

Research Article

# Analysis of Railway Vehicle Wheel Wear in Relation to Dynamic Behaviour on Curved Tracks

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## Abstract

Wheel wear in railway vehicles is a critical factor affecting operational safety, maintenance costs, and ride comfort. Curved tracks impose complex dynamic interactions between wheels and rails, leading to accelerated wear due to increased lateral forces, slip, and contact stresses. This study examines the influence of wheel wear due to yaw and track irregularity on vehicle dynamic behavior demonstrated in terms of wear depth, derailment coefficient and ride index. Software simulation-based and wear data validation was used. The research utilized an integrated approach combining computational modeling and experimental wear measurements for comprehensive analysis. The wheel-rail contact interaction was modeled using Hertzian contact theory, while multibody dynamics and wear depth calculations were performed using SIMPACK and a custom MATLAB implementation of the Archard wear model. Key parameters examined included curve radius, operating speed, wheelset yaw, and track irregularities, with their effects quantified in terms of wear depth and dynamic performance metrics such as derailment coefficient and ride index. A re-profiling analysis conducted up to 60, 000 km, with wear depth measurements extracted at 10, 000 km intervals. The simulated wear depths closely matched the collected experimental data. Additional case studies revealed that curves with a 50-meter radius produced the most severe wear (6.18 mm), along with an elevated derailment coefficient (1.01) and poor ride comfort - even with the presence of yaw dampers or track irregularities. However, the track irregularities alone had only a minor impact on the derailment coefficient and ride index, their combination with yaw motion significantly worsened both metrics. Consequently, proactive measures should be implemented to mitigate the compounded effects of yaw and track irregularities.

## Keywords

Multibody Dynamic System, Archard Wear, Wear Depth, Curved Track

## 1. Introduction

Railway wheels are subject to continuous wear due to rolling contact fatigue (RCF), sliding, and abrasive forces, particularly on curved tracks where lateral dynamics dominate. The interaction between wheel and rail on curves leads to non-uniform contact patches, increased creep forces, and

material loss, affecting vehicle stability and track longevity. Understanding the dynamic behavior of railway vehicles on curved tracks is essential for predicting and mitigating wheel wear [1-3].

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## 1.1. Background

Wear is one of the most prevalent modes of material damage in machinery parts. According to [4] material loss connected with friction and wear in machines is estimated to take 3-5% of the gross national product. Also, about a quarter of the energy input in the industry is spent on overcoming friction, having direct relation to wear in most of the materials. Wear mechanisms can generally be Abrasive, Adhesive, surface fatigue, and tribo-chemical wear depending on the cause and means of wear propagation. It is also classified as being mild or severe [5].

In the railway Industry, wear is a very crucial subject because of a metal running on another metal scenario which poses excessive material loss, noise, and discomfort due to this effect. The term 'wheel wear' is generally used to denote any damage occurring on the rolling surface of railway wheels, which involves material loss from the surface. This can be viewed as thread wear or flange wear based on the location of the wearing surface and can be caused by the effect of adhesive and/or abrasive wear, rolling contact fatigue (RCF), and sometimes due to material relocation due to plastic deformation [5]. Material is usually removed from the existing profile of the wheel, mostly from the flange contact point due to cornering and hunting effects giving rise to a new wheel profile with reduced flange thickness (15 mm min. limit for AALRT Case) as shown in Figure 1. 5. Since the new profile is undesirable and causes unwanted dynamic effects, wheels are re-profiled after inspections are performed until their dimensional constraints set by the manufacturer (580 mm min. limit for the AALRT case) are reached so that there is no further re-profiling recommended unless the wheel is disposed [6].

## 1.2. Wear and Wear Mechanisms

Generally, wear refers to the loss of material from an object's surface caused by mechanical, thermal, or chemical factors. Mechanical wear occurs primarily through friction between two or more surfaces, where the harder material leaves an imprint on the softer one or where material transfer happens, with one surface removing particles from the other. The specific wear mechanisms depend on the type of motion between the interacting surfaces, known as tribo-pairs [7]. These include:

**Abrasive wear:** In this wear mechanism, a substantial difference in hardness between the contacting materials or the presence of a third body between them leads to material loss. As shown in Figure 1, it occurs when fragments break off from the softer surface under load and motion [7].

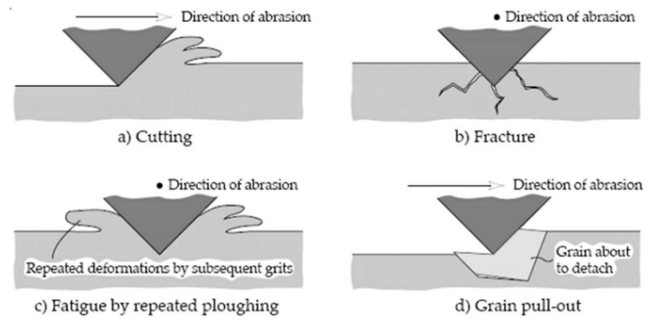


Figure 1. Abrasive wear modes [7].

