Influence of Switches and Crossings on Wheel Wear of a Freight Vehicle

Master of Science Thesis

by

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Abstract

Turnouts (Switches & Crossings) are important components in railway networks, as they provide the necessary flexibility for train operations by allowing trains to change among the tracks. But the turnout’s geometry with discontinuity in rail profiles and lack of transition curve causes additional wear both on track and on vehicle.

The main goal of this MSc thesis is to investigate the influence of turnouts on wheel wear of a freight vehicle. This will be obtained by simulations in the commercial MBS software GENYSYS. The wheel-rail contact is modelled according to Hertz’s theory and Kalker’s simplified theory, with the FASTSIM algorithm, and the wear calculations are performed according to Archard’s law.

Wheel wear is estimated by considering variations in parameters which have effect on wheel-rail contact. All these variations are common in daily rail operation, and they are caused by it, i.e. worn wheel profiles, worn crossing nose and different stiffness of the stock and the switch rails at the beginning of the turnout. Moreover, the wheel wear is calculated for both possible directions which a vehicle can run, the diverging and the straight direction of the turnout. Especially for the straight direction, various running speeds have been tested as the speed limit when the vehicle follows the straight direction is higher than for the diverging part.

Running with worn wheel profiles has the greatest impact in terms of increasing the wheel wear, especially on the outer part of wheel tread. In addition, the worn crossing nose results in increased wheel wear in this area. The results of the simulations concerning the different stiffness showed that the wheel wear caused by the contact of wheel and stock rail increases whereas the wear caused by the contact with the switch rail is kept at about the same level or decreases. It is concluded that turnouts have a significant impact on wheel wear, mainly because of the discontinuity in rail geometry and all the investigated parameters increase this impact. Moreover, great differences in wear values for areas close to each other are observed, mainly because of the wear coefficient values chosen in Archard’s wear map.

Keywords: dynamic vehicle-track interaction, turnout, switch & crossing, wheel wear, wheel-rail contact geometry
Preface

This MSc thesis corresponds to the final part of my studies on MSc Vehicle Engineering. The thesis work has been carried out at the Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering at KTH Royal Institute of Technology in Stockholm.

First of all I would like to thank my supervisor, Carlos Casanueva, for giving me the opportunity to accomplish this very interesting subject and for his continuous support and guidance during this work.

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Chapter 1

Introduction

Railway systems evolve into higher speeds and increased axle loads, which often results in increased contact forces between wheel and rail. As a result, significant wear appears on both wheels and rails despite the considerable improvements on rail materials in recent years. For reducing the extensive costs, wear predictions are performed through simulations, in order to optimize the maintenance procedure and modify the infrastructure or the rolling stock designs.

Wear is the material removal of wheels and rail and, to some extent, their plastic deformation. The main consequence of the wear is that the wheel and rail profiles are continuously being changed and may affect the desirable performance of the rail vehicle. Wheel wear is usually divided into two categories, the uniform and non-uniform wear. The first one consists of flange wear which results in reduced flange thickness and increased flange inclination, and wheel tread wear which causes increase of flange thickness and increase of flange height. The second category is the non-uniform wear which is divided into out-of-round wheels, where eccentricity and corrugations are found, polygonization and wheel flats.

Turnouts are the switches and crossings in a railway network, which enable trains to change track. They are the most complex and expensive parts to be maintained and, eventually, replaced. For this reason, most studies, so far, concern the wear which appears in the turnout. The new contribution of this MSc thesis is that the main consideration is the wheel wear generated when running through a turnout.

In this context, wheel wear is investigated, especially how it is influenced by the special geometry of the turnout, i.e. the introduction of the switch rail and the crossing nose which will be described in the related chapter. In combination with this, the possibilities of worn wheel profiles, worn rails and different stiffness among various independent parts of the turnout will be studied.
Chapter 2

Background

In some measurements heavy wear at the end of the wheel tread has been seen after running a vehicle along the rail track. This means that there are moments in which the contact point reaches the outer end of the wheel. But it is a phenomenon very unusual for tangent or even curved tracks. The only case where the contact points reach the tread-end of the wheel is when running on turnouts, or switches and crossings (S&C) [1]. The geometry of a turnout gives the probability for these type of phenomena to appear.

In this chapter, definitions will be given for important issues concerning this study. Such subjects are wheel wear, turnout geometry, vehicle characteristics and wheel profile. In the last section the objectives of this study will be described and thesis contents as well.

2.1 Wear

As mentioned in the introduction, wear is mainly the material removal of wheels and rails and to some extent the plastic deformation of these two parts [2]. Wheel wear can be divided into two categories, the uniform and non-uniform wear. Uniform wear is also divided into flange wear where the flange thickness reduces, whereas the flange inclination increases and the opposite result in flange thickness is caused by wheel tread wear where the flange height increases as well. Non-uniform wear includes eccentricity, wheel flats, corrugation, etc.

Flange thickness is the main criterion for measuring and judging flange wear. It is mainly influenced by curve radii, wheel-rail friction and running gear design. In the case of a turnout there is usually no transition curve so when the curve radius decreases, the contact point of the outer wheel moves towards the flange and it may reach it. Furthermore, the geometry of the turnout with the stock rail going away and a new one appearing (switch rail) may lead to flange contact too.

Wheel tread wear is caused by the normal forces between wheel tread and rail top surface in combination with the creepages and creep forces [2]. Creep or creepage is normalized the sliding velocity between wheel and rail in the contact zone. Possible tread braking is also responsible for the tread wear. As will be presented in the results, the shape of the turnout increases the creepages at some parts, which results in even higher tread wear.
CHAPTER 2. BACKGROUND

Non-uniform wear can appear due to several reasons. For example, wheel flats arise when the wheels are locked during braking and pure sliding appears at the wheel-rail contact. Usually, there are two wavy patterns of corrugation. One presents shorter wave length and smaller amplitude than the other one. Turnouts may also be a reason of the appearance of corrugation on the wheels, as the wear along a turnout appears with high amplitude and in certain positions, affecting a limited part of the wheel perimeter.

2.2 Turnout

Turnouts provide the necessary operational flexibility to railways. They are composed of a switch panel, a crossing panel and a closure panel (Figure 2.1). The switch panel is the part of the turnout which enables the change of the direction of a moving vehicle. For this function, there are two movable switch rails, switching machines and two wing rails. The pair of switch rails lies between the diverging outer rails. The crossing panel is where the two tracks intersect and the closure panel is the part of the turnout between the switch and the crossing panels. The Swedish railway network has over 12000 turnouts in its about 15000 km of railway track [3].

![Figure 2.1. Components of a turnout [4].](image)

When the train is entering a turnout, the wheels of the one side have to deal with its varying geometry. The contact points of the wheels move from the stock rail to the switch rail at the beginning of the turnout. As a result there is a transition area where the wheel contacts both the stock and the switch rail. Almost the same phenomenon appears when the wheels pass from the crossing nose. They run on the wing rail which is going off the running track and the crossing nose appears. So there is also a transition area where the wheels contact both rails. As a consequence, wheel wear is increased in these areas.

The dynamic interaction between vehicle and track presents a greater complexity than on tangent or curved tracks. Multiple wheel-rail contacts and large lateral wheelset movements are common. The usual damage mechanisms are wear, rolling contact fatigue and large accumulated plastic deformations [4]. Because of this, turnouts become very costly
2.3. VEHICLE CHARACTERISTICS

for the railway administrators in terms of maintenance, which is needed more often than the other parts of the track. As a result, most of the studies evaluate the wear on the turnouts and not on the wheels, which seems to be considerable as well.

The turnout which is used for the simulations in this study is a medium size turnout about 60 m long and with standard track design UIC60-760-1:15 (UIC60 rails, curve radius 760 m and crossing angle 1:15) [4].

2.3 Vehicle Characteristics

The investigated vehicle is a two-axle freight wagon with Unitruck running gear which transports timber logs in central Sweden. The axle load is 5.8 tons for a tare vehicle and between 22.5 and 25 tons for a laden vehicle.

![Figure 2.2. Unitruck running gear and expanded view of its components [1].](image)

The Unitruck running gear is shown in the Figure 2.2 and it is a single-stage suspension system, which is composed by four nested coil springs for the connection between the carbody (1) and the saddle (2). The nested coil springs allow having different stiffness values in the cases of laden and tare vehicle. Furthermore, friction elements attached to the wedge (3) provide the damping. The wedge is a small component which enables the coupling between the vertical preload and the longitudinal, lateral and the vertical friction. The inner coil springs are connected to the wedge, which transmits vertical load to the carbody through an inclined friction surface (4). Because of this inclination a longitudinal force is generated and transmitted between wedge and saddle through a vertical friction surface (5). As a result, friction damping is created in longitudinal, lateral and vertical direction. The saddle is mounted on the axle box (6) through a rocket seat coupling (7) that enables it to have a relative sway angle with respect to the axle box. Bumpstops also limit the possible displacements between carbody and saddle [1].
CHAPTER 2. BACKGROUND

This type of suspension was used in the vehicles until the year 2005. The main disadvantage was that the vehicles presented high flange wear and for this reason the running gear was modified (Figure 2.3). The two modifications that were introduced were to soften the inclined suspension by replacing the longitudinal friction surface with a roller, and also the nested springs were shortened and a plate was introduced to connect the centre position of both springs, in order to avoid spring buckling due to this rolling coupling. After these modifications, flange wear was reduced by up to 50%. This modified suspension system is the one that the investigated vehicle is equipped with.

