GSJ: Volume 10, Issue 1, January 2022, Online: ISSN 2320-9186
www.globalscientificjournal.com

ANALYZING WEAR CONDITION OF WHEEL ON CROSSING A CASE OF ADDIS ABABA LIGHT RAIL TRANSIT LINE

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KeyWords
Multi-body system, crossover, wear, creepage, wear index, modeling, articulation

ABSTRACT

Wheel wear is one of the common factors that interrupts the vehicle movement and may cause derailment and overturning risk. The evaluation and prediction of wheel wear is a fundamental issue in railway field, both in terms vehicle, and economic cost. Wheel Wear effect is the main challenge of AALRT which takes more operation and maintenance cost, and also reduces the life of track and vehicles.

The computational tools developed here determine wear index of wheels consists of using commercial multi-body software (MBS) to study the railway dynamics problem and purpose built code for managing its pre and post processing data in order to compute the wear index. According to this strategy an initial wear profile is provided and the software runs a simulation and for necessary data from the dynamic analysis results and calculation the wear; the wear rate can be calculated. In real situation many vehicles are operated on different tracks, geometrical geometry. Therefore when predicts wear evolution of the wheel different track geometries and other factors has to be considered to wear studies. In this work the geometric component of the track such as crossing and straight alignments are considered to calculate wear of the wheel and compared under different loads and speeds.

The study tries to assess whether the crossing alignment is the cause of excessive wheel wears of AALRT, determine wear index and wear rate prediction. This study tries to assess the cases for wheel wear and also focuses on wear conditions at crossing under different loads and varying speed.

1. INTRODUCTION

The aspects of wear, friction and fatigue are very important in the operation of railways. Significant researches exist that deal with identifying the wear and fatigue mechanisms within wheel-rail contact. Different approaches are necessary to identify different phases on wear and fatigue behavior in wheel-rail contact, eventually making it possible to minimize the problem of unexpected wear and fatigue failure [15].

When railway vehicles run on tracks, contact forces are transferred between wheels and rails through the contact patches [21].These tangential and normal contact forces allow the vehicle to run over the track. Nevertheless when sliding occurs, it can lead to material removal or plastic deformation, due to this both the rails and wheels produce a change in geometry of the profiles [2]. This change in the geometry of the profiles is a really important issue since it can change the dynamic behavior of the vehicle [5]. One of these causes of change in the geometry and material removal can nowadays be predicted through wear prediction tools. These wear prediction tools are codes developed to know the evolution of the wear, thus the evolution of the profiles, without having to run thousands of kilometers and take profile measurements at every specific distance [1]. Railway crossings are the fundamental elements used in railway track structures. It is used in rail transportation to change direction of train and makes flexibility [8]. There are discontinuities between rails on crossing. These discontinuities cause a critical stress environment due to a higher dynamics going on in the crossing, and result in elevated damages like the mostly observed ones of rolling contact fatigue and wear on wheel and rail it[3]. Analyzing and predicting wear is very complex but things can be simplified and researched to get a very good deal of information about wear index determination and wear prediction [18].

1. **Wear measurement:**

It can be seen from published work that a number of computational models have been developed for the prediction of wheel wear. These models generally fall in two main categories:[23] Models which assume the material loss is proportional to the frictional energy dissipated in the contact patch (Tγ). Tγ is expressed as the sum of the products of the creepage and creep force for the lateral, longitudinal and spin components, as illustrated below. [9]

$$T\gamma = [Ty\gamma\ y]+[Tx\gamma\ x]+[M\ z\omega z] \quad \textbf{(1)}$$

Where; *T, M* = Longitudinal, lateral creep force and spin creep moment components
γ, ω = Longitudinal, lateral and spin creepage components [2].

A number of factors, based on the rate of the Tγ, have been developed to account for the different wear regimes (e.g. mild, severe and catastrophic. [25]

Sliding models according to Archard, where the material loss (*Vw*) is proportional to the normal force (*N*) and the sliding distance (*s*) divided by the material hardness (*H*), as illustrated below:

$$V_w = K\frac{N*S}{H} \quad \textbf{(2)}$$

The wear coefficient (k) differs depending on the governing wear regime (e.g. mild, severe and catastrophic) [24].

