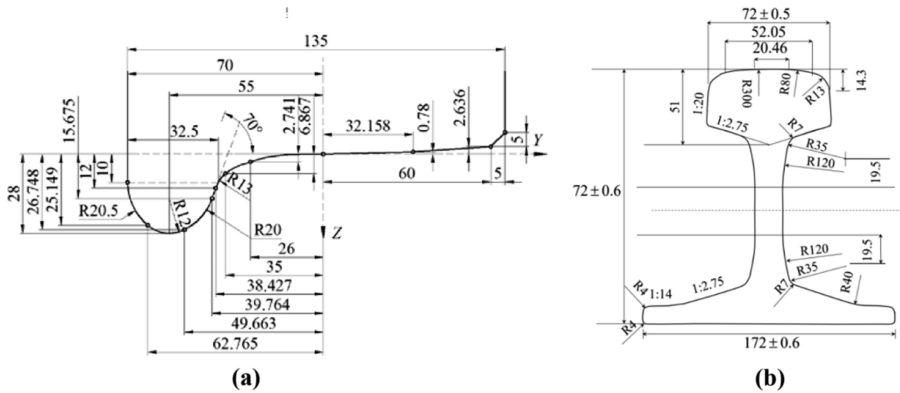


Figure 6. Geometric profiles adopted in the study model: (a) standard UIC – ERRI S1002 wheel profile; (b) standard UIC 60 rail profile. **Source(s):** Bizic *et al.* (2015)

framework is organised into 3 principal stages. First, local contact parameters are determined from the global contact quantities obtained through the multibody simulation. Second, the distribution and depth of wear on the wheel surface are calculated based on the prevailing contact conditions. Finally, the wheel profile is updated and smoothed to ensure geometric continuity before being used as input for the subsequent dynamic simulation cycle (Ignești, Innocenti, Marini, Meli, & Rindi, 2013a, b; Ignești, Malvezzi, Marini, Meli, & Rindi, 2012a; Ignești, Marini, Meli, & Rindi, 2012b).

In this study, the wear depth and distribution over the wheel surface are evaluated using Archard’s wear model. This approach enables the estimation of material removal at each discrete element of the initial wheel profile based on the wheel-rail contact state derived from the dynamic simulation. The wear prediction process is implemented as a simulation tool in which the wheel profile is discretised into small regions and the wear at each region is computed according to the corresponding load and slip conditions. This method allows the geometric evolution of the wheel profile to be simulated over the course of service. Various wheel wear models have been proposed in the literature, including twin-disc experimental models, Archard-based formulations and empirical approaches relating wear rate to material loss and rolling distance. Within the scope of this work, Archard’s model is selected owing to its direct linkage between contact parameters and wear progression, as well as its suitability for numerical implementation (Pombo *et al.*, 2011).

Previous studies have developed wear evolution models based on the USFD wear formulation, implemented in MATLAB in conjunction with multibody dynamics software such as SIMPACK, and validated against experimental data collected from the ALn 501 “Minuetto” train operating on the Aosta-Pre Saint Didier line (Auciello, Ignești, Malvezzi, Meli, & Rindi, 2012). In the present study, the wear distribution is estimated using Archard’s model to establish a direct connection between the contact parameters obtained from the dynamic simulation and material degradation at the wheel surface.

Wheel wear evolution is influenced by track geometry, the friction coefficient at the wheel-rail interface—where distinct friction levels may exist between the tread and flange regions—and lubrication conditions at the wheel flange and rail (Pombo *et al.*, 2010). Beyond simulation-based approaches, wear has also been assessed experimentally through relationships between the volume of removed material and the frictional power generated by tangential forces within the contact patch (Braghin, Lewis, Dwyer-Joyce, & Bruni, 2006; Ignești *et al.*, 2012a, b).

3.2.4 Contact model investigation. The tangential force distribution at the wheel-rail interface is assumed not to significantly influence the normal displacement of the contact

region. This assumption was originally proposed by Mindlin and subsequently demonstrated by De Pater to provide high accuracy when the 2 contacting bodies possess identical elastic constants (Kalker, Ingenieur, & Te's-Gravenhage, 1967). On this basis, the normal force is used to fully determine the contact pressure distribution, and the tangential force problem is solved using the results obtained from the normal contact solution.