![Figure 2.3. Modified Unitruck running gear [1].](image)

2.4 Wheel Profile

As already mentioned, the vehicle consists of a two-axle wagon and the wheel profile is the standardized type UIC/ORE S1002 (Figure 2.4). The wheel profile is expressed in $y$- and $z$-coordinates, where $y=0$ is the nominal running circle and the positive values of the $y$-coordinate point towards the flange and thus the centre of the track. The nominal wheel radius is $r_0 = 0.46$ m.

2.5 Objectives

The objectives of this study are to theoretically analyse and examine the different conditions of a freight vehicle running through a turnout in terms of wear prediction of the wheels. Certain parameters of the turnout will be studied in order to analyse how they influence wheel wear. First of all, the geometry of a turnout will be investigated. The
The main goal of this work is to investigate how the wear is distributed across the \( y \)-coordinate of a wheel profile and also differences around the wheel perimeter, as it has been noticed through measurements that high tread wear appears on the wheels and turnouts seem to be the main reason of this phenomenon.

2.5.1 Thesis Contents

After the 'Background' (Chapter 2) covering the basic information on wear, turnout, vehicle characteristics and wheel profile, the 'Methodology' (Chapter 3) is described, with the wear modelling and the wheel wear prediction tools. Furthermore, a brief summary of the previous work in this field will be given and the simulation cases with the input data is presented.

For vehicle dynamic simulations the commercial software GENSYS version 1301 [5] is used. Apart from the existing modelled turnout which is a straight track - curve deviation and the vehicle runs through the curve, the geometry of the turnout has also been modified in order for the vehicle to run in the straight track. Furthermore, the model is an unworn turnout, so the turnout has also been modified for examining the influence of a worn turnout.
turnout on the wheel wear, both in straight track and in diverging track. A worn turnout, especially in the nose part, may increase the normal forces in the tread-end contact point as the height of the nose will be decreased.

Another parameter that is investigated is the speed of the vehicle and how it affects the wear on the wheels, especially when the vehicle runs through the straight track of the turnout. Moreover, simulations with worn wheel profiles and different stiffness at the stock and switch rails are performed. The various simulation cases are presented in Chapter 4.

The wear of the wheels is calculated through a wear calculation programme developed in MATLAB and it is compared for all the above different cases and the most interesting results are presented in the 'Results and Discussion' (Chapter 5) section of this report. In the end the results are summarized and ideas for further work are given in the 'Conclusions and Future Work' (Chapter 6).
Chapter 3

Methodology

This chapter first describes the simulation procedure that is followed in this study for obtaining the wheel wear and also its lateral distribution piecewise along the perimeter of each wheel, as the vehicle runs through the turnout. Then a brief description of the simulation cases is presented and some remarks and definitions are given for better determining the different cases and interpreting the results in the next chapter (see Section 3.3).

3.1 Simulation Procedure

The following Figure 3.1 shows the flow chart of the followed simulation procedure. The wheel wear calculation begins with the initial wheel profile and the various rail profiles along the turnout. The variation in rail profiles along the turnout is accounted for by sampling the rail cross-sections at several positions. In this study, rail profiles at 30
different positions are used. After this, the Contact Point Functions (KPF in GENSYS) create the wheel-rail geometry functions that will be used in the simulation. The results are obtained after only one run of the wagon along the turnout.

Then the track design and the vehicle characteristics are imported. At this point, dynamic simulations of the vehicle running and along the turnout are performed, giving creepages and creep forces at the wheel-rail contact. The creep forces are calculated using the Kalker’s simplified theory \[6, 7\]. Having the results of the creepages and the creep forces, the wheel wear is calculated by using Archard’s wear model \[8\]. Instead of obtaining a uniform wheel wear for the whole run of the vehicle through the turnout, the results are examined piecewise around the perimeter for analysing the specific positions on the turnout which cause the highest wheel wear.

### 3.2 Wear Modelling

#### 3.2.1 Creep

Creepages or sliding in the wheel-rail contact is always more or less present, as the suspension etc. prevent the wheelset from pure rolling. So apart from inertia forces, normal forces and suspension forces, the equation for the lateral dynamics of a wheelset contains creep forces as well. These creep forces arise when the wheelset does not roll ideally on the track, and are the main reason for wear appearance on the wheels. They are dependent on the wheel-rail geometry, material, normal forces, creepages and wheel-rail friction.

In reality both wheel and rail are elastic and, when they are pressed together with a certain force, a contact area arises. Both normal and shear stresses exist in the contact area. Creepage is defined as the sliding velocity between wheel and rail in the contact zone, normalized by the vehicle speed. The creepages can be divided into three elements, a longitudinal creep, a lateral creep and an angular sliding velocity around an axis normal to the contact patch, which is called spin or spin creep \[2\]. Creepages and spin in all three directions in the contact patch have a great influence on the amplitude of the creep forces.

For defining the sliding velocities, creepages and creep forces, a coordinate system \((\xi – \eta – \zeta)\) attached to the contact point and tangent to the contact patch is used (Figure 3.2). The \(\xi\) coordinate is defined positive to the rolling direction, \(\eta\) is on the contact plane and perpendicular to the travel direction and \(\zeta\) is perpendicular to the contact plane.

The creepages are defined as the quotients of sliding velocities at the wheel-rail contact and vehicle’s speed (Equation 3.1 to 3.3).

\[
\begin{align*}
\text{Longitudinal creep } v_\xi &= \frac{v_\xi}{v_{\text{vehicle}}} \\
\text{Lateral creep } v_\eta &= \frac{v_\eta}{v_{\text{vehicle}}} \\
\text{Spin creep } \phi &= \frac{\omega}{v_{\text{vehicle}}}
\end{align*}
\] (3.1) (3.2) (3.3)
3.2. WEAR MODELLING

Figure 3.2. Coordinate system for the wheel profile \((x - y - z)\) and for the contact surface \((\xi - \eta - \zeta)\).

3.2.2 Normal Contact

For calculating the position of the wheel on the rail and the rigid body creepages, it is assumed that the contact occurs in a point. The wheel-rail contact problem has a complexity which needs several simplifications in order to execute the calculations in a reasonable amount of computational time. Although the normal and the tangential forces are influenced by each other, the normal and the tangential contact problem are usually solved separately. The normal contact problem is solved according to the Hertzian contact theory [9, 2]. For separating the problem into two and solve the normal contact problem, the following assumptions are taken into account [2]:

1. Displacements and strains are small.
2. The contact patch is relatively small in comparison with the typical dimensions of the contact partners, \(e.g.\) the rolling radius of the wheels.
3. The surfaces in the vicinity of the contact area are described by constant curvature in order to be able to calculate the shape and the magnitude of the contact area.
4. The surfaces are smooth.
5. Only elastic displacements exist.
6. The bodies consist of homogeneous, isotropic material.
7. The bodies are geometrically and elastically the same, so the normal and tangential contact problems can be calculated separately.

For high contact pressures also plastic deformations can occur but Hertzian theory [9] has been the most practical way of analysing the problem of normal contact in railway applications. According to this theory the contact patch will be elliptic. The size of this ellipse depends on the normal load, the geometry of the wheel and the rail and the material.
3.2.3 Tangential Contact

As already mentioned, the tangential forces in the wheel-rail contact are inseparable with the creepages. The longitudinal creep force leads to compressive stresses on the wheel in front of the contact patch and tensile stresses behind it. The creep forces are non-linear functions of the creepage and the maximum possible creep force is $\mu N$, $\mu$ being the friction coefficient and $N$ the normal force; though it can be regarded as linear as long as the creepage is small enough.

For calculating the tangential forces, some more assumptions are taken into account:

1. Inertia can be neglected.
2. There is a dominating speed direction in the contact which coincides with one of the axles of the contact point’s coordinate system.
3. Friction coefficient is constant along the whole contact patch.

The contact area could be divided into two sub-areas: The first one is an area of adhesion without slip, and the second is an area of slip where contacting particles of wheel and rail move relatively to each other. Several theories for determining the creep forces have been developed and some of them will be presented briefly below.

Kalker’s linear theory [10] determines creep forces for small creepages. This theory has limited accuracy because of the approximation of the linear relation of the creep forces and the spin moment with the creepages. However, this theory can be used for studies of lateral dynamics on tangent track and especially for linear stability analysis. It is also the basis for other theories. One of them is the widely used approximate nonlinear creep force theory which has been suggested by White et al. [11] and Shen, Hedrick, Elkins [12]. This theory gives more realistic results but neglects the spin moment which may result in unsatisfactory outcome in case of high spin creepages.

Kalker’s non-linear creep force theory [6, 13, 10], which assumes the contact patch to be elliptic, is a more accurate theory. A numerical algorithm called CONTACT implements this theory. Though, its disadvantage is the high computational time which is needed. In order to solve this problem Kalker introduced a simplified theory [6, 7], which is commonly used for modelling the tangential contact between wheel and rail. The main difference in the simplified theory is that the deformation in one point only depends on the load at that point whereas in the previous theory this deformation depends on the load at all locations. The simplified theory is implemented in a numerical algorithm called FASTSIM [7]. FASTSIM calculates the creep forces in the contact area by using the results from the normal contact problem as input.