Table 1: equation for wear function [21]

Wear regime	Wear range Tγ/A[N/ Tγ)	Wear rate μg/m/mm^2
Mild	Tγ/A<10.4	5.3* Tγ/A
Serve	10.4 Tγ/A≤77.2	53.3
Catastrophic	Tγ/A≥77.3	60.9* Tγ/A

1.1 Wear prediction***:***

Ward, Lewis and dwyer Joysce; It uses rail interface to calculate contact position, the force and slip using wheel and rail profiles wheel set design, friction coefficient and material properties[13]. Based on an energy approach it used wear coefficient as an input. The contact between wheel and rail is assumed to be an ellipse [4].

$$\text{Wear rate } (\mu g/m/mm^2) = \frac{ke * T * v}{A} \quad \textbf{(3)}$$

Where k_e is the wear coefficient in (μg/Nm), T is tractive force (N), v is total creepage between wheel and rail ,A is contact area in mm^2

2. Methods and simulation

2.1 Methods

The following steps used during the analysis of wear indices

1. Prepare the input data of computation
2. Prepare the wheel-rail contact data file
3. Run the multi-body dynamic analysis
4. Read the multi-body dynamic analysis output: creapage and creep forces [6].
5. Compute the wear indices, wear rate
6. By varying loads and speeds repeat the same procedure

2.2 Simulation

Three-dimensional (3D) models represent passenger rail vehicles by using a collection of points in 3D space, connected by various geometric entities such as wheels, wheel sets (axles), primary and secondary suspension component force elements, axle box, markers, joint mechanism and articulation components like; fixed hinge lower articulations to each adjecent car bodies, flexible and free hinng articulation at the top of each adjecent car bodies, which balansses weight and supporting components and the three car body modules[17]. The modeling process takes Ethiopia light railway transient project data. From starting wheels and wheel sets up to the whole component model, the force elements, joints, constraints and body markers also included [11]. For the modeling of the passenger rail vehicles, the wheel profiles used s1002 and for rail profiles UIC 50 are used. According to this specification the maximum value of the flange angle $\gamma=70^0$ for the S1002/UIC50 wheel/rail profiles commonly used by the railway corporations in the European countries and in china railway industry.

Table 2. Vehicle geometry specification

No	Basic parameter	Value
1	Track gauge	1.435m
2	Bogie base	10.4m
3	Wheel set base for motor bogie	1.9m
4	Wheel set base for trailer bogie	1.8m
5	Bogie mass for motor bogie	5120kg
6	Bogie mass for trailer bogie	3120kg
7	Car body A or B mass (front and rear car body)	17500kg
8	Car body movement of inertia along x direction	4375
9	Car body movement of inertia along y direction	8750
10	Car body movement of inertia along z direction	8750
11	Car body B mass	12325
12	Car B movement of inertia along x direction	3081.25
13	Car B movement of inertia along y direction	6162.5
14	Car B movement of inertia along z direction	6162.5
15	Bogie movement of inertia along x direction	1250
16	Bogie movement of inertia along y direction	1870
17	Bogie movement of inertia along z direction	2182
18	Front and rear car body length	11.800m

19	Middle car body length	3.600m
20	Height car body including pantograph	3.750m
21	Width of car body	2.650m
22	Vertical mass center of Car body	-0.9m
23	Vertical mass center of bogie	-0.528m
24	Primary spring stiffness along x/y/z	5.8*10^5/5.8*10^5/1.45*10^6
25	Primary damping ration along z	1.25*10^5
26	Secondary spring stiffness value along x/y/z	3*10^5/3.5*10^5/5x 105
27	Secondary damper x/y/z	0/2.0 x 105/4*10^5
28	Minimum radius of vertical curve	1000m
29	Minimum radius of horizontal curve	50m

2.2.2 Crossover specification

Turnout type to be optimized is#9 which is used for speeds less than or equal to 30Km/h like 20Km/h speed limit in the AALRT turnout system;[10]

Table 1-3 crossover dimensions

Frog angle (deg.)	Lead curve central radius (m)	Distance between beginning and center of turnout (m)	Distance from center of turnout to heel of frog (m)	Total turnout length (m)	permissible speed (Km/h)
6020'25''	180	13.839	15.009	28.848	30

