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Addis Ababa University
Addis Ababa Institute of Technology
African Railway Center of Excellence

Investigation of Wheel Wear Behaviors Subjected Under Dry and Wet Loading Conditions: A Case study of Addis Ababa Light Rail Transit

A Master's Research Thesis

By

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the award of Degree of Masters of Science in Railway Engineering
(Rolling stock)

Advisor

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June 2020

Addis Ababa Ethiopia

APPROVAL

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DECLARATION

I hereby declare that the work undertaken in this thesis entitled “Investigation of Wheel Wear Behaviors Subjected Under Dry and Wet Loading Conditions: A Case study of Addis Ababa Light Rail Transit” is my original work and it has not been presented for the award of a degree to any other university and all the other resource materials used in this work has been referred.

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ABSTRACT

Wheel/rail wear continues to be an important factor that limits the service life of rolling stock. Wheel-rail surfaces are subjected to contact stresses that results in surface damage which is observed as either wear or contact fatigue damage mechanisms. The contact stresses are the result of train and passenger loads, speed, friction, and other operating conditions, and the effect of the stresses on the damage depends on the loading conditions (lubricated or dry). The analysis of wheel damage due to wear is very important to determine the service life of the rolling stock and railway track. The aim of this research was to investigate the wear behavior of the train wheels due to wet and dry loading conditions for the case of Addis Ababa Light Rail Transit (AALRT) through friction, speed and creep forces analysis. All forces and wear parameters were analyzed using a train simulation model in SIMPACK software. The effect of overloading condition during dry and wet contact on wear rate, wear volume and wear resistance of the wheel materials were determined and compared. The higher coefficient of friction (CoF) value, which is in the case of dry sliding contact condition, resulted in the formation of higher longitudinal and lateral creep forces that increased the wear values in terms of volume loss, wear rate and wear depth, and wet sliding contact (with low CoF) resulted in low wheel wear parameters. Speed variation showed that an increase in the speed increased the wheel wear volume wear depth and wear rate, and these are relatively lower in the case of low speed operations.

Key words: Wheel-rail contact, contact conditions, wheel damage, wear parameters

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ABBREVIATIONS

AALRT	Addis Ababa Light Rail Transit
CoF	Coefficient of Friction
ERC	Ethiopian Railways Cooperation
MBS	Multi-Body Simulation
RCF	Rolling Contact Fatigue
ToR	Top of Rail

1 INTRODUCTION

1.1 Overview

With the continuous development of railway, in the last decades, it plays a vital role in transporting passengers and merchandise which is considered as the best and safest among the transportation systems [1]. Addis Ababa Light Rail Transit (AALRT) provides an additional transportation system to the public of Addis Ababa city and majority of the passengers use the railway transportation mode (relatively cheaper compared to others). However, this shifting has led to overcrowding of the trains at peak hours, during the morning and evening times, and this condition consequently accelerates the train wheel wear rate.

Like any high stress concentration zone, the wheel-rail contact interface is subjected to various damage in the form of wear and fatigue which highly depend on the train load and coefficient of friction. The contact between the wheel and the rail surfaces results in stick-slip wear that leads to crack initiation and subsequent propagation, then final catastrophic damage. The contact condition in terms of contact pressure which is a result of train and passenger loads, train running speed and the friction coefficient are found to be more important factors for surface damage than the grade of steel [2].

Due to excessive wheel-rail contact stress distribution, wear damage and rolling contact fatigue (RCF) of wheels and rails has become an inevitable problem in railway operation. In the railway transportation industry, the ability of the rail vehicle to move at higher travelling speeds and to carry heavier axle loads has been achieved through the advances in technology that result in increase in traction power and contact stress much more rapidly than ever before thereby accelerating progressive wear and RCF at the contact surface of the rails and wheels [3]. Figure 1.1 represents wear damage at the wheel surface in form of surface fatigue.

Surface fatigue mechanisms is a result of stress fluctuations and repeated loading, and conditions under which the two solid materials have sliding contact. Despite of applying a lubricant to the wheel/rail, crack initiation and propagation is formed due to repetitive

alternating forces thereby causing surface fatigue which causes destruction of parts which may begin with a defect or surface cracks [4].

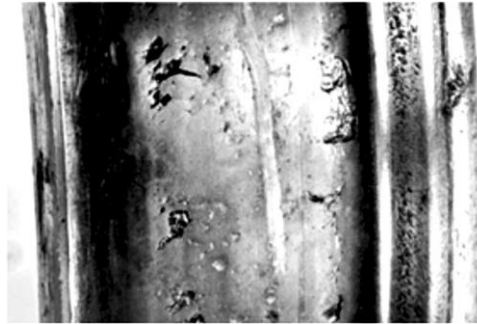


Figure 1.1: Surface damage on the wheel due to wear [4]

The structure of railway and the rolling stock in general form a very complex mechanical system. The interaction between the rolling stock and the rail provides a wear damage both at the wheel and rail profiles. As a result, functional deterioration of track and wheel surface occurs due to the interaction forces applied at the wheel/rail contact area. This thereby results into expensive maintenance requirements on the rails and wheels [5].

The wheel-rail contact is very crucial in the successful and safe operation of railways. Managing the wheel-rail contact effectively is therefore probably the most important aspect of rail infrastructure and rolling stock operations. Due to the many different rail and wheel profiles, a large variety of loading conditions and contact geometries exist, rail cant and curve radii, and railway vehicles running on a network.

Contact conditions vary considerably between the two main contact areas that is the wheel tread-rail head and wheel flange-rail gauge corner, but are usually more severe in the latter, where greater wear and fatigue cracking is seen to occur. Friction and creepage in the contact are also highly variable [6].

The presence of third body materials in the contact either complicate the situation further or enhances the functionality. These materials may occur naturally such as humidity, precipitation or leaves, some may be products that are applied to the rail or wheel surface to perform a particular purpose, and these may include friction modifiers, greases, and traction gels among others. Even in dry contacts that are uncontaminated, a third body layer exists, this layer is usually made up of oxides and wear debris.

The damage processes and adhesion are influenced by all of these factors namely, natural materials such as precipitation which can negatively influence the adhesion thereby causing braking problems and wheel slip interaction, and sanding as well can increase wear or rolling contact fatigue (RCF) in the wheel-rail contact. Measures to balance these additives have to be of most importance for smooth maintenance and better railway operations for example using applied lubricants to reduce wear in curves as well as friction modifiers or traction enhancers to increase adhesion [6].

1.2 Background

The history of railway transportation system in Ethiopia goes as back as 1917, with the construction of a 784 Km railway that links the capital city Addis Ababa with the port of Djibouti [7]. The Ethio-Djibouti Railway began operating between Dire Dawa and Djibouti in 1901 and was later extended to Addis Ababa city [8].

In Ethiopia, railway transport can currently be evidenced by the operation of the Addis Ababa Light Rail Transit which opened in 2015, this is characterized by an electrified standard gauge rail line and the Addis Ababa-Djibouti Railway which commenced freight operations in October 2016 [8].



Figure 1.2: Legehar train station [8]

AALRT is operated on two main lines that is the East-west line (from Ayat to Torhayiloch) and the North-South line (Kality to Menilik II square). The length of the first line is 16.99 km and the second is 16.689km long, the total length of the track is 31.025 Km. A section is shared by both lines and it is about 2.662km.

AALRT vehicle is a 70% low-floor tram supplied by CNR Changchun. It operates at a speed of 20 to 40 km/h on straight line and at 17 km/h on curved line, the two routes are designed to carry up to 15,000 passengers/h in each direction. Services are operating between 06.00 and 22.00 each day, and the train's independent power supply network includes four substations with a total rating of 160 MW [8].

Since the wear at the wheel/rail interface plays a vital role in determining the reliability of railway transportation, the increase in speed and axle loads has caused the wear damage to become more severe, which results in a significant decrease of wheel/rail system service life time. Researchers [3] [4] and others have therefore expanded, explored and discussed the knowledge about the wear mechanisms of the wheel/rail system and the methods of alleviating the wear of wheel and rail materials.

Through friction studies, researchers have indicated that one of the ways of effectively reducing wear of wheels and rail materials is through lubricating the wheel/rail contact.

The interest for investigation of wear mechanism or behavior is not only aimed at reducing wear losses but also at developing effective methods of prediction of wheels service life, and ensuring of reliable wheel operation especially in extreme conditions. The mechanism of wear of railway wheels tread presents a combination of mechanical, thermo-physical and chemical effects. This connects with wear particles and micro cracks formation in areas of intensive plastic deformation and corrosive products of wheel steel.

To investigate the mechanisms behind wheel–rail RCF and wear, there are a lot of theoretical studies that have been experimented and conducted. Wheel–rail contact wear was simulated using Archard's law. It is only in the slip region of the contact patch that Archard's law is applicable and assumes that the volume of worn material V_{wear} [m³] is proportional to the normal contact force N [N] and the sliding distance d [m], and inversely proportional to the hardness [N/m²] of the softer of the two materials in contact. The amount of wheel–rail contact wear is therefore calculated using Archard's law [9].

This study therefore aims at identifying the wear behavior of the wheel subjected to wet and dry conditions with varying loads for the case of AALRT.

1.3 Problem statement

Wheel wear is generally used to describe any kind of damage occurring on the rolling surface of railway wheels which involves material worn-out, surface and subsurface fracture. These surface fracture and material removal, which is a result of mechanical action of the counter body, have a significant effect on the service life of railway wheels and maintenance cost. However, the presence of water or a lubricant on the contacted wheel/rail surfaces may enhance the resistance to surface cracks occurrence, severe wear and damage at overloading condition in particular. In addition, the applied loads that are significantly higher than the design capacity can cause damage to the surface of the wheel at the contact area. The wear resistance of wheel materials depends on wet and dry contact condition at the wheel/rail interface, operating speed and applied load and may be important to investigate for AALRT cases. It was observed from the month of July 2019 that the wheels of the trains operating from the North-South line (Kality to Menilik II square) were getting worn much faster than the trains operating from the East-west line (from Ayat to Torhayiloch) and this therefore needed to be analyzed. The purpose of this research is therefore to investigate the wheel wear behavior under dry and wet loading conditions for the case of AALRT.

1.4 Objectives of the study

1.4.1 General Objective

The main objective of this study is to investigate the wheel wear behavior subjected under dry and wet loading conditions for the case of AALRT transportation system.

1.4.2 Specific objectives

- To analyze the effect of train over loading on the wheel wear volume, wear rate and wear depth of AALRT.
- To analyze the effect of wet and dry contact conditions on the wheel wear volume, wear rate and wear depth.
- To compare the wheel/rail contact pressure results analyzed using Matlab® and SIMPACK modeling for pressure validation.

1.5 Significance of the study

The availability of a railway system depends on the condition of its infrastructure and rolling stock. This study will give an insight understanding of the rate at which the rolling stock wheel diminishes because of wear in form of surface damage or cracks as well as the damage behavior from which counter measures will be discussed to reduce this effect and as well prevent damage to the track due to faulty wheels.

1.6 Scope and limitations of the research

This research will mainly be useful in advancing the knowledge in the field of wheel wear. Therefore; the scope of this study will be limited to investigating the wheel wear behavior under dry and wet loading conditions for the case of AALRT by analyzing wear parameters in form of wear volume wear rate and wear depth. This will be done accordingly to successfully acquire the advancement of knowledge required in this field.

The work will be completely based on computer simulation of the train wheelset model. Experimental investigations on the other causes of wheel wear as well as investigations on rail wear, either to validate the SIMPACK results or to gain an understanding on the wear behavior of wheels will not be conducted. This work will therefore be solely drawn and conducted from simulation analysis using SIMPACK and contact pressure numerical analysis using MATLAB.

1.7 Organization of the proposal

This thesis is presented in seven (7) chapters organized as follows:

In Chapter 1, a brief introduction to the problem and objectives of the research are given. In Chapter 2, the existing literature has been reviewed on wear of wheel rails, including modes of wheel damage, sources of train wheel wear damage and wear mechanisms. Chapter 3 describes the methodology that will be used to achieve the stated objectives in Chapter 1. Chapter 4 presents the research results and discussion; chapter 5 presents the conclusion and recommendations and finally chapters 6 and 7 presents the references and appendices respectively.

2 LITERATURE REVIEWS

2.1 Train Wheels

Railroad car wheels are fixed to a straight axle such that both wheels rotate in unison. This is called a wheelset. A train wheel or rail wheel is a type of wheel specially designed for use on rail tracks. A rolling component is typically pressed onto an axle and mounted directly on a rail car or locomotive or indirectly on a bogie. Wheels are cast or forged and are heat-treated to have a specific hardness. New wheels are trued using a lathe to a specific profile before being pressed onto an axle.

All wheel profiles need to be periodically monitored to ensure proper wheel-rail interface. Improperly trued wheels increase rolling resistance, reduce energy efficiency and may create unsafe operation. A railroad wheel typically consists of two main parts: the wheel itself and the tire around the outside (tread and flange). A rail wheel is usually made from steel and is typically heated and pressed onto the wheel where it remains firmly as it shrinks and cools. Monobloc wheels do not have encircling tires while resilient rail wheels have a resilient material such as rubber between the wheel and tire.



Figure 2.1: Wheel profile [2]

The lifetime of railway wheels and rails is limited by wear and rolling contact fatigue (RCF), both of which are deterioration phenomena. A competition exists between wear and surface-initiated RCF in a way that wear can worsen the contact geometry between wheel and rail which may accelerate crack growth. At higher wear rates, RCF does not have the opportunity to develop further. Cracks can initiate, but will be worn off due to the

high wear rate and will not be able to propagate beneath the surface. Care must therefore be taken, when optimizing to reduce the wear, since RCF can in that case become the dominant problem [10].

2.2 Wheel/rail contact conditions

Both rolling and sliding occur in the contact zone in wheel–rail contact. The wheel tread is in contact with the rail head on straight track but in curves the wheel flange may be in contact with the gauge corner of the rail with the rail head. As a result of the wheel profile conicity, flanging results in a large sliding motion in the contact. The contact area can therefore be divided into stick (no slip) and slip regions. With the increase in tangential load, the slip region increases and the stick region decreases, resulting in a rolling and sliding contact. The stick region disappears when the tangential load reaches its saturation [11].

2.2.1 Mathematical model for wheel – rail contact

The steady wheel-rail rolling contact condition is based on the above - mentioned assumptions of Hertz's contact theory.

According to [12], if two elastic non - conforming bodies are pressed together, the contact area assumes an elliptical shape with a semi major axis 'a' and a semi minor axis 'b'. Figures 2.2 and 2.3 show the elliptical shape of the contact area between wheel and rail and the distribution of contact pressure in the elliptical area respectively [13] [14].

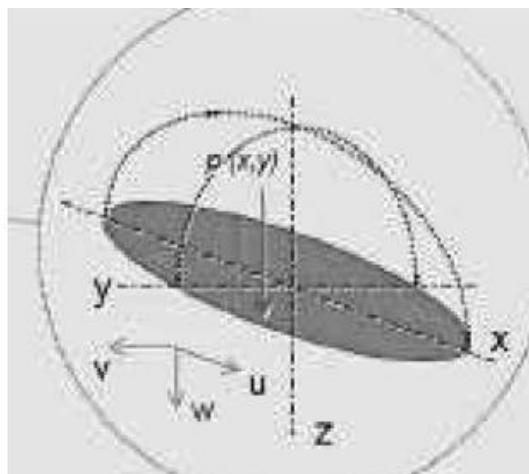


Figure 2.2: Elliptical shape of the contact area between wheel and rail [13]

Wheel load is transmitted to the rail through a tiny contact area under high contact stresses. This result in repeated loading above the elastic limit that leads to plastic deformation.

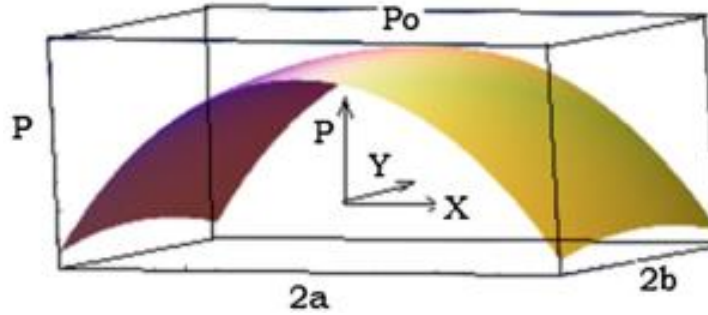


Figure 2.3: Pressure distribution across elliptic area [14]

According to [2], the maximum pressure P_o acting at the area of contact is given by;

$$P_o = \frac{3}{2} \frac{P}{\pi ab} \quad 2.1$$

Where, P is the normal load/ force applied at the contact area; a and b are the semi axes of the elliptic boundary of the surface of contact.

$$a = m \left[\frac{3\pi P(K_1 + K_2)}{4(A + B)} \right]^{\frac{1}{3}} \quad 2.2$$

$$b = n \left[\frac{3\pi P(K_1 + K_2)}{4(A + B)} \right]^{\frac{1}{3}} \quad 2.3$$

K_1, K_2 are constants given by;

$$K_1 = \frac{1 - V_1^2}{\pi E_1} \quad 2.4$$

$$K_2 = \frac{1 - V_2^2}{\pi E_2} \quad 2.5$$

Where; V_1 and V_2 are poisson's coefficients of the wheel and rail respectively, E_1 and E_2 are modulus of elasticity for the wheel and rail respectively

The calculation of the contact area requires the knowledge of some geometric constants used in equations 2 & 3. With respect to wheel -rail configurations, the following curvature combinations are related as in the equations below [13].

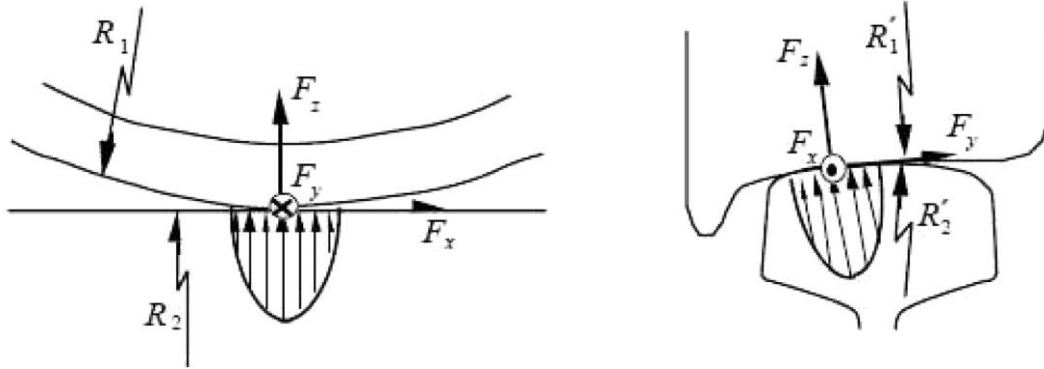


Figure 2.4: Major curves of the wheel and rail- F_z vertical force and F_x and F_y are the contact forces in contact with wheel and rail [13]

$$A + B = \frac{1}{2} \left[\frac{1}{R_1'} + \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_2'} \right] \quad 2.6$$

$$B - A = \frac{1}{2} \left[\left(\frac{1}{R_1} - \frac{1}{R_1'} \right)^2 + \left(\frac{1}{R_2} - \frac{1}{R_2'} \right)^2 + 2 \left(\frac{1}{R_1} - \frac{1}{R_1'} \right) \left(\frac{1}{R_2} - \frac{1}{R_2'} \right) \cos 180 \right]^{\frac{1}{2}} \quad 2.7$$

Where;

R_1 – is the rolling radius of the wheel

R_1' – is the lateral curve radius of the width of the wheel profile at the contact area

R_2 – is the major rolling radius of the rail (which is normally infinite in rails)

R_2' – is the lateral curve radius of rail profile curve.