3.2.4 Archard’s Wear Model

In order to investigate the wheel-rail wear, software tools have been developed. A widely used wear model is Archard’s wear model [8] [14]. The model assumes that sliding is necessary for wear to take place. The wheel-rail contact patch, for a given application
3.2. WEAR MODELLING

and a given instant in time, should therefore first be divided into an adhesion zone and a slip (sliding) zone.

![Contact patch discretisation](image)

**Figure 3.3.** Contact patch discretisation [2].

For an element $\Delta \xi \Delta \eta$, in the slip zone, the Archard wear model gives a wear depth $\Delta \zeta$.

$$\Delta \zeta = k p \cdot \frac{\Delta s}{H}$$

(3.4)

where $k$ is the wear coefficient, $p$ is the contact pressure in the element, $\Delta s$ is the magnitude of the sliding distance of an element and $H$ is the material hardness.

The wheel wear during one revolution is obtained by summing the contribution $\Delta \zeta$ from each element $\Delta \xi \Delta \eta$ along the longitudinal strip on the wheel. In the same way the wear from parallel strips are obtained and then the contributions of each wheel can be summed to obtain the wheel uniform wear for longer travelled distances.

However, in this study and as the influence of the turnout should be investigated, the wear will also be calculated and presented piecewise around the wheel and as it runs along the turnout. In this way, it will be more obvious which parts of the turnout result in a higher wheel wear.

The wear coefficient $k$ is a complex parameter as it is not constant for a given material but it depends on contact pressure and sliding velocity. In order to apply a reasonably adequate wear coefficient wear maps are utilized. They describe the wear coefficient, as function of sliding velocity and contact pressure. There are different maps for various wheel and rail materials.

The wear map in Figure 3.4 presents four approximate regions, in which the wear coefficient varies within certain values. These values have been defined by laboratory measurements at dry conditions with usual values for tangent track and curves. Under the pressure limit of $0.8H$, which corresponds to 80% of the hardness, the wear coefficient
depends on amplitude of the sliding velocity. Above this limit the wear coefficient is one order of magnitude greater than below the limit and as a result the wear conditions become catastrophic for the material.

### 3.3 Wheel Wear Prediction Tools and Previous Work

In this section, a short review of previous work on wheel and rail wear prediction is presented and also previous work on rail wear along turnouts.

**Chudzikiewicz and Kalker**

Chudzikiewicz and Kalker have investigated the evolution of wheel profile wear [7]. Various wheel-rail contacts and dynamic models have been used for this study. Kalker’s complete theory was firstly combined with Hertz’s theory in a algorithm called CONTACT and then Kalker’s simplified theory has been combined with Hertz’s theory in an algorithm called FASTSIM. The aim of this study was to compare the results of the two algorithms and determine whether FASTSIM, which is a faster algorithm, is a good tool in the context of wear. The conclusion from this study was that FASTSIM had acceptable results. The difference between the predicted wheel wear with the two different algorithms was about 10%.

**Kalker**

Later on Kalker used the previous method to calculate the wheel wear [15] and simulate the wheel tread wear of a metro train in Amsterdam. He also compared the results with measurements. In this study there is only one rail profile and the wear on the rails is neglected. The algorithm FASTSIM has been used for calculating the creep forces, and the contact area with the normal pressure are calculated according to Hertz’s theory.
It is assumed in this study that the volume removed per contact patch area is proportional to the frictional work in the wheel-rail contact.

\[ V = k \frac{T d}{H} = k \frac{W}{H} \]  

where \( V \) is the volume of the worn off material, \( k \) is the wear coefficient, \( d \) is the sliding distance, \( T \) the frictional force, \( H \) the hardness of the material and \( W = T d \) is the frictional work. From the results of this study, and according to Kalker, the Hertzian contact model is not adequate to predict severe wear.

\section*{Jendel}

Jendel has developed a wheel profile wear prediction tool which has been applied to a X10 rail vehicle which is operating on Stockholm’s commuter rail network. The model is built in the GENSYS software and it considers only the uniform wear. Track irregularities are also included in the study. The wheel-rail contact is modelled by the Hertzian theory in combination with Kalker’s simplified theory (FASTSTIM) for the tangential solution.

The results obtained from the previous method are compared with the results taken by the calculation with the CONTACT algorithm. The outcome of this study is that the general worn profiles of the two methods are very close despite some differences in the results and as the CONTACT method is much slower, FASTSIM is considered adequate for these calculations.

The outputs from the vehicle-track simulations are the inputs to Archard’s wear model (see Section 3.2.4). According to Archard’s wear model the sliding distance depends on where it is evaluated either in adhesion zone, in which there is no wear, or in the slip zone.

Jendel performed also a simulation with a running distance of 200000 kilometres and the resulting worn wheel profiles have been compared with the measured profiles. The conclusion was that the simulation results were very close to those of the measurements.

\section*{Enblom}

The main goal of Enblom’s PhD thesis [16] is to determine a suitable tool to predict wheel-rail wear for any condition. A sufficient accuracy for the use of the obtained profiles in vehicle dynamics simulations is a requirement in order to calculate the uniform wear by numerical simulation.

From a further investigation into the already proposed methods, Enblom concludes that Jendel’s method has the greatest potential in terms to generality and accuracy. So taking Jendel’s method as a base, several tests are performed regarding influence of disc braking, contact environment and contact modelling. Furthermore, Jendel’s wear method, Pearce and Sherratt’s and the one of Ward, Lewis and Dwyer-Joyce are compared regarding the energy dissipation.

Pearce and Sherratt assumed that the wear rate is proportional to the dissipated energy in the contact zone, taken as a product of creep force and creep [17]. Ward, Lewis and Dwyer-Joyce assumed the contact model to be approximately an ellipse, which is
separated into longitudinal strips corresponding to the wheel strips. Each strip is divided into equally sized cells and the profile wears out uniformly around the wheel [18].

The results of the above comparison showed that the tread wear is significantly underestimated by the Ward, Lewis and the Dwyer-Joyce method as compared to Jendel’s model. The flange wear predictions are of similar magnitude although a small difference for high friction in the case of Jendel’s method. Generally, Jendel’s and Pearce and Sherratt’s methods present similar results in several cases despite the totally different tribological approaches.

A conclusion of Enblom’s study is that the tread wear depth and distribution in the simulation agree with the measurements very well. Moreover, Hertz’s contact theory is considered valid and the flange wear results could be improved by further development of wear maps.

Orvnäs

Orvnäs investigated the wear of the rail profiles for the Swedish light rail line Tvärbanan in Stockholm [19]. Simulation results were compared with measurements for four different curves of the line. The wear prediction tool which was used for this study is Jendel’s method and the wheel-rail contact mechanics is modelled according to the Hertzian theory combined with the simplified Kalker’s theory through the commercial software GENSYS.

Wear calculations are performed by a programme developed in MATLAB which implements Archard’s wear model. The wear coefficients are already determined by laboratory studies at dry conditions and they are reduced for natural and deliberate lubrication.

The presented results of the rail wear prediction tool do not agree with the measured rail profiles very well, since the simulated rail wear is more extensive than the measured one, especially on the outer rail. However, the simulated worn rail profiles seems to have a relatively similar shape as the measured rail profiles.

Sánchez

Sánchez studied the wear prediction tool of Archard’s model, which is implemented in SIMPACK [20] vehicle dynamics simulation environment, and simulates the Flexible Swift vehicle (A32) in Stockholm’s commuter service on the line Tvärbanan [21].

The calibration of the wear prediction tool carried out by testing various settings and determine which of them give better match with the real conditions of the vehicle. Furthermore, wheel and rail wear as well as profile evolution results were compared with existing measurements.

The conclusion of this study was that the simulated wear at the far tread and flange parts of the wheel were similar to the measurements. On the other hand, the results for the middle part of the wheel were not so good, as the measurements showed a quite evenly distributed wear along the profile, while the simulated results showed larger differences between the extremes and the middle part of the wheel. So more tests would be necessary in order to obtain an optimal solution.
Kassa

The main goal of Kassa’s PhD thesis was to simulate and investigate the dynamic interaction between train and turnout [4]. For this reason two models for simulation have been developed. The first one is derived by using a commercial software for dynamics of multi-body systems and the second is based on a detailed model of track dynamics and multi-body dynamics formulation that accounts for excitation in an extended frequency range. Hertzian theory and FASTSIM are used for the normal and tangential wheel-rail rolling contact respectively.

For a given nominal layout of the turnout, the influence of various parameters on wear and on rolling contact fatigue is investigated. Four of the parameters (axle load, wheel-rail friction coefficient and wheel and rail profiles) were identified as the most significant.

The simulation results were compared with measured data in the field. The conclusion was that there was a good agreement between measured and calculated contact forces and the influence of train speed, moving direction and route on the measured wheel-rail contact forces is quantified.

Pålsson

Pålsson studies in his licentiate thesis [22], the dynamic interaction between vehicle and turnout by using numerical tools for multi-body dynamics with focus on laying a foundation for robust optimization of turnout geometry.