Within the local contact model, the contact parameters at each point are derived from the global contact quantities obtained from the multibody dynamics simulation, with vehicle speed treated as a key input parameter. This approach has been implemented in various multibody simulation environments, including Bond Graph formulations (Banerjee & Karmakar, 2007; Banerjee, Saha, Karmakar, & Bhattacharyya, 2009; Banerjee & Karmakar, 2007), SIMPACK, ADAMS (VI-Rail) and VAMPIRE (Pombo *et al.*, 2011).

In previous studies, wheel-rail contact forces have been evaluated using different approaches, including approximate Hertzian and non-Hertzian solutions proposed by Kalker (Kalker, 1973, 1982; Kalker & Johnson, 1993) and by Kik and Piotrowski (Piotrowski & Kik, 2008), as well as exact solutions based on the elastic half-space assumption developed by Kalker (Kalker, 1973, 1982).

In the present study, the normal and tangential contact forces are determined using Kalker's simplified rolling contact theory (Kalker, 1982), assuming an elliptical contact patch. The contact region is described in a local coordinate system, and the relevant contact concepts are introduced in the following sections.

The computational procedure for wheel wear prediction is illustrated in Figure 7. The process begins with the vehicle dynamic model, which provides global contact parameters such as creepage, normal contact force and contact patch characteristics. These parameters are then used to derive local contact conditions through discretisation of the contact patch, enabling the evaluation of normal pressure distribution, tangential stresses and adhesion-slip behaviour. Based on these local interactions, the slip velocity and sliding distance are calculated for each discretised cell, forming the basis for wear depth estimation using the adopted wear model. The accumulated wear is then extrapolated over the simulated track length and scaled to represent long-term operation, followed by smoothing and updating of the wheel profile. This iterative procedure establishes a predictive framework for analysing the evolution of wheel wear under high-speed operating conditions.

3.2.5 Normal pressure distribution within the contact patch. Under elastic contact conditions between the wheel and the rail, the contact region assumes an elliptical shape and is described in a local coordinate system (x , y), where the x -axis is aligned with the rolling direction and the y -axis with the lateral direction. The normal pressure within the contact patch follows the Hertzian distribution (Hertz, 1882), attaining its maximum value at the centre of the contact area and decreasing smoothly to zero at the elliptical boundary.

The normal pressure at an arbitrary point (x , y) within the contact patch is given by Equation (4):

$$p(x, y) = p_{\max} \sqrt{1 - \left(\frac{x}{a}\right)^2 - \left(\frac{y}{b}\right)^2} \quad (4)$$

where $p_{\max}$ is the maximum contact pressure, a and b are the semi-axes of the elliptical contact patch in the rolling and lateral directions, respectively. The maximum pressure is related to the total normal force F_n through Equation (1).

An accurate determination of the normal pressure distribution and the corresponding local forces at each discrete element within the contact patch forms the basis for evaluating local contact parameters. These, in turn, enable the assessment of micro-slip, tangential stresses and the resulting wear distribution over the wheel surface in the subsequent stages of the simulation.