2.2.3 Modeling of passenger rail vehicle

Three-dimensional models represent a passenger rail vehicle by using a collection of points in 3D space connected by various geometric entities such as wheels; wheel sets (axles), primary and secondary suspension component force.. The modeling process takes Ethiopia light railway transient project data. From starting wheels and wheel sets up to the whole component model. The force elements, joints, constraints and body markers also included. For the modeling of the passenger rail vehicles, the wheel profiles used s1002 and for rail profiles UIC 60 are used. According to this specification the maximum value of the flange angle ᵞ=70^0 for the S1002/UIC60 wheel/rail profiles commonly used by the railway corporations in the European countries and in china railway industry.

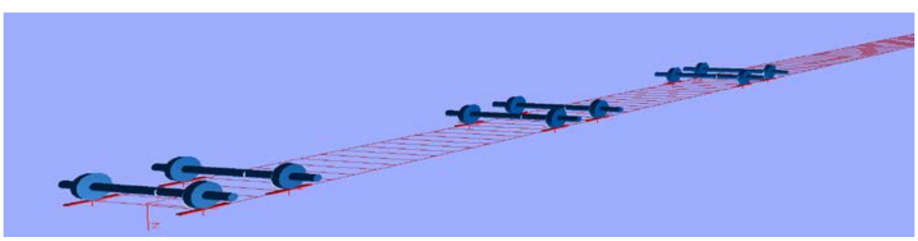

Figure 1 six wheel arrangments including axles

After modeling wheels and axles, then, modeling the axle boxes, primary suspension components, and bogie frames by using graphical user interface window was made. The primary suspension force elements start from wheel set axle box to bogie frame and the force type is spring damper parallel component forces.

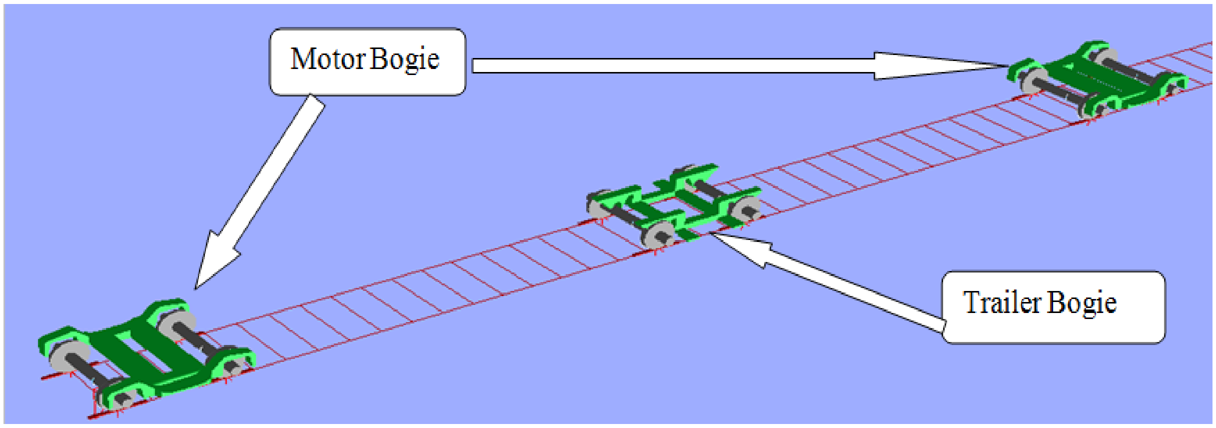

Figure 2 Wheel sets, axle boxes and bogie frames

The bogie frame is the intermediate component between wheel set and car body module. This system connects to the wheel set and car body by force element which means primary suspension connects bogie frame to wheel sets and secondary suspension connects car body to bogie frames. This frame has many marker points and force elements to connect different components like wheel set to bogie frame, bogie frame to car body in longitudinal, lateral and vertical point of contact, stopper, traction, weight balance and etc.
The following figure shows that, the two side lower fixed articulations and upper flexible and free articulation including middle car body-B and middle bogie frame and also the lower articulation damping components.