The influence of wheel profile wear on wheel-rail interaction in a turnout is also studied and it is concluded that equivalent conicity is the characterization parameter with the best correlation to rail wear of the investigated parameters. The influence of hollow-worn wheels on rail damage is investigated as well and the result was that this type of wheels profiles display a different and probably more harmful running behaviour at the crossing. A good correlation between the friction and the lateral contact forces and the wear in the diverging route was shown. Good agreement between the simulation model and field measurement data has been observed. It is also concluded that the use of more resilient rail pads can reduce wheel-rail impact loads during the crossing transition.

Present Study

In this study, Jendel’s method is used and the wheel-rail contact dynamics are modelled in the commercial software GENSYS v.1301 [5]. The wheel wear calculations are performed by a programme developed in MATLAB which implements Archard’s wear model.

The main goal of this MSc thesis is to simulate and calculate the wheel wear for a freight rail vehicle which passes a turnout and either follows the diverging or the straight direction. The investigated factors are apart from the geometry of the turnout, the worn wheel profiles, the worn crossing nose, the different stiffness of the stock and the switch rails at the beginning of the turnout, and the running speed for the straight direction.
Chapter 4

Simulation Cases and Input Data

In this chapter, the various simulation cases are described along with all the parameters which have been changed for the different cases for analysing their contribution to wheel wear. But first the multi-body system model (MBS) of dynamic interaction between the freight wagon and a standard turnout design is also presented. This includes wheel-rail geometry, track model, vehicle input data and load case.

4.1 Wheel-rail Geometry

Two of the main factors that affect wear and consequently wear simulation are the wheel and rail profiles. As already mentioned in Section 2.4, the wheel profile used in this study is the standardized type UIC/ORE S1002 (Figure 2.4), a very typical wheel profile for freight wagons in Sweden.

![Figure 4.1. Rail profiles of the turnout’s left rail.](image)
For the rail geometry, and in order to simulate a moving vehicle along a turnout, there are several rail profiles along the track presenting the changing rail shape of the turnout. More precisely, there are thirty rail profiles (see Figure 4.1). They are presented in a straight line as they are modelled and positioned at certain distance from the centre of the track; then, curvature can be introduced for the track design to account for the diverging track curve radius. Some of the profiles are located very close to each other as the geometry of the turnout changes rapidly in a short distance. The two sections where this happens are at the beginning of the turnout and at the nose part. One is when the switch rail appears and the second when the switch rail goes away from the track and the other stock rail appears through the nose part, which can be seen in detail in the Figure 4.2. The diverging part of the simulated turnout is a right turn.

4.2 Track Model

The turnout chosen for the simulation is a standard design UIC60-760-1:15. This means UIC60 rails with no inclination, the diverging part of the turnout has curve radius 760 m and the crossing angle is 1:15. This geometry implies a speed limit of 70 km/h for trains running through the diverging part.

The wheel wear is investigated for both possible directions (curve and straight) that the vehicle can move on the turnout. A right curve is used for the simulations, but in order to alter the direction of the vehicle in the turnout, the curve radius is set to infinite and the rail profiles stay the same. So, the simulated turnout for the train which goes straight is a left turn turnout and the switch rail and the nose part interact with the left wheels of the train (Figure 4.3). This also simplifies the fact that the wheels which are affected by the switch rail and the crossing nose are the same (the left wheels) in both cases.
4.2. TRACK MODEL

Figure 4.3. The two turnout geometries used for the curve (A) and straight (B) directions in the simulation cases.

In all simulations, track irregularities are neglected. The main reason for this is to study the influence of turnout geometry on wheel wear without additional disturbances and how this wear changes when certain parameters are altered, like a worn nose part.

The dynamic model of the track is shown in the Figure 4.4. The track model is a mass-spring-damper model which is moving along the track under each wheelset in the vehicle model. This track model is a standard GENSYS track model that includes linear springs and viscous dampers in the lateral and vertical directions for each rail. The lateral movement of the track piece is modelled with a stiffness $k_{ytg} = 30 \cdot 10^6$ N/m and a damping $c_{ytg} = 300 \cdot 10^3$ Ns/m. The rails are massless and attached on the track. Each one of the rails has two degrees of freedom, a lateral and a vertical displacement. The lateral stiffness is $k_{yrt} = 42 \cdot 10^6$ N/m, lateral damping $c_{yrt} = 400 \cdot 10^3$ Ns/m, vertical stiffness $k_{zrt} = 75 \cdot 10^6$ N/m and vertical damping $c_{zrt} = 1600 \cdot 10^3$ Ns/m [4]. These values are also consistent with the literature where values of the same magnitude are used [23].
CHAPTER 4. SIMULATION CASES AND INPUT DATA

4.3 Vehicle Setup and Load Case

In this section the vehicle setup and the load case used in the present study are presented. As mentioned in the vehicle description in Section 2.3, the utilized vehicle is a two-axle freight wagon with Unitruck running gear. The load case of the wear simulations is for 22.5 tons axle-load. The wagon is considered fully loaded, so the mass of the carbody can be calculated as the permitted load minus the weight of the wheelsets and the saddles (Eq. 4.1).

$$m_c = 2 \cdot 22500 - 2 \cdot m_a - 4 \cdot m_s$$  \hspace{1cm} (4.1)

The components included in the vehicle are the carbody, wheelsets and saddles. The centre of gravity of the carbody is defined as the height of its mass centre above the track plane. The moments of inertia of the carbody with respect to body’s centre are also needed. All parameters are listed in Table 4.1.

4.4 Simulation Cases

For investigating the influence of the turnout on wheel wear, various simulation cases with altered parameters have been defined. In this section these different cases will be described. These parameters are referred either to the turnout geometry or to the vehicle properties. In order to be able to compare the wheel wear results, each comparison applies to cases that have only one different parameter for investigating the effect of this particular parameter when the running on a turnout.

In this way there will be three general groups of simulation cases (Table 4.2). The first group concerns a new unworn turnout, which will be called 'normal case' hereafter; and the second group will concern a worn turnout geometry. More specifically the nose part of
4.4. SIMULATION CASES

Table 4.1. Vehicle and Track Data

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbody</td>
<td>Mass [kg]</td>
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</tr>
<tr>
<td></td>
<td>Moment of Inertia [kg m²]</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$J_{xx}$</td>
<td>62760</td>
</tr>
<tr>
<td></td>
<td>$J_{yy}$</td>
<td>584923</td>
</tr>
<tr>
<td></td>
<td>$J_{zz}$</td>
<td>584923</td>
</tr>
<tr>
<td>Saddle</td>
<td>Mass [kg]</td>
<td>140</td>
</tr>
<tr>
<td></td>
<td>Moment of Inertia [kg m²]</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$J_{xx}$</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>$J_{yy}$</td>
<td>400</td>
</tr>
<tr>
<td></td>
<td>$J_{zz}$</td>
<td>200</td>
</tr>
<tr>
<td>Wheelset</td>
<td>Mass [kg]</td>
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</tr>
<tr>
<td></td>
<td>Moment of Inertia [kg m²]</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$J_{xx}$</td>
<td>1000</td>
</tr>
<tr>
<td></td>
<td>$J_{yy}$</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>$J_{zz}$</td>
<td>1000</td>
</tr>
<tr>
<td></td>
<td>Wheel radius [m]</td>
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</tr>
<tr>
<td></td>
<td>Wheel profile</td>
<td>ORE-S1002</td>
</tr>
<tr>
<td></td>
<td>Longitudinal semi-distance between wheelsets [m]</td>
<td>4.6</td>
</tr>
<tr>
<td>Track</td>
<td>Turnout type</td>
<td>UIC60-760-1:15</td>
</tr>
<tr>
<td></td>
<td>Track gauge [mm]</td>
<td>1435</td>
</tr>
<tr>
<td></td>
<td>Track cant</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>Rail inclination</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>Rail profile</td>
<td>30 sampled cross-sections</td>
</tr>
</tbody>
</table>

the turnout has been modified and in terms of modelling, the rail profiles at the nose part appear on the track with a delay, which means that they are repositioned longitudinally. Only the rail profiles of the nose are delayed and not the profiles of the wing rail. In this way the geometry of the nose is practically relocated to another longitudinal position.

As mentioned in a study concerning the deformation and damage of a crossing nose due to wheel passages performed by Wiest et al. [24], the maximum vertical displacement for a steel crossing, which was caused by the wheels passing the crossing, is -0.058 mm after the first five cycles. This study concluded that the failure in composite steel crossings is probably due to other mechanisms than plasticity-induced damage. Though there is greater damage on the crossing nose. So through the relocation of the nose part, a difference of 2 to 3 mm is created between the unworn and the worn crossing nose, as shown in Figure 4.5.

Moreover, the third group of simulation cases concerns a turnout in which the switch rail has different vertical stiffness than the stock rail between the rail and the sleeper. This study is more focused on the beginning of the turnout where the switch rail is introduced and there is a part where stock and switch rail are in parallel positions and very close to each other. As a result, the wheel can have contact with both of them.