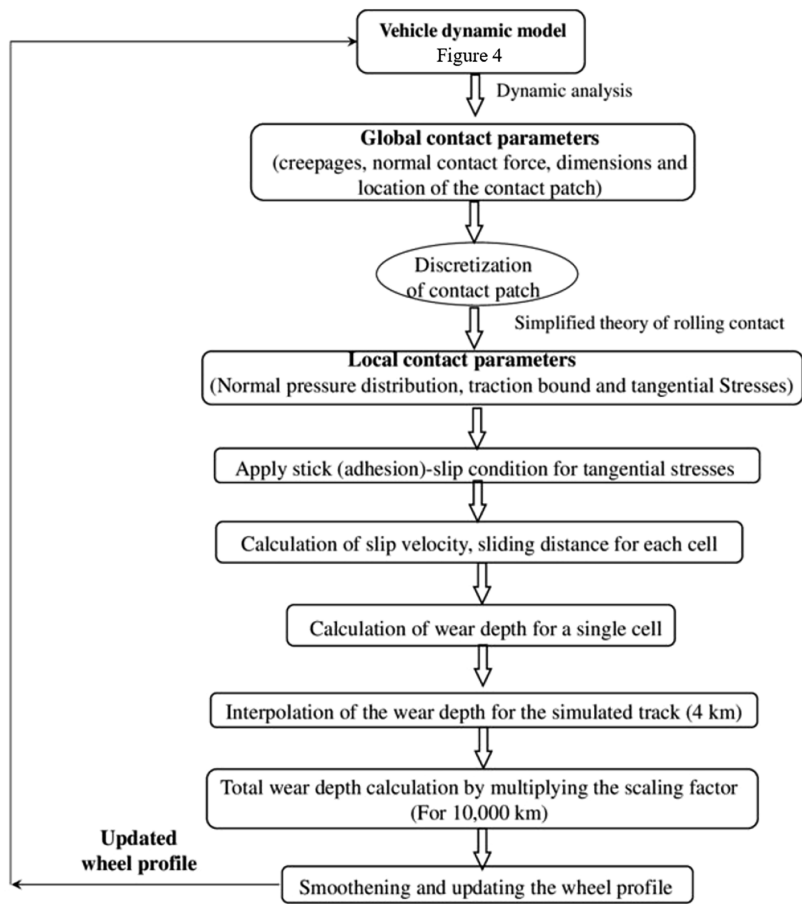


Figure 7. Detailed workflow of the wear model. Source(s): Pradhan *et al.* (2019)

Figure 8 illustrates the discretisation of the elliptical wheel-rail contact patch in the local coordinate system for the purpose of determining local contact quantities. The contact region is defined in the contact plane, with the x-axis aligned with the rolling direction and the y-axis in the lateral direction. It is then subdivided into parallel strips along the y-direction with a width of Δ_y . For each strip j , the geometric limits of the contact region in the x-direction are defined by the boundaries $x_l(y_j)$ and $x_r(y_j)$, thereby forming local contact elements with finite area. The resulting normal pressure distribution directly determines the local normal force through integration of the pressure over the area of each element and serves as the basis for solving the tangential contact problem. The total normal force acting at the wheel-rail interface is obtained by summing the local normal forces over all elements within the contact patch. Based on the established normal pressure distribution, the tangential forces and stresses at each element are evaluated according to stick-slip conditions, with the tangential force constrained by the adhesion limit governed by the local normal pressure and the friction coefficient. This tangential force distribution reflects the micro-slip state within the contact patch and plays a decisive role in the mechanisms of wear and RCF. Consequently, the discretisation of the contact region, as depicted in the figure, constitutes a key step in

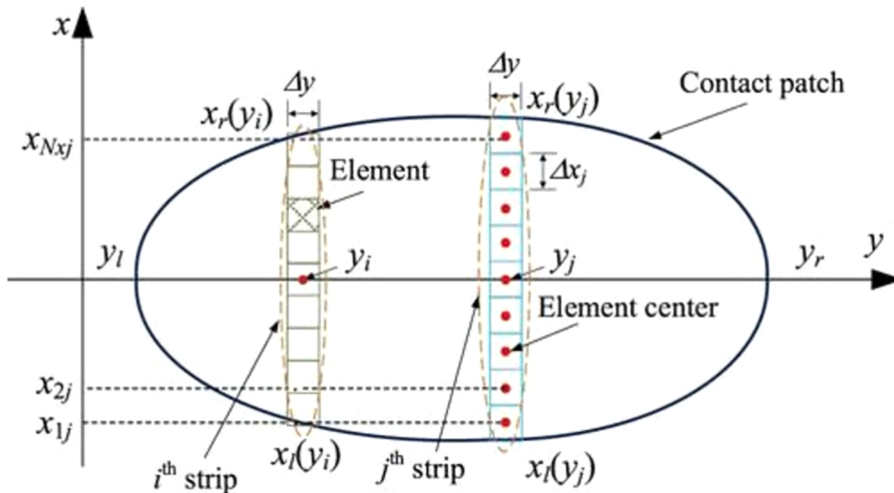


Figure 8. Discretisation of the elliptical wheel-rail contact patch into strips and local contact elements in the local coordinate system. **Source(s):** Sun and Ling (2022)

consistently and physically linking the multibody dynamics simulation with the wheel-rail wear model (Kalker, 1973, 1982; Kalker & Johnson, 1993; Sun & Ling, 2022).

3.2.6 Evolution of wear in wheel-rail contact. Numerous studies have shown that, despite differences in wear modelling approaches, wheel profiles tend to converge towards similar geometrical shapes after prolonged wear evolution. The wear distribution along the wheel profile is directly related to the frictional work generated within the wheel-rail contact patch, that is, the product of the tangential force and the associated micro-slip components (Fries & Davila, 1985; Jendel, 2002; Kalker, 1973, 1982).

In the present study, Archard's wear model is employed to describe material degradation at the wheel surface due to sliding. The model enables wear to be evaluated using a local approach, in which the location of the contact patch—whether on the tread or flange—is explicitly identified, allowing the corresponding material removal to be estimated at each position along the wheel profile.