The ellipticity parameter $\left(\frac{a}{b}\right)$ is related to geometric parameter $\left(\frac{A}{B}\right)$ by means of the coefficients ‘m’ and ‘n’ from the notation $\cos \theta = \frac{B-A}{A+B}$, therefore, the values of ‘m’ and ‘n’ for different values of θ can be obtained using the equations below. [2]

$$n = 3E - 05\theta^2 + 0.0045\theta + 0.334 \quad 2.8$$

$$m = 62.19\theta^{-0.914} \quad 2.9$$

2.2.2 Hertzian contact of the Wheel/rail

Wheel/rail contact is typically assumed to be Hertzian, in which case the shape of the contact patch on the wheel and rail is elliptical and the contact pressure is distributed elliptically over the contact area. The length and width of the ellipse are similar in magnitude when contact occurs on the tread of the wheel. For contact on the flange of the wheel, the ellipse tends to become long and thin, with the longest axis oriented along the rail [15].

The contact between the wheel and the rail result into stick slip wear which brings about wear that results into crack initiation and propagation and this is very catastrophic. The contact condition in terms of contact pressure and the velocity at which the wheel slides on the rail are found to be more important factors than the grade of steel its self [2].

The wear that occurs on the wheels and continuous usage of the rails ultimately results into profiling these components so as to regain their conicity and to ensure reliable operation of the train, profiling the wheel results into reduction to the wheel profile radius and thus this results into a change in the contact area thereby influencing the stress concentration at the contact patch.

Investigation of the contact stresses and resulting damage arising at the wheel-rail interface is therefore an important issue in the wheel-rail contact phenomena. This problem can be numerically or theoretically analysed using the theory of Hertz which was developed in 1881 to solve the problem of pressure distribution between two elastic spherical bodies in contact [2]. According to Hertz's contact theory, the contact stress model is based on four assumptions that are made in order to determine the solutions of Hertzian contact problems. These assumptions are as stated below;

- i. The strains are small and within elastic limit
- ii. The area of contact is much smaller than the characteristic radius of the body.
- iii. The surfaces are continuous and non - conforming
- iv. The bodies are in frictionless contact.

2.3 Wheel Materials

In order to increase the time between wheel re-profiling operations, to improve safety, and to reduce total wheelset life cycle costs, new specifications are being imposed on railway wheel wear and reliability. In parallel with these requirements, railway vehicle missions are changing due to the following;

- (a) The need to operate rolling stock on track with low radius curves as well as the high-radius curves on the high-speed line;
- (b) Increasing speeds;
- (c) Track quality decrease due to a reduction in maintenance.

The above reasons lead to an increase in the severity of the wheel–rail contact conditions which will increase the likelihood that wear occurs. Excessive wear can affect the dynamic behavior of the railway vehicle and reduce ride comfort, impact upon the potential for derailment, and reduce the integrity of the wheel material. New wheel materials are therefore being developed to give greater durability in order to deal with these demands. To aid such development an improved understanding is required of the wear mechanisms and regimes apparent in wheel steels [16].

Railway Group Standard GM/RT2466 specifies the approved material grades for monobloc wheels used on Network Rail controlled infrastructure [17]. This requires particular BS5892 part 3-wheel material grades to be used for particular vehicle applications.

Table 2.1: Wheel material grades

Wheel type	Wheel material grade (BS 5892 part 3 and UIC812-3)
Freight, integral brake disc wheel	R7E
Freight, cheek mounted brake disc wheel	R8E
Other freight wheels	R7T or R8T
All passenger vehicle and other wheels	R8T

Table 2.2: Mechanical and geometrical properties of different wheel types used by AALRT

	Mechanical property		Geometrical parameters	
Wheel type	Modulus of Elasticity (E)	Poisons coefficient(v)	Wheel Radius (R ₁₂)	Wheel radius of curvature (R ₁₁)
CL60	210GPa	0.3	330mm	1098mm
ER7 = UICS1002	206GPa	0.27	330mm	1098mm
R8T	206GPa	0.285	330mm	1098mm

Source: Ethiopian railway corporation

Table 2.3: Chemical composition of standard wheel steels (% by weight)

Cast Composition	Class D (typical)	Grade R7 (max)	Grade R8 (max)
C	0.61	0.52	0.56
Si	0.33	0.40	0.40
Mn	0.70	0.80	0.80
P	0.027	0.04	0.04
S	0.022	0.04	0.04
Cr	0..30	0.30	0.30
Cu		0.30	0.30
Mo	0.05	0.08	0.08
Ni	0.20	0.30	0.30
V		0.05	0.05
Cr+Mo+Ni		0.60	0.60

2.4 Wear of railway wheels

“Wheel wear” includes any kind of damage occurring on the running surface that involves loss of material. Wheel wear can be categorized into three main classes, deeply described as follows [18];

1. **Flange wear**, this involves the reduction of the flange thickness that leads to a reduction in the strength and a worsening in wheel-rail contact, this is shown in Figure 2.5.

2. **Wheel tread wear**, this involves an increase in the flange height and flange thickness which in turn might cause severe problems in turnouts or crossings.
3. **Out-of-round wheel wear**, this is often caused by the presence of a mixed microstructure within the wheel tread as a result of heat treatment issues during wheel manufacturing.

All the above wear classes lead to a significant change in the shape of the wheel profile, and as a result could severely affect the dynamics of the vehicle and the safety against derailment. However, these damages or modifications may be recovered by reprofiling the wheel.

A number of wheel reprofiling operations may be carried out before reaching the maximum height and thickness of the rim beyond which reprofiling does not comply anymore with the mechanical resistance of the wheel and as this limit is reached, the wheel must be replaced [18].

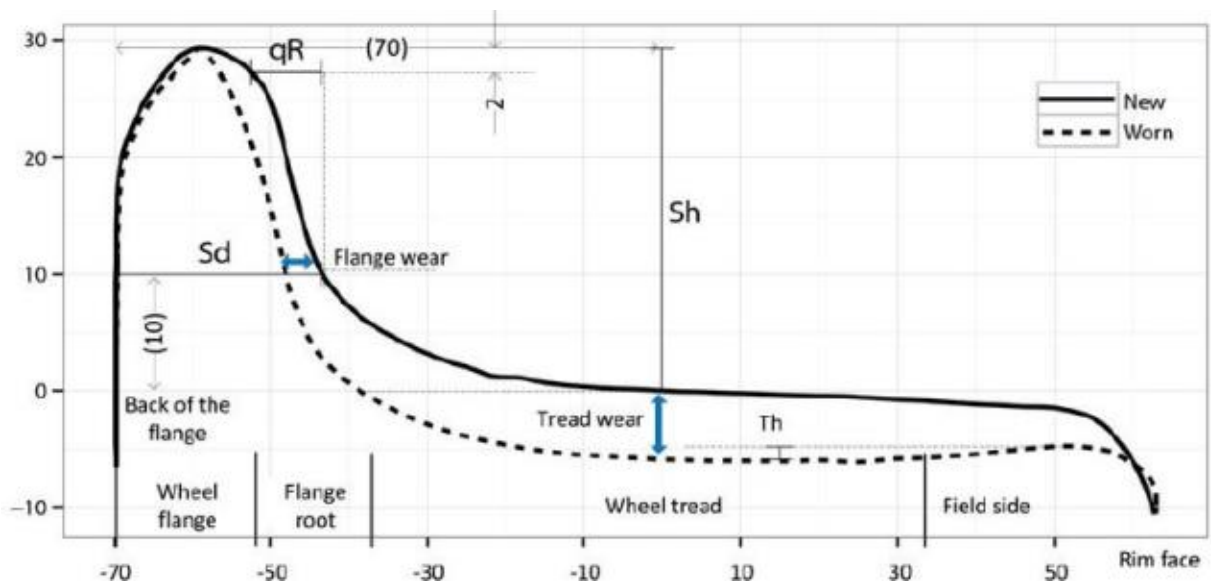


Figure 2.5: Typical railway wheel profile. The picture shows also flange and tread wear phenomena [18]

2.5 Wheel damage modes

It is crucial to know how each damage can be recognized, how they occur, what influence they have and what happens if continuing to run with them. In order to investigate wheel wear and identify ways of reducing wear of wheels, it is crucial to identify the modes of wheel damage. These are presented in this section.

2.5.1 Wheel flat

Wheel flats are the most common form of wheel tread damage. They are caused by the sliding of the wheel along the rail that occurs as a consequence of a blockage or partial blockage of the wheelset. Wheel sliding may be either because of a fault in the brake system or an excessive applied braking force. Consequently, the dissipated friction energy turns into heat and makes the tread surface locally flat. This local flat defect is known as wheel flat.

Under significantly high temperatures like 800°C reached during the generation of the flat, the pearlitic microstructure of the wheel steel transforms into martensite which is a very hard and brittle form of steel microstructure. As a consequence of further cyclic loads arising from the wheel-rail contact, cracks can develop in the brittle material and grow considerably until the brittle surface starts spalling out from the wheel tread.

The cavities on the tread surface such as RCF clusters and wheel flats with spalling, produce a local deviation from the nominal wheel radius which may generate impact loads in the wheel-rail contact that further aggravate the irregularity [18].

A wheel flat can as well be described as a scraped-off part of the wheel tread as shown in Figure 2.6. This is caused by a blocked wheelset in a moving vehicle in a way that, as the vehicle keeps moving, the material in the contact patch is grinded off. Some reasons for a wheelset to become blocked are listed below [19].

- Too much applied brake force, for example due to emergency braking
- A brake block has frozen to the wheel after braking
- Brake has not been released when the driver starts moving the train

In the case of partially blocked, the result could be several subsequent small flats as seen in Figure 2.7



Figure 2.6: Wheel flat [19]

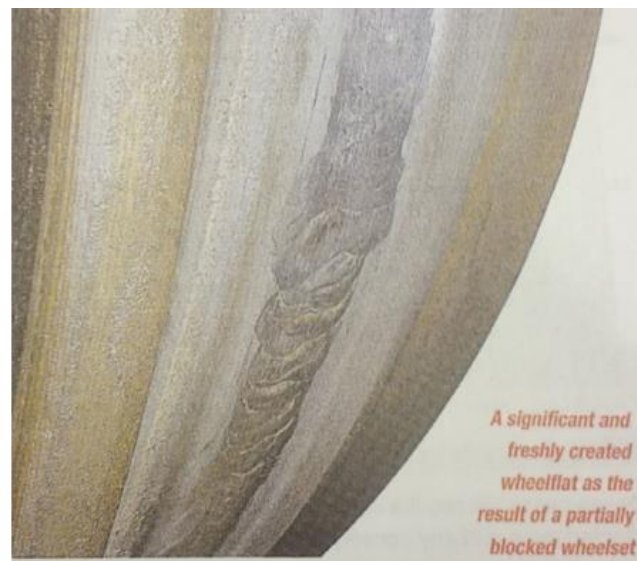


Figure 2.7: Wheel flat due to partially blocked wheelset [19]

An experimental study [20] where a real trainset locked the brakes to generate wheel flats, found that, in the initial seconds flat growth is fast and thereafter slows down significantly. As an example, they found that a 5 second locking time in 20 km/h gave a flat of 40 mm, while 25 seconds locking just had it extended 10 mm more. The sheared-off material piles up behind the flat afterwards as shown in Figure 2.8, even if it often falls off after some rolling. With continued operation, the flat will increase in length but keep its depth as shown in Figure 2.9. “Small” flats are however worn or pounded off.



Figure 2.8: Huge freshly made flat, with sheared off material piled up [19]

A wheel flat causes impact loads on the rail, the amount of which depend on length and depth of the flat, speed, axle load, un-sprung mass and properties of the track [21]. The peak contact force however depends mainly on the depth of the flat and not so much on the length [22]. Wheel flats have medium derailment potential and will cause out-of-round wheels over time. Other issues with wheel flats are that, they can cause damage on the bearings and axles if left unattended [23].

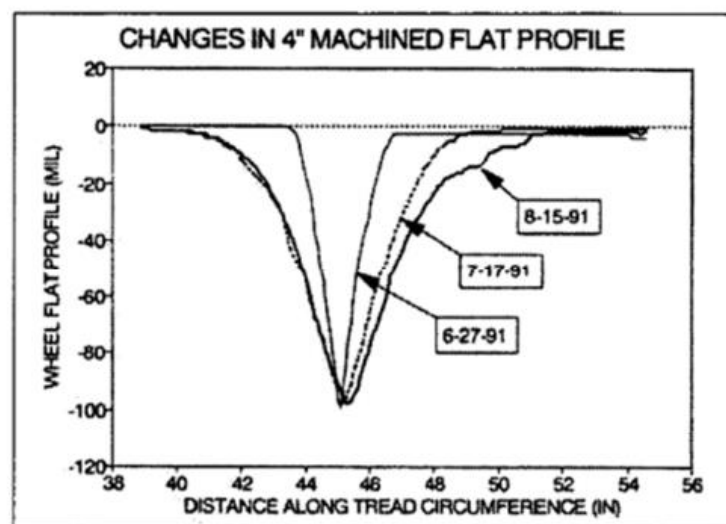


Figure 2.9: Flat evolution over time [19]

2.5.2 Tread cracks

Tread cracks is a type called surface induced RCF, it is initiated close to or on the surface and it appears on the surface due to frictional stresses. When the wheel/rail contact is

subjected to large frictional loads, such as braking and curving, plastic deformation will occur if the stresses become sufficiently big to deform the material. After several deformations, this will finally lead to fatigue cracks forming. Where the cracks meet, material will fall out.

Another possibility of tread crack is that it is formed close to the surface. In this case, voids form within a few millimeters below the surface in the cyclic compression, which then propagates into a crack [24]. The crack forms a shallow angle out of the material which is often called shelling in literature [25]. Shelling is often happening in loaded wagons, where the sliding is low and the load is big [24]. To the right in Figure 2.10 a combination of these damages can be seen.



Figure 2.10: Surface induced rolling contact fatigue [24]

It is possible to study the deformations and cracks on the surface to determine what caused them. First of all, there are four zones on the wheel tread each representing the contact zone in different cases as seen in Figure 2.11. Braking is mainly done in the middle of the tread which is zone 3, the low rail wheel in curving is in contact in zone 1 [26], and the high rail wheel in curving is pressed against the rail with its flange and this is mainly in zone 2. Damages in zone 4 are the result of a large lateral force. Secondly, as also can be seen in Figure 2.11, due to the fact that the fatigue cracks resulting after enough deformation cycles will form perpendicularly to the prevailing deformation direction, the direction of the cracks in each case varies. For example, in the case of under radial curving, illustrated in Figure 2.12.

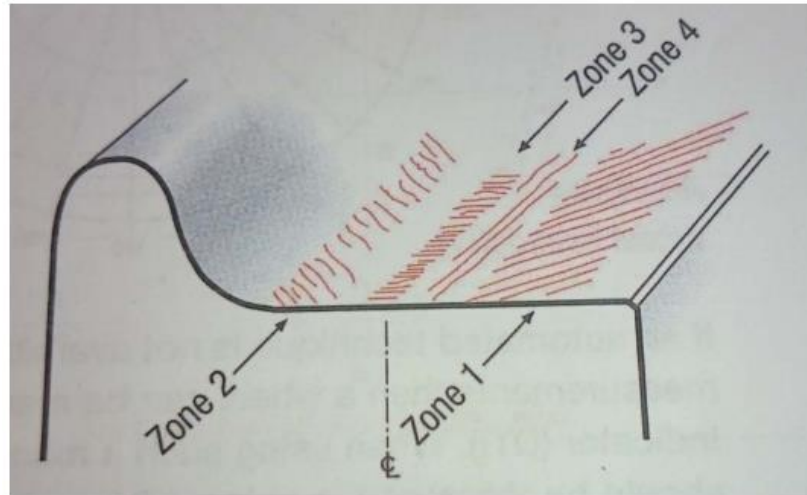


Figure 2.11: Wheel tread damage zones

It is shown that braking moments cause cracks in wheels, while traction causes cracks in rails. Under radial position the high rail wheel experiences traction while the low rail wheel experiences braking [26]. Therefore, zone 1 RCF is more common than zone 2 RCF.

The consequences of surface induced RCF are a deteriorated rolling contact, a risk of developing into an out-of-round wheel that will later be described [25] and a low risk of derailment.

Spalling is another damage classified tread crack. It manifests through material fall out in a limited area. Spalls are initiated through cracks from martensite spots. Hence, they are not uniform around the wheel. These cracks start radially and they could later deviate into circumferential direction instead [24]. Also, in the cracks caused by spalling, fluid can be trapped in the cracks resulting in an increased growth rate in wet or snowy conditions [25]. The problem of spalling is mainly seen in empty cars where adhesion is low and there is a high braking ratio due to the low weight [24]. If left unattended to, spalling might develop into out of round formation.

In the cracks caused in both surface induced RCF and spalling, certain conditions speed up the crack growth. For-example, if the wheel is first exposed to dry conditions where cracks are initiated due to the high friction, and then gets wet, the moisture inside the cracks during the next load cycle will speed up the crack growth [27]. It is beneficial if lubrication is applied in the beginning before cracks occur. A lubricated surface reduces friction and thereby also the stresses in the material causing the cracks.



Figure 2.12: Damage under radial curving

2.5.3 Thermal damage

Wheels can be damaged from a too high heating of the outer layer of the wheel, often related to tread braking. In a mild form, cracks very similar to surface induced RCF can appear. RCF and thermal damage often promote each other. A steep inclination of the cracks indicates a big influence from thermal loading, while a shallower one indicates mostly RCF [28].

In a severe heating the cracks can extend radially into the wheel, eventually causing a brittle fracture [28]. Therefore, such cracks have a moderate derailment risk [25]. Thermal damage may also be because of too much friction between the wheel and rail, overload conditions and sudden application of the brakes.

2.6 Wear mechanisms

The two most dominant wear mechanisms in wheel/rail contact mechanics are adhesive and delamination wear. Adhesive wear is relatively mild, produces thin flakes on the surface over a large number of cycles [29]. Wheel and rail surfaces remain shiny under adhesive wear. However, there are other wear mechanisms that occur on the wheel/rail surface that is; abrasive wear, tribo-chemical wear, fretting wear, surface fatigue wear and impact wear [4].

2.6.3 Adhesive wear

Adhesive wear mechanism occurs due to the non-homogeneities in the surfaces of the contacting bodies as a result of being manufactured by different processes. This mechanism is divided into two forms and that is moderate (mild) and severe wear.

Mild wear appears in straight lines and gentle arches in which lubrication is not used. In dry flanges, a more severe adhesive wear occurs during lubrication and this is because the tendency to form adhesive connections depends on the chemical and physical properties of the materials [4].

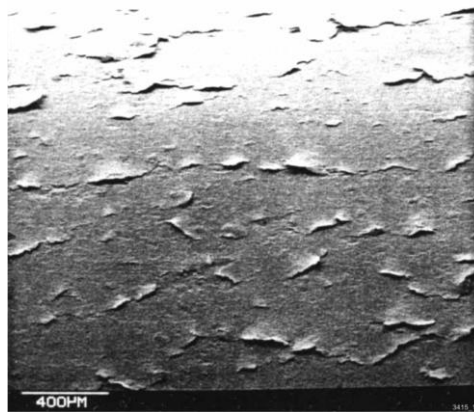


Figure 2.13: Surface resulting from mild wear [30]

2.6.4 Delamination wear

Delamination wear begins when a crack is initiated at the surface. The crack propagates under the surface until it turns up and breaks through the surface, allowing a flake of material to become detached [31]. Delamination wear is more severe than adhesive wear. It is characterized by light grey wear debris that is entirely metallic.

Delamination wear can cause microscopic wear. The wear rate can be reduced if the plastic deformation surface layer is prevented since the surface of the material is thought to be separated by the wear process. Delamination wear occurs in three steps that is by plastic deformation, germination of cracks and crack propagation. Figure 2.14 shows delamination wear mechanism.