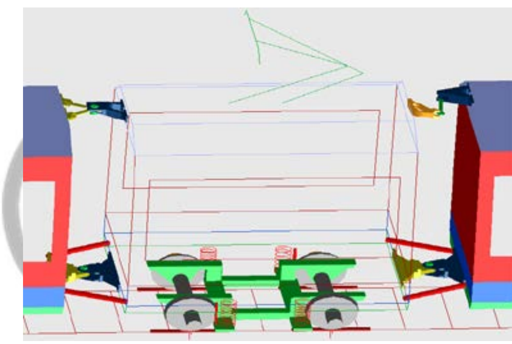

Figure 3 the two side lower and upper articulations and car body-B including middle bogie

The 3D assembled articulated passenger vehicle including some of marker points and corresponding force elements shown as follows in the following figure.

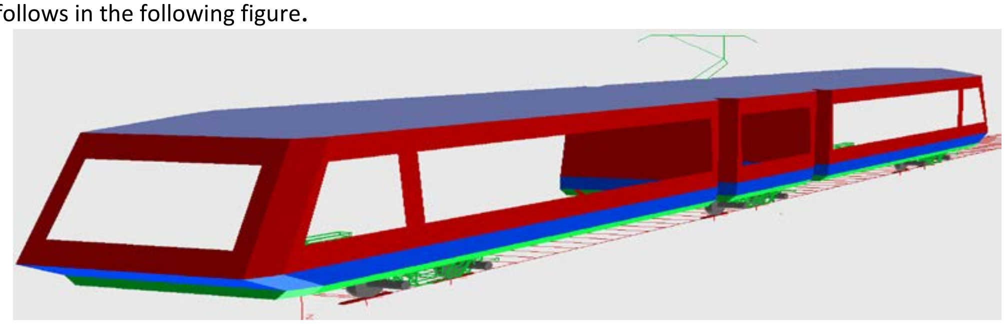

Figure 4 : Model of car body

3. RESULTS AND DISCUSIONS

In this part simulation and the analysis of the simulation runs were conducted in order to find wear index and predict wear [14]. The mathematical calculation was made to determine the wear rate, which is directly proportional to traveled mileage [20]. By testing material hardiness any researcher can calculate the material removal at certain travelled distance, which is not included here due to certain limitations. And finally summary has been given about the discussions made on this section.

In all presented cases, the tram simulations has been performed for the 30m straight track with 60m crossover distance and 30m intermediate length with default #9 frogs[10]. The total animation length is 1000m. The vehicle allowed traveling with minimum speed 30kmh and max 70kmh.

This section contains the analyzed results in 2D plots that are explained and discussed in the next section. All the analyzed results related to our main objectives, wear rate investigation at crossing, such as: creep forces, creepage, wheel-rail contact force, wear index and wear rate prediction are included[7].