The three simulation case categories are then divided into sub-cases. So the cases of unworn, worn and with different stiffness turnouts are studied for the two possible movements of the trains, i.e. the diverging direction and when the train runs on the straight
part of the turnout. For the diverging part of the turnout there is a speed limit of 70 km/h, but for the straight part the limit is not so low. For that reason, various running speeds of the train have been tested for the straight part in order to determine if the running speed results in a higher wheel wear in certain parts of the turnout. Furthermore, the case of the unworn turnout will also be simulated with worn wheel profiles for both turning and straight part of the turnout.
Chapter 5

Results and Discussion

This chapter includes the presentation, analysis and comparison of the results of the various simulations, starting with the simulation cases of the unworn turnout and unworn S1002 wheel profile for both the diverging and the straight direction of the turnout (simulation cases 1.1 and 1.2). These cases are called initial cases. Then these results will be compared with the results of simulations with worn wheel and then with the results of simulations with worn crossing nose for unworn wheels. In the end, the influence of different stiffness of the stock and the switch rails on wheel wear will be investigated in the beginning of the turnout. Especially for the straight direction of the turnout, another parameter will be investigated, the speed.

5.1 Simulation Case 1.1 - Initial Curve

Simulation case 1.1 deals with the diverging direction of the unworn turnout and the wagon is equipped with original (unworn) wheels. The speed of the vehicle \( v \) is 50 km/h and the friction coefficient \( \mu \) is 0.3.

The vehicle speed is thus set below the speed limit of the diverging direction of the turnout (70 km/h), for checking the influence of the turnout for a more common speed and avoiding possible extreme behaviours. Results from different speeds will be presented in the next simulation case with the straight running of the vehicle on the turnout.

The wheel wear results are shown in the following Figures 5.1-5.4 for the four wheels of the freight wagon. In the longitudinal direction zero determines the beginning of the turnout while the simulation begins with the position of the rail vehicle (centre of gravity) 10 m before the beginning of the switch. Furthermore, the crossing nose appears at about 46 m after the turnout entry and the first contact with the crossing nose is at about 47 m after turnout entry. On the \( y \)-axis the lateral position of the contact points on the wheel is shown. Additionally, wear is presented as iso-lines with the same wear depth value. There are several lines in some areas and this means that wear is spread in the lateral direction. Also, the wear at some sections (switch and crossing nose) is so high that the scale is not very good, but allows the representation of the lateral position of the contact patch area in the same figure. When \( y = 0 \) the wheel is rolling at its nominal running
circle and positive $y$ represents positions towards the flange. All figures concerning the right wheels are in different scale than the figures of left wheels as wear values are smaller.

**Figure 5.1.** Wear prediction for the *front left* wheel towards diverging direction, initial case.

**Figure 5.2.** Wear prediction for the *front right* wheel towards diverging direction, initial case.
5.1. SIMULATION CASE 1.1 - INITIAL CURVE

It is obvious that there are two mainly areas of the turnout which cause high wheel wear, especially high tread wear. These two areas are the beginning of the turnout where the switch rail appears and also the area of the crossing nose in combination with the area before that, where the wing rail goes out of the track centre. High wear is observed for
both outer wheels which pass this rail geometry. These two areas are also mentioned in Kassa’s PhD thesis [4] and Pålsson’s licentiate thesis [22], in which both vertical and lateral contact forces present an increase and also the leading wheelset is moving significantly in the lateral direction (outwards).

Specifically in Figures 5.1 and 5.3, which correspond to the outer wheels, the large wear cannot be fully visualized. In the contour plots the magnitude has been adjusted in order for the other wear areas to be visible. More detailed and focused 3D-plots in these areas will be presented below. Furthermore and for example in Figure 5.4, there is a lack of wear iso-lines in some areas. This is because the wear is very small and it is smaller than the used interval.

The inner (right) wheels have a smoother distribution of the wheel wear along the turnout. The main reason of this is that the inner wheels do not pass from the switch rail and the crossing nose, but they are following constantly the right stock rail. Higher values of wheel wear appear in the same areas like the wear of outer (left) wheels. The rear wheels present a movement of the contact point and, as a result, a spread wear at the moment when the front wheels enter the turnout, despite the fact that they are still running on tangent track.

Figures 5.5 and 5.7 are more detailed figures that focus on the area between 4 and 8 metres in the longitudinal dimension of the turnout and on the area of the crossing nose respectively. The red part in the figures depicts the flange area of the wheel profile (lateral position 34 to 60 mm from the nominal running circle).

As shown in Figures 5.5 and 5.6 there are two contact points in the area when the switch rail appears at the longitudinal position on the turnout from about 4 to 7.5 m. Flange contact and high wear are also two main characteristics of this part. The wheel hits the switch rail with high wear in two sections. In the first section, where the initial contact with the switch rail is, there are two contact points and the wear is greater. After 7.5 m, the stock rail loses contact, the wheel changes rail and the wear returns to more normal
values. This phenomenon presents some differences for the front and the rear wheel. At the front wheel the high wear values appear about 4 m from the beginning of the turnout whereas for the rear wheel it starts at about 6 m. This is normal as the freight wagon has already changed direction of running and the rear wheel does not hit the switch rail as abruptly as the front wheel. The area between 5.5 and 6.5 m has lower wear values and this is probably because of the model and the wear coefficient. If the contact pressure or the sliding velocity change, we might jump to another area of the wear map in Figure 3.4. In the catastrophic wear area, over $0.8H$, the wear coefficient could be even 10 times higher than its common values.
The other part of the turnout which presents high wear is the crossing nose (Figure 5.7), in which wear on the outer part of the wheel tread appears when the switch rail goes away from the track centre and increases until the wheel contact with the crossing nose. Then high wear values are detected for the first contact points with the crossing nose. The values are so high at this specific area that they have been limited in the figure in order for the rest of the wear to be visible. The highest calculated wear depth value in this part is $1.047 \cdot 10^{-6}$ m corresponding to the first contact point between the wheel and the crossing nose. This phenomenon is gradually reduced, but the wear values are still at least one order of magnitude greater than in other parts of the turnout. This implies a very concentrated wear on the wheel flange which might cause out-of-round wheels.

As already mentioned, the simulation cases are taking into account a laden freight wagon. In case of a tare vehicle, the wheel wear will be lower (Figure 5.8).

![Figure 5.8. Wheel worn-off area for a single run along the turnout and for front left wheel, comparison for laden and tare freight vehicle.](image)

The figure shows that the worn-off area (the removed area of the wheel profile) of the laden vehicle for the initial curve case is about five times greater than for the tare vehicle. The axle load for a laden vehicle is 22.5 tons and for a tare vehicle 6 tons. The relative increase of the wear between these cases is this greater than the relative increase of the axle load. An increase of the axle load will result in an even greater increase of the wheel wear. The total wear volume for the tare vehicle is $3.707 \cdot 10^{-10}$ m$^3$ and for the laden case $21.83 \cdot 10^{-10}$ m$^3$. 
5.2 Simulation Case 1.2 - Initial Straight

The second initial case investigates the wear on the wheel while the freight wagon runs on the straight part of the turnout. There are not many differences in the general geometry of the track with which the vehicle has to deal, apart from the fact that the direction of travel does not change so the centrifugal forces are avoided. The left wheels though have to pass from the stock rail to the switch rail and then to the crossing nose for continuing to the straight track. As already mentioned in Section 4.2, the simulation for the straight direction of the turnout is generated by changing the curve radius and by keeping all the rail profiles the same. So it concerns a turnout with the diverging track towards the left. In this way the comparison of the results with the initial curve case will be between the same wheels which will pass the same geometry; for example, left wheels in both cases pass through the crossing nose.

As for other input parameters, the friction coefficient remains the same like in the previous case, \( \mu = 0.3 \), while the speed of the vehicle \( (v) \) is the same for the first run and then increased for investigating the influence of the speed while passing the turnout. The speed limit for the trains which are not taking the diverging direction of the turnout is greater than 70 km/m, so two more simulations have been carried out with 100 km/h and 160 km/h respectively.

In the following Figures 5.9, 5.10, 5.11 and 5.12 the wear prediction results for the four wheels are presented, for a speed of 50 km/h.

![Wear Prediction](image.png)

**Figure 5.9.** Wear prediction for the front left wheel towards straight direction, initial case (50 km/h).
If the results of the wear prediction for the straight part of the turnout are compared with the results from the diverging part, and for the same vehicle speed of 50 km/h, a very similar behaviour of the position of the wheel-rail contact point can be observed. For the left wheels which pass from the wing rail to the crossing nose there is a faster alteration
5.2. SIMULATION CASE 1.2 - INITIAL STRAIGHT

Figure 5.12. Wear prediction for the rear right wheel towards straight direction, initial case (50 km/h).

of the position of the contact point. This has an impact on greater tread wear in the area just before the crossing nose, whereas there is no flange contact when the freight wagon follows the straight track due to the lack of centrifugal forces.

The wear in the area close to the crossing nose can be seen in Figure 5.13 and a lateral displacement of the wheel contact point is visible. Again, the top values in the figure have been limited in order for the wear to be visible along the whole area. The contact point is moving laterally while being on the wing rail and, as a result, the wear in this region is greater than in the previous case. Furthermore, when the contact point moves on the crossing nose the high wear values appear in the first part but also reappear after a while instead of phasing out like in the initial curve case (Figure 5.7). This is due to the dynamic impact against the nose.

In addition the right wheels do not have to deal with the switch rail and the crossing nose and that is the reason why the wear predictions along the turnout for front and rear wheels as also for the curve and the straight case are very similar.