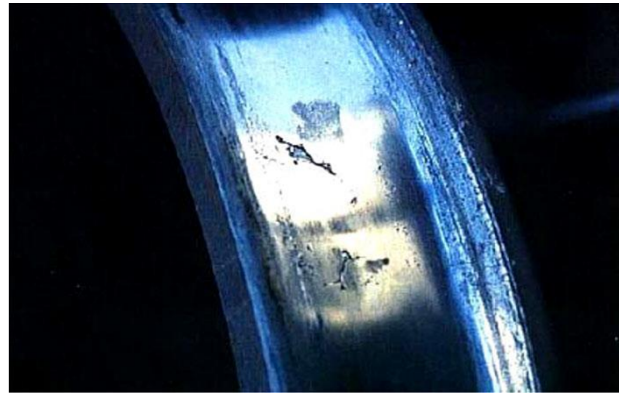


Figure 2.14: Delamination wear [4]

Due to repeated loading, a smoother surface can be easily deformed when there is contact between the sliding surfaces. The surface can relatively become smooth when the heterogeneity of the surface is modified. Plastic shear deformation is accumulated and some points are deformed at the surface and below the surface due to alternate loading of harder heterogeneity on the softer surface. The wear particle that are formed have a thickness that depends on the growth of cracks in the surface and are controlled by the vertical and tangential forces which are a result of friction [4].

2.6.5 Tribo-chemical Wear

Tribo-chemical wear results from environmental reactions, the surrounding environment can be in a state of gas or liquid. Tribo-chemical wear is seen as the reaction products of the reaction layer at the interface when wear layers are produced through a continuous separation process. Abrasion-resistant materials that are used as additives for lubricants to lower metal to metal contact friction as a way of reducing adhesive wear leads to the formation of surface layers. As a result, the use of lubricants increases chemical activity levels that happen by increasing the thickness of the protective layer.

In the process of reducing adhesive wear by increasing the thickness of the brittle surface layers, the crack tendency increases. The tribo-chemical reactions that produce a harder surface layer to reduce wear on the other hand increases abrasive wear [4].

2.6.6 Fretting Wear

Fretting wear mechanism is caused when two surfaces would have tangential and oscillating movements under the applied loads with low relative amplitudes and the slip is caused by vibratory or cyclic stresses. This phenomenon is often associated with corrosion

and oxidation. The deterioration due to fretting is affected by factors such as hardness, lubrication atmospheric environment force, the number of cycles and frequency.

2.6.7 Impact Wear

Impact wear mechanism is produced when a solid surface is continuously in contact with another solid surface. This wear mechanism in wheel/ rail is related to;

- Contact tension range: such as contact fatigue in comparison with corrosive fatigue
- Speed: increasing speed makes corrosive wear with low stress to the adhesive wear.
- Type of loading: for example, tangential load causes a slip or vertical causes impact [4].

2.7 Sources of train wheel wear/damage

2.7.3 Leaves on the Line

One of the major sources of wheel damage in temperate climates with deciduous trees is fallen leaves. Fallen leaves can really disrupt rail services across the whole world causing the scale of the leaf-fall problem and the cost of keeping services running smoothly huge.

According to [32], It is impossible to predict exactly when the leaf fall season will start and how long it will last, but the weather can provide a guide to its likely onset and how serious it is likely to be for the railway. Leaves on the line affect trains in such a way that they are like black ice on roads, they are a hard-slippery layer that coats the rails and is very difficult to remove.

Leaves are swept onto the track by the slipstream of passing trains, light rain falls and train wheels crush the wet leaves at a pressure of over 30 tons per square inch which compacts and carbonizes the leaves forming a hard, Teflon-like coating on the rails. As a result, trains have to operate at slower speeds to ensure safety and to reduce the potential for wheel slip and spin. This means that drivers have to brake earlier for stations and signals and move off again more slowly, this consequently delays train services.

If a train can't move because its wheels can't grip the rails, often there is no alternative route, therefore following trains are delayed or have to be cancelled. In addition to causing severe disruption to passengers, the damage inflicted on train wheels during sliding and

spinning on rails is considerable and means some trains have to be taken out of service for expensive repair. The rails too can be damaged, costing many thousands of money to repair each year [32].

2.7.4 Third body materials

The third body materials that appear in the wheel/rail interface by accident or due to environmental conditions, such as **water, oil, leaf, iron oxides**, among others strongly influence the adhesion of wheel/rail and cause low adhesion phenomena. Traction enhancers like **sand, alumina**, and others are often sprayed into the wheel/rail interface for improving the adhesion coefficient when the adhesion in the wheel/rail interface is poor. However, various traction enhancers can increase wear rate of wheel and rail materials and aggravate surface damage correspondingly.

Furthermore, low adhesion is unlikely to occur while thick oxides with a rough surface give an extremely high adhesion and wear. While these previous studies on causes of wear and wear prevention have been wide ranging, very specific contact conditions of wheel/rail are always adopted and there is not a wide range of wheel/rail contact information which is what is really needed for improving the wear prediction [33].

2.8 Friction in the wheel rail interface

Friction force is defined as the force acting tangentially to the interface resisting motion, when, under the action of an external force, one body moves or tends to move relative to another. The friction force F may be associated with sliding motion or with pure rolling motion of the bodies.

$$F = \mu N \quad 2.10$$

The coefficient of friction μ is a dimensionless number which is defined as the ratio F/N between the friction force F and the normal force N acting to press the two bodies together [34].

Train operation is a result of the imposed friction that is between the wheel and the rail steel surfaces. Lack of enough friction causes poor adhesion during braking and this is an issue of safety as it leads to extended stopping distances. On the side of performance,

inadequate friction affects traction in a way that delays occur when a train passes over areas with poor adhesion while in service [35].

In addition, inadequate friction of wheel/rail interface can make the wheel to slip on the rail surface during braking or traction processes and this results into serious wear and damage of the wheel/rail interface [36].

2.8.3 Adhesion in the wheel/rail contact

According to W.J. Wang, loss of adhesion in the wheel/rail contact highly influences both the traction and braking abilities of the rolling stock. If the adhesion between the wheel and the rail is poor, consequences such as wheel skidding on the rail surface occurs during the traction process and this causes wheel/rail surface damages such as skidding marks on the rail surface and wheel flats on the tread surface [37].

According to various researchers, adhesion loss is a result of coated slippery layers formed by leaves, humidity, oil, and wear particles. Adhesion is mainly enhanced by applying sand however, even though applying sand to the rail surface is effective and easy, it can cause complex and costly problems for the rolling stock and track infrastructure as it has been shown to substantially elevate the wear rates of both the wheel and rail materials [35]. According to W.J. Wang, wheel/rail adhesion under water or oil conditions is significantly lower compared with dry condition [36].

Poor adhesion may as well lead to unpredictable stopping distance, adhesion as much as is complicated, it is a very important phenomenon as regards to wheel/rail contact and it is affected by factors like water, leaves, contamination, wear debris, speed and surface roughness.

The presence of water, dew, oil contamination or leaves on the wheel/rail surface results into loss of adhesion at the contact zone. Water and oil are good lubricants and therefore they lead to significant reduction in the adhesion coefficient when applied to the wheel/rail contact surfaces. To add on, the contaminants on the contact surface of the wheel and rail could significantly affect the adhesion characteristics and can as well damage or protect the wheel/rail surfaces [37].

2.8.4 Influence of adhesion on temperature at the wheel/rail contact zone

When the traction forces or braking forces are much more than the available adhesion, wheel slip occurs which generates heat at the wheel/rail contact patch. The rise in temperature at the contact patch most likely increase the wear of both wheels and rails. If wheel slipping is extreme, the generated temperature may be very high that it can cause metallurgical changes to the material of the wheel or rail. The on subsequent quenching produces a surface layer of brittle material which even though it might be very thin, it can initiate cracks that propagate into the bulk material leading to wheel spalling and rail breakage [38].

M. A. Tanvir deduces that it is very difficult to measure the temperature at the wheel/rail contact zone and therefore theoretical predictions of the rise in temperature for different wheel loads, vehicle speed and slip are necessary. The heat flow rate problem involving a moving source of heat on a dry surface is definitely different from that on a wet surface. The rise in temperature due to a lubricated contact surface may differ from that due to a dry contact surface [38].

2.8.5 Influence of water on adhesion in the wheel/rail interface

According to G. Trummer [39], in Great Britain, several incidents like station over runs and signals passed at danger occur every year during autumn period (October-November) and these incidents are related to low adhesion conditions.

A proportion of these incidents are related to water on the rail head that is caused by the prevailing environmental conditions with most of the incidents occurring around dew point conditions in the morning and evening [39].

Experimentally, Beagley provides evidence that low amounts of water combined with iron oxides on the surface reduces adhesion in rolling contacts without the presence of other contaminants [40].

The adhesion in rolling contacts in interfacial fluids presence is governed by boundary lubrication and hydrodynamic lubrication. The shape of creep curves that is adhesion as a function of creep in dry condition differs from creep curves in wet condition and this is with respect to the adhesion level, the shape of the curve and the initial slope. Wetting the surface with water has an effect of reducing the adhesion level shifting the adhesion

maximum to higher creep values and reducing the decrease of adhesion with increasing creep force. In the presence of water, maximum adhesion values drop compared to that in dry conditions because of the low viscosity of water [39].

Researcher W. J Wang discussed under the research carried out using a JD-1 wheel/rail simulation facility on adhesion that, the adhesion coefficient of wheel/rail decreases largely under oil and water conditions with an increase in speed. When the oil contamination exists between the wheel/rail surfaces in the field, there would be very low adhesion coefficient of wheel//rail, this poor adhesion would affect both the traction and braking of the rolling stock and therefore aggravate the surface the surface damage of the wheel/rail material.

A mixture of water and oil when applied, further decreases the adhesion coefficient of the wheel/rail surfaces as compared to applying only oil. It was found that the mixture of water, oil and sand enlarges the wear width of the contact surface that is under the oil condition. Due to oil being a good lubricant, the oil condition brings no obvious surface damage due to its good lubrication performance.

The use of sand may cause small spalling spots on the wheel/rail surfaces which results into wear mechanisms when sand is applied. Fatigue wear may take place whereby large pieces of material are removed from the surface as a result of subsurface cracking initiated at the dents caused by sand fragments [37].

2.8.6 Friction under dry conditions

Wheel/rail rolling contact is highly influenced by the coefficient of sliding friction at the wheel rail interface. Friction between the wheel and the rail strongly depends on the operating conditions such as wheel load and creep as well as the state in which the surface is in [41].

High temperatures of the wheel rail contact resulting from the slip would cause thermal softening of material transition of rail material wear results from different contact conditions. For instance, in the contact between the wheel tread and the rail head, mild and severe wear are likely to occur. Large or full slip may be present when the wheel flange and rail gauge corner contacts [33].

2.8.7 Friction under wet conditions.

As it is already known, friction plays a vital role in the wear of wheel and rail and investigations and applications have indicated that, lubrication of the wheel-rail contact is an effective method for reducing the wear of wheel and rail materials.

According to the investigations carried out on wear effects of greases used in curves for lubrication, the use of a lubricant significantly changes the wear rate of wheel/rail materials and the contact conditions for wear transitions under the water condition compared to dry or other grease conditions [33].

2.8.8 The effect of friction modifiers on the wheel/rail friction and adhesion.

Friction modifiers can control the wheel/rail friction coefficient and the level of friction is significantly dependent on the amount applied to of rail (TOR) friction modifier.

The adhesion of wheel/rail is greatly influenced by the third body materials that are present in the wheel/rail interface either by accident or due to environmental conditions such as water, oil, leaves, iron oxides among others, these not only strongly influence the adhesion of wheel/rail but also cause ow adhesion phenomena.

Traction enhancers such as sand and alumina are often sprayed into wheel/rail interface when the adhesion in this interface is poor for a purpose of improving the adhesion coefficient. On the other hand, various traction enhancers can increase wear rate of wheel and rail materials and further aggravate surface damage correspondingly. In addition, low adhesion is unlikely to occur while thick oxides with rough surfaces give an extremely high adhesion and wear.

According to the experiment carried out by Wang, poor adhesion can occur when the water or oil contamination exists in the wheel-rail interface. Therefore, different friction enhancers are usually used to improve poor adhesion level to ensure adequate tractive and braking force. It was clear in the experiment that the use of sand or alumina particles obviously increases the wear rate of rollers under water conditions compared to dry conditions. The wear rate of wheel and rail materials under sand and alumina particles have a similar influence on the wear of wheel and rail [33].

2.9 Wheel/rail wear behavior

Beagley and Bolton carried out a research on wear behavior of railway steels and it was identified that a number of wear regimes existed. According to the investigation that was carried, [42] [43], it was demonstrated that two regimes existed and these were designated as mild and severe. Later research led to identification of a third regime. A designated catastrophic wear was identified beyond the severe regime. It was thought that the wheel-tread would likely fall within the mild wear regime and the wheel flange would be within the severe region, these regions were characterized in terms of wear rate and wear debris [44].

Wheel/rail contact behavior prediction is very complicated reason being its ability to entrain contaminants. The substances found within the contact can be classified into four main categories;

1. Naturally found substances; these include iron oxides or substances that are not deliberately entrained into the contact for example leaves, water or ballast dust.
2. Substances that enhance adhesion such as sand and traction gels.
3. Lubricants and these are predominantly grease for gauge face application in curves.
4. Friction modifiers; these can either be solid that is they are applied directly to the wheel or liquid which are applied to the wheel or rail [45].

Hardwick affirms that a natural third body layer always exists and it is an interfacial layer that is formed within the bounds of the wheel-rail contact under high contact pressure and rolling/sliding conditions. The Figure 2.15 illustrates a third body layer remaining on the wheel material after a twin disc test in dry condition was carried out. According to the investigations, this third body layers was suggested to be made up of iron oxides and wear debris from the wheel-rail [45].

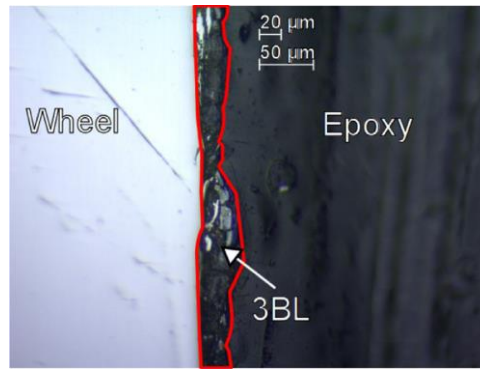


Figure 2.15: Natural third-body layer (3BL) on the surface of an R8T wheel specimen from a twin disc test run [45].

In wheel-rail contact, both rolling and sliding occur in the contact zone. The effect of sliding in the contact area induces wear under poorly lubricated conditions that are typical in wheel-rail contact. The form of the contacting surfaces of a rolling- sliding contact can be changed by wear into an unfavorable way.

Other than contact pressure, wear rate is also influenced by the size of the sliding component, lubrication micro structure and hardness. According to an observation that can be made about sliding wear is that, an increase of the severity of loading (normal load, sliding velocity or bulk temperature) leads to a sudden change in wear rate (volume loss per unit of sliding distance). The different types of wear that produce different wear rates can simply be classified into “mild wear” and “severe wear”.

Mild wear results in a smooth surface that is often smoother than the original surface. Severe wear is a form that is often associated with seizure wear. The top of the rail is always subjected to low levels of sliding but still high contact pressure on a levels of sliding and on a straight track. In this case, an oxidative mild wear process can be the dominating wear mechanism. In train dynamic simulations and detailed contact mechanisms measurement, the coefficient of friction is used as an input [11].

Due to the fact that the wheel and the rail system works in an open environment in which natural and/or artificial lubricants as well as third body operational and climatic contaminants like sand, dead leaves, ballast stones and self-produced particles are introduced inevitably, all the wear mechanisms in the wheel- rail contact have not been summarized regardless of the many systematic and thorough tests and research made the wear performance and mechanisms of the wheel-rail system [46].

3 NUMERICAL MODELLING AND SIMULATION

This section presents the Modelling of the wheelset developed using SIMPACK software package and the behavior of the wheel (pressure, stress, contact area, creepages, creep forces and wear index) under loading and frictional contact that were analyzed. In addition, this section also presented a numerical model (code) developed using MATLAB software to validate the contact pressure results determined from SIMPACK simulation.

3.1 The wheel-rail rolling contact problem

Due to the elasticity of the wheel and the rail and the externally applied load, some points on the surfaces in the contact region may slip while others may stick when the two bodies move relative to each other. The difference between the tangential strains of the bodies in the adhesion area leads to small apparent slip (creepage). Creepages generate tangential creep forces and spin moment. In the case of wheel-rail contact, tangential forces and spin moments are generated because the motion of the wheel relative to the rail is a combination of rolling and sliding.

In case of a wheel rolling on a rail, the rail coordinate system is defined by the fixed Cartesian coordinate system X_r , Y_r and Z_r , as shown in Figure 2.16. If the wheel rolls over the rail in the direction of positive X_r with a rolling velocity vector of the wheel center, the magnitude of the rolling velocity is defined as $V=|v|$. In addition, a coordinate system, $X Y Z$, which moves with the contact point can be used so that $r X = X_r - Vt$, where t is the time. In the point of contact, the wheel has a circumferential velocity c , relative to the wheel center, that is almost opposite to the rolling velocity in the point of contact. Therefore, a rigid slip can be defined as the sum of these velocities as follows: $\dot{s} = v + c$ [47]

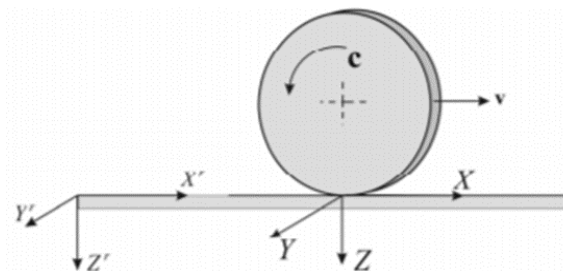


Figure 3.1: Wheel rolling over a rail [47]

3.1.1 Wheel-rail contact creepages

According to classical mechanics, the relative motion between two solids is classified in two ways namely;

1. Based on pure rolling without slipping.
2. Based on the pure slippage in which the force between the tangential two bodies reached the limit of friction, highlighting the fact that below this limit, there is no slipping.

However, there is an intermediate state where the elasticity of body contact can divide the interaction area in an adhesion area and sliding region, which have a geometry elliptic response to chart in Figure 3.2. It is appreciated that the axial axis of the elliptical adhesion zone, coincides with the axial axis of the contact ellipse, and both ellipses coincides at their most extreme point [48].

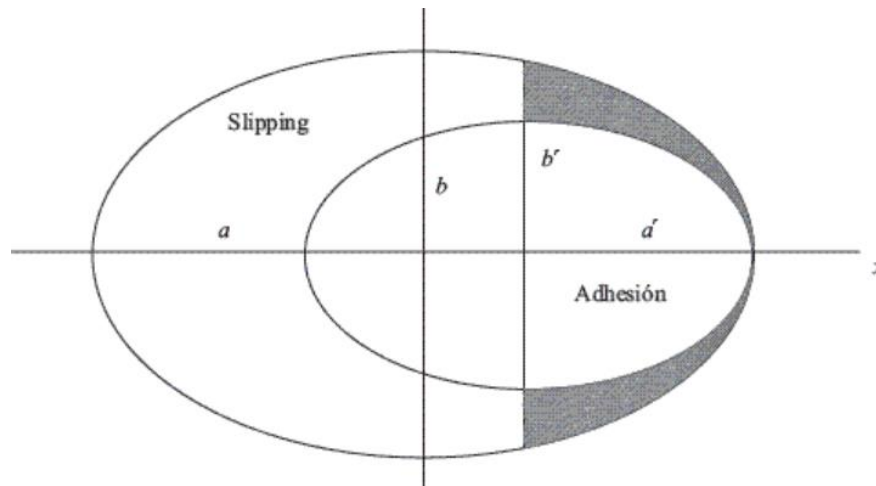


Figure 3.2: Adhesion and slipping in an elliptical area [48]

Below the level of the limiting value of friction there is a finite amount of slip between the two bodies. The magnitude of this slip can be determined by analyzing the elastical and frictional properties of the two bodies. Such slipping below the limiting value of friction is known as creepage.