The volume of material removed is determined according to Archard's law shown in Equation (2). In this study, the wear coefficient is selected on the basis of standard experimental data for wheel-rail steel pairs (Ignesti *et al.*, 2013a; Jendel, 2002), with a representative value of $k = 10^{-3}$. The hardness of the wheel and rail material is taken as 2,943 MPa, corresponding approximately to a Brinell hardness of 300.

Under the operating conditions considered, wear is assumed to arise from dry friction with a constant friction coefficient of 0.4. The distribution of normal load within the contact patch is obtained from the normal pressure solution. On this basis, the sliding velocity at each contact element is determined from the corresponding micro-slip components under steady rolling conditions, and the sliding distance of each element is calculated as the product of sliding velocity and the associated contact duration.

The relationship between contact pressure, slip velocity and wear coefficient is illustrated in Figure 9. The contact conditions are divided into distinct regions corresponding to tread and flange interactions, each characterised by different ranges of wear coefficients. In the central region associated with stable tread contact, moderate wear coefficients are observed, reflecting relatively uniform contact conditions. In contrast, the flange contact regions exhibit significantly higher wear coefficients due to increased slip velocity and elevated contact stresses. This regional variation highlights the non-uniform nature of wheel-rail interaction

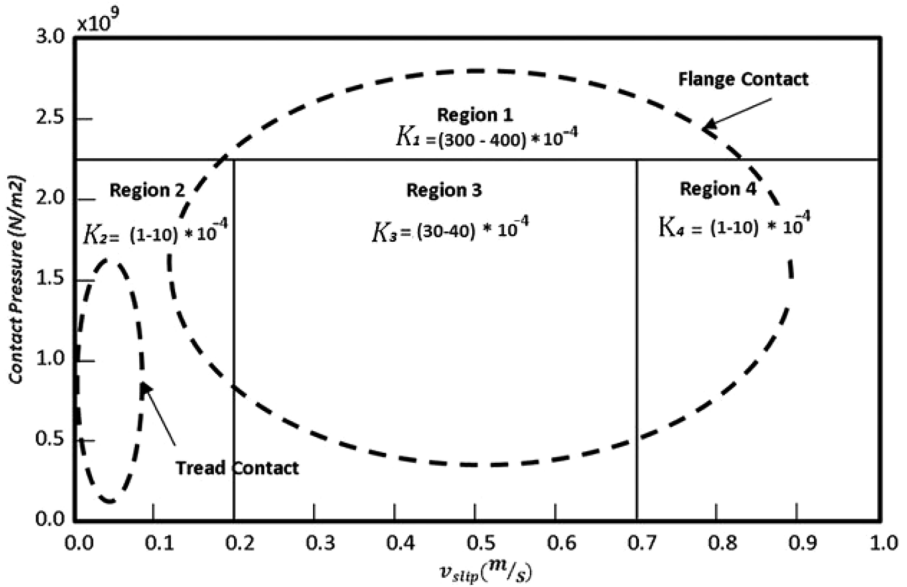


Figure 9. Archard wear maps for tread and flange contact. **Source(s):** Archard (1953)

and demonstrates that local contact conditions play a critical role in determining wear intensity. The classification of wear coefficient ranges in these regions provides a practical basis for implementing Archard-based wear modelling under varying contact scenarios.

The local wear depth at the centre of each contact element (x, y) is then evaluated according to Equation (5) (Jendel, 2002):

$$\Delta_{z(x,y)} = \frac{\mathbf{k}p_n(x,y)|s|}{\mathbf{H}} \quad (5)$$

where $p_n(x, y)$ denotes the local normal pressure and s the corresponding sliding distance of the contact element. The resulting local wear depth is distributed along the wheel circumference, and the total wear is obtained by summing the wear contributions of all elements located within the slip region of the contact patch.

This approach enables a consistent representation of the relationship between normal pressure distribution, tangential forces, micro-slip conditions and the progressive development of wheel wear during operation. It also provides the basis for updating the wheel profile in subsequent simulation cycles.

3.2.7 Profile smoothing and wheel geometry update. Once the local wear distribution within the wheel-rail contact patch has been determined, the wheel profile is updated to reflect material removal during operation and is subsequently used as input for the next dynamic simulation cycle. To achieve this, the local wear values are mapped from the contact patch onto the entire wheel profile through a curvilinear coordinate system.