**Adhesive wear:** This type of wear happens when two sliding surfaces form bonds between their outer layers. Under applied load, the contacting asperities (microscopic high points) fracture and detach from one surface, transferring to the other, resulting in material loss [7]. Figure 2 shows the adhesive wear.



Figure 2. Adhesive wear [7].

**Fatigue wear:** It called surface fatigue, occurs due to repeated cyclic loading, which initiates subsurface fatigue cracks in one or both contacting surfaces. See Figure 3 below. As these cracks grow, they eventually lead to material detachment. Additionally, the subsurface layers may undergo plastic deformation if the material's fatigue strength is insufficient to withstand the applied stresses [7].



Figure 3. Fatigue wear [7].

**Corrosive wear:** In this mechanism, material loss begins with a chemical reaction (corrosion) between the surfaces. The subsequent rubbing action then dislodges these corroded particles, which further abrade and damage the surface [7]. Corrosive wear is shown in Figure 4 below.

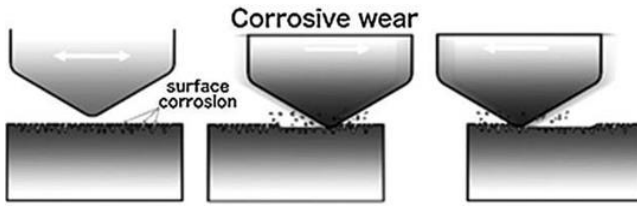


Figure 4. Corrosive wear [7].

### 1.3. Contact Mechanics Analysis

In real-world applications, wear mechanisms rarely act in isolation; instead, they interact synergistically to cause substantial material degradation. To determine the dominant wear mechanisms affecting a surface, examining the motion type—such as sliding, rolling, impact, or oscillation—can provide valuable insights. However, the most effective approach involves a thorough analysis of the worn surface to identify the specific wear processes at play. Additionally, each wear mechanism has tailored mitigation strategies to

minimize its impact and prevent severe damage to the interacting surfaces.

Adhesion plays a pivotal role in ensuring the safe, efficient, and reliable operation of trains within a railway network. Maintaining an optimal adhesive force is essential for acceleration, braking, and stopping, making it a key focus in railway engineering. The wheel-rail contact is particularly critical, as it governs the tractive and braking forces that can be applied while minimizing slip [8]. Figure 5 shows the description wheel/rail contact surfaces. Given that both the train wheels and rails are metallic, their actual contact area known as the contact patch is extremely small relative to their surface area. This results in exceptionally high stresses at the interface. Furthermore, unlike vehicles with steering mechanisms, trains navigate curves by sliding the wheels along the rails, which further amplifies interfacial stresses. The primary objective of this analysis is to evaluate the stress distribution, deformation, and geometry of the contact patch under these dynamic operating conditions.