According to [49], the creepages equations are calculated as given in the following equations basing on the geometry of the wheelset.

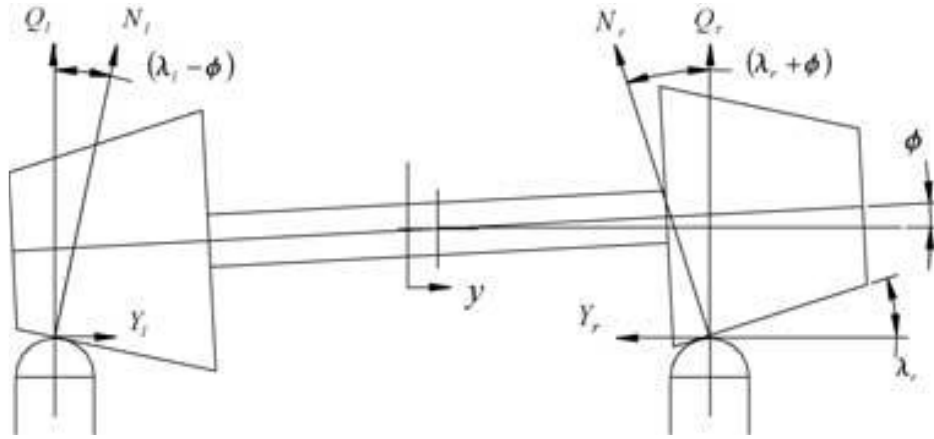


Figure 3.3: An idealized wheelset [49]

$$\text{Longitudinal creepage } \gamma_1 = \frac{v_1' - v_1}{v} \quad 3.1$$

$$\text{Lateral creepage } \gamma_2 = \frac{v_2' - v_2}{v} \quad 3.2$$

$$\text{Spin creepage } \omega_2 = \frac{\Omega_3' - \Omega_3}{v} \quad 3.3$$

where v_1 , v_2 and Ω_3 are the actual velocities of the wheel; v_1' , v_2' and Ω_3' are the pure rolling velocities (velocity when no creep occurs at the same forward velocity) calculated from the wheel motion and v is the forward velocity of the wheelset.

In terms of lateral displacement of the wheelset center of gravity, the velocities at each wheel are obtained and the yaw angle and for small displacement and conicity, the creepages for both wheels can be calculated as follow:

$$\text{longitudinal creepages on the right wheel} = \gamma_{1r} = -\frac{l_0 \dot{\psi}}{v} - \frac{\lambda y}{r_0} - \frac{l_0}{R} \quad 3.4$$

$$\text{longitudinal creepages on the left wheel} = \gamma_{1l} = +\frac{l_0 \dot{\psi}}{v} + \frac{\lambda y}{r_0} + \frac{l_0}{R} \quad 3.5$$

$$\text{lateral creepages on the right wheel} = \gamma_{2r} = \frac{\dot{y}}{v} - \psi \quad 3.6$$

$$\text{lateral creepages on the left wheel} = \gamma_{2l} = \frac{\dot{y}}{v} - \psi \quad 3.7$$

$$\text{spin creepages on the right wheel} = \omega_{3r} = -\frac{\lambda}{r_0} - \frac{\dot{\psi}}{v} \quad 3.8$$

$$\text{spin creepages on the right and left wheel} = \omega_{3l} = -\frac{\lambda}{r_0} - \frac{\dot{\psi}}{v} \quad 3.9$$

$y, \dot{y}$ are the lateral displacement and velocity of the wheelset, $\psi, \dot{\psi}$ are the yaw angle and yaw velocity of the wheelset.

3.1.2 Wheel-rail contact creep forces

It is stated in [50] that when a rail vehicle moves on a straight track, it is in equilibrium under inertia forces, normal forces, suspension forces and creep forces. Creep forces arise if wheel-axle is not rolling ideally on the track. A well-known example is accelerating or braking a car on an icy street, if the wheels are blocked during braking, they are obviously sliding on the ice. Sliding or creep more or less always exists in railway vehicles because normal and suspension forces prevent the wheel-axle from pure rolling.

In the absence of pure rolling, a contact patch is formed between wheel/rail interface, and in this contact region, both wheel surface and rail surface have difference in their sliding velocities which is termed as creepage or creep. The creep forces are generally nonlinear functions of wheel-axle displacement and velocity. Kalker's linear creep theory describes linear creep force and creepage relationship under the small creepage [50].

According to research carried out by Ivan Y. Shevtsov about Wheel/Rail interface optimization, the linear creep coefficients (small creepages) can be obtained from;

$$f_1 = \frac{8a}{3C_{11}G}, \quad f_2 = \frac{8a}{3C_{22}G}, \quad f_3 = \frac{\pi a \sqrt{a/b}}{4C_{23}G} \quad 3.10$$

Research done by Khaled E. Zaazaa and A. L. Schwab shows that Kalker determined the values of the coefficients, and as functions of the ratio of the contact ellipse semi-axes and Poisson's ratio. For two bodies with different material properties, Kalker proposed the following expressions for the combined material properties of the two bodies [47].

$$\frac{1}{G} = \frac{1}{2} \left[\left(\frac{1}{G^w} \right) + \left(\frac{1}{G^r} \right) \right] \quad 3.11$$

$$\text{Modulus of rigidity } (G) = \frac{E}{2(1 + \nu)} \quad 3.12$$

The creep forces for the lateral and longitudinal direction at each wheel are then combined to give a lateral force and a yaw torque acting on each wheelset:

$$\text{Lateral force, } Y_w = 2f_{22} \left(\frac{\dot{y}}{v} - \psi \right) - f_{23} \frac{\dot{\psi}}{v} \quad 3.13$$

$$\text{Yaw torque, } M_w = 2f_{23} \left(\frac{\dot{y}}{v} - \psi \right) - f_{23} \frac{\dot{\psi}}{v} - \frac{2f_{11}l_0^2\dot{\psi}}{v} - \frac{2f_{11}l_0}{r_0} y \quad 3.14$$

The general equation of motion for the wheelset obtained by the combination of creep force and gravitational stiffness with the inertia force terms is denoted as 3.15 and 3.16. The wheelset is taken as having the two degrees of freedom of lateral translation motion and yaw rotation:

$$m\ddot{y} + 2f_{22} \left(\frac{\dot{y}}{v} - \psi \right) + f_{23} \frac{\dot{\psi}}{v} + \frac{w\delta_0 y}{l} = 0 \quad 3.15$$

$$I_z\ddot{\psi} - 2f_{23} \left(\frac{\dot{y}}{v} - \psi \right) + 2f_{33} \frac{\dot{\psi}}{v} + \frac{2f_{11}y\delta_0 l}{r_0} + \frac{2f_{11}l^2\delta_0\dot{\psi}}{v} = 0 \quad 3.16$$

These equations are used in simulations of complete railway vehicles according to [49]. This further notes that the creepage creep/force relationship is further complicated by the fact that the three creepages do not act independently and according to Kalker, the creep forces depend on the creepages as follows:

$$Fx = -f_{11}\gamma_1\theta \quad 3.17$$

$$Fy = -f_{22}\gamma_2 - f_{23}\omega_3 \quad 3.18$$

$$Mz = f_{23}\gamma_2 - f_{33}\omega_3 \quad 3.19$$

According to Johnson and Vermeulen, the tangential forces Fx and Fy can further be modified as Fx' and Fy' .

$$Fx' = \frac{Fx}{Fs} \left[\frac{Fs}{\mu N} - \frac{1}{3} \left(\frac{Fs}{\mu N} \right)^2 + \frac{1}{27} \left(\frac{Fs}{\mu N} \right)^3 \right] \mu N \quad 3.20$$

$$Fy' = \frac{Fy}{Fs} \left[\frac{Fs}{\mu N} - \frac{1}{3} \left(\frac{Fs}{\mu N} \right)^2 + \frac{1}{27} \left(\frac{Fs}{\mu N} \right)^3 \right] \mu N \quad (\text{for } Fs \leq 3\mu N) \quad 3.21$$

$$\text{And } Fx' = \mu N \frac{Fx}{Fs}, \quad Fy' = \mu N \frac{Fy}{Fs} \quad (\text{For } Fs > 3\mu N), \quad 3.22$$

$$\text{Where } Fs = (Fx^2 + Fy^2)^{\frac{1}{2}} \quad 3.23$$

μ = the coefficient of friction at the contact patch, N = the normal force at the contact patch, Fx is the longitudinal creep force, Fy is the lateral creep force, ξ is the longitudinal creepage, η is the lateral creepage and θ is the yaw angle

3.2 Wear prediction model.

Wheel/rail wear depends on the creep forces and creepages at the contact patch. There are many factors that influence creep forces and creepages and these parameters depend on a large number of interdependent factors like, wheel/rail friction coefficient, wheel profiles (initial profile and state of wear), rail profiles (initial profile and state of wear), traction and braking forces, material properties of wheel and rail, track geometry quality, vehicle configuration (axle load, wheel diameter, wheelbase), suspension design in particular primary yaw stiffness, curve radius and cant deficiency (depends on speed, curve radius and cant) [51].

Pioneering work was carried out on British Rail in order to research, understand and model wheel/rail wear behavior. Empirical studies by McEwen et al; and Pearce et al; both on a full-scale laboratory test rig and in the field have shown that wheel/rail wear depends on the rate of dissipation of energy within the contact patch. It was concluded that wheel wear rate could be related to the frictional energy expended through creepage in each wheel/rail contact.

This therefore shows that wheel wear index is the sum of the products of the individual creep forces and creepages in the longitudinal and lateral directions. In most cases, the contribution from the spin term is assumed to be small and is ignored. [52]. Wheel wear is therefore estimated using the wear index W that is taken from the English Normative (British Rail) that reads [51]:

$$W = F_x \cdot \xi + F_y \cdot \eta \quad 3.24$$

Inconsideration of Johnson and Vermeulen modification of tangential forces, wheel wear index (W) can be obtained from British Rail normative as:

$$W = F_{x'} \cdot \xi + F_{y'} \cdot \eta \quad 3.25$$

3.2.1 Archard's wear model

Many quantitative models have been developed for sliding wear but one of the simplest is Archard's which is very useful for comparing wear rates and material behavior under different conditions [34].

The Archard wear equation to examine the phenomenon of adhesion is presented as follows.

Wear rate, W_{ad}

$$W_{ad} = \frac{V}{S} = K \frac{F_N}{H} \quad 3.26$$

In the above relation, W_{ad} is wear rate (volume per unit of distance traveled worn sliding), K is the coefficient of wear which is sometimes known as specific wear rate, V (m^3) is the worn volume, S (m) is the sliding distance, F_N (N) is the vertical or normal load and H (N/m^2) is the hardness of the softer material [4].

Wear volume, V_{wear}

According to Archard's wear model, the wear volume wear V_{wear} (m^3) can be calculated with the equation:

$$V_{wear} = K \frac{S \cdot F_N}{H} \quad 3.27$$

The depth of wear can be calculated from;

$$\Delta z = K \frac{S \cdot P}{H} \quad 3.28$$

Where P is the contact pressure (N/m^2) [53].

Using the quantitative assessment of wear

Wear resistance, (m^{-3})

$$wear\ resistance = \frac{1}{wear\ amount} = \frac{1}{wear\ volume} \quad 3.29$$

From Tables, [16] [54] the hardness (H) of R8T wheel steel is 2.6 GPa

Discussion of wear on a macro scale can as well be started with Archard's equation which states;

$$W = K \times s \times P \quad 3.30$$

where W is the worn volume, s is the sliding distance, P is the applied load and K is the wear per unit load per sliding distance. According to Archard, K may be described as the coefficient of wear and, in experimental series with the same combination of materials, changes in K denote changes in surface conditions [55].

3.3 AALRT Vehicle Specifications and Environment Conditions

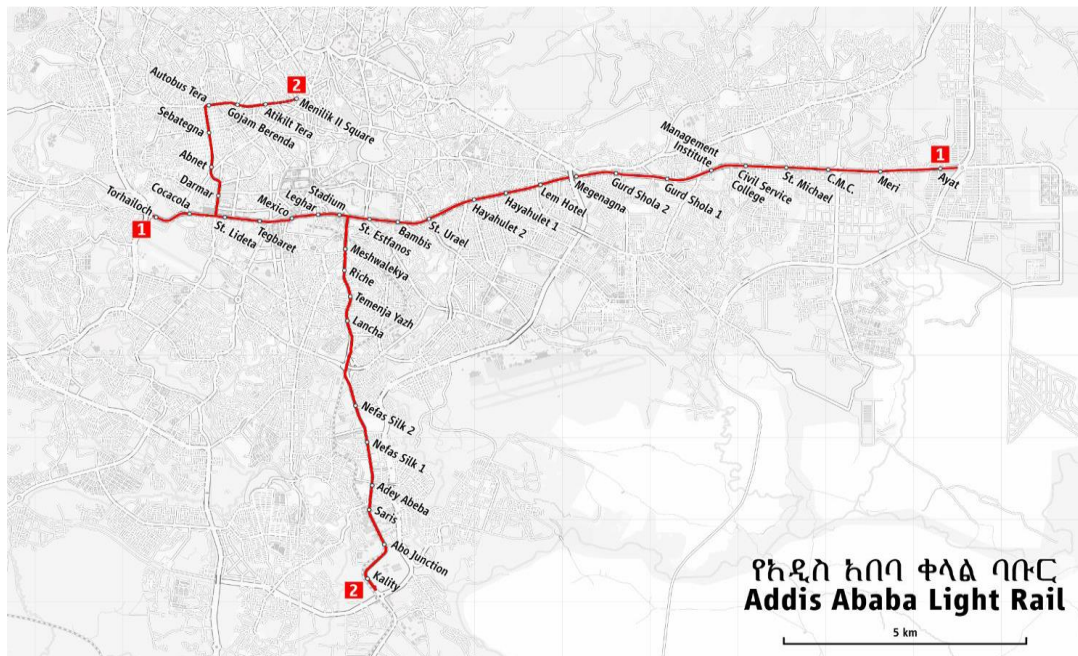


Figure 3.4: AALRT line Map showing the two main routes [Source: Ethiopian Railway Corporation]

AALRT vehicles offer transportation services in two main lines, namely;

1. East-west line (from Ayat to Torhayiloch)
2. North-South line (Kaliti to Menilik II square)

The length of the first line is 16.99 km and the second is 16.689km long, the total length of the track is 31.025 Km. A section is shared by both lines and it is about 2.662km. AALRT track gauge is 1435mm (standard gauge). The minimum radius of the horizontal curve for mainlines between sections is 50 m and for yard line the radius goes down to 30m. The minimum radius of vertical curve for the lines is 1000 m while the maximum gradient is about 55%. The main line and the depot rail are 50 kg/m with a maximum

super-elevation of 120 mm and inclination at rail bottom 1/40. The axle load is ≤ 11 ($1 \pm 3\%$) tons.

3.3.1 AALRT vehicle configuration and dimensions.

AALRT vehicle consists of three cars that is, two motor cars (Mc) with a driver's cab and a trailer car (Tc), this is positioned in the middle of the motor cars as shown in the figure below.

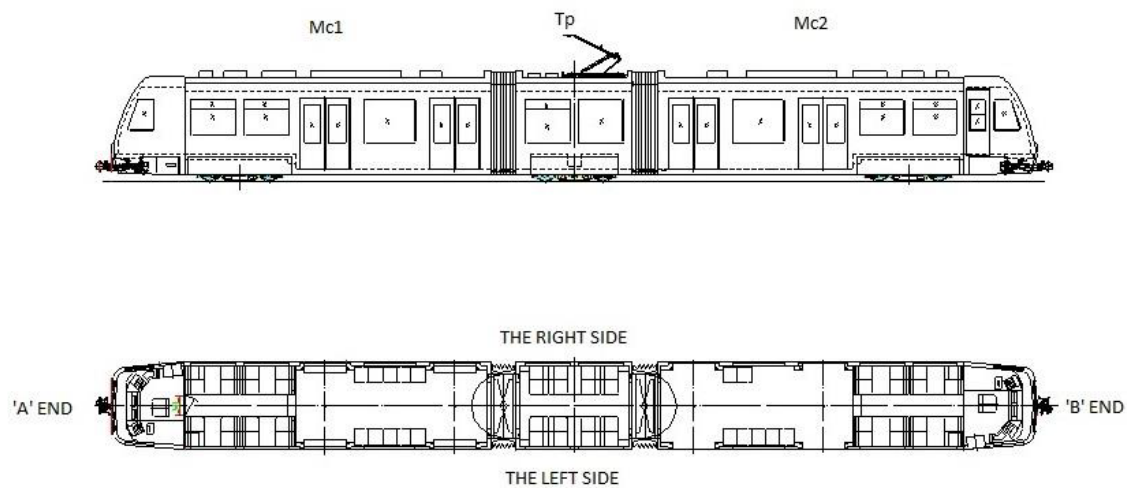


Figure 3.5: AALRT vehicle configuration [Source: Ethiopian Railway Corporation]

Important dimensions for AALRT vehicle

Table 3.1: AALRT vehicle dimensions

Specification	Dimensions
Length of car body	≤ 30000 mm
Height of vehicle roof from top of rail (excluding pantograph)	≤ 3700 mm
Maximum car body width	2650mm
Height of vehicle floor from top of rail (low floor area, new wheels and empty load)	≤ 380 mm
Height of vehicle floor from top of rail (exit and entry areas, new wheels and empty load)	≤ 350 mm
Height of vehicle floor from top of rail (raised floor area, new wheels and empty load)	≤ 900 mm
Fixed wheelbase	1900 mm

Wheel diameter	660 mm (New wheel) 580 mm (worn out wheel)
Axle load	≤ 11 ($1 \pm 3\%$) t.
Track gauge	1435mm
Maximum operation speed	70 km/hr
Average travelling speed	≥ 20 km/hr
Maximum test speed	80 km/hr
Vehicle ride quality	< 2.5
Derailment coefficient	< 0.8

Source: Ethiopian Railway Corporation.

Table 3.2: AALRT wheelset and bogie properties (moment of inertia)

	Mass (kg)	I_{xx} (kgm^2)	I_{yy} (kgm^2)	I_{zz} (kgm^2)
wheelset	880	176	76	176
Bogie frame	4200	1215	1875	2182

Source: Ethiopian Railway Corporation.

3.3.2 AALRT vehicle seating capacity and weight.

Tables 3.3 and 3.4 below show the seating capacity and weight of AALRT vehicle

Table 3.3: Vehicle seating capacity

Capacity	Seated (person)	Standing (person)	Total (Person)
Seats (AW1)	65	0	65
Seating capacity (AW2) (standing: 6 persons/ m^2)	65	189	254
Overload capacity (AW3) (standing: 8 persons/ m^2)	65	252	317

Source: Ethiopian Railway Corporation.

Table 3.4: Vehicle weight in tons

loading conditions	Car body weight (ton)	Passenger weight (ton)	Total weight (ton)
Empty vehicle	44	0	44
Seating capacity	44	15.24	59.24
Overload capacity	44	19.02	63.02

Source: Ethiopian Railway Corporation

Table 3.5: Natural environment in Addis Ababa region

Altitude	≤2500m
Ambient temperature	0°C~+29.7°C
Average daily highest temperature in years	25.5°C
Average daily lowest temperature in years	6.1°C
Average relative humidity in years	95%
Average annual rainfall	1000-1600mm
Maximum daily rainfall	47mm

Source: Ethiopian Railway Corporation.

