Simulation of Wheel and Rail Profile Evolution
Wear Modelling and Validation

by

Roger Enblom

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Preface

The research reported in this thesis has been carried out in the course of continued work on computer aided wheel-rail wear simulation, a part of the vehicle-track research field at the Royal Institute of Technology (KTH), Division of Railway Technology. The objective has been to refine the methods applied to uniform wheel wear simulation by inclusion of braking and improvement of the contact model. Further a tentative application to uniform rail wheel simulation has been proposed and tested.

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Roger Enblom
Abstract

Numerical procedures for reliable wheel and rail wear prediction are rare. Recent development of simulation techniques and computer power together with tribological knowledge do however suggest computer aided wear prediction. The objective of the related research field at the Royal Institute of Technology (KTH) is to arrive at a numerical procedure able to simulate profile evolution due to uniform wear to a degree of accuracy sufficient for application to vehicle dynamics simulation. Such a tool would be useful for maintenance planning as well as optimisation of the transport system and its components.

The research contribution accounted for in this thesis includes, in addition to a literature review, refinement of methods applied to uniform wheel wear simulation by inclusion of braking and improvement of the contact model. Further a tentative application to uniform rail wheel simulation has been proposed and tested.

The first part addresses issues related to braking and wheel-rail contact conditions in the context of wheel wear simulation. The KTH approach includes Archard’s wear model with associated wear maps, vehicle dynamics simulation and railway network definition. In previous work at KTH certain variations in operating conditions have been accounted for through empirically estimated average scaling factors. The objective of the current research is to be able to include such variations in the set of simulations. In particular the influence of disc braking and varying friction and lubrication conditions are investigated. Both environmental factors like moist and contamination and deliberate lubrication need to be considered. As part of the associated contact analysis the influence of tangential elastic deformation of the contacting surfaces on the sliding velocity has been separately investigated and found to be essential in case of partial slip contact conditions.

In the second part validation of the improvements related to wheel wear simulation is addressed. Disc braking has been included in the simulation set and a wear map for moist contact conditions based on recent tribometer tests has been drafted and tested. It has been shown that the previously used braking factor accounts for the combination of the contributions from surface elasticity and braking. Good agreement with measurements from the Stockholm commuter service is achieved. It is concluded that the model improvements accounted for are sufficient for adequate simulation of tread wear but that further development of the flange / gauge corner contact modelling may be needed.

In the final part a procedure for simulation of rail wear and corresponding profile evolution has been formulated. A simulation set is selected defining the vehicles running on the track to be investigated, their operating conditions, and contact parameters. Several variations of input data may be included together with the corresponding occurrence probability. Trial calculations of four non-lubricated curves with radii from 303 m to 802 m show qualitatively reasonable results in terms of profile shape development and difference in wear mechanisms between gauge corner and rail head. The wear rates related to traffic tonnage are however overestimated. It is believed that model refinements in terms of environmental influence and contact stress calculation are useful to improve the quantitative results.

Keywords: contact, wear, wear prediction, wear model, wheel profile, rail profile, simulation, vehicle-track interaction, multibody simulation, MBS.
Outline of Thesis

The scope of this thesis is simulation of wheel and rail profile evolution due to wear. Railway vehicle dynamics and tribology are combined using multi body simulation and wear modelling in an iterative process. The thesis includes an introduction and summary and the following appended papers:

A  Enblom R, Berg M: *Simulation of Railway Wheel Profile Development due to Wear – Influence of Disc Braking and Contact Environment*. Accepted for publication in Wear.


C  Enblom R, Berg M: *Tentative Formulation of Simulation of Rail Profile Evolution due to Uniform Wear*. To be submitted for publication.

All papers have been written by Roger Enblom and reviewed by Prof. Mats Berg.
Contribution of Thesis

A few different approaches to simulation of wheel-rail profile evolution are reported in the literature. The combination of Archard’s wear model, widely accepted in the tribological society, multibody simulation of the complete vehicle, systematic representation of an arbitrary network or vehicle fleet, and validation by field measurements of real world applications is however not found elsewhere. The research accounted for in this thesis takes its starting point in the results reported by Jendel [12]. Improved contact modelling, increased generality in wheel wear simulation, and pioneering efforts in rail profile evolution are detailed as follows:

- A fundamental improvement of contact modelling has been achieved by including the tangential flexibility of contacting surfaces in determining relative sliding velocities and distances. This has been shown to be essential for the adequate determination of the wear distribution over a contact area subjected to partial slip. These additional calculations have been efficiently implemented in the numerical procedure Fastsim by Kalker.