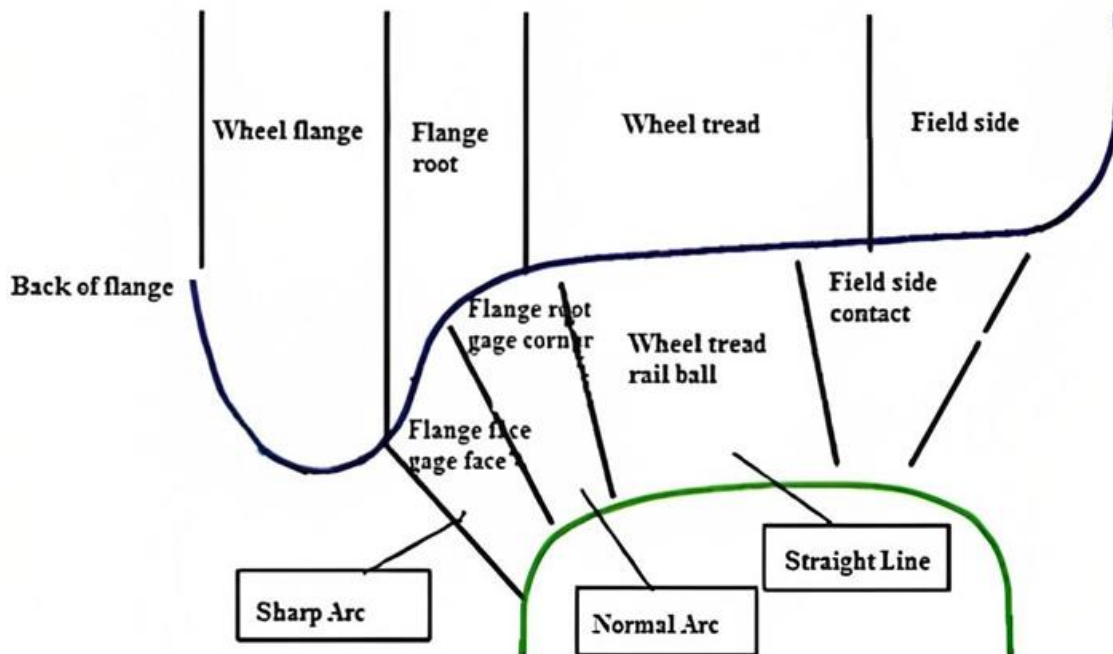


Figure 5. Contact surfaces description [9].

In the following contact mechanics analysis's,  $R_w$  and  $R_r$  are the radius of curvature of wheel tread and rail head at the contact point,  $r$  is radius of wheel,  $N$  is the normal force acting on the contact area,  $a$  and  $b$  are the major and minor axis semi-lengths given by [10] and shown in Eq 1 and Eq 2 as follows:

$$\left(\frac{a}{m}\right)^3 = \left(\frac{b}{n}\right)^3 = \frac{3N(1-\nu^2)}{E\left(\frac{1}{r} + \frac{1}{R_r} - \frac{1}{R_w}\right)} \quad (1)$$

$$\cos \beta = \frac{\left(\frac{1}{r} + \frac{1}{R_r} - \frac{1}{R_w}\right)}{\left(\frac{1}{r} - \frac{1}{R_r} + \frac{1}{R_w}\right)} \quad (2)$$

Where:  $E$  = Youngs modulus  $\nu$  = Poisson Ratio,  $m$  and  $n$  = hertzian parameters tabulated in terms of hertzian contact constant beta.

In general, based on summarized viewpoints of various researchers, even though most of the related published research papers consider many factors influencing wheel wear and vehicle dynamics and were reasoned out well, yaw and

track irregularities were considered less often, except Ye and Zhu [11-13]. Yet, even though the previous considered the effect of yawing on the wear rate based on a field test-based validation by continuous measurement of wheel profile, they were both concerned with high-speed trains rather than low or medium- speed [14-22].

## 2. Materials and Methods

This phase focuses on gathering essential input data for subsequent analysis phases. Primary sources include relevant technical literature and operational data obtained from rolling stock depots.

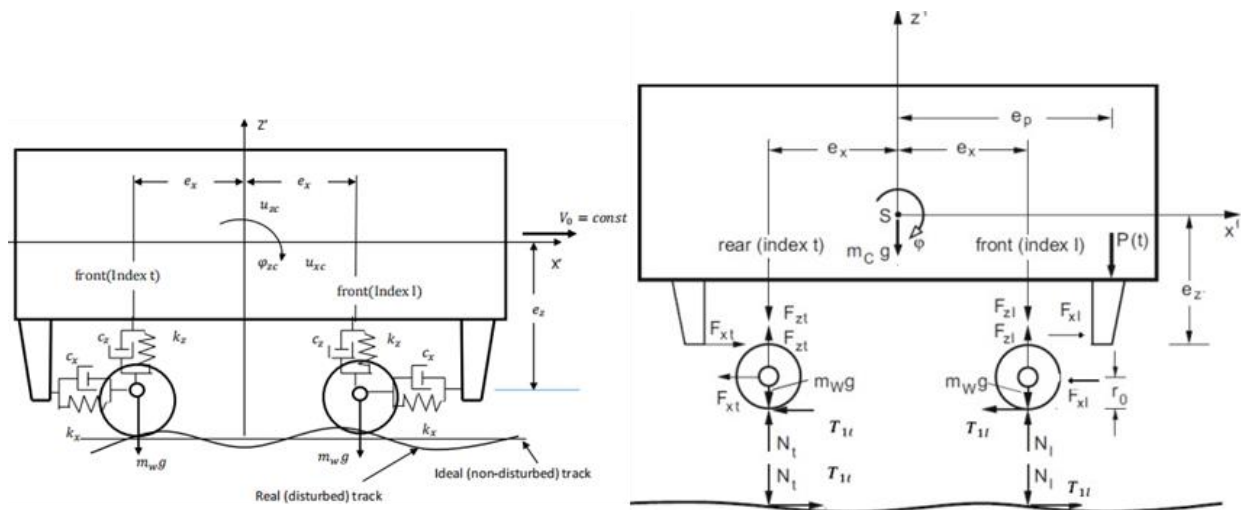


Figure 6. Free body model of Train [13].