3.3.3 Applied wheel load for model simulation

The applied load is the normal force applied on the wheel in the downward direction. The wear investigation was carried out with consideration of vehicle overload capacity. From the total of persons inside the tram and by taking average of 60kg/person and given six (6) number of axles, carrying capacity is calculated as;

Total carrying capacity of the train

$$= \text{total of persons inside the train average individual weight} \quad 3.31$$

$$= 317 \times 60 = 19.02 \text{ t}$$

Total train weight = empty car body weight + total train carrying capacity

$$\text{Total train weight} = 44\text{t} + 19.02\text{t} = 63.02\text{t}$$

$$\text{Total axle load} = \frac{\text{total train weight}}{\text{number of axles}} \quad 3.32$$

$$\text{Total axle load} = \frac{63.02t}{6} = 10.503(1 \pm 3\%)t$$

$$\text{Total axle load} = 10.82t$$

Applied load on each wheel.

$$\begin{aligned} \text{Applied load on each wheel} &= \frac{\text{Total axle load}}{\text{number of wheels on the axle}} \\ &= \frac{10.82t}{2} \end{aligned} \quad 3.33$$

$$\text{Applied load on each wheel} = 5.41t$$

$$\text{Applied load on each wheel} = 5.41 \times 9.81 \times 1000 = 53072 \text{ N}$$

3.3.4 Rotational Velocity of vehicle

Given the maximum operating speed of the vehicle as 70Km/h, the angular velocity of the wheel at maximum operating speed is:

$$\omega = \frac{V}{R_w} \quad 3.34$$

Where: V = maximum operating speed of the vehicle

R_w = Rolling radius of the wheel

$$\omega = \frac{70 \text{ Km/hr}}{0.33m} = \frac{70 \times 1000}{3600 \times 0.33}$$

$$\omega = \frac{19.44 \text{ m/s}}{0.33 \text{ m}} = 58.92 \text{ rad/s}$$

3.3.5 Applied rotational velocity for model simulation

The simulation was carried out with consideration of 35Km/hr. and 20 Km/hr. as operating speeds. The angular velocity (ω) of the wheel at this operating speed is therefore:

$$\omega_{@35 \text{ km/hr}} = \frac{35 \frac{\text{Km}}{\text{hr}}}{0.33 \text{ m}} = \frac{35 \times 1000}{3600 \times 0.33}$$

$$\omega = \frac{9.72 \text{ m/s}}{0.33 \text{ m}} = 29.46 \text{ rad/s}$$

$$\omega_{@20 \text{ km/hr}} = \frac{20 \text{ Km/hr}}{0.33 \text{ m}} = \frac{20 \times 1000}{3600 \times 0.33}$$

$$\omega = \frac{5.56 \text{ m/s}}{0.33 \text{ m}} = 16.84 \text{ rad/s}$$

3.3.6 Coefficient of friction at the wheel/rail contact patch

The coefficient of friction between the wheel and the rail according to [56] has to be provided by the user/operator but this coefficient depends highly on parameters like rolling velocity, normal load, microscopic surface roughness of the wheel and rail among others. The influence of these parameters strongly depends on the lubrication state of the contact that is to say whether the contact is humid (boundary lubricated) or is in a dry or mixed state where there is enough fluid present in the wheel/rail contact to build a positive fluid pressure regime [56]. The friction coefficient on the wheel/rail interface also varies significantly due to environmental conditions.

The Coefficient of Friction (CoF) between wheel and rail varies in operation between a minimum of almost zero and a maximum of up to 0.8 (the steel to steel CoF (static) is 0.8 for dry conditions and 0.16 for lubricated conditions). The recommended or ideal adhesion coefficient μ in the contact area between wheel and rail on the running surface is between 0.30 and 0.35, whereas the value at the wheel flange/gauge corner contact should be kept as small as possible [57].

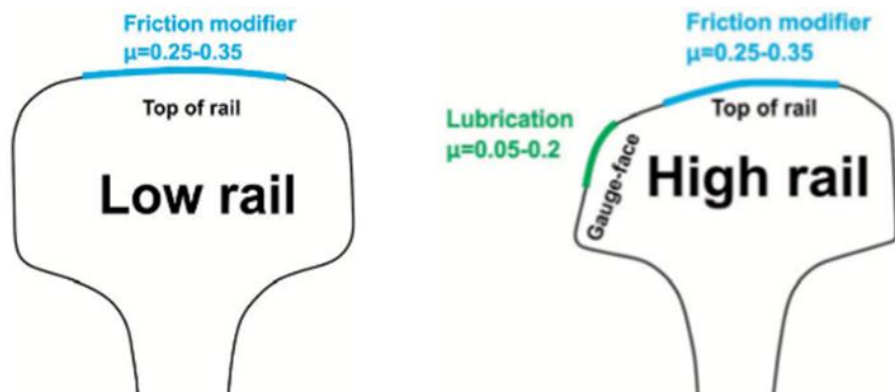


Figure 3.6: Friction management targets for low and high rails [57]

Effective friction modifiers for railway applications must be able to produce the required top of rail (ToR) intermediate CoF of 0.35–0.4 by reducing friction from a high-friction state to a controlled friction level in order to ensure the achievement of appropriate adhesion values and to maintain lateral forces at safe levels [57].

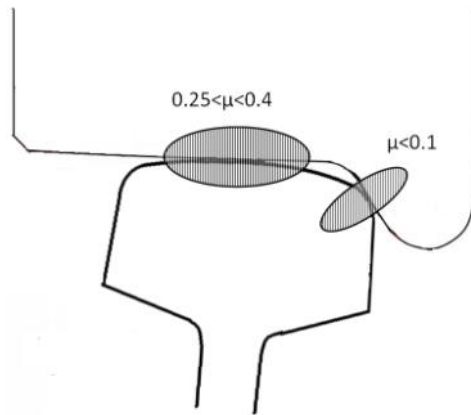


Figure 3.7: Ideal friction coefficients in the wheel–rail contacts [58]

Simulation cases for selected coefficient of friction at the wheel/rail contact

The CoF for simulation of AALRT vehicle operation under dry and wet contact condition was considered following the research and studies carried out by various researchers on friction and adhesion of wheel/rail [57] [58] [59].

Table 3.6: Friction coefficients measured using a hand-push tribometer [58]

Conditions	Temperature (°C)	Friction coefficient
Sunshine dry rail	19	0.6–0.7
Recent rain	5	0.2–0.3
Substantial grease on rail	8	0.05–0.1
Damp leaf film on rail	8	0.05–0.1

Table 3.7: Simulation CoF values

	Conditions	Coefficient of friction (μ)
Case 1	Dry contact	0.4, 0.45, 0.5, 0.55
Case 2	Wet contact	0.25, 0.3, 0.35

3.4 Geometry Modelling

Following the constraints on computational hardware and software, compromises are usually made in order to decide on an optimum number of elements or parts that should be used. The geometry therefore needs not to be over defined and should be modelled only if very accurate results are required for those regions. With these geometric approximations in mind, the simulation results can be interpreted [60]. As a result, a simple wheelset model was used to simulate the wear behavior of the wheel using SIMPACK software.

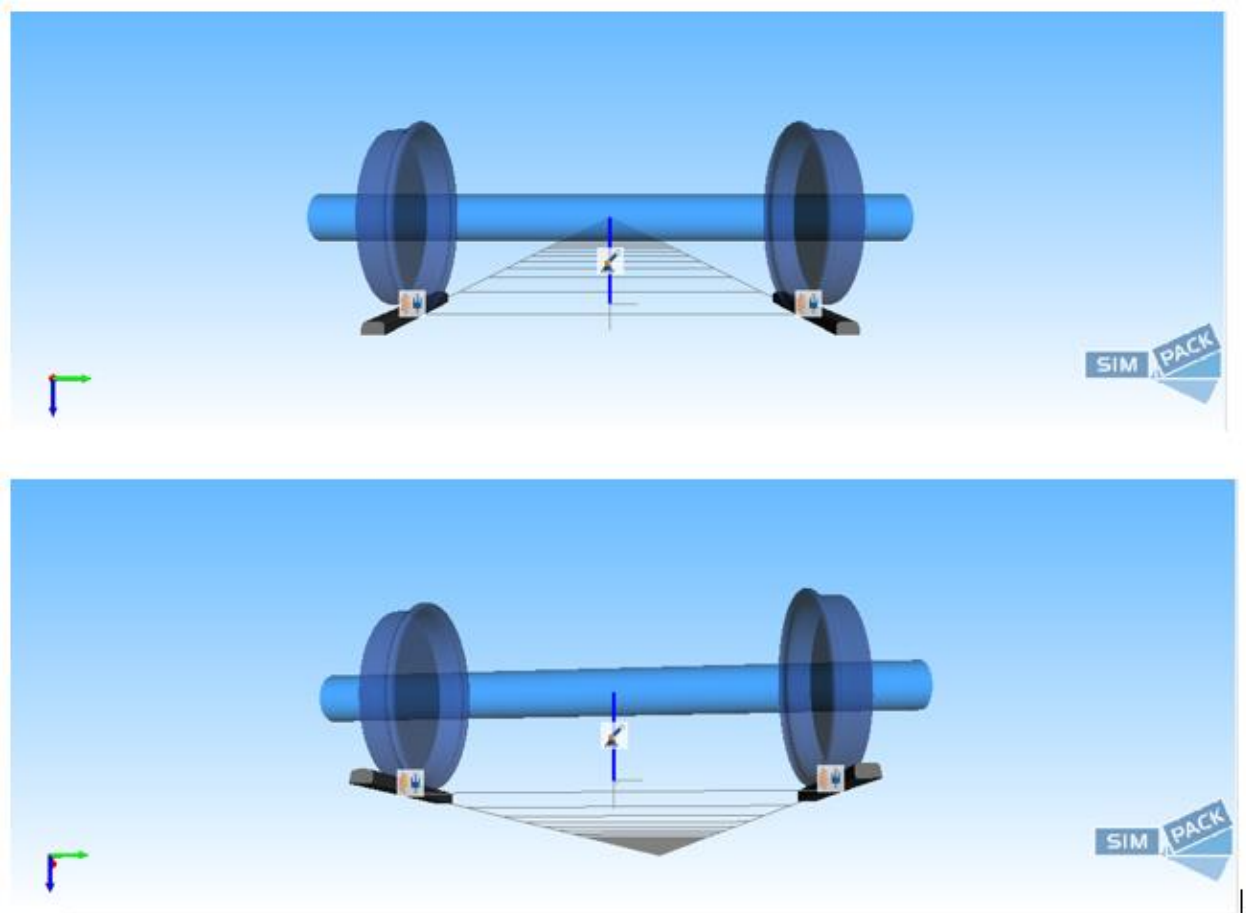


Figure 3.8: Wheelset model views

4 RESULTS AND DISCUSSION

4.1 Effect of coefficient of friction on wear

The effect of CoF was simulated in SIMPACK by varying the friction coefficient from 0.25 to 0.55 with variations of 0.25, 0.3, 0.35, 0.4, 0.45, 0.5 and 0.55. The longitudinal creepage, lateral creepage, longitudinal creep force and lateral creep force were obtained from the simulation for each coefficient of friction as depicted from Figures 4.1 to 4.7.

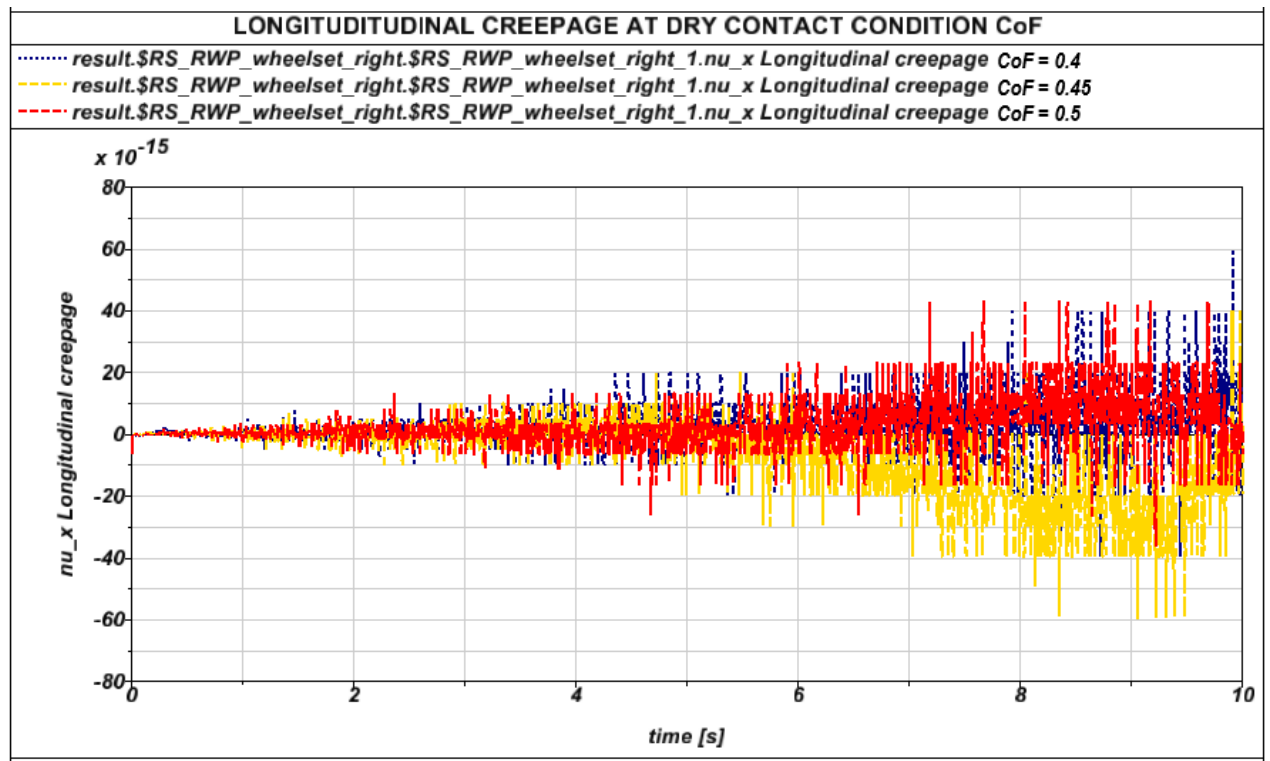


Figure 4.1: Longitudinal creepage at different CoF values for dry contact condition.

The Figures 4.1 and 4.2 showed that the longitudinal creepage increased with increase in the coefficient of friction, in other words, a higher CoF value led to higher longitudinal creepage and a lower CoF value led to a lower creepage value. As observed from the Figure 4.1, dry contact condition induced a higher longitudinal creepage and lower longitudinal creepage values were observed from the wet contact condition as showed in Figure 4.2.

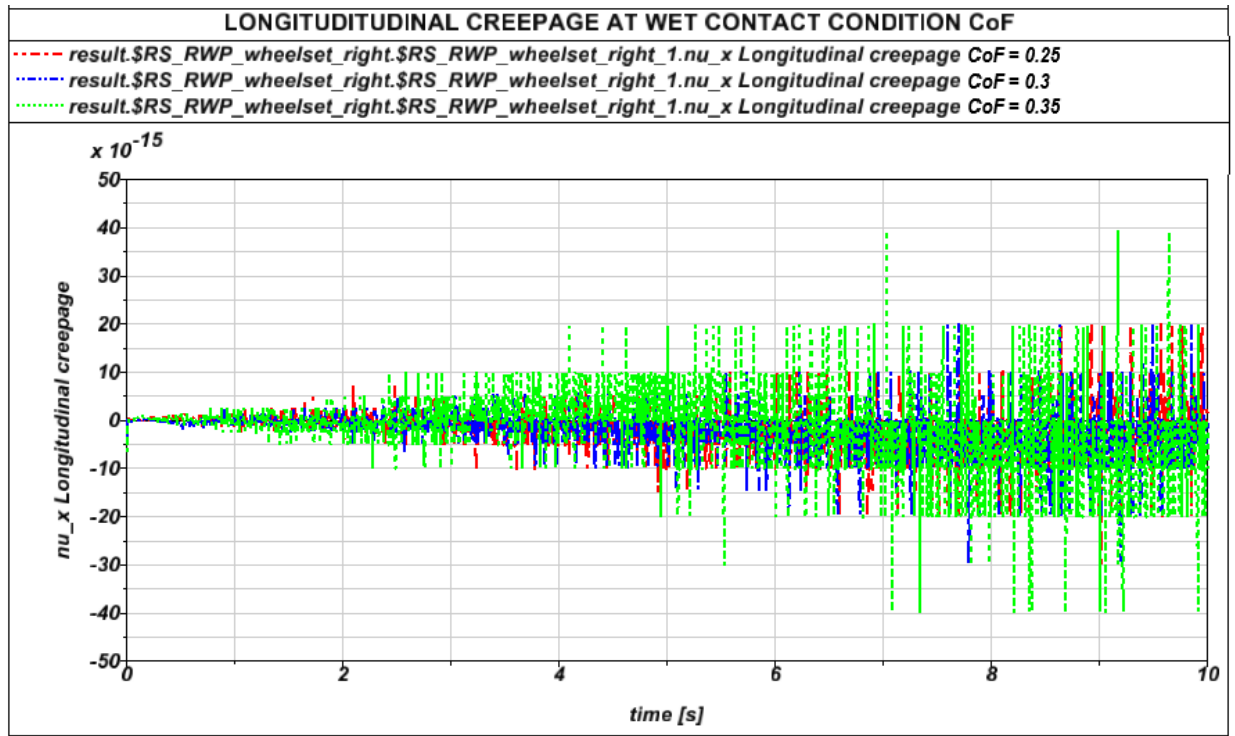


Figure 4.2: Longitudinal creepage at different CoF values for wet contact condition.

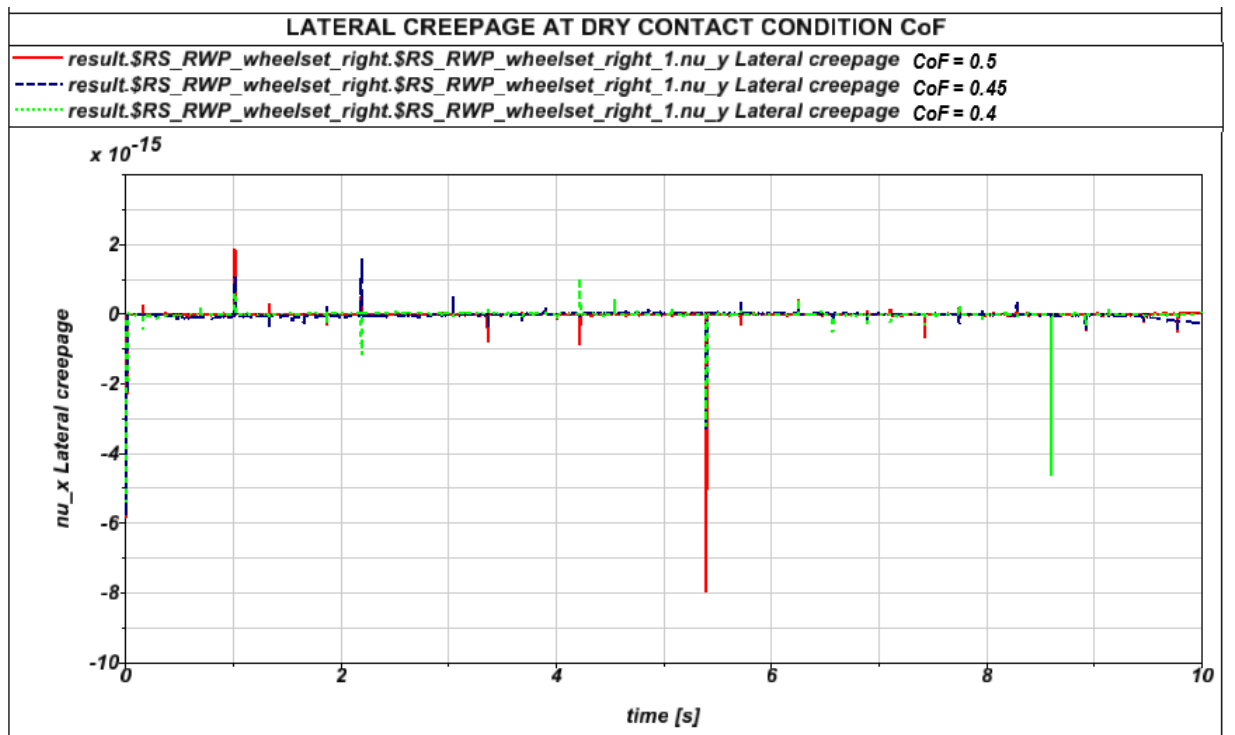


Figure 4.3: Lateral creepage at different CoF values for dry contact condition.