Despite the fact that there is no track curvature, the freight wagon has to deal with the same geometry as when it follows the diverging direction of the turnout. So it is quite reasonable that the wear distribution and the position of contact points have several similarities between these two simulation cases.

The wear around the crossing nose in Figure 5.13 is quite high and very concentrated. As the whole longitudinal distance of this phenomenon is about 2.5 m and the perimeter of the wheel is about 2.89 m, this wear is not evenly distributed around the perimeter of the wheel. As mentioned before, this might lead to out-of-round wheels and corrugation.
CHAPTER 5. RESULTS AND DISCUSSION

Figure 5.13. Wear prediction for the front left wheel towards straight direction and around the crossing nose.

Continuing, the influence of the speed is now investigated. The impact of the turnout is about the same in both cases while taking the diverging part and while running on the straight part, but the speed limit is not the same for both possible ways. So Figure 5.14 depicts the maximum wear depth, which could be at different lateral position of the wheel profile, along the turnout for the speeds 50, 100 and 160 km/h.

Figure 5.14. Maximum value of wheel wear in each simulation’s time step and towards straight direction.

As seen, an increase of the vehicle speed results in an increase of the maximum value of wear in each time step. Especially for the speed of 160 km/h, there is a significant
5.2. SIMULATION CASE 1.2 - INITIAL STRAIGHT

increase of wear in comparison with the wear concerning the speeds of 50 and 100 km/h. This difference is mainly located in the area where the vehicle changes from the stock rail to the switch rail. Thus for higher speed, harsher dynamics appear and the impact at the discontinuities of the turnout is greater. By looking at the wear chart, we might jump to catastrophic wear area above the pressure level $0.8H$. For further comparison of wear depth, the two most different cases will be considered: 50 km/h and 160 km/h.

In both cases of 50 and 160 km/h, there is two-point contact between wheel and rail when the switch rail appears. As shown in Figures 5.15 and 5.16, the two-point contact for 50 km/h is observed after 7.5 m from the beginning of the turnout, whereas for the speed of 160 km/h the contact point remains on the stock rail for some more distance and then the contact with the switch rail is more aggressive and a higher maximum wear depth is displayed. On the other hand and for the speed of 50 km/h, the wear on the outer part of the tread is higher than the wear on the tread closer to the flange. This is a phenomenon that is not seen for the speed of 160 km/h in which there are lower values of wear on the outer part of the tread.

![Figure 5.15. Wear for the front left wheel and for the area where the switch rail is introduced, with vehicle speed of 50 km/h.](image)

Figures 5.17 and 5.18 represent the wheel wear depth in the area around the crossing nose for the speeds of 50 km/h and 160 km/h and have no great differences. There is no (relatively) high wear on the outer part of the tread in any case. However, a small increase of the wheel wear can be observed when the contact point is transferred to the crossing nose for the speed of 160 km/h. For the lower speed the wear has similar values for about 30 cm of running on the crossing nose, whereas for the higher speed this distance becomes greater than 40 cm and there are two peaks in the values of wear. One is higher than the average values of wear with the lower speed and the other one is lower. So also in this case the greater wear values are concentrated to small areas of the wheel perimeter which leads to an unevenly worn wheel.
The rear left wheel has similar behaviour, passing through the same rail geometry; however the results presented in this report are limited to the front left wheel where the greatest wear is observed. The wear for the right wheels is very low in comparison to the left wheels, as they follow constantly a stock rail without changes in the rail profile geometry along the turnout.
5.3 Simulation Case 1.3 - Curve with worn wheel

An important factor which has a great impact on the contact points and consequently on the wear is the wheel profile geometry. So in the simulation of this section the wear that is caused to an already worn wheel when it interacts with the diverging part (curve) of the turnout is studied.

As before, the simulation speed is set to 50 km/h. The shape of the worn wheel profile comes from measurements of the present freight wagon (Figure 5.19) [1]. The same nominal rolling radius has been used for the worn wheel profile as well. The additional wheel wear is shown in the following Figures 5.20-5.23 for the four wheels of the freight wagon.

In all four figures the contact points move more abruptly in the same areas as in the initial curve case. The contact points of the left wheels are moving from the very end of the tread until the flange, with high wear too. Moreover the contact point of the right wheels, which are moving on the stock rail, are located closer to the flange and there is flange contact in several cases along the turnout, which did not happen for unworn wheels.

The most important observation is that the worn wheels result in high tread wear and particularly high wear on the outer part of the tread. The contact point variations in the area around the crossing nose are extreme and the wear presents high peaks on the outer part of the tread as well. Because of the changing rail geometry at this point and the worn wheel profiles the results present an abrupt fluctuation. Furthermore, when the wheels contact the wing rail and in comparison with the initial curve case (Section 5.1), the contact point moves towards the very end of the tread. This phenomenon causes high wheel wear values and it lasts for longer distance. After the discontinuities have ended, the contact point returns to around the nominal point and the wear to lower values.
CHAPTER 5. RESULTS AND DISCUSSION

Figure 5.19. Worn wheel profile.

Figure 5.20. Wear prediction for the front left wheel towards diverging direction with worn wheels.
5.3. SIMULATION CASE 1.3 - CURVE WITH WORN WHEEL

Figure 5.21. Wear prediction for the *front right* wheel towards diverging direction with worn wheels.

Figure 5.22. Wear prediction for the *rear left* wheel towards diverging direction with worn wheels.

A more detailed image of the lateral position of contact points and the amplitude of wheel wear in the beginning of the turnout and around the crossing nose are shown in Figures 5.24 and 5.25 respectively. In the first figure, two-point contact can be observed and also flange contact with high wear values. Furthermore in Figure 5.25, very high wear
values appear in the outer part of the tread close to the contact with the crossing nose (about 47 m from turnout’s entry). In the figure there are some areas where the wear is not obvious as it is much lower than the wear in the areas next to them. The highest wear value is about $2 \cdot 10^{-5}$ m, which is an order of magnitude higher than the initial case.
This simulation case presents the greatest wheel wear values so far, and they do not appear at single positions but they last for some distance with the same magnitude. The contact point is changing position very quickly and there is two-point contact in several areas. So the worn wheel profile has a great impact on wheel wear and it accelerates this procedure by causing higher wheel wear in certain areas of the turnout.

5.4 Simulation Case 1.4 - Straight with worn wheel

Worn wheels result in high wheel wear for the diverging part of the turnout, so in simulation case 1.4 the influence of the worn wheels for the straight direction of the turnout will be investigated. As in initial straight case (Section 5.2), speed is a varying parameter and there are three simulations with speeds 50, 100 and 160 km/h.

The following Figures 5.26 - 5.29 depict the wear distribution on the wheel and along the turnout’s longitudinal direction for the speed of 50 km/h.

The wheel wear prediction is very similar for the straight part of the turnout as for the diverging part of it, despite the absence of centrifugal forces. This phenomenon was observed for the initial cases and it is also followed for the cases with worn wheel profiles. As mentioned in previous cases, there is high tread wear and especially in the outer area of the tread (Figures 5.26 and 5.28). The position of the contact point is moving abruptly in the parts of the introduction of the switch rail and in the crossing nose part.

The right wheels, which are not passing through changing geometries of the track, do not show high wear values; but because of the worn profiles, the position of the their contact points is closer to the flange, and there are several areas along the turnout where the flange contact lasts for some distance.
The variation of contact point, its relocation to the outer part of the tread and its high values after the introduction of the switch rail shown in the figures have a great impact on the wheel as they keep these characteristics for several metres. About 10 m after the beginning of the turnout the contact point returns close to the wheel’s nominal running circle and the wear has more average values.
Figure 5.28. Wear prediction for the rear left wheel towards straight direction with worn wheels (50 km/h).

Figure 5.29. Wear prediction for the rear right wheel towards straight direction with worn wheels (50 km/h).

Figure 5.30 depicts the maximum wear values for the left front wheel of the freight vehicle at three different speeds 50, 100 and 160 km/h. As the speed gets higher, the wheel wear increases. The high wear values appear about 10 m from the beginning of the turnout for all the three running speeds. The highest wear value around the crossing nose and for the speeds 100 and 160 km/h is a little repositioned in comparison with the case with the speed of 50 km/h which appears earlier. This is probably because of higher speed and
inertia, the wheel continues on the wing rail for some distance more and then changes to the crossing nose more abruptly. Generally, the maximum wear values are the greatest of all the simulation cases concerning the straight part of the turnout.

5.5 Simulation Case 2.1 - Curve with worn crossing nose

The common characteristics of the simulations in the second group, and at the same time the main difference from the previous simulation cases, is the worn crossing nose. As already mentioned a study concerning the deformation of a crossing nose due to wheel passages [24] concluded that the maximum vertical displacement for a composite steel crossing is -0.058 mm after the first five cycles of wheel passing. And also it is mentioned that the failure in composite steel crossings is probably due to other mechanisms than plasticity induced damage. So the simulated worn crossing nose, which has been presented in the Figure 4.5, has about 2 to 3 mm difference from the normal one. This difference is created by only relocating the rail profiles of the crossing nose and not the rail profiles of the wing rail.