### 2.1. Modelling Multibody Dynamics System

The foundational step for any software-based dynamic and wear evaluation of multibody systems involves creating a digital model within multibody simulation (MBS) software. For this study, we utilize SIMPACK, which offers two distinct modeling approaches. In the vertical dynamic analysis, the important factors to consider are suspensions (primary and secondary), masses, springs, and dampers, irregularities of rails and external forces applied on them. First the system is

modelled as a spring, mass and damper system as shown in Figure 6 below.

### 2.2. Vehicle and Track Parameters

The development of an accurate multibody dynamic model in SIMPACK requires careful consideration of both vehicle and track parameters. Vehicle-specific data has been primarily obtained through secondary research sources and direct measurements of primary components. Table 1 shows the parameters of track.

Table 1. Track model parameters [23].

| No | Track Parameter                    | Unit | Value             |
|----|------------------------------------|------|-------------------|
| 1  | Track Profile                      | -    | UIC 60            |
| 2  | Track Gauge                        | m    | 1.435             |
| 3  | Track Length                       | m    | 1000              |
| 4  | Track Super elevation              | m    | 0.1               |
| 5  | Minimum radius of horizontal curve | m    | 50, 100, 150, 190 |
| 6  | Operating Speed (Normal/Maximum)   | Km/h | 18/36/72          |
| 7  | Friction Coefficient               | -    | 0.4               |

### 2.3. Track Condition

The track was modeled in a way to contain the short critical curves of the North to South line of AALRT. The minimum value, i. e., 50, 100, 150, and 190 meters of curve radiuses with 100 m long approach curves of 1200 m radius for transition purposes in between and a total length of 1 km, has been used for analysis. The curve length has been analyzed based on Hallade method found from Eq 3 the Indian railway manual guideline as follows in order to limit the running simulation path at 1 km for all radiuses.

$$R = \frac{125C^2}{V} \quad (3)$$

Where: R = Radius, C = Chord length, and V = Versine

Track irregularities for the curved track have also been imported to the excitation part of the track layout and used during the analysis with a fade in and out the length of 10 m. Maximum superelevation of 0.1 meters was also applied to one of the track sides on each curve based on SIMPACK recommendation on the curving direction (right or left) to aid the centrifugal effect of the loaded car bodies during the simulation to prevent early derailment of the vehicle's wheelsets.

## 3. Results and Discussion

The simulation followed the prescribed methodology through successive iterations, continuing until reaching the specified annual operational distance of 60, 000 kilometers.

For every 10, 000 km analysis, i. e. 1 km analysis with 10, 000 distance factors, the lateral and longitudinal creepage values, contact patch area, wheel contact position, and worn wheel profile were extracted from the SIMPACK post-processor file and imported to the MATLAB, each value in. txt format. Once the resulting profile has been generated from, the value is plotted and is imported to the SIMPACK wheel profile. prw file format so that it can be smoothened.

### 3.1. 50 m Curve Run

Considering a 50 m radius run (the actual minimum radius of curve at AALRT) simulation of the AALRT model for a track length of 1 km and distance factor of 10, 000, the maximum wear depth for the right wheel was found to be close to 0.952 mm on the case of yaw and track irregularity (+Y+Tr). For the left wheel, on the other hand, this value was found to be 6.18 mm which is a very high wear rate since the radius of curvature was too tight at the given curving speed of 36 km/hr. As a result shown in Figure 7, the wheel is highly wearing on the left side, but the presence of the yaw damper has not improved much of the wear depth rather increased by 1.8 and 0.88% compared to without yaw (-Y+Tr) and no track irregularity (+Y-Tr) cases, respectively. It can also be noticed that the wear type on the right is purely thread type while on the left wheel a higher flange and thread wear were observed. As a result of this, care shall be taken to curvature speed at tight curves of AALRT since they aggravate wear rate in a great deal.

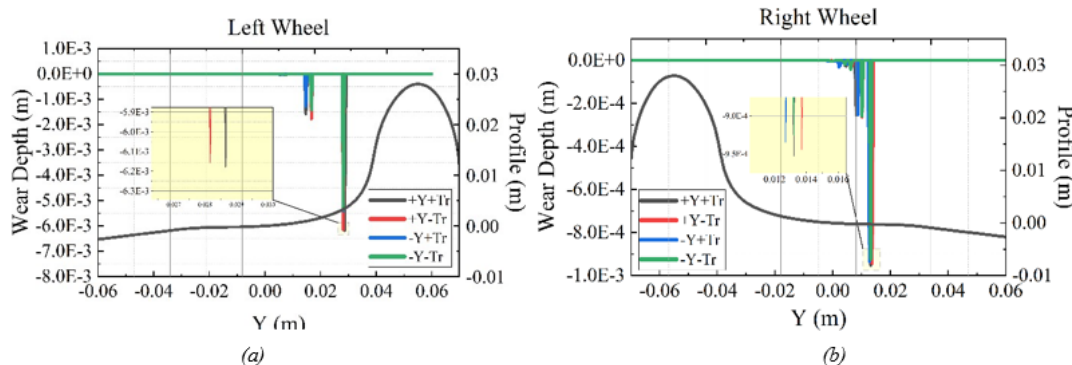


Figure 7. Wear Depth result of MC1WS1 (a) Left and (b) Right wheels 50 m Curve.