According to Figure 4.3, the highest lateral creep force is observed at a time of 5.4 seconds because the train assumes a velocity/time relationship in such a way that it starts from rest

at zero speed and attains maximum speed at half the distance of the total journey, it thereafter starts decelerating until it comes to stop at the end of the journey with zero speed.

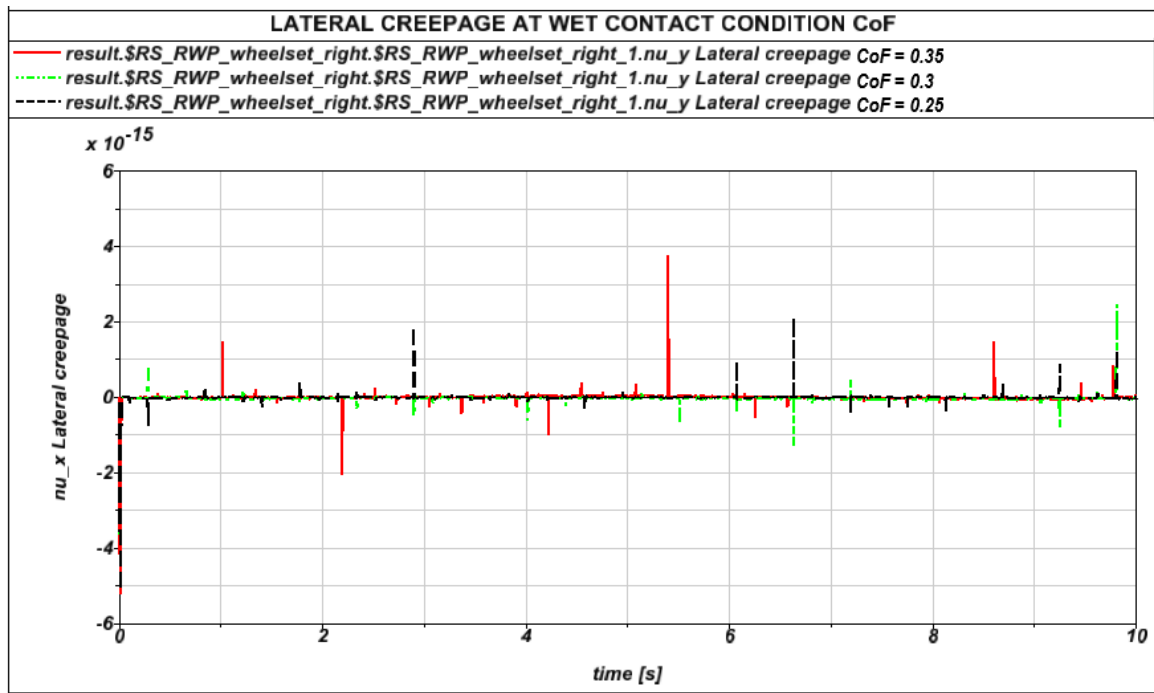


Figure 4.4: Lateral creepage at different CoF values for dry contact condition.

As observed from the Figure 4.3 and 4.4, dry contact condition induced a higher lateral creepage and lower lateral creepage values were observed from the wet contact condition. It was seen in Figure 4.3 that a higher CoF value led to higher lateral creepage and a lower CoF value led to a lower lateral creepage as shown in Figure 4.4 during wet contact.

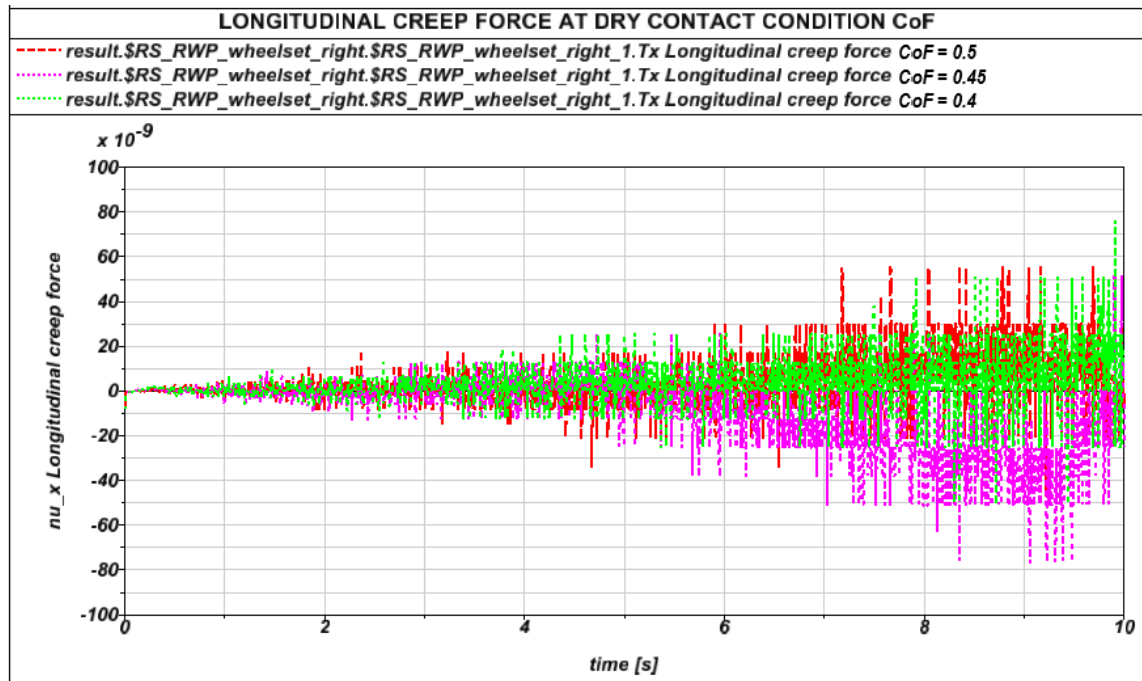


Figure 4.5: Longitudinal creep force at different CoF values for dry contact condition.

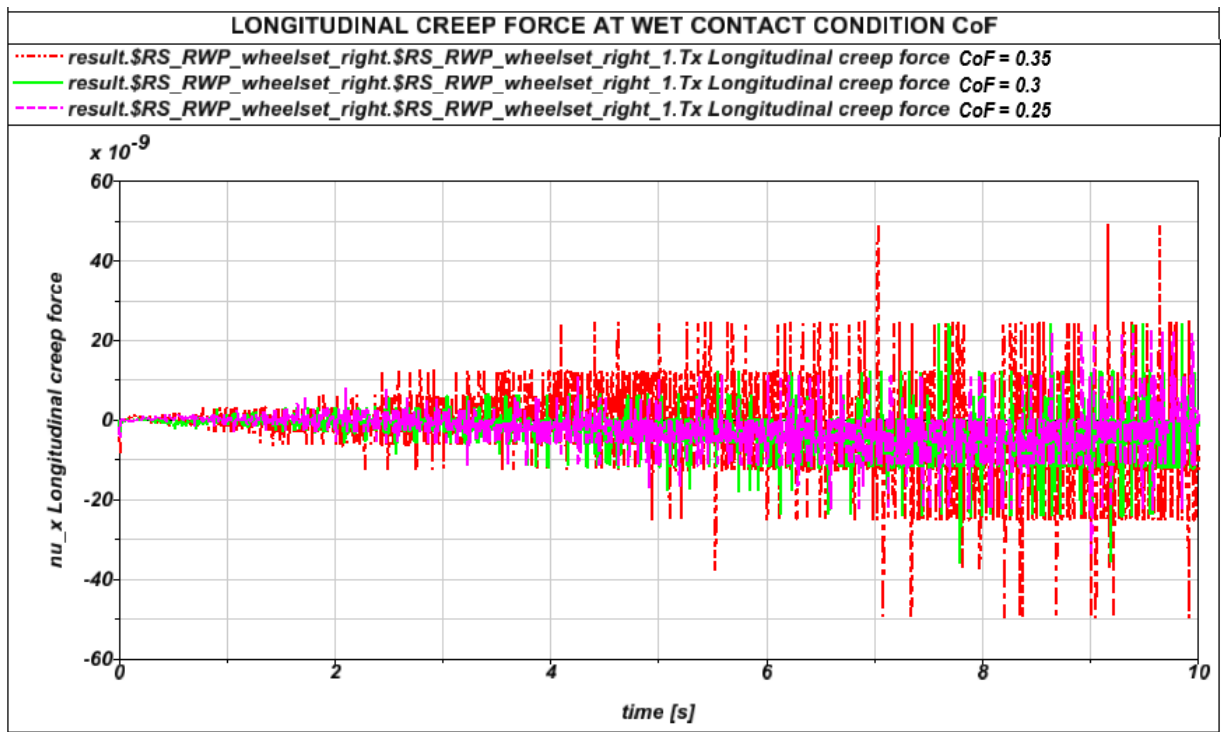


Figure 4.6: Longitudinal creep force at different CoF values for wet contact condition.

Figures 4.5 and 4.6 illustrate the longitudinal creep force relationship with the CoF, it was observed that at higher CoF values, the longitudinal creep force was high and at lower CoF

values, the longitudinal creep force was relatively low. It was observed the longitudinal creep force less at wet contact and high at dry contact.

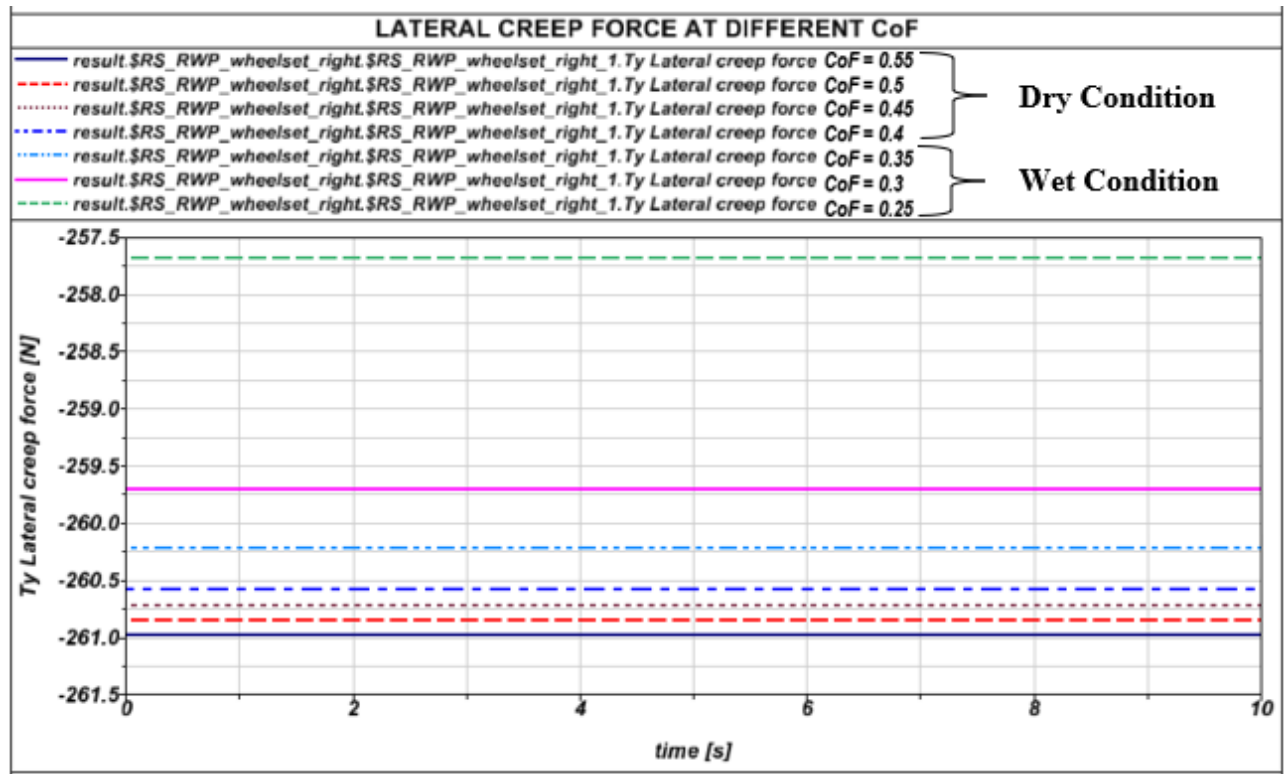


Figure 4.7: Lateral creep force at different CoF values

An observation was made as shown in Figure 4.7 that lateral creep force from wet contact was relatively low compared to that from dry contact. It was observed that higher CoF values induce higher lateral creep force and lower CoF values induce lower lateral creep force.

The wear behavior was calculated by exporting the wear number data obtained from the simulation results and summarized in Figure 4.8. The Figures 4.9, 4.10 and 4.11 show the wear number, wear volume, wear rate and wear depth against time respectively for varying coefficients of friction at speed of 35 km/hr. It was seen that there was an increase in the wear behavior with increase in the CoF.

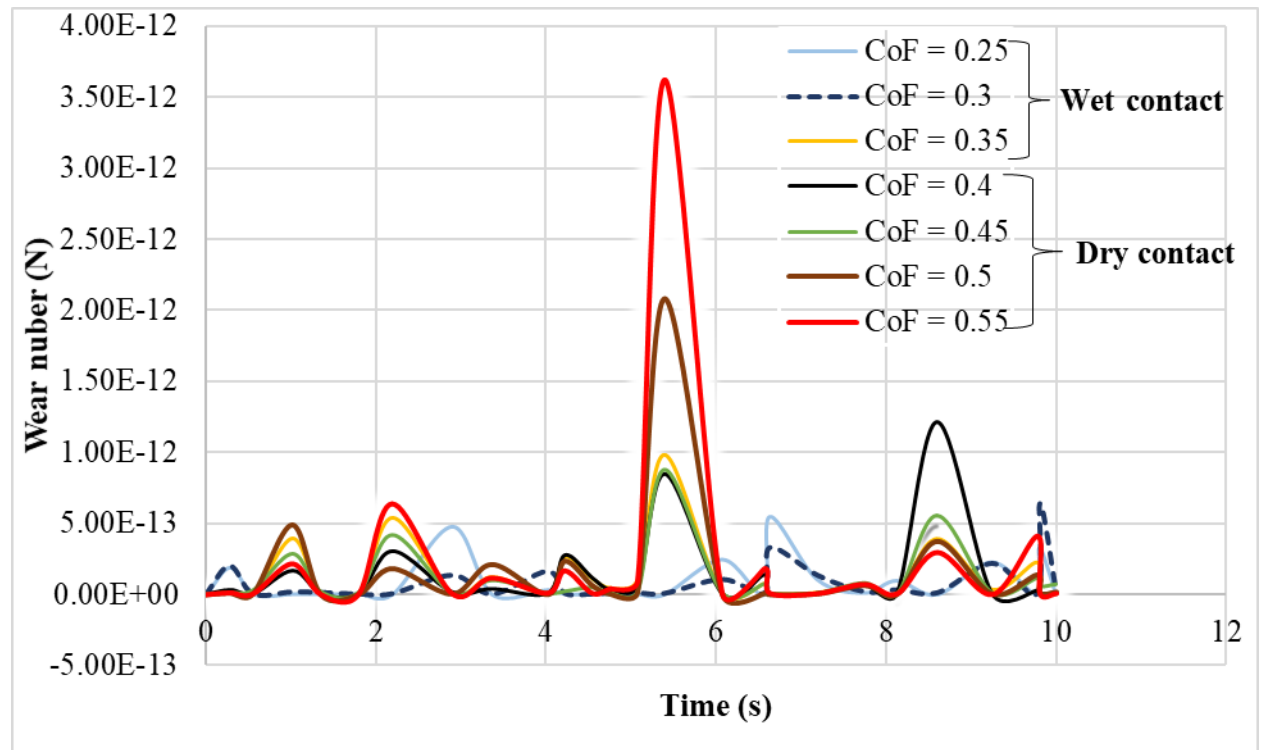


Figure 4.8: Wear number against time at different CoF values

Figure 4.8 shows that the higher wear number or index was obtained at high friction coefficient values (dry contact) and low CoF at wet contact values led to a low wear number/index.

According to the Figure 4.8, the highest wear number is observed at a time of five (5) to six (6) seconds because the train assumes a velocity/time relationship in such a way that it starts from rest at zero speed and attains maximum speed at half the distance of the total journey thereby causing maximum wear to occur at this point of time, it thereafter starts decelerating until it comes to stop at the end of the journey with zero speed. This also applies to Figures 4.9 to 4.11.

The above wear number results agree to the wear number results obtained by researchers M. Ansari, I. A. Hazrati, E. Esmailzadeh and S. Azadi on their research carried out on

Wear rate estimation of train wheels using dynamic simulations and field measurements [61].

The wear behavior was obtained as shown in the Figures 4.6 - 4.8 for different coefficients of friction on dry and wet sliding contact.

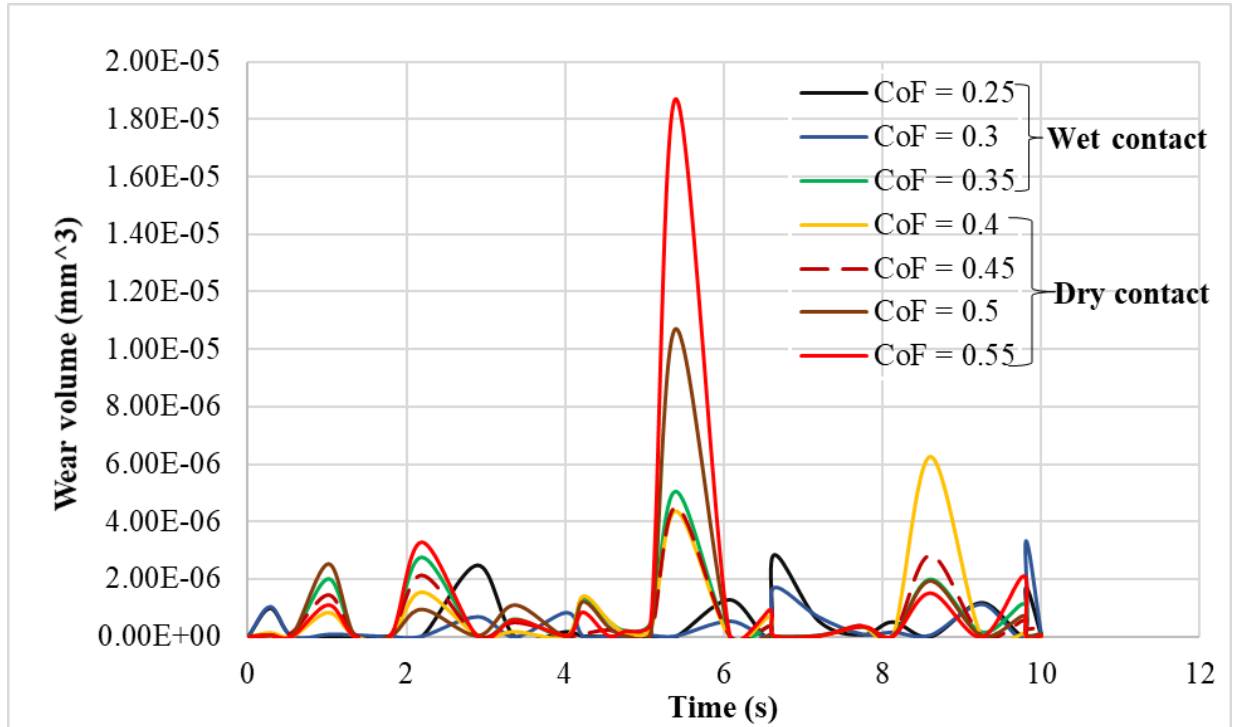


Figure 4.9: Volume of worn material against simulation time at different CoF values

Figure 4.9 shows that the highest wear volume of $1.87 \times 10^{-5} \text{ mm}^3$ resulted at the highest coefficient of friction of 0.55 during dry sliding and the lowest wear volume of $2.84 \times 10^{-6} \text{ mm}^3$ resulted at the lowest coefficient of friction of 0.25 during wet sliding contact condition.