Since the same point on the wheel comes into contact with the rail during each revolution, the wear rate of the wheel is typically significantly higher than that of the rail. The present study therefore focuses on describing and predicting the wear distribution along the wheel profile. The wear distribution in the rolling direction is obtained by integrating the local wear depth over the contact region, and the total wear is calculated by summing the contributions from all contact patches occurring during the dynamic simulation.

In practical operation, a considerable running distance is required for significant wheel wear to develop. Consequently, the simulation is conducted over a representative short track section, and the results are subsequently scaled to correspond to longer service distances. An adaptive update strategy is adopted, in which the length of each update step depends on the maximum accumulated wear obtained during a simulation cycle.

During the scaling procedure, numerical oscillations and non-physical spikes may arise in the wear distribution. To mitigate these effects, the wear profile is smoothed using piecewise polynomial interpolation combined with a Gaussian filter, ensuring geometric continuity and numerical stability of the updated wheel profile.

Finally, the new wheel profile is obtained by displacing the original profile along the surface normal direction, with a displacement magnitude corresponding to the smoothed wear depth. The updated profile is then reintroduced into subsequent simulation cycles, forming a closed-loop procedure for predicting wheel wear evolution under high-speed railway operating conditions (Braghin *et al.*, 2006; Ignesti *et al.*, 2013a, b; Innocenti, Marini, Meli, Pallini, & Rindi, 2014).

3.2.8 Influence of acceleration, steady running and braking regimes on wheel wear. In practical operation, railway wheels do not function solely under constant-speed conditions but are frequently subjected to phases of acceleration and braking. Accordingly, this study examines the influence of different motion regimes on wheel-rail contact conditions and the resulting wear distribution on the wheel.

Braking is assumed to be achieved by means of a disc braking system, in which frictional heat is generated primarily at the interface between the brake disc and brake pads. Owing to heat dissipation through conduction and convection, the temperature rise at the wheel surface—including both the tread and flange—is considered negligible in comparison with the braking surfaces. The material properties of the wheel are therefore assumed to remain unchanged during braking, allowing wear to be evaluated under isothermal conditions. On this basis, the analysis focuses on the influence of wheel-rail contact parameters during acceleration, steady running and braking phases.

The constant-speed segment of the velocity profile is intentionally defined over a limited duration to reduce computational cost. However, this segment can be extended to represent longer operating periods, as the wear evolution is assumed to scale proportionally with running distance under quasi-steady conditions.

The velocity profile adopted in the simulation is shown in Figure 10, representing a typical operational cycle consisting of acceleration, steady-state cruising and deceleration phases. This profile is designed to capture realistic variations in vehicle speed, which directly influence the dynamic response of the vehicle-track system and the resulting wheel-rail contact conditions. During the acceleration phase, increasing speed leads to progressive changes in creepage and contact forces, while the steady-state phase provides a stable

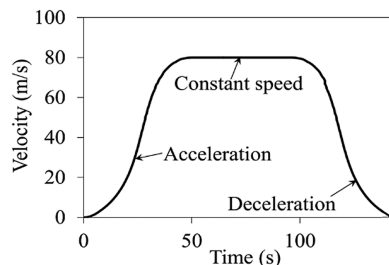


Figure 10. Velocity profile of the simulated vehicle capturing acceleration, steady-state cruising and deceleration phases to represent realistic high-speed railway operating conditions. **Source(s):** Pradhan *et al.* (2019)

condition for evaluating wear accumulation under constant operating parameters. In contrast, the deceleration phase introduces transient effects associated with braking forces, which can significantly alter slip conditions and increase local contact stresses. The inclusion of these distinct operating regimes ensures that the simulation captures both steady and transient behaviours, providing a more comprehensive assessment of wear evolution and RCF under realistic high-speed railway conditions.

In the multibody simulation, the data output interval is defined to correspond to 1 complete wheel revolution, ensuring that the same point on the wheel circumference returns to contact with the rail after each cycle. Owing to the rotational symmetry of the wheel, other circumferential points are assumed to experience equivalent wear conditions. Following each simulation run, the characteristic wheel-rail contact quantities—including contact location, normal force, contact patch dimensions and the longitudinal, lateral and spin creepages—are extracted and employed as input for the wear model.

The simulation results indicate that longitudinal creepage arises predominantly during the acceleration and braking phases, with magnitudes significantly greater than those observed during steady running, as shown in Figure 11. Moreover, the tangential forces and longitudinal creepage during acceleration and braking exhibit comparable absolute values. The clear correlation between longitudinal creepage and tangential force highlights the dominant role of transient operating regimes in governing wear mechanisms.