- For wheel wear calculation typical simulations are defined representing the network properties, operations, and contact conditions. This simulation set has been extended with simulation of disc or dynamic braking. The braking effort is applied directly in the multibody analysis, thus the resulting creep and creep forces represent the combined effect of dynamic interaction, curve negotiation, and braking. It has been shown that an empirical scaling factor to account for this effect is not needed and that the scaling factor proposed by Jendel approximately represents the combination of surface elasticity and braking.

- The generation of tribological data in terms of wear rates under different conditions applicable to wheel-rail contact is a tedious process. The knowledge of environmental influence on the contact conditions is however emerging and the amount of test data steadily increasing. Recent results from wear tests at high sliding velocity and different levels of humidity have been used to simulate typical weather conditions. The efficiency of trackside lubrication has been evaluated from available field measurements of rail wear.

- All model improvements listed above have been applied to the Stockholm commuter service for validation. Good agreement between simulations and available measurements has been achieved using the generalised models.

- For application to rail wear simulation a general structure defining the simulation set in terms of relevant parameters and occurrence probabilities has been proposed. A three-level hierarchy is used with the vehicle definitions at the uppermost level, the desired operations at the next, and varying contact conditions at the bottom level.

- Rail wear prediction is more difficult than corresponding wheel wear simulation due to the limited variation of contact conditions and corresponding averaging effect. The results are highly sensitive to the contact modelling in particular at the gauge corner. The limited applicability of traditional elliptic contact modelling is indicated and desired improvements pointed out.
1 INTRODUCTION

Wear of wheels and rails has been of concern in the railway business for several decades. With current trends towards increased axle loads and higher speeds, the phenomenon becomes even more accentuated despite significant achievements in material development and vehicle design. The focus on infrastructure maintenance and rolling stock life cycle costs also draw attention to the possibilities of wear control.

In general, the course of events usually called wear is a complicated process involving several modes of material deterioration and contact surface alteration. Thus may material removal or relocation, plastic flow and phase transformation take place at, just below, or in-between the contacting surfaces.

From a tribological point of view, the wheel-rail contact is an open system dependent on both design features and environmental conditions. Different wear mechanisms may be activated in response to the actual loading, slip, and lubrication. Lubrication may in this context be understood as intentional or as a result of the ambient state. The process of metallic material removal usually obeys some threshold function of the operation parameters where a limited change may influence the rate of wear dramatically.

From a solid mechanics point of view the railway operation imposes cyclic loading on the wheel and rail. Depending on the wheel load, contact stress distribution and subsurface stress, plastic deformation and shake down as well as fatigue crack initiation and propagation, known as rolling contact fatigue (RCF), may occur. The initiation of fatigue cracks in steel is again a threshold phenomenon.

These different mechanisms of deterioration manifest themselves through a number of damage patterns. Wheel damage occurs as fatigue cracks, initiated at or below the surface, which may result in material fall-out like shelling or spalling. Micro-level material removal, possibly in combination with plastic deformation, may cause both out-of-roundness and profile alteration reasonably constant around the circumference. Rail damage shows a corresponding variety of damages like fatigue cracking, in severe cases leading to rail fracture, and longitudinally constant profile related wear. A periodic deformation pattern, known as corrugation, is also common.

Traditionally, corrective actions in order to reduce wear have been based on experience and measurements on in-service vehicles and track, occasionally supported by laboratory testing. Theoretical and predictive methods are rarely developed to the level required for engineering application, in particular suitable for application in early design phases.

The focus of the current work is on the prediction of profile related wear, altering the wheel-rail profile match but considered constant around the wheel and along the track. Nevertheless, even this process only constitutes a tribological system depending on several parameters requiring input from different disciplines (figure 1). To assess the states of interaction in the wheel-rail interface the forces and motions originating from the dynamic response of the vehicle are required. Furthermore the location of contact, contact stress distributions, and a corresponding wear model are needed. Finally the accumulated effect of different types of services and ambient conditions shall be considered.
Research activities aiming at the application of numerical methods to the problem of wheel-rail wear prediction begun to be published in the mid-eighties. An immediate conclusion became that the solution of this problem would be demanding both in terms of computer power and cross-discipline mathematical models. The early simulations were thus limited to few degrees-of-freedom vehicle models and simplified contact and wear models, typically modelling only one of the related disciplines in some detail. The obtained results were however classified as “promising”, showing qualitatively reasonable behaviour.