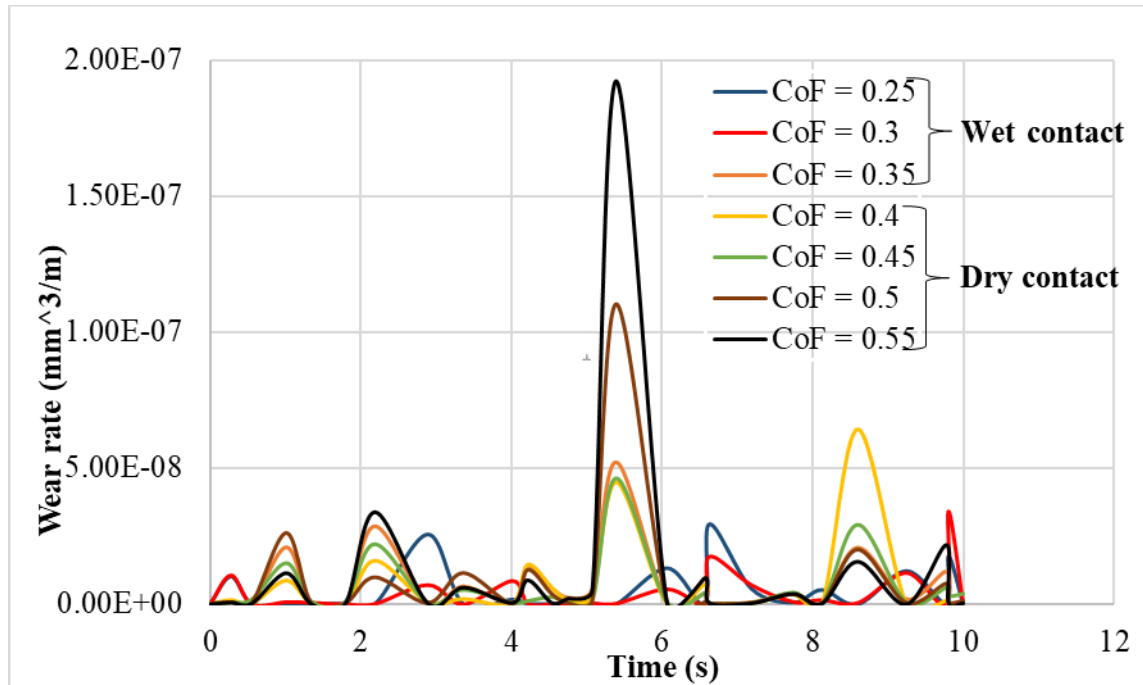


Figure 4.10: Wear rate against simulation time at different CoF values

Figure 4.10 shows that the highest wear rate of 1.92×10^{-7} occurred at the highest CoF of 0.55 during dry sliding contact condition and the lowest wear rate of 2.92×10^{-8} resulted at the lowest CoF of 0.25 during wet sliding contact condition.

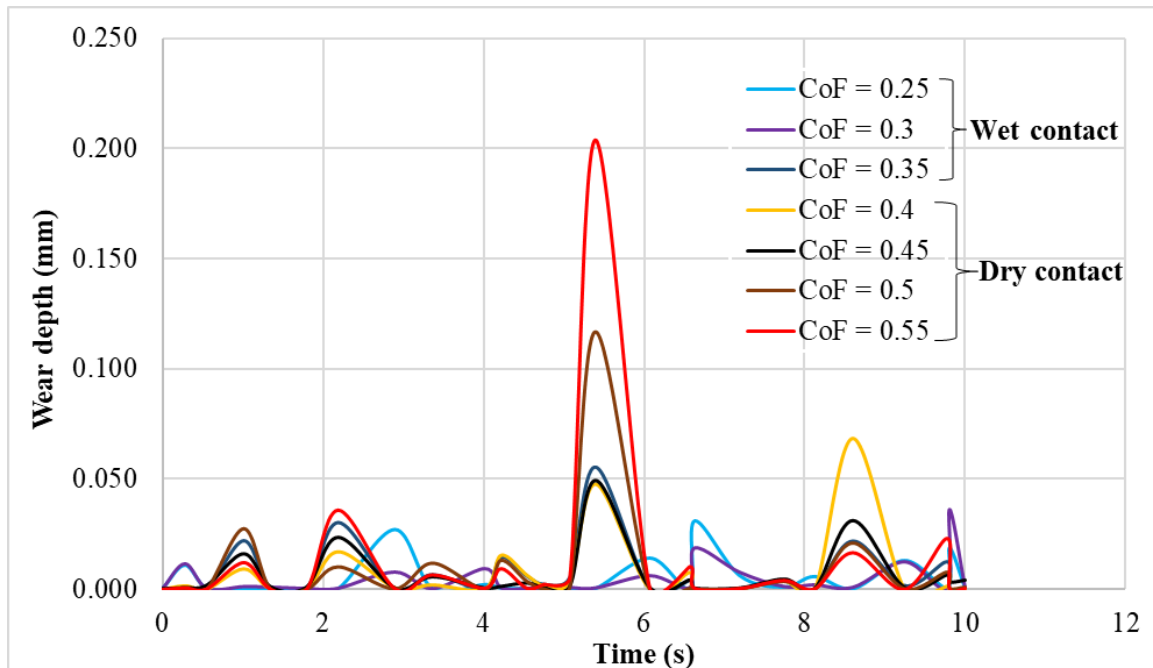


Figure 4.11: Wear depth against simulation time at different CoF values

As observed in Figure 4.11, the depth of wear during wet sliding contact is low and it is high during dry sliding contact. Highest wear depth of 0.204×10^{-1} mm resulted during dry sliding at CoF of 0.55 and the lowest depth of wear of 3.1×10^{-2} mm resulted during the lowest wet sliding CoF of 0.25. In general, the CoF is a very important factor in wheel/rail interaction as it critically influences the wear behavior of train wheels.

4.2 Effect of CoF with varying speed on wheel wear.

The effect of CoF with speed variation was simulated in SIMPACK for speeds of 20 Km/hr. (5.56 m/s) and 35 km/hr. (9.72 m/s) at CoF of 0.25 and 0.35 for wet contact condition, 0.5 and 0.55 for dry contact condition. The longitudinal creepage, lateral creepage, longitudinal creep force and lateral creep force were obtained from the simulation for each coefficient of friction and can be seen in appendix C. The wear behavior was calculated by exporting the wear number data obtained from the simulation results to excel as summarized in Figure 4.12 to 4.17. The Figures 4.12 and 4.13, 4.14 and 4.15, 4.16 and 4.17 show the wear volume, wear rate and wear depth against time a wet and dry condition respectively for varying coefficients of friction and speed of 20 and 35 km/hr. It was evidenced that higher longitudinal and lateral creepages appeared at higher coefficient of friction and higher speed. In addition, lower coefficient of friction at the wheel/rail interface and lower speed resulted into lower longitudinal and lateral creepages. It was seen that there was an increase in the wear behavior with increase in the CoF and increase in speed.

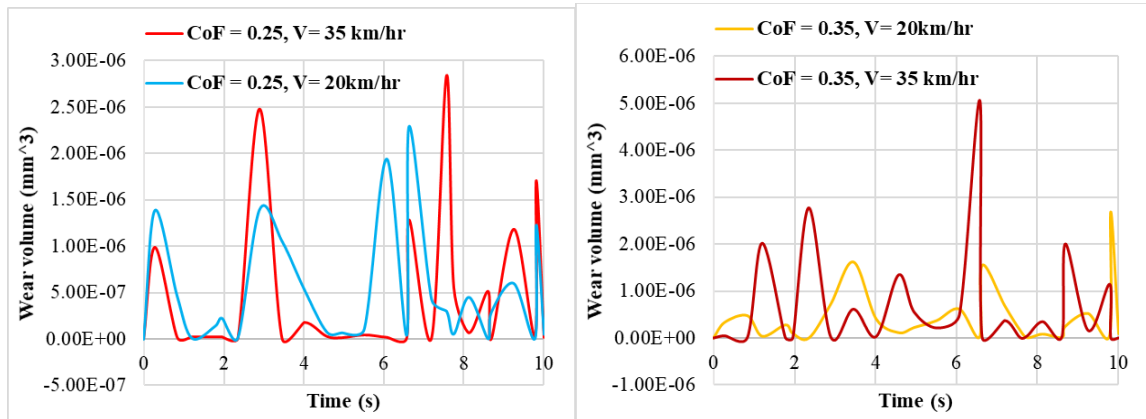


Figure 4.12: Wear volume against time for different CoF and speed for wet sliding contact condition.

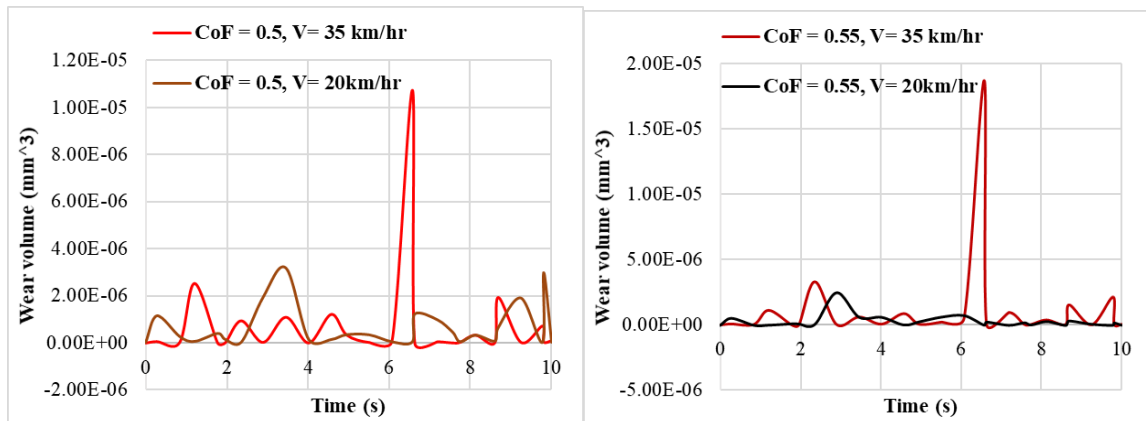


Figure 4.13: Wear volume against time for different CoF and speed at dry sliding contact condition.

It was observed that the wear volume increases with increase in speed and friction. Figures 4.9 and 4.10 show that the wear volume is relatively low at lower CoF and speed. It as well shows that an increase in speed and CoF in the wheel/rail contact interface increases the wear volume of the wheels, maximum wear volume was observed as at speed of 35 km/h with CoF of 0.55 during dry sliding contact condition and the lowest wear volume was observed at speed of 20 km/h and CoF of 0.25 during wet sliding contact condition.

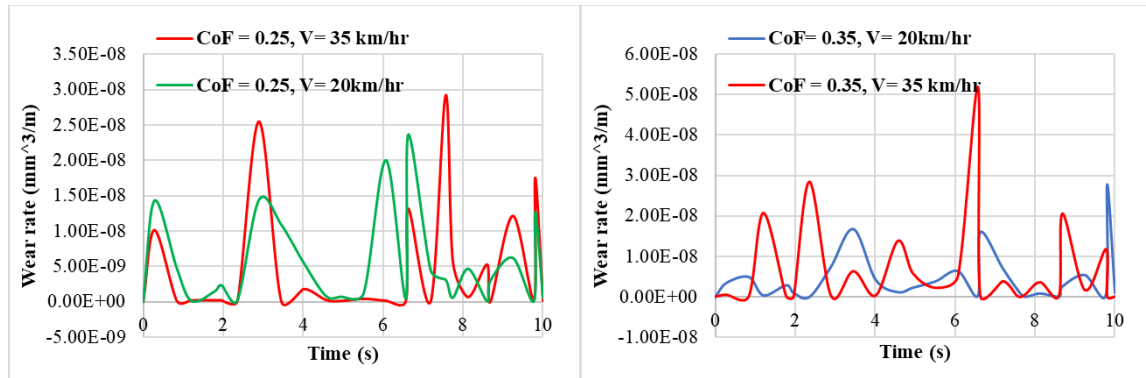


Figure 4.14: Wear rate against time for different CoF and speed at wet sliding contact condition.

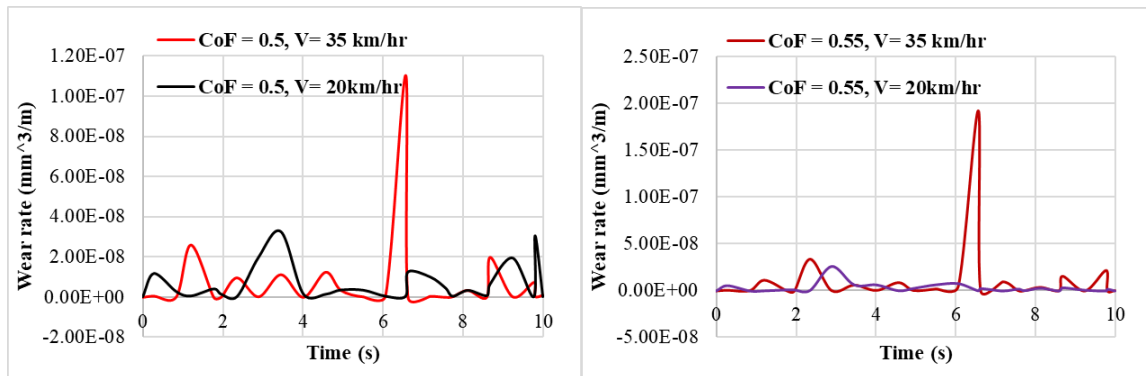


Figure 4.15: Wear rate against time for different CoF and speed at dry sliding contact condition.

Figures 4.11 and 4.12 show that the wear rate is low at low speed and CoF as compared to higher speed and CoF. It was observed that at higher speed and CoF, the wear rate of the wheels was high. The highest wear rate was observed as at speed of 35 km/h with CoF of 0.55 at dry contact condition and the lowest wear rate was observed at speed of 20 km/h with CoF of 0.25 at wet contact condition.

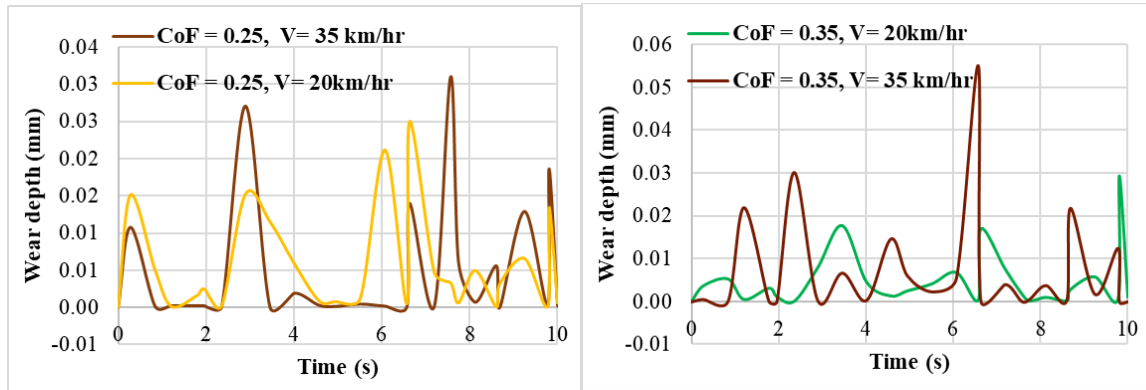


Figure 4.16: Wear depth against time for different CoF and speed at wet sliding contact condition.

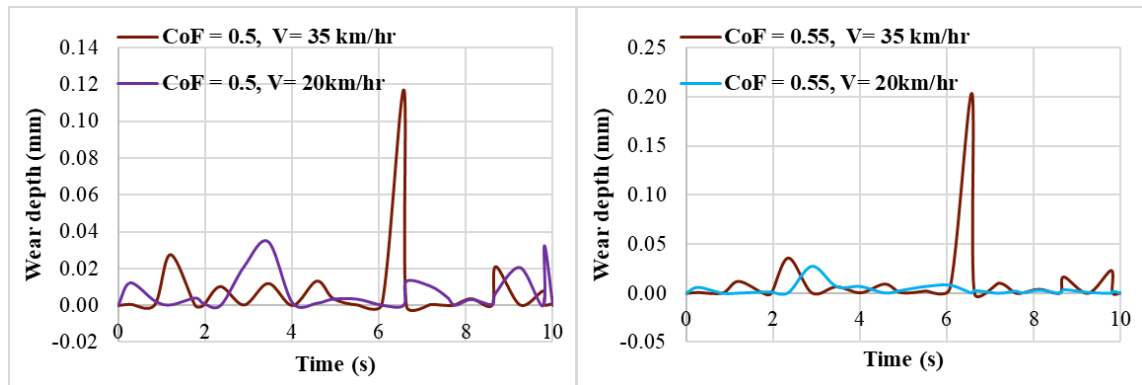


Figure 4.17: Wear depth against time for different CoF and speed at dry sliding contact condition.

Figure 4.13 and 4.14 also depict that the wear depth is low at low speed and CoF as during wet sliding contact condition compared to higher speed and CoF. It showed that at higher speed and CoF, the depth of wear of the wheels is much as observed at speed of 35 km/h with CoF of 0.55 (dry sliding contact condition), a reduced wear depth was observed at speed of 20 km/h with CoF of 0.25 during wet sliding contact condition.

4.3 Validation of simulation results.

Validation was done by comparing the simulation maximum contact pressure with the mathematical contact pressure obtained using MATLAB.

Table 4.1: Simulation contact prssure, contact area and semi axes ratio for different CoF at 35 km/hr speed

CoF	Maximum Pressure (Mpa)	Contact Area (m ²)	Semi axes ratio a/b
0.25	579.032	1.11633E-04	1.54658
0.3	579.026	1.11630E-04	1.54658
0.35	579.024	1.11629E-04	1.54658
0.4	579.023	1.11629E-04	1.54658
0.45	579.022	1.11629E-04	1.54658
0.5	579.022	1.11628E-04	1.54658
0.55	579.021	1.11628E-04	1.54658

From Table 4.1, the maximum contact pressure can be summarized as 579 Mpa. According to the MATLAB CODE in appendix D that was generated to numerically evaluate the maximum contact pressure at the wheel/ rail contact patch using the Hertzian contact theory discussed in chapter 2.2.2, following the equations in this very chapter, the maximum contact pressure at the wheel/rail interface was obtained as 505.617MPa. There is therefore a close correlation between both results.

Comparison was also done on the relationship of the obtained simulation results to the work done by some researchers and it was seen that the wear number results are in line with the wear number results obtained by researchers M. Ansari, I. A. Hazrati, E. Esmail zadeh and S. Azadi on their research carried out on Wear rate estimation of train wheels using dynamic simulations and field measurements [61]. Wear behavior results in terms of wear rate, wear depth and wear volume have the same relationship with the research results obtained by R. Lewis¹, M. Cavalletti, R.S. Dwyer-Joyce, A. Ward, S. Bruni, K. Bel Knani and P. Bologna on Railway wheel wear predictions with Adams/rail [62] and Na Wu and Jing Zeng research on Parametric analysis of wheel wear in high-speed vehicles [63].