Owing to the asymmetric distribution of carbody mass and underfloor equipment, the axle loads are not perfectly uniform. In addition, because the wheel tread possesses a conical profile and the rail is installed with an inclination, the resultant lateral force at the wheel-rail interface is largely balanced by friction within the adhesion region. This leads to the presence of a baseline lateral creepage component that persists under all operating regimes, including acceleration, steady running and braking. Variations in lateral creepage about this baseline depend on the lateral slip state of the wheelset.

To obtain wear levels of practical significance, a wheel must accumulate a substantial running distance, typically on the order of tens of thousands of kilometres. It is therefore assumed that, within each extended operating interval (for example, 10,000 km), variations in vehicle dynamic behaviour remain negligible. On this basis, the contact parameters, creepages and tangential forces are treated as quasi-constant over each interval, allowing short-term

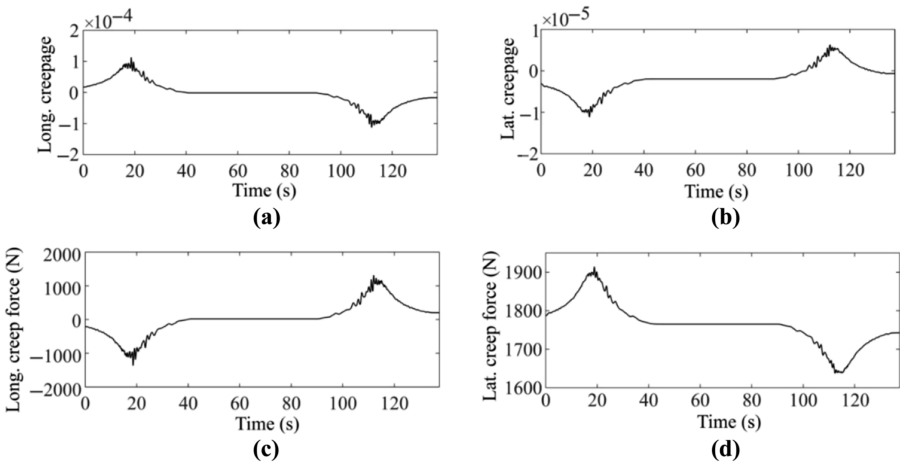


Figure 11. Global contact quantities: (a) longitudinal creepage, (b) lateral creepage, (c) longitudinal creep force, and (d) lateral creep force of the wheel obtained from the multibody dynamics simulation. **Source(s):** Pradhan *et al.* (2019)

4. Simulation results and discussion

Before presenting the detailed simulation results, the validity of the proposed modelling framework is assessed through comparison with established findings reported in the literature.

First, the predicted wear distribution along the wheel tread is consistent with previously reported results, which indicate that wear is primarily concentrated in the nominal contact region and gradually decreases towards the flange and field side (Pradhan *et al.*, 2019; Yan *et al.*, 2023). The present simulation reproduces this characteristic distribution pattern, as illustrated in Figure 12, confirming the ability of the model to capture the dominant wear mechanisms.

Second, the evolution of effective conicity and its influence on vehicle dynamic stability follow trends reported in earlier studies. Specifically, previous research has shown that moderate wear may initially improve stability by smoothing contact geometry, while excessive wear leads to a reduction in critical speed and increased risk of hunting instability (Gao *et al.*, 2025; Wu *et al.*, 2024). The non-monotonic variation of critical speed observed in Figure 13 is in agreement with these findings.

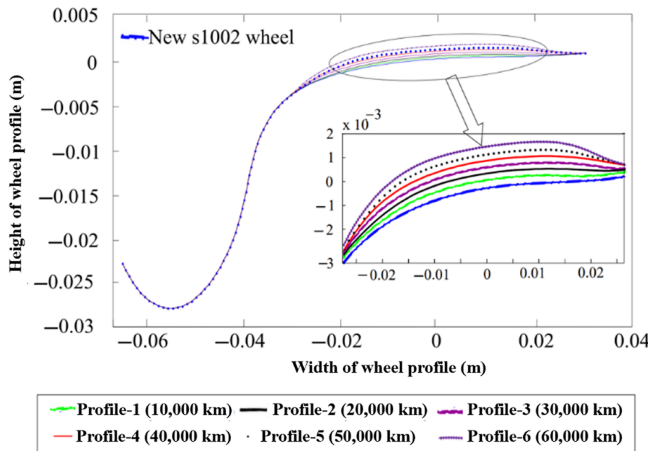


Figure 12. Evolution of the wheel profile with accumulated running distance under multibody dynamics simulation conditions. **Source(s):** Pradhan *et al.* (2019)