The speed of the vehicle has been kept the same as in the previous simulations through the curve, 50 km/h. The results of wear for the four wheels while the freight wagon follows the diverging part of the turnout are presented in the Figures 5.31-5.34.
5.5. SIMULATION CASE 2.1 - CURVE WITH WORN CROSSING NOSE

As expected, the wear until the part of the crossing nose is exactly the same as in the initial curve simulation case (Section 5.1), whereas a difference in the wear prediction can be observed especially for the left wheels which are passing from the worn crossing nose.

The main alteration is that the wear is more extended along the $y$-coordinate of the wheel in the introduction of the crossing nose. The gap between the wing rail and the crossing

![Figure 5.31.](image1.png) Wear prediction for the front left wheel along the diverging part of the turnout with worn crossing nose.

![Figure 5.32.](image2.png) Wear prediction for the front right wheel along the diverging part of the turnout with worn crossing nose.
nose has been increased and as a result the wear around this specific point is greater and the contact point is moving a little more. The flange contact is more intense but generally the lateral movement of the contact point caused by the worn crossing nose is fading out in a short distance, and the wear results end up having the same values like the initial curve case.
5.6 SIMULATION CASE 2.2 - STRAIGHT WITH WORN CROSSING NOSE

The right wheels are not affected so much from worn crossing nose as the contact points have not changed and lateral movements of the freight wagon remains the same. So the right wheels have very similar behaviour and wear results as in the initial curve simulation case.

5.6 Simulation Case 2.2 - Straight with worn crossing nose

The influence of the worn crossing nose is here investigated while the freight wagon is running along the straight part of the turnout. Figures 5.35-5.38 present the wear for the four wheels of the wagon running at 50 km/h. Furthermore, the maximum wear values are compared along the turnout for the speeds of 50, 100 and 160 km/h. As mentioned before the speed limit of 70 km/h corresponds only to the diverging part of the turnout.

![Figure 5.35](image)

**Figure 5.35.** Wear prediction for the front left wheel towards straight direction with worn crossing nose.

For this case, wear values have increased around the crossing nose. More specifically, high wear values can be observed in the outer part of the tread, and the contact point has also moved towards the outer part of the tread compared to the normal case with the unworn crossing nose. Moreover, the lateral movement of the contact point right after the contact with the crossing nose is greater and, as a result, wear appears in a wider region on the wheel surface.

Generally, a worn crossing nose results in a movement of the contact point towards the outer part of the tread, and also an increase of the wear values. This is also obvious in the comparison of the maximum values of wear for the three different running speeds (Figure 5.39). In this case, the highest wear value with worn crossing nose and for speed 160 km/h is about $2.4 \cdot 10^{-6}$ m and the same value for the initial case is $1.6 \cdot 10^{-6}$ m.
While the speed increases, wear peaks have greater values. As a result, the freight wagon needs more running distance until the wear values fall to the same values as in the initial straight case with the unworn crossing nose. A worn crossing nose results in higher wear values in an area larger than the longitudinal dimension of the worn nose.
Figure 5.38. Wear prediction for the *rear right* wheel towards straight direction with worn crossing nose.

Figure 5.39. Maximum wear value towards straight direction for *worn crossing nose* and for three different running speeds (*front left* wheel).
5.7 Simulation Case 3.1 - Curve with reduced switch rail stiffness

In the last two simulation cases, the influence of reduced stiffness for the switch rail is investigated. These cases are focused on the part of the turnout beginning, where the freight wagon passes from the stock rail to the switch rail. These two rail blades may have different vertical stiffness in their connection with the track, as one is connected permanently to the sleepers while the second one is a moving part and their stiffness may differ. The part of the crossing nose can not be investigated for this phenomenon as usually the crossing nose is a cast part with part of the switch rail which ends there and part of the nose and the stock rails which start at this point.

As referred in [23] and according to measurements in a turnout in Sweden, the vertical stiffness of the switch rail in the beginning is up to 70% lower than the stiffness in the end of the switch rail where the rail is stationary. In this study it is assumed that the difference in stiffness between the stock and the switch rail is 50% and that this difference is applied in the stiffness between wheel and rail and not between rail pad and sleeper. The reason for this assumption is that the modifications which were needed in the software for implying different stiffness for the two rails against the sleeper in the same position, were exceeding the scope of this master thesis. Still, the present results of the wear will give a good image of the phenomenon and how the difference in stiffness influences the wear.

So the vertical stiffness of the switch rail used in the simulation cases 3.1 and 3.2 is $1200 \cdot 10^6$ N/m, whereas the stiffness concerning the stock rail is $2400 \cdot 10^6$ N/m. In Figures 5.40 and 5.41 the wear results are presented for the left wheels. The right rail has no change, the running behaviour of right wheels is thus not significantly affected by the variation on the left wheels and the wear results are exactly the same as the initial case.

If Figures 5.40 and 5.41 are compared with Figures 5.5 and 5.6 from the initial curve case, it is obvious that the wear values concerning the contact of the wheel with the stock rail have increased, whereas the values concerning the contact of the wheel with the switch rail have slightly decreased and the transition is smoother. For example, the increase of flange wheel wear which is shown in this area with the different stiffness, presents a smoother wear increase instead of steep peaks which appear in the initial case. In the initial case the first peak reaches the value of $3 \cdot 10^{-6}$ m, whereas in the present case it is less than $2 \cdot 10^{-6}$ m.

So when the stiffness of the switch rail is 50% lower of the stock’s rail, there is an increase of tread wear and a slight decrease of the wear closer to the flange. The transition from the stock rail to the switch rail is more progressive than a sudden geometry change.
Figure 5.40. Wear prediction for the *front left* wheel along the beginning of the turnout towards diverging direction, with reduced stiffness in the switch rail.

Figure 5.41. Wear prediction for the *rear left* wheel along the beginning of the turnout towards diverging direction, with reduced stiffness in the switch rail.
5.8 Simulation Case 3.2 - Straight with reduced switch rail stiffness

The final simulation case concerns the straight part of the turnout with different vertical stiffness in the two rail blades, the stock and the switch rail. For investigating also the influence of the running speed while there is lower stiffness at the switch rail, three simulations have been carried out for running speeds of 50, 100 and 160 km/h.

The stiffness values have kept the same as in the simulation case 3.1, which means $1200 \cdot 10^6$ N/m for the switch rail and $2400 \cdot 10^6$ N/m for the stock rail. Figures 5.42- 5.44 present the calculated wear in the beginning of the turnout which appears in the front left wheel of the freight wagon.

![Figure 5.42. Wear prediction for the front left wheel along the beginning of the turnout with reduced stiffness in the switch rail and towards the straight direction with speed 50 km/h.](image)

In comparison with the case of the diverging direction, there is no flange contact in these cases. Also, high wear values appear about 7 m from the beginning of the turnout, whereas in the previous case they appeared earlier at about 4 m from the beginning of the turnout. This is caused by the different stiffness as the train runs towards the straight direction of the turnout, the wheel keeps contact with the stock rail and the higher stiffness. The contact with the switch rail gives low wear values because of the reduced stiffness. When finally the stock rail goes away from the running track the wear increases in both contact points with the stock and the switch rail.

Furthermore, there are high wear values in the left part of the tread for the speed of 50 km/h, whereas this phenomenon disappears for higher speeds (100 and 160 km/h). Still, greater wear values can be observed in the part of the wheel closer to the flange, which is the wear caused by the contact with the switch rail.
5.8. SIMULATION CASE 3.2 - STRAIGHT WITH REDUCED SWITCH RAIL STIFFNESS

Figure 5.43. Wear prediction for the rear left wheel along the beginning of the turnout with reduced stiffness in the switch rail and towards the straight direction with speed 100 km/h.

Figure 5.44. Wear prediction for the front left wheel along the beginning of the turnout with reduced stiffness in the switch rail and towards the straight direction with speed 160 km/h.

This simulation case has many similarities with the initial straight case (Figures 5.15 and 5.16). In both cases the high tread wear values caused by the contact with the stock rail are reduced when the freight wagon runs with higher speed. But this case with the
different stiffnesses presents higher tread wear caused by the contact with the stock rail for 50 km/h than in the initial straight case. For the higher speeds the different stiffnesses have practically no effect, as the calculated wear values are very similar.

5.9 Comparison

In this section the different simulation cases are compared, in terms of worn-off area and position of the worn-off material of the front left wheel. The integral of the computed wheel wear across the width of the wheel (worn-off area) has been calculated, and the wear values for the same angular position of the wheel have been added (integer number of revolutions) in order to have an overall wear around the wheel’s perimeter for one passage of the turnout.

The compared simulation cases are the initial case (1.1-1.2), the case with worn wheel profiles (1.3-1.4), the case with worn crossing nose (2.1-2.2) and the one with different stiffnesses for stock and switch rails (3.1-3.2). In the following Figures 5.45, 5.46 and 5.47, the case for the diverging direction and the cases for the straight direction of the turnout with running speeds of 50 km/h and 160 km/h are compared.

The figures show the calculated wheel worn-off area as a function of angular position. For visualization purposes, the calculated worn-off area has been scaled (x10^8) and then subtracted from the wheel nominal circle (r₀ = 0.46 m).

![Figure 5.45](image_url)

**Figure 5.45.** Angular position on the wheel of worn-off area for the front left wheel after passing the diverging part of the turnout.
In all figures, the worn-off area of the worn wheel profiles presents the maximum wear value and it has a significant difference with the other cases, which are quite similar. The initial case and the worn crossing nose case have very similar material removal for the speed of 50 km/h, whereas for the worn crossing nose and speed 160 km/h an additional curve appears which shows that the speed is a parameter which increases the wheel wear and also changes the position where the largest worn-off area appears.