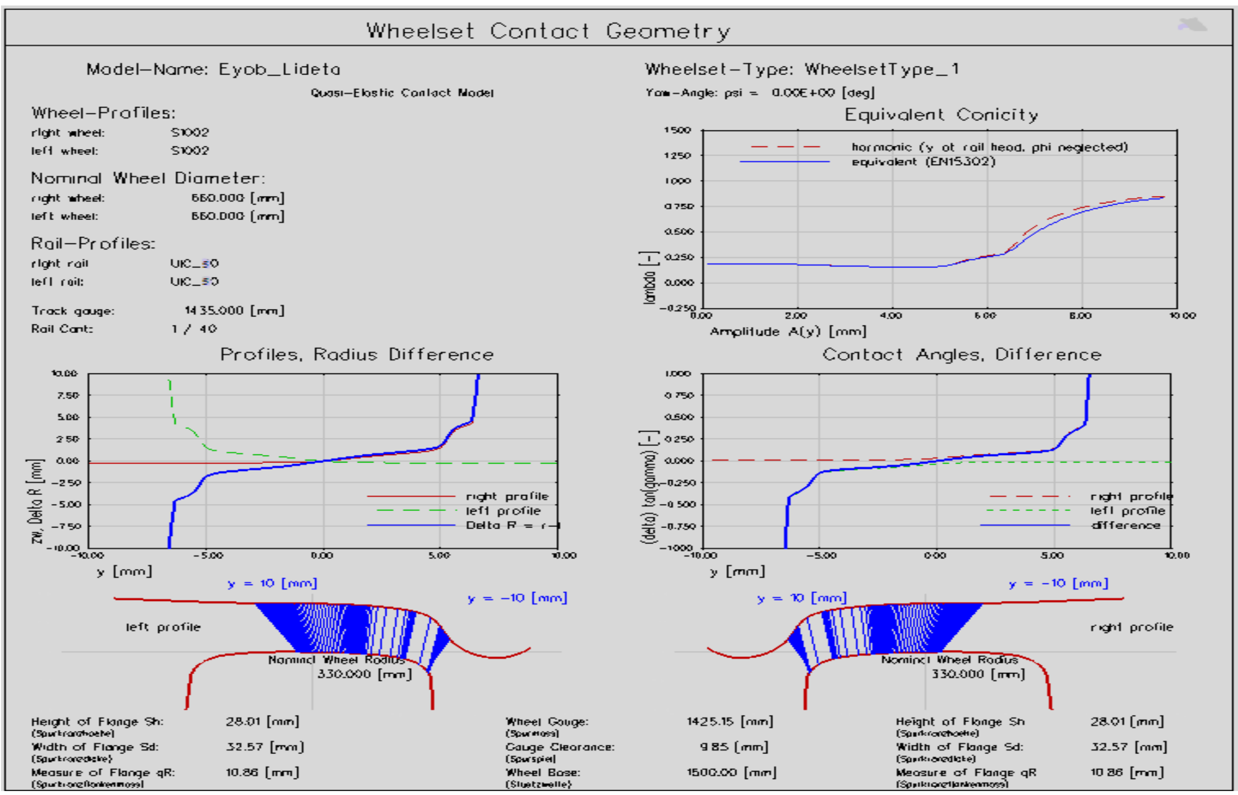

Figure 5 wheel-rail contact

From the above figure, Lateral contact point location on the wheel with the relative lateral shift y (-10) between wheel and rail is not varied. This indicates there is no sinusoidal movement. The locations of the contact points and the corresponding contact angles determine the locations and the orientations of the contact forces, respectively. Contact forces acting in the point of contact are subdivided into normal forces and tangential friction (creep) forces [19]. The default assumption of just one point of contact per wheel can be replaced by a multiple contact model, allowing up to three different points of contact per wheel: one on the tread, one on the flange and a third one again on the flange or on the back of the wheel. There is high contact per wheel on the flange of wheel this indicates the wheel wears more on flange.

Wear index : Wear index from the following figure is determined SIMMPACK, 8.9 simulation model; the Models which assume the material loss is proportional to the frictional energy dissipated in the contact patch.[22] Wear index is expressed as the sum of the products of the creepage and creep force, wear energy, in the lateral, longitudinal directions.. The degradation of wheels through wear and/or fatigue is often the primary driver behind the maintenance and replacement requirements for this critical component. The requirements and costs associated with these processes are closely linked to the operating conditions, which are a direct reflection of the track and rolling stock designs, the ongoing maintenance procedures and the way in which these are managed. Due to higher impact occurrence at crossing it is very logical to be convinced that the stresses at crossing to be higher than the straight alignments. It is very visible on 2D results plotted in figure bellows. The index in table below indicates the location of from starting point of simulation. About 3000 point which we call mesh in finite element is taken along the rail from animation of vehicle movement.