In the same manner, as shown in the Figure 8 below, the derailment coefficient for the left and right wheels between the curving periods of 46.5 to 49 seconds were 1.01 and 0.408 respectively, the left being 147.5 % higher than the right wheel. Since the limiting value is based on Nadal's stability

condition of 0.8, the left wheel is expected to be unstable and possibly experience derailment to the rolling stock. As a result, care shall be taken while passing tight curves like this so that derailment can be avoided.

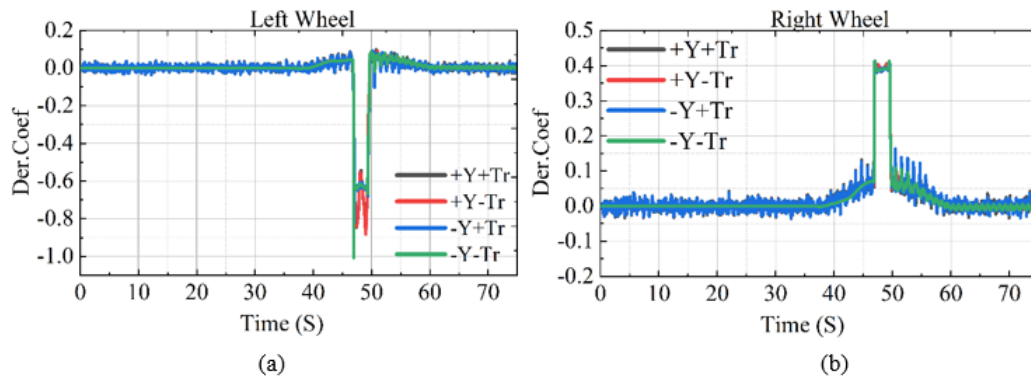


Figure 8. Derailment Coefficient result of MC1WS1 (a) Left and (b) Right wheels 50 m.

### 3.2. 100 m Curve Run

In this simulation as well, a 100 m curve was used. The maximum wear depth has reduced for the left wheel to 0.989 mm for the case of yaw and no rack irregularity (+Y-Tr), improving by 84% and 0.658 mm for the right wheel (-Y-Tr case) thereby decreasing by 30.9%, which is due to the radius

of curvature being increased by 50 m. As a result, the spiked wear depth in the 50 m radius of curvature has been reduced significantly, especially for the left wheel. Yet here also, the presence of the yaw damper made a 2.1% increment to the wear depth compared to no yaw case, as shown in the Figure 9 below.

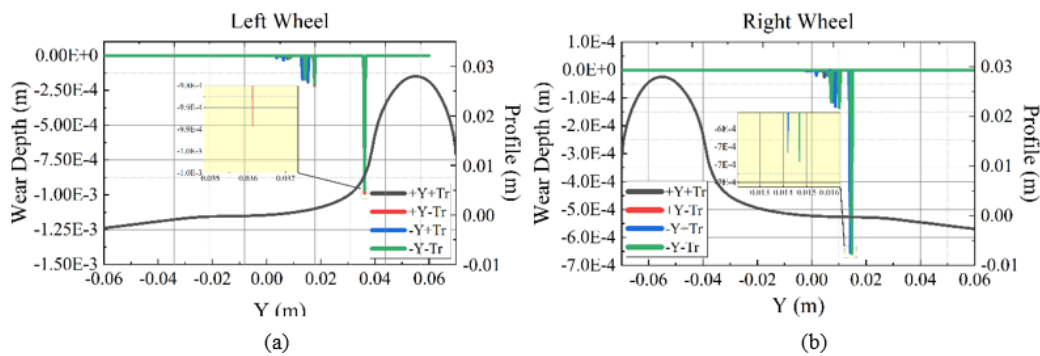


Figure 9. Wear Depth result of MC1WS1 (a) Left and (b) Right wheels 100 m Curve.

This in turn also resulted in a reduced maximum derailment coefficient of 0.78 and 0.41 for the left and right wheels, respectively. This is a 22.77% reduction to the left wheel, while the right one increased by 0.5%. This shows that the

wear region on the right wheel is still the thread region, and no significant increment has been seen on the derailment coefficient, as shown in Figure 10 below.