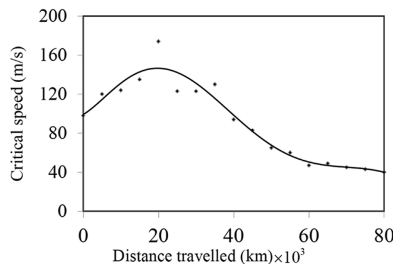


Figure 13. Variation of critical speed along the straight flexible track **Source(s):** Authors' own work

Third, the increase in creepage and tangential forces under track irregularities is consistent with established multibody simulation results. Similar magnitudes and trends of creepage growth have been reported in studies considering track excitation effects (Wu *et al.*, 2024; Yan *et al.*, 2023), supporting the validity of the dynamic response predicted by the present model.

From a quantitative perspective, the order of magnitude of the simulated creepage and contact forces remains within the typical ranges reported in the literature for high-speed railway applications, indicating that the model produces physically realistic results.

Although direct validation against field measurements is not feasible due to the absence of operational high-speed railway data in Vietnam, the agreement between the present results and established trends reported in the literature provides confidence in the reliability of the proposed modelling framework for preliminary assessment purposes.

The multibody dynamics simulations in this study are carried out using the VI-Rail software, employing the standard ERRI passenger bogie model in conjunction with a new wheel profile. The track system is represented as an elastic straight track, incorporating vertical geometric irregularities to reflect realistic operating conditions. The vehicle is assumed to run at constant speed for the majority of the simulated distance, while acceleration and braking phases are included within a representative operating cycle in order to assess the influence of motion regimes on wheel-rail contact characteristics and wear development.

Under the influence of track irregularities, the creepage magnitude increases by approximately one order of magnitude compared to ideal track conditions, reaching values on the order of 10^{-3} . Correspondingly, the tangential contact forces increase significantly, leading to an estimated increase in wear rate of approximately 20–30% under irregular track conditions (Figure 11). A comparison between elastic and rigid track models shows only limited differences in the global contact quantities. However, given the actual mechanical behaviour of railway track structures, the elastic model is considered more appropriate for wear prediction.

Within the scope of this study, thermal effects arising from frictional heating during braking are not taken into account. An isothermal assumption is adopted to focus on the role of mechanical contact parameters, including normal force, tangential force and micro-slip components. After each dynamic simulation cycle, the contact quantities are extracted and supplied as input to the wear model, enabling the evolution of the wheel profile to be determined as a function of running distance.

The simulation results indicate that wear is predominantly concentrated within the nominal contact region on the tread surface, with the maximum wear depth reaching approximately 0.2–0.4 mm depending on the simulation conditions. The wear depth decreases progressively towards both the flange root and the field side, with reductions of approximately 40–60% relative to the peak value. This distribution is consistent with the non-uniform contact pressure and micro-slip conditions within the contact patch (Figure 12). During numerical processing, smoothing techniques are applied to eliminate non-physical numerical noise, thereby ensuring the stability of the global contact model.

In addition to wear evolution, the results also provide insight into the potential risk of RCF. The increase in contact pressure and tangential forces under irregular track conditions leads to higher stress levels within the contact patch, which may approach or exceed the material yield strength. This indicates an elevated risk of crack initiation under repeated loading cycles.

Furthermore, the interaction between wear and RCF is evident. Moderate wear may reduce RCF risk by removing surface-initiated cracks, whereas excessive wear can alter contact geometry and increase stress concentration in localised regions. This wear-fatigue competition mechanism highlights the importance of evaluating both degradation modes concurrently in high-speed railway applications.

The critical speed of the vehicle is determined through simulation on a short elastic straight track segment, where small lateral perturbations are introduced to excite hunting oscillations. The critical speed is identified at the onset of sustained lateral hunting motion. The results show that the critical speed varies non-monotonically with the degree of wheel wear

(Figure 13). At the initial stage of wear, the critical speed increases slightly, by approximately 3–5%, which can be attributed to the smoothing of the contact geometry. However, as wear progresses further, the critical speed decreases by approximately 10–15%, indicating a reduction in dynamic stability and an increased susceptibility to hunting instability.