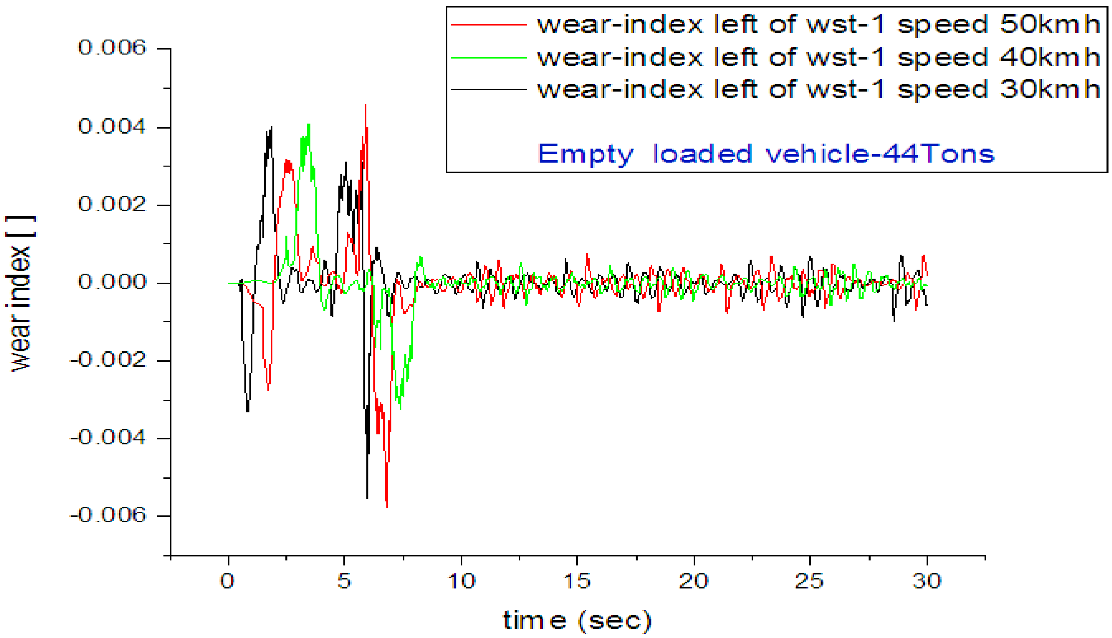

Figure 6 wear index for empty loaded vehice

From the above figure wear indexes is direct proportional to load and vehicle deriving speed. The Figure clearly shows the influence of train speed for empty loaded vehicle to be rather small [13]. This observation is in accordance with the findings of the parametric study reported in [27]. We concluding that, the influence of train speed on contact pressure is small relative to fully loaded train. For the presented vehicle speed range of 30–50 km/h, it can be observed that the influence on wear energy development at the flange contact is limited.

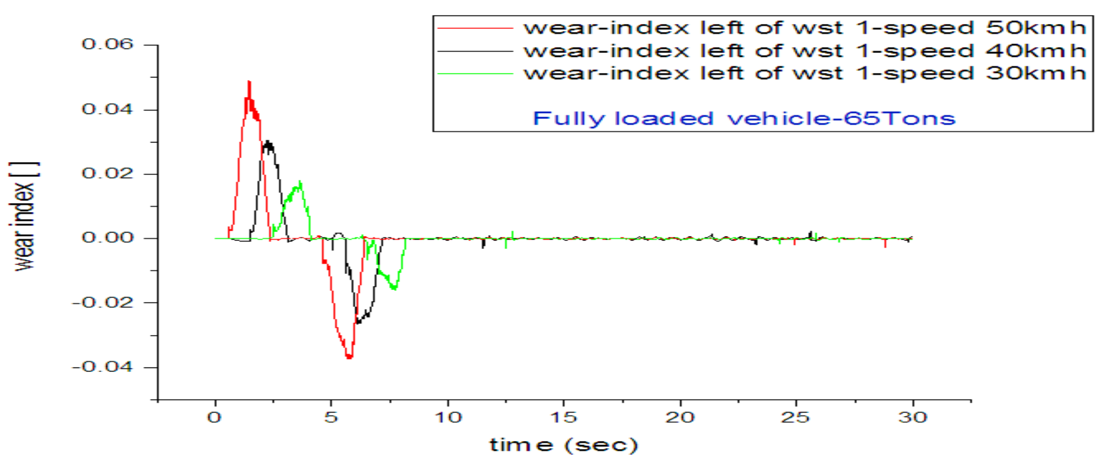

Figure 7 wear index for fully loaded vehicle

Figure 7, presents the effect of train speed in relation to wear index of fully loaded vehicle, which is related with contact flange, Tγ, development at the leading wheel of the leading bogie [16]. The result indicates that the wear index increases as driving speed increases. The effect is visible only at crossing. When negotiating the switch panel, distinct peaks for Tγ are seen to arise. These occur from changes in contact position and corresponding changes in locations and orientations of contact forces and slip. Upon entering the switch, the first peak for Tγ arises due to the appearing flange contact. A second peak occurs when the wheel load fully transfer from stock rail to switch rail. From table 5 the leading wheel set of the bogie has high amount of wear rate, the wear rate is maximum at wheel set 3.