Subsequent contributors refined the models in different aspects. The numerical models were allowed to grow as the computers became more powerful. Much attention was paid to find reliable relationships between contact conditions and wear rate, replacing the traditional approach of energy dissipation in the contact patch as a measure of the wear. Others improved the contact stress modelling, especially close to the wheel flange and rail gauge face where the Hertzian assumptions may be a poor approximation. Inclusion of plasticity models has been reported only recently, basically in connection with periodic wear pattern. There is also a trend towards improved system modelling in terms of model integration and more realistic traffic scenarios.

A particular feature of this type of simulation is the huge difference in time scales involved. The vehicle dynamics problem needs to be solved with millisecond resolution, while the profile evolution time typically counts in months. Some authors have defined a formal mathematical relation between the two time-scales while others have adopted the concept of wear step (figure 2), de-coupling the two calculations and leaving some freedom in specifying their interaction.

The scope of the current research is prediction of uniform wear and its influence on the evolution of wheel and rail profiles. The four cornerstones of the modelling procedure are (i) definition of the service to be investigated by appropriate selection of a simulation set, (ii) dynamic simulation of the track-vehicle interaction, (iii) calculation of wear depth and profile shape, and (iv) validation through in-service measurements. Modelling of plastic deformation of the contacting surfaces is not included at this stage.

The following two sections give an overview of related published results on wear modelling and profile evolution prediction. A more comprehensive literature survey is given by the author in [8].
The last two sections summarise the current research and indicate future directions respectively.

**Figure 2:** General wear prediction flow chart. The dotted box indicates desirable future extension.
2 WEAR MODELS

In this section some general results are quoted addressing material loss due to adhesive and abrasive wear as well as the influence of lubrication. Research directly related to profile evolution is accounted for in next section.

Kimura [18] gives an overview of recent research on both wear and fatigue from a tribological point of view. It is pointed out that both phenomena have elemental processes in common on the micro-slip level.

2.1 Material loss

Published results addressing material loss through wear may be divided into three major categories: field measurements, laboratory research, and theoretical prediction model development. In model development, two main streams can be observed:

1. An one-parameter model assuming the material loss is proportional to dissipated friction energy in the contact. Different proportionality factors may be used for different wear regimes, the transition still determined by the rate of dissipated energy.

2. A two-parameter model according to Archard [1], where the material loss is taken proportional to the normal force times the sliding distance divided by the material hardness. The proportionality factor is dependent on the wear regime expressed in terms of contact pressure and sliding velocity.

In the mid-eighties, a large amount of laboratory testing was carried out using a quarter scale simulation facility at the Illinois Institute of Technology in Chicago. Kumar et al. have studied material loss under different conditions and loading [21], [22].

In the first paper, material loss under zero angle of attack and different lateral force levels is discussed, i.e. representing ideally radial aligned wheelsets. Earlier wear indices using the angle of attack as governing parameter are questioned since the wear contribution at zero angle is significant. It is also concluded that the wear-work principle is a reasonable approach for clean contact surfaces, taking both longitudinal and lateral forces into consideration. Different wear coefficients should however be used for crown and gauge side or tread and flange respectively. In case of contaminated or lubricated surfaces or when significant plastic flow is present, the wear-work principle should be used with care.

In the second paper, wear rates for freight car and locomotive wheels are investigated for different levels of wheel loads and adhesion coefficients. It is concluded that the wear rate can be reasonably described with a bilinear function both with respect to wheel load and adhesion coefficient. The wear rate is approximately one magnitude higher for tractive wheels than for free rolling freight car wheels.

McEven and Harvey [32] suggest a wear prediction model for curves based on full scale testing. The model relies on the dissipated energy hypothesis, here applied to the severe wear regime. A linear relationship is proposed between the wear rate and the dissipated energy per unit area, amended with a constant off-set term.

Markov reports extensive laboratory testing to determine wear rates for different operating conditions using twin-disc equipment [31]. Wear rates are given for different
combinations of roller hardness, load levels, and slip rates. The results discussion is focused on high slip rates, distinguishing between mild wear below 5% slip, severe wear, and catastrophic wear, the latter measured approaching 100% slip (figure 3).

Zakharov et al. have carried out twin-disc tests as well, here with focus on wear rate at different loads and lateral creep levels [65]. The rollers, 40 mm in diameter, were loaded from 300 MPa to 1100 MPa with lateral creep levels ranging from 2.5% to 10%. It is concluded that the product of contact pressure and creep, $p \cdot \nu$, is the relevant parameter for identification of wear regimes. Several other authors distinguish between mild, severe, and catastrophic wear. In this particular investigation however, a fourth regime, heavy wear, is identified between the severe and catastrophic regimes. Field observations of worn wheels and tracks lead to the conclusion that scale effects are important.