Vertical track geometric irregularities, as illustrated in Figure 14, generate continuous dynamic excitations acting on the wheel-rail system during vehicle motion. These excitations induce relative oscillations between the wheel and the rail, leading to time-varying normal forces, tangential forces and micro-slip components. As a consequence, the wheel wear distribution becomes laterally non-uniform, while the dynamic stability conditions are altered, as evidenced by the variation in critical speed at different stages of wheel wear.

In addition, the simulation results indicate that during the acceleration and braking phases, longitudinal creepage and longitudinal creep force increase markedly compared with steady constant-speed operation (Figure 11). In the lateral direction, a baseline creep force component persists throughout the operating cycle, arising from the combined effect of wheel conicity and rail inclination. The non-uniform distribution of axle loads, as well as left-right wheel load imbalance, further contributes to the asymmetry of the contact quantities.

Taken together, the results demonstrate that wheel wear arises from a complex interaction between vehicle dynamic conditions, contact geometry characteristics, track structural properties and operating regimes, with measurable variations in wear depth (up to approximately 0.4 mm), creepage magnitude (on the order of 10^{-3}) and critical speed (variation of approximately –10% to +5%).

Overall, the quantitative results demonstrate that wheel wear and dynamic stability are strongly coupled, with measurable variations in wear depth, creepage magnitude and critical speed that can be directly linked to changes in operating conditions and wheel profile evolution.

The simulation conditions adopted in this study are intentionally simplified to enable a controlled investigation of fundamental wheel-rail interaction mechanisms. In practical high-speed railway operation, vehicles are subjected to a wide range of speeds, track geometries and loading conditions. The extension of the present model to multi-scenario simulations, including varying speeds and curve radii, constitutes an important direction for future research.

5. Conclusions and recommendations

This study has developed a multibody dynamics-based computational framework to investigate progressive wheel wear and RCF risk under prospective high-speed railway operating conditions in Vietnam, with a nominal service speed of 350 km/h. In the absence of in-service high-speed operational data, the proposed simulation-based methodology provides

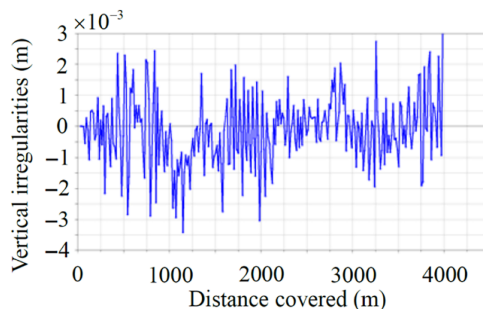


Figure 14. Measured tyre irregularities (from ADAMS VI-Rail) **Source(s):** Authors' own work

a physically consistent approach for linking vehicle dynamics, wheel-rail contact mechanics and degradation indicators within a unified modelling structure.

The adopted framework integrates an ERRI-based vehicle model implemented in VI-Rail with a local contact formulation derived from simplified rolling contact theory and an iterative Archard-based wear evolution model. Through closed-loop updating of the wheel profile, the approach enables the progressive modification of wheel geometry and its feedback effect on dynamic stability to be quantified. The results indicate that progressive wheel wear leads to measurable geometric changes in the tread profile, particularly a gradual reduction in effective conicity with increasing running distance. These geometric modifications influence operational safety margins in a non-monotonic manner. While limited initial wear may slightly increase the critical speed, continued profile degradation results in a reduction in critical speed and a corresponding deterioration of dynamic stability. Track vertical irregularities significantly amplify micro-slip and tangential forces, accelerating wear development, whereas acceleration and braking phases produce markedly higher longitudinal creepage than steady running, highlighting the importance of operational regimes in degradation accumulation.

Within the investigated wear range, ride comfort indices remain within acceptable limits. However, stability-related indicators exhibit greater sensitivity to profile evolution. The findings therefore underline the necessity of periodic wheel profile monitoring and the concurrent evaluation of stability criteria in high-speed railway operation.

The present study is based on deterministic numerical modelling under idealised contact and environmental assumptions. Future research should incorporate friction variability, thermo-mechanical effects and probabilistic parameter sensitivity analysis, and extend the framework to coupled wheel-rail profile evolution. The integration of simulation results with future field measurements will be essential for enhancing predictive reliability and for supporting safety-orientated maintenance strategies in the development of high-speed railway systems in Vietnam.

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Further reading

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