Wear rate

The Royal Institute of Technology Stockholm model is used to predict the wear rate. This model is based on an energy approach it uses wear coefficient as an input. This model is one of the most common models which have been used in recent works for wear modeling; accordingly, it has the same drawbacks as most of the classical wear models, which are dependent on the value of certain constants. The contact between wheel and rail is assumed to be an ellipse. [4]

Wear rate (µg/m/mm2) =(ke*T*γ)/A ………………. 4-5

Where, ke is the wear coefficient in (µg/Nm), T is tractive force (N), v is total creepage between wheel and rail, A is contact area in mm^2, the war condition through the total wheelsets are mild, the wear coefficient ke =5.3 [21].

Table 4 Wheel-rail tangential and global creepage force analyzed

Wheel set	wst1	Wst2	wst3	Wst4	wst5	wst6
Contact force, [N]	35300	26700	30300	31800	18000	28000
Global creepage ,γ, [N]	6582	6580	7990	7350	6910	7840
Contact area[mm^2]	52.86	45.4	49.2	50.81	34.77	46.61

Table 5 Wear rate of more laded wheels

Wheel set (left)	Wst 1	Wst 2	Wst 3	Wst 4	Wst 5	Wst 6
Wear rate ($\frac{T\gamma}{A}$) µg/m/mm2	5.3*4.32= 24.192	5.3*4.027 =21.3431	5.3*4.969 =26.3357	5.3*4.6 =24.38	5.3*3.57 =18.921	5.3*4.712=1 6.973

According to (Joao Pombo, Jorge Ambrósio, Manuel Pereirathe) twin disc experimental data acquired from the contact between discs made of R8T wheel material and UIC50 900A rail materials the wheel wear of all wheelsets are under mild regimes[21]. According to the impute data **mitigations of excessive wear**: the allowed material and design geometry is not the main case of excessive early weal wear. Application of wheel and rail to lubricants containing extreme pressure (EP) additives, such as molybdenum disulphide (MoS2) or graphite,[12] is by far the most effective approach to mitigating wheel wear and the one that can be implementing in LRT.

Altering wheel material by hard graded material is on option that can be considered to mitigate wheel wear. Mitigation by wheel-rail interface management is another mitigation method. This includes, examining regularly and limiting corrugation growth, vehicle stability and ride quality, friction management strategy, manage according to implementation and managing plan.

Conclusions

The study has performed the task of determining creepage and creep forces between wheel-rail contacts. It uses the wheel type s1002, and rail USC50 on crossing, by using MBS software. Varied speeds 30kmh, 40 and 50kmh is used, by considering empty and fully loaded vehicle under maximum load of 65 ton. The simulation was made and the results are used to determining wear index by using SIMPACKv8.9 MBS function.

Ward, Lewis and Dwyer-Joyce, wheel prediction tool, which uses wear coefficient based on energy approach is used to determine wear rate by using numerical method. The contact area between wheel and rail is assumed to be an ellipse.

The study investigate wheel wear regime by using Pombo, J., Ambrosio, J., Pereira, M., Lewis, R., Dwyer-Joyce, R., Ariaudo, C., & Kuka, formula wheel wear is not catastrophic, and it is under Mild wear regime, this indicates that crossing geometry and material properties allowed

ACKNOWLEGMENTS

First of all, I am very delightful to praise the Almighty God who has enabled me to tis papaer. Next, I would like to thank my Advisor Mr. Mequanent Mulugeta for his patience, support and guidance. And I would like to express my gratitude to Ethiopian Railway Corporation (ERC) for providing my tuition fee. They gave me valuable information about AALRT track conditions and vehicle specifications as well as provided me the necessaries data and specifications. Lastly, I extend my sense of gratitude to all my friends who directly or indirectly helped me in my thesis work.

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