5 CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

Poor friction management coupled with overload conditions are primary sources of train wheel wear. There are high possibilities of wear increase during dry sliding contact conditions compared to its counterpart (wet sliding contact) and especially since it is likely to be the condition for AALRT as there is lack of a solid lubrication strategy on the rails for managing friction on AALRT vehicles. As simulation results showed, high friction values (during dry sliding contact) escalate wheel wear volume, wear rate and depth. Higher CoF value (dry sliding contact condition) showed that it causes higher longitudinal and lateral creep forces which as a result causes higher wear values and the reverse was true for low CoF values (wet sliding contact condition). Variation in speed was also studied and showed that an increase in the speed increased the wheel wear volume, rate and depth and at low speed, it showed that the wear behavior was relatively low. It can be concluded that dry sliding contact condition causes high wear values especially given the passenger traffic in Addis Ababa which in most cases leads to overload train operation, this coupled with insufficient friction management is costly in the long run. Also, higher operating speeds leads to higher wheel wear.

5.2 Recommendations

The following are recommended as ways of reducing the identified causes of wheel wear in a way of reducing maintenance costs that are associated with wheel reprofiling and also to prolong the service life of the wheels.

- i. Generation of a reliable friction management plan which incorporate lubricating both the wheels and the rails. This can be in form of using friction modifiers with a moderate CoF to stabilize the lubrication the wheels and on top of the rail even in cases of very high contact temperatures.
- ii. From the outcomes of the study, it can be recommended that operating contact coefficient of friction should be maintained between 0.2 and 0.4 as they yield a relatively low wheel wear. Operating speed should also be $20 \text{ Km/h} \leq \text{speed} \leq 40 \text{ Km/h}$ for purposes of minimizing wear losses.

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7 APPENDICES

7.1 Appendix A: Simulation parameters

Table 7.1: Table of simulation parameters

Parameter	value
Wheelset Mass (m)	880 kg
wheelset inertia, (I_{xx})	176 kgm ²
wheelset inertia, (I_{yy})	76 kgm ²
wheelset inertia, (I_{zz})	176 kgm ²
Wheel radius (r_o)	0.33 m
Simulation speed (V)	9.72, 5.56 m/s
Distance between each wheel contact (l_o)	0.75 m
Applied normal load (W)	53072 N
Conicity of the wheel ($\delta\theta$)	0.5
Youngs modulus of wheel steel (E_w)	206GPa
Youngs modulus of rail steel (E_r)	210GPa
Poison ratio of wheel (ν_w)	0.29
Poison ratio of rail (ν_r)	0.3
COF at the contact area (μ)	(0.4, 0.45, 0.5, 0.55) dry contact (0.25, 0.3, 0.35) wet contact

Table 7.2: Wheel materials hardness [16]

Material	Hardness (GPa)
R8T (wheel)	2.6
W8A (wheel)	2.7

7.2 Appendix B: Kalker's creepage and spin coefficients.

	C ₁₁			C ₂₂			C ₂₃			
g	$\sigma=0$	$\frac{1}{4}$	$\frac{1}{2}$	$\sigma=0$	$\frac{1}{4}$	$\frac{1}{2}$	$\sigma=0$	$\frac{1}{4}$	$\frac{1}{2}$	
$\downarrow 0$	$\pi^2/\{4(1-\sigma)\}$			$\pi^2/4$			$\frac{\pi\sqrt{g}}{3(1-\sigma)}\{1+\sigma(\frac{1}{2}\Lambda+\ln 4-5)\}$			
b>a	0.1	2.51	3.31	4.85	2.51	2.52	2.53	0.334	0.473	0.731
	0.2	2.59	3.37	4.81	2.59	2.63	2.66	0.483	0.603	0.809
	0.3	2.68	3.44	4.80	2.68	2.75	2.81	0.607	0.715	0.889
	0.4	2.78	3.53	4.82	2.78	2.88	2.98	0.720	0.823	0.977
	0.5	2.88	3.62	4.83	2.88	3.01	3.14	0.827	0.929	1.07
	0.6	2.98	3.72	4.91	2.98	3.14	3.31	0.930	1.03	1.18
	0.7	3.09	3.81	4.97	3.09	3.28	3.48	1.03	1.14	1.29
	0.8	3.19	3.91	5.05	3.19	3.41	3.65	1.13	1.15	1.40
	0.9	3.29	4.01	5.12	3.29	3.54	3.82	1.23	1.36	1.51
	1.0	3.40	4.12	5.20	3.40	3.67	3.98	1.33	1.47	1.63
a>b	0.9	3.51	4.22	5.30	3.51	3.81	4.16	1.44	1.59	1.77
	0.8	3.65	4.36	5.42	3.65	3.99	4.39	1.58	1.75	1.94
	0.7	3.82	4.54	5.58	3.82	4.21	4.67	1.76	1.95	2.18
	0.6	4.06	4.78	5.80	4.06	4.50	5.04	2.01	2.23	2.50
	0.5	4.37	5.10	6.11	4.37	4.90	5.56	2.35	2.62	2.96
	0.4	4.84	5.57	6.57	4.84	5.48	6.31	2.88	3.24	3.70
	0.3	5.57	6.34	7.34	5.57	6.40	7.51	3.79	4.32	5.01
	0.2	6.96	7.78	8.82	6.96	8.14	9.79	5.72	6.63	7.89
	0.1	10.7	11.7	12.9	10.7	12.8	16.0	12.2	14.6	18.0
$\downarrow 0$	$\frac{2\pi}{(\Lambda-2\sigma)g}\left\{1+\frac{3-\ln 4}{\Lambda-2\sigma}\right\}$			$\frac{2\pi}{[(1-\sigma)\Lambda+2\sigma]g}\left\{1+\frac{(1-\sigma)(3-\ln 4)}{(1-\sigma)\Lambda+2\sigma}\right\}$			$\frac{2\pi}{3g\sqrt{g}}\frac{1}{(1-\sigma)\Lambda-2+4\sigma}$			

$\Lambda = \ln(16/g^2)$; $g = \min(a/b, b/a)$; $\ln 4 \approx 1.386$.

Figure 7.1: Kalker's creepage and spin coefficients.

The above analytic forms involving Λ are accurate for very small g only [64]

7.3 Appendix C: Longitudinal and lateral creep forces for different CoF at speed of 20 km/hr

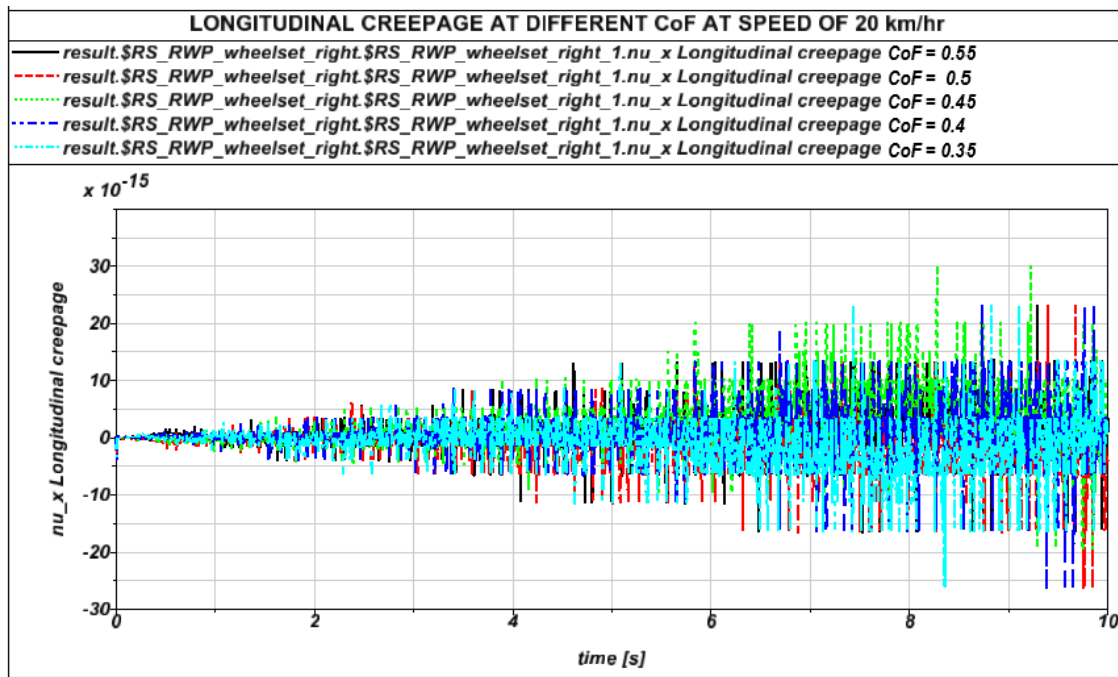


Figure 7.2: Longitudinal creepage at speed of 20 km/hr

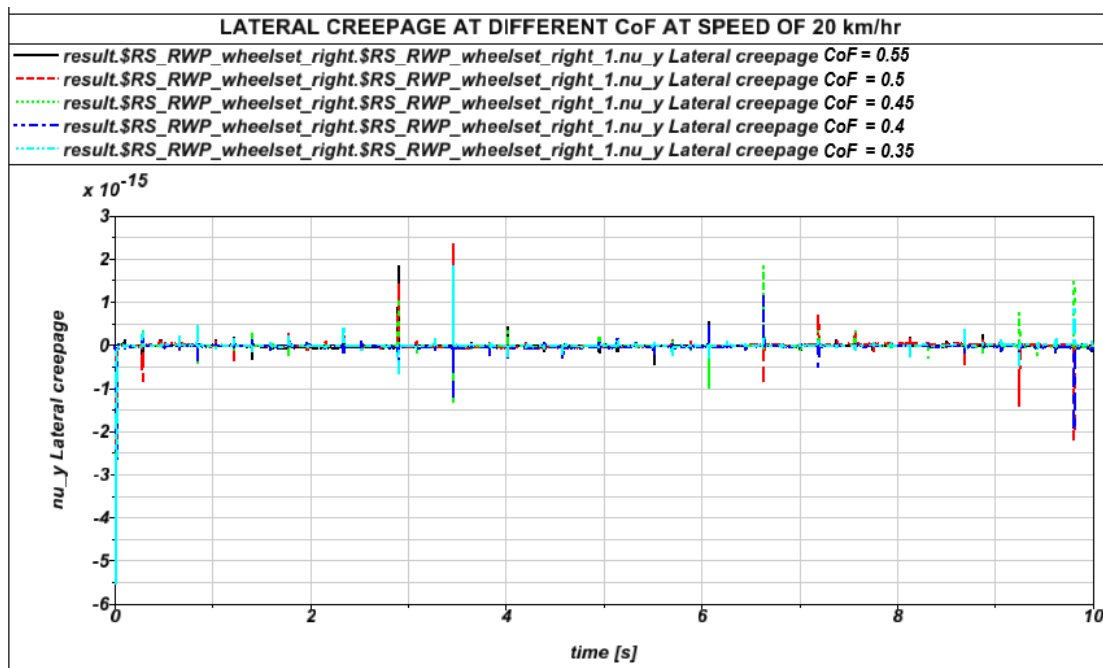


Figure 7.3: Lateral creepage at speed of 20 km/hr

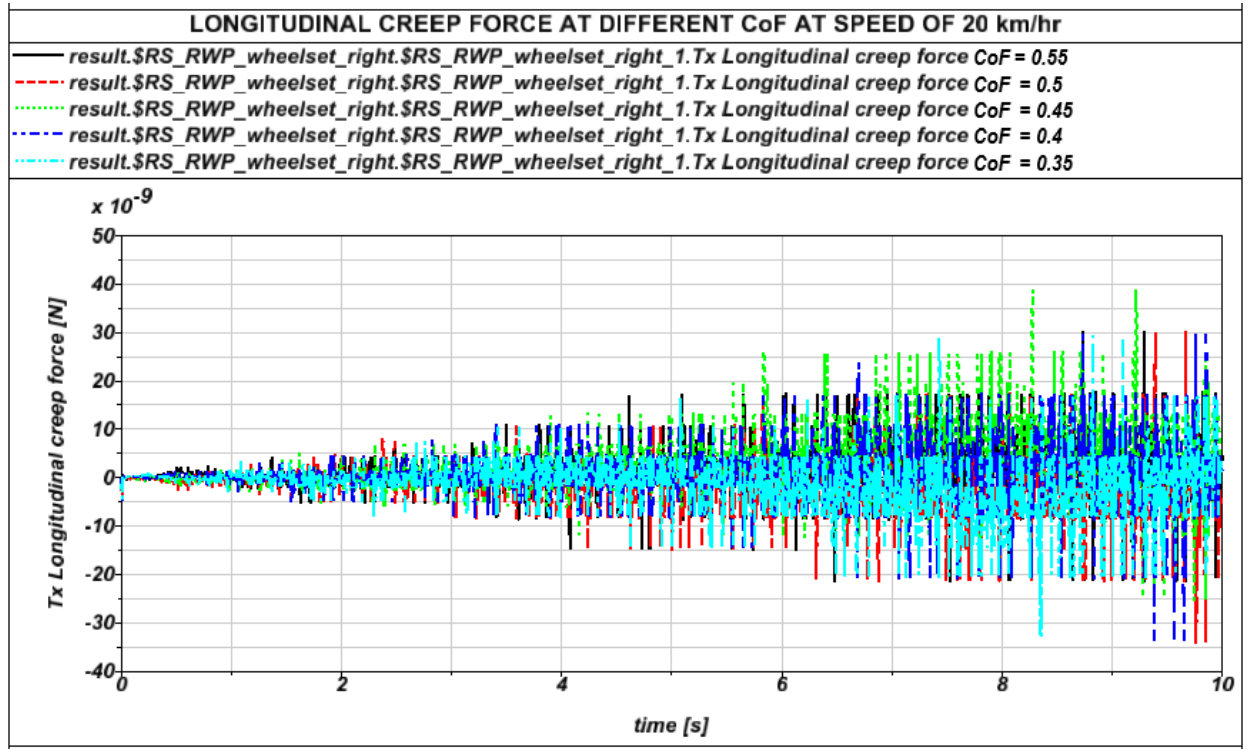


Figure 7.4: Longitudinal creep force at speed of 20 km/hr

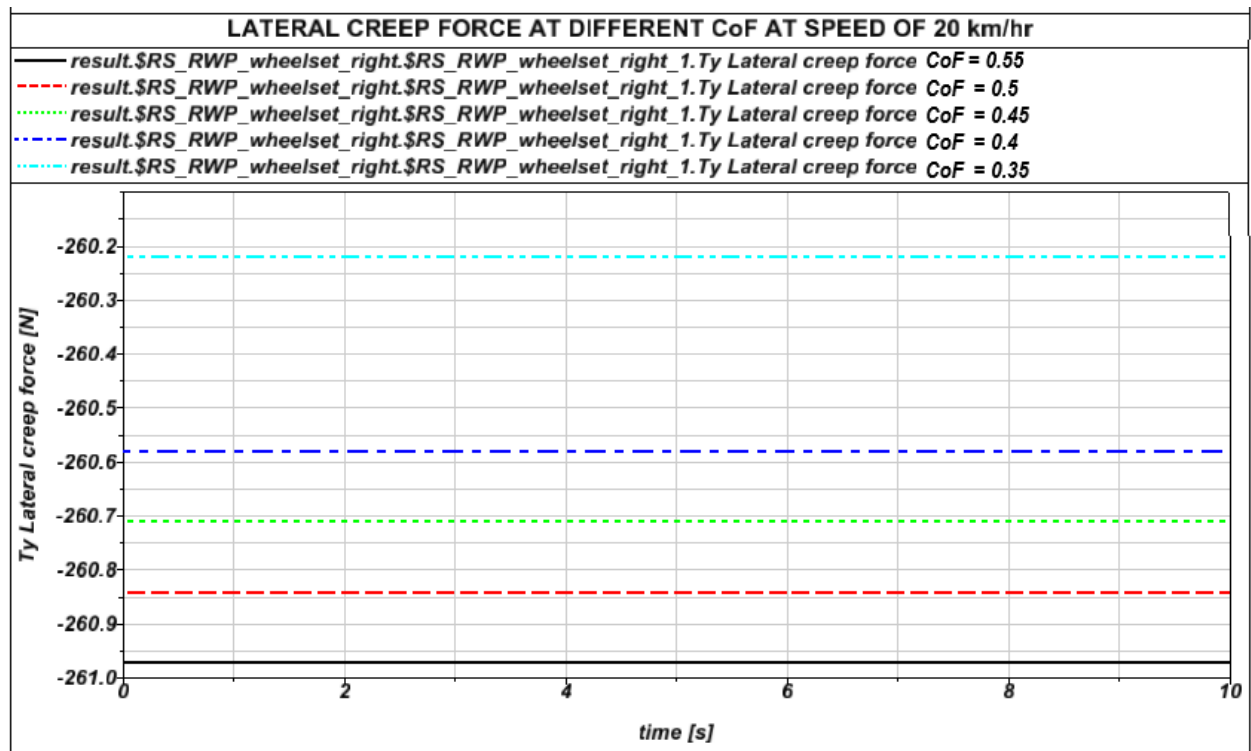


Figure 7.5: Lateral creep force at speed of 20 km/hr

7.4 Appendix D: MATLAB CODE for maximum contact pressure computation.

```
clc; close all;
% R21 equals to infinity therefore 1/R21 is approximately equal to zero
and thus it is eliminated from our equation.
% Because MATLAB is programmed to read angles in radians, angle 180 =
pi
% R1=R11, R1'=R12, R2=R21, R2'=R22

P=53072; E1=206*10^9; E2=210*10^9; R11=-0.32; R12=0.33; R22=0.3;
V1=0.29; V2=0.3;

cs=cos(pi);

syms A1 B
% Solving the simultaneous equations (6) and (7) gives;
eqn1=A1+B==(1/2*(1/R12 + 1/R11 + 1/R22 ));
eqn2=B-A1==(1/2*((1/R12 - 1/R11)^2+(1/R22 )^2+2*(1/R12 - 1/R11)*(1/R22
)*cs)^(1/2));

sol=solve([eqn1,eqn2],[A1,B]);
A1Sol = sol.A1;
BSol = sol.B;

% The values of A and B are represented below as aa and bb respectively
aa= vpa(A1Sol);
bb=vpa(BSol);

% To find the angle theta which is represented in this code as (S), we
programmed MATLAB in such a way that it converts it from radian to
degrees. cos (S) is also written as r .
r=(bb-aa)/(aa+bb);
S= vpa (radtodeg( acos(r)));

% K1 and K2 are poisson's coefficients
K1=((1-V1^2)/(pi*E1));
K2=((1-V2^2)/(pi*E2));

% m and n are Hert's coefficients
m=62.19*S^(-0.914);
n=(3*10^-05)*S^2+0.0045*S+0.334;
m=vpa(m);
n=vpa(n);

% a and b are ellipticity parameters that define the contact area at
the wheel-rail interface, these values are in millimeters and thus they
are multiplied with 10^-3
a= vpa(m*((3*pi/4)*(P*(K1+K2)/(aa+bb)))^(1/3));
b= vpa(n*((3*pi/4)*(P*(K1+K2)/(aa+bb)))^(1/3));

% semi axis ratio a/b = r
r=a/b;
```

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```
% the elliptical area of contact of an ellipse of semi-major axis of  
length 'a' and semi-minor axis of length 'b' is given by Area = pi*a*b  
and is written in this code as x  
x=(pi*a*b);  
  
% the maximum contact pressure Po acting at the area of contact is  
given by;  
po= vpa((3/2)*(P/(pi*a*b)))
```