The case with the different stiffness of stock and switch rails, presents some differences with the three other cases, but also these differences show lower peaks and higher average values in the figure. As a result, it creates a more uniform wear on the wheel than the other cases. For these cases, it is assumed that the switch rail has different stiffness from the stock rail along all its length, but in reality the stiffness increases from the beginning of the switch rail until the crossing nose part which is a cast part, as mentioned earlier. Therefore, the overall calculated worn-off area for this case may differ, but it allows to show the positions of the concentrated wear on the wheel surface.

In addition, the wheel worn-off area computed for the straight direction is much less than in the diverging direction of the turnout for all the simulation cases (Figure 5.46). However, the case with the speed of 160 km/h presents worn-off area, almost half of the volume of the diverging part despite the different speed. The higher speed for the straight case increases the wear volume around the perimeter of the wheel with more peaks but also the wear values at the position of the peak values of 50 km/h are smaller. In a railway network, a train passes along the straight part of a turnout much more often than along...
CHAPTER 5. RESULTS AND DISCUSSION

Figure 5.47. Angular position on the wheel of worn-off area for the front left wheel after passing the straight part of the turnout with speed of 160 km/h.

the diverging part and at higher speeds, so this can rapidly lead to worn wheels which increase the wheel wear even more.
Chapter 6

Conclusions and Future Work

Conclusions

The goal of this study was to assess the influence of turnouts on wheel wear of a freight vehicle. For this purpose several simulations were carried out alternating some key parameters of the turnout. The results of the main simulation categories of unworn, worn and different-stiffness turnout as well as worn wheel profiles have been presented in the thesis.

When a rail vehicle runs through a turnout, it deals with an alternating rail geometry. The wheels of one side of the vehicle first contact with the stock rail. Then, the switch rail appears and the contact point moves to this rail when the stock rail goes away. While this movement, there is an area where the wheel contacts the rails at two points, one with the stock rail and one with the switch rail. As a result high wear values have been observed in this area. The last part of the turnout is the crossing nose, where the wheel leaves the wing rail and contacts with the crossing nose. This is another area with high wheel wear values.

In general, it is concluded that the turnout has a significant impact on wheel wear, especially where there is change in rail geometry. This is happening in the beginning of the turnout in which the switch rail is introduced and also in the crossing nose part. The rail sections in between do not have a remarkable impact on wheel wear in comparison to the two mentioned areas. High wear values have been observed for the wheels passing from the switch rail and the crossing nose, which are the outer wheels when the freight wagon follows the diverging part of the turnout.

Similar influence of the turnout on wheel wear has also been observed when the train follows the straight direction. The speed limit of the straight direction is higher than 70 km/h which is the speed limit of the present diverging direction. The investigation has been carried out for speeds of 50 km/h, 100 km/h and 160 km/h. The lower speeds cause higher wear values towards the outer part of the tread, whereas an increased speed presents lower wear in the outer part of the tread but a little higher wear values appear closer to the flange.

The greatest impact on wheel wear is the case of the worn wheels. Worn wheel profiles are very common results of train operation. As shown, they cause even higher wear values
in the very end of the tread and they increase the variability in the lateral position of the contact point. This was obvious in both running cases of the vehicle towards the diverging part and towards the straight part of the turnout. Worn wheel profiles result also in higher additional wear of the wheels which do not pass through the switch rail and the crossing nose, the right wheels in the simulations. A higher lateral movement of the contact point of the right wheels has been observed in the areas where the left wheels pass through the switch rail and the crossing nose.

Another parameter which affects the wheel wear is worn crossing noses. A worn crossing nose presents a greater gap between the wing rail and the crossing nose. As a result the wheel contacts with the wing rail for longer distance and then it contacts with the crossing nose more abruptly. This causes higher wear values in the outer part of the tread, and a variability in the lateral position of the contact point when the wheel comes in contact with the crossing nose.

Finally, in the beginning of the turnout, the stock and the switch rails may have different vertical stiffness against the sleeper. The stiffness of the switch rail in the beginning could be only half of the stock rail stiffness. This difference increases the wear that is caused by the contact of the wheel with the stock rail. On the other hand, it could be observed a smoother alteration of the wear caused by the switch rail. The wear values reach the same levels but in the case with the same stiffness, wear peaks are more irregular and they do not increase gradually as in the case with different stiffnesses.

Future Work

This MSc thesis was limited to investigating how wheel wear was influenced by turnouts in combination with some parameters affecting the wheel-rail contact and being natural consequences of train operation. Such parameters are worn wheel profiles, worn crossing nose, difference in stiffness for the stock and the switch rails and the speed when negotiating the straight direction of the turnout. Some ideas for future work are described in this section.

All parameters were investigated separately with the turnout, for determining their influence independently. An option would be to study the influence of the turnout for a combination of altered parameters and to determine if there is an even higher increase of wheel wear.

Measured worn rail profiles should be studied. The worn rail profiles which were used in this study for the crossing nose were assumed to be worn by about 2-3 mm. Despite the fact that the simulations gave the tendency of increased wheel wear, there will be more accurate results with real data from a worn crossing nose.

The influence of wheel-rail wear coefficient should also be further investigated, especially for the areas where flange contact appears with the switch rail and the crossing nose. In these areas and while the flange contact appears, there are areas of wear with values about ten times greater than nearby areas. This is because of changing wear coefficient in the wear map and the difference from an area in the map to the other is very big.
Furthermore, the introduction of different vertical stiffnesses of stock and switch rails could be modelled between the sleeper and the rail where it is actually happening. In order for this to be modelled, two independent rails for the stock rail and the switch rail are needed. This would give the possibility to implement two different stiffnesses between the rail and the sleeper. But it requires much more work which exceeds the scope of this thesis. Also, the stiffness of the switch rail against the sleeper could be increased along the $x$-coordinate of the turnout until reaching the value of the stock rail, as observed in field measurements in Sweden and in [23].

According to Kalker [15], Hertzian contact model is not adequate to predict severe wear. Especially for the cases with worn wheel profiles, which presented high wear values in the extreme outer part of the tread, another contact model should also be tested.

All the simulations were with constant running speed, but if the vehicle is either accelerating or braking along a turnout, the wheel wear should be affected reasonably, as the simulations showed that it is affected by different constant speeds. Longitudinal creepage will increase at braking and block braking will even more increase the wheel (tread) wear. And finally, other types of vehicles and other types of turnouts, for example with curves with smaller radius, should be tested.
Bibliography


Appendix A

Notations

Latin Letters:

- $b_0$: Lateral distance between track centre line and the nominal rolling circle of the wheel [m]
- $c_{yrt}$: Lateral damping between rail and track [Ns/m]
- $c_{ytg}$: Lateral damping between track and ground [Ns/m]
- $c_{zrt}$: Vertical damping between rail and track [Ns/m]
- $d$: Sliding distance [m]
- $H$: Material surface hardness [N/m$^2$]
- $J_{xx}$: Moment of inertia around the longitudinal axis, roll [kgm$^2$]
- $J_{yy}$: Moment of inertia around the lateral axis, pitch [kgm$^2$]
- $J_{zz}$: Moment of inertia around the vertical axis, yaw [kgm$^2$]
- $k$: Wear coefficient in Archard’s wear model [-]
- $k_{yrt}$: Lateral stiffness between rail and track [N/m]
- $k_{ytg}$: Lateral stiffness between track and ground [N/m]
- $k_{zrt}$: Vertical stiffness between rail and track [N/m]
- $m_c$: Mass of carbody [kg]
- $m_a$: Mass of wheelset [kg]
- $m_s$: Mass of saddle [kg]
- $p$: Contact pressure [N/m$^2$]
- $r$: Wheel rolling radius [m]
- $r_0$: Wheel radius at nominal running circle [m]
- $s$: Sliding distance [m]
- $\Delta s$: Magnitude of sliding distance in an element [m]
- $T$: Tangential force [N]
- $v$: Running speed [m/s]
- $V$: Volume of wear [m$^3$]
- $W$: Frictional work [N·m]
- $x$: Longitudinal direction
- $y$: Lateral direction
- $z$: Vertical direction
Greek Letters:

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<thead>
<tr>
<th>Symbol</th>
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<tbody>
<tr>
<td>ζ</td>
<td>Coordinate direction perpendicular to contact plane</td>
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<tr>
<td>Δζ</td>
<td>Wear depth for a contact surface element [m]</td>
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<tr>
<td>η</td>
<td>Lateral coordinate direction in the contact plane</td>
</tr>
<tr>
<td>Δη</td>
<td>Width of a contact element [m]</td>
</tr>
<tr>
<td>ξ</td>
<td>Longitudinal coordinate direction in the contact plane</td>
</tr>
<tr>
<td>Δξ</td>
<td>Length of a contact surface element [m]</td>
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<tr>
<td>μ</td>
<td>Friction coefficient [-]</td>
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<tr>
<td>νη</td>
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<tr>
<td>νξ</td>
<td>Longitudinal sliding velocity [m/s]</td>
</tr>
<tr>
<td>νvehicle</td>
<td>Vehicle speed [km/h]</td>
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