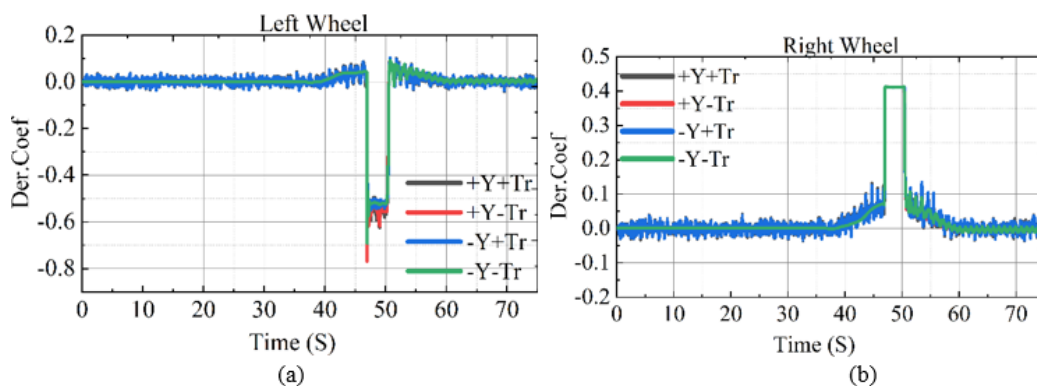


Figure 10. Derailment Coefficient result of MC1WS1 (a) Left and (b) Right wheels 100 m.

### 3.3. 150 m Curve Run

The wear depth for the 150 m curve radius run is found to be 0.26 mm and 0.306 mm for the right and left wheels, respectively, as shown below for a case of no yaw and no track irregularity (-Y- Tr) case for the left wheel while no yaw and track irregularity (-Y+Tr) case for the right wheel. As shown

in Figure 11, this is 69.1% lower than the wear depth registered for the left wheel, while the right wheel's reduced by 60.5%. This is also a great improvement registered due to increased radius of curvature to 150 m with the other factors kept constant. Unlike the previous cases, the presence of yaw resulted in a 6.7% improvement in wear depth to the right wheel.

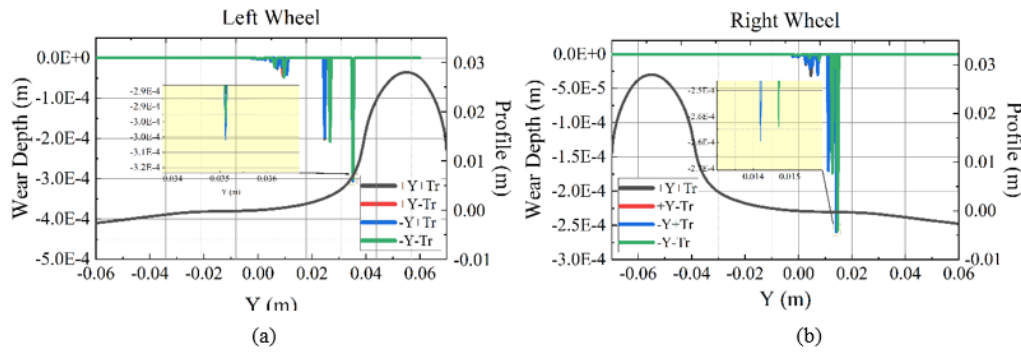


Figure 11. Wear Depth result of MC1WS1 (a) Left and (b) Right wheels 150 m Curve.

On the other hand, the derailment coefficient improved for this case to 0.4 and 0.65 for the right and left wheels, respectively as shown in Figure 12. This is a reduction of 16.7% and 2.44% for the left and right wheels. In this series of cases, it

can be seen that the derailment coefficient is reducing steadily with a significant amount, while for the right wheel, only slight improvement is observed since the wear region hasn't yet changed.

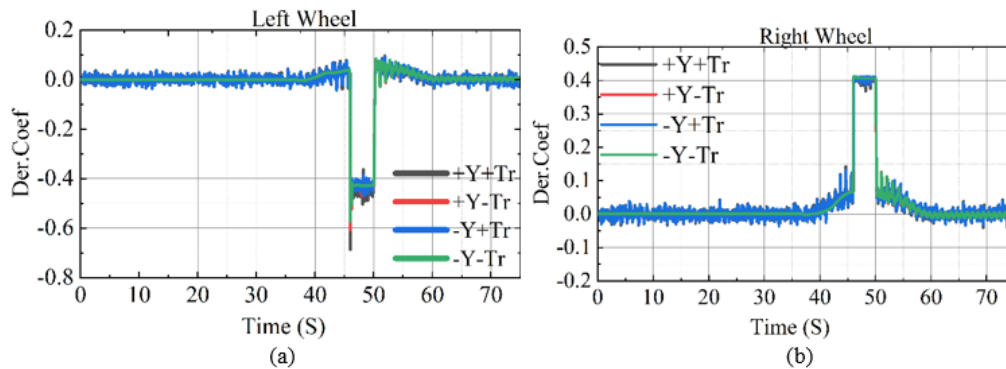


Figure 12. Derailment Coefficient result of MC1WS1 (a) Left and (b) Right wheels 150 m Curve.

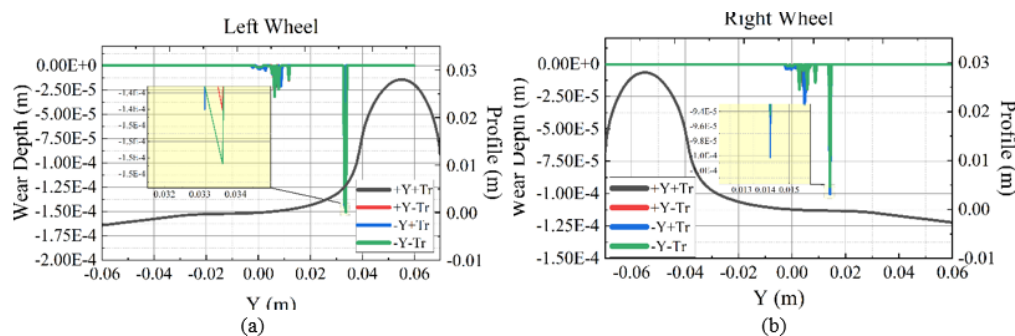


Figure 13. Wear Depth result of MC1WS1 (a) Left and (b) Right wheels 190 m Curve.

### 3.4. 190 m Curve Run

Finally, for this specific scenario of curve radius variation, the wear depth improved to 0.151 and 0.1 mm for the left and right wheels, respectively as shown in the Figure 13 below. From the previous cases, this radius of curvature resulted in a 50.7% and 61.5% reduction to the wear depth, which is significantly lower and it can also be observed that the left wheel flange wear has become very much reduced. This is due to the reduction of lateral and longitudinal creep forces at the flange

location when the radius of curvature increases from 50 m to 190 m. This is very important information that shall be considered to reduce the wear depth during the design of railway infrastructure.

On the other hand, as shown in Figure 14, derailment coefficient of left wheel can be seen for zero yaw with track irregularity (-Y+Tr) case increased to 0.78 (20%) and right wheel is still close to 0.41, yet still below the Nadal's stability and safety criteria and we can consider derailment is well minimized for this scenario.

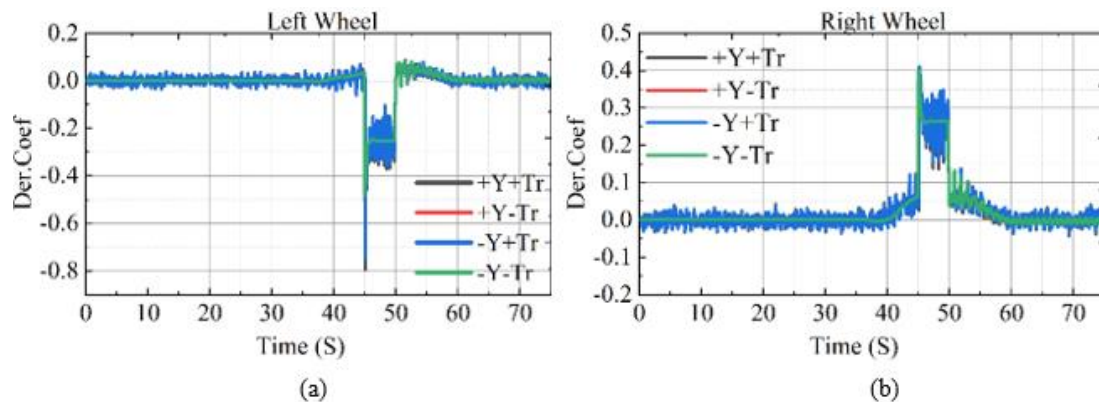


Figure 14. Derailment coefficient result of MC1WS1 (a) Left and (b) Right wheels 190 m Curve.

### 3.5. Comparison of Maximum Wear Depth

The analyzed data reveals that the wear depth is notably higher at the smaller radius of curvature (50 m), with the left wheel experiencing the most significant wear. This occurs because the simulation involves a right-hand curve, and due to the conical shape of the railway wheel, the wheel shifts laterally during steering, leading to increased flange wear on the left wheel. These findings are clearly illustrated in the maximum wear depth comparison shown in Figure 15 below.

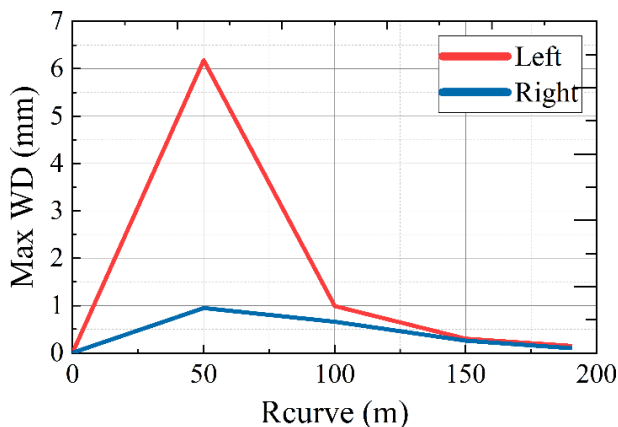


Figure 15. Maximum Wear depth variation for different Radiuses.

### 3.6. Ride Index Variation

The ride index was calculated for each curve variation, revealing similar values across different radius cases. Although minor fluctuations occur due to track irregularities on all car bodies, the yaw damper has little influence on the vertical ride index. This is because the yaw damper primarily operates in the lateral and longitudinal directions rather than the vertical direction. However, it still has some effect, albeit minimal, as illustrated in Figure 16 below.

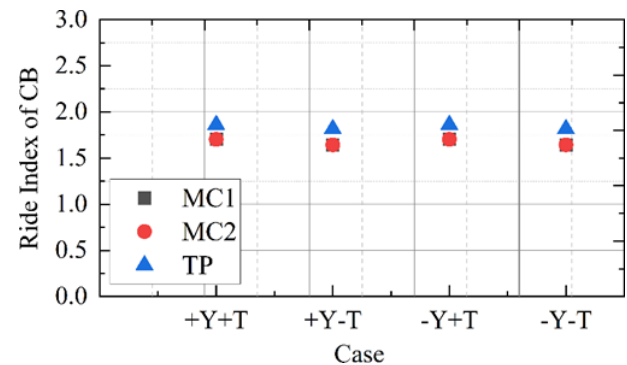


Figure 16. Ride index Variation for different cases of Yaw and track irregularity.

The RI values were approximately 1.7 for MC1 and MC2,

while the TP recorded a slightly higher value of 1.85, indicating a moderately comfortable ride though slightly above the recommended comfort threshold of 1.6. The greatest discomfort was observed at the TP due to its shorter, more articulated car body. This occurs because the turning forces from the rail are transferred through both front and rear couplings, increasing wheel slip and reducing ride comfort.

## 4. Conclusion

The dynamic interaction between railway wheels and curved tracks plays a crucial role in wheel wear development. Through comprehensive evaluation of contact mechanics, suspension performance, and track geometry, optimal wear mitigation approaches can be developed.

Research findings indicate that expanding curve radii substantially reduces wheel wear by minimizing creep forces and their resulting tangential effects. This geometric optimization not only decreases the derailment risk but also improves passenger comfort by reducing lateral force magnitudes. Furthermore, track surface imperfections were found to cause slight variations in derailment coefficients while producing negligible increases in ride index measurements.

Future research should focus on real-time adaptive systems and advanced materials to enhance wheel lifespan and operational efficiency.

## Abbreviations

|       |                                 |
|-------|---------------------------------|
| AALRT | Addis Ababa Light Rail Transit  |
| RCF   | Rolling Contact Fatigue         |
| RI    | Ride Index                      |
| UIC   | International Union of Railways |
| WS    | Wheelset                        |

## Author Contributions

**Mazuri Erasto Lutema:** Conceptualization, data curation, investigation, methodology, resources, software, validation, writing original draft, writing review & editing

**Faraja Nyangasa:** Methodology, writing review & editing

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## Data Availability Statement

The data is available from the corresponding author upon reasonable request.

## Conflicts of Interest

The authors declare no conflicts of interest.

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## Biography



**Mazuri Erasto Lutema** is a qualified mechanical engineer and academic, serving as an Assistant Lecturer in the Faculty of Transport Engineering and Technology at the National Institute of Transport since 2020. Currently, he holds the position of Acting Head of the Automotive and Mechanical Engineering Department. Eng. Lutema earned his MSc in Railway Engineering from Addis Ababa University and completed his undergraduate studies at the National Institute of Transport. His research expertise spans vehicle system dynamics, railway vehicle design, and wheel-rail interaction, covering theoretical, design, and practical applications. He has contributed to the field with two publications in prestigious mechanical engineering journals. With five years of teaching experience, Eng. Lutema has delivered courses in rolling stock technology, railway vehicles, urban railway systems, dynamics, engineering materials, project management and machine mechanics to both diploma and undergraduate students.

## Research Field

**Mazuri Erasto Lutema:** Vehicle system dynamics, railway vehicle design, wheel-rail interaction and mechanics of materials.

**Faraja Nyangasa:** Finite element method, automotive design, plastic deformation.