![Figure 3: Influence of slip on total wear rate for wheel steels [31]. Maximum contact pressure: 650 MPa. Hardness: (1) 700 HV, (2) 500 HV, (3) 250 HV.](image)

A more metallurgical approach has been applied by Singh et al. [48] in determining wear properties for pearlitic rail steels. Twin-disc and pin-on-disc tests have been carried out using three grades of rail steel and one grade of wheel material. The results were evaluated with respect to hardness and interlamellar spacing. It is concluded that reduction of pearlite interlamellar spacing has a strong positive effect on the wear resistance.

A method for identification of profile related wear coefficients based on field measurements combined with simulations is presented by Krettek and Szabo et al. [20], [53]. The wear coefficients along the lateral profile shape are determined from measured radial change by a least-square procedure and related to the friction work calculated at similar operating conditions.

Yet another wear modelling approach is accounted for by Williams [63], featuring continuum mechanics and tribology. An overview of traditional models is given including the Archard wear equation and the concept of wear maps. Various models of abrasive wear are addressed as well and the effect of asperity shakedown is commented. It is concluded that no universal mechanism of wear exists nor any simple correlation between surface degradation and friction coefficient.
2.2 Lubrication

Lubrication of the wheel-rail contact is often deliberately applied, predominantly at the high rail in narrow curves, in order to reduce wear or noise. From a tribological point of view, non-deliberate contamination and weather related moist might be considered as lubricants as well. Lubrication-related investigations often focus on efficient application of lubricants, properties of different lubricants, and the effect on wear and adhesion. For the purpose of wear prediction, however, the effect of non-deliberate contact environment modification is of importance as well.

Steele [52] and Reiff [43] report lubrication tests carried out on the Facility for Accelerated Service Testing (FAST) of the Association of American Railroads (AAR).

Steele has investigated the effect of lubrication of the high rail gauge face in curves. A considerable reduction of wear rates measured as gauge face wear (GFW) has been shown. In addition reduced head height loss (HHL) has been observed at both rails. The benefit is less pronounced for high-grade steel (figure 4).

![Figure 4: Lubrication benefit ratio with respect to gauge face wear (GFW) and head-height loss (HHL) related to initial material hardness [52]. The lowermost curve for the inner rail.](image)

Reiff has studied the efficiency of different lubricants and lubrication systems. In general it is concluded that proper lubrication can be used both to control wear and reduce fuel consumption (figure 5).

It is observed that the wear reduction is highly dependent on the efficiency of lubricant application as well as less pronounced for high-grade rail steel. Despite the type application device, grease tends to migrate to the railhead reducing traction and causing locomotive slip.

Waara [62] has been comparing the performance of environmental friendly lubricants with that of traditional types as well as water lubricated and dry rail. It is concluded that the tested graphite-free environmental friendly lubricants can be used without increased
risk for wear. Applied to field conditions in northern Sweden, it is assumed that the gauge face wear can be reduced 3 – 6 times by full year lubrication.

Figure 5: Effect of rail lubrication on fuel consumption, shown as engine throttle positions [43].

Nilsson reports extensive in-field follow-up of rail wear in narrow curves [35]. Both lubricated and non-lubricated rails as well as seasonal variations have been investigated. The wear rates vary over the year probably due to changing weather conditions. In particular the precipitation correlates well with the wear rate for non-lubricated curves (figure 6).

Figure 6: Wear rate as function of average daily precipitation [35].

Track-side lubrication can reduce the rail wear substantially. The reported lubrication benefit factors are approximately 9 for a 300 m curve and about 4 for the measured 600 - 800 m curves. Lubrication is however effective only for a limited distance ahead of the application device.
Motivated by observed adhesion loss at increased speed, Ohayama undertook adhesion measurements under wet conditions [37]. The experiments were carried out with a large-sized rolling contact testing machine. The background was measured adhesion coefficients for Shinkansen on wet rails, dropping significantly at high speed. The test results were evaluated and compared to tribology based theoretical reasoning.

In a Dutch case study by Savkoor and van der Schoor [46], the effect of contaminants on traction and braking has been investigated. A low viscosity contaminant standardised by UIC was used together with a test rig over a wide range of rolling speeds with slip ranging from 0% to 80%. The results show a reduced traction coefficient at increased speed but also a transition to a significantly higher traction coefficient at high slip. Beyond the threshold, the traction force and the wear rate are significantly larger and the surface becomes distinctly coarser. The threshold behaviour shows a hysteresis, possibly an opening to design antiskid brake systems with the potential to boost traction while keeping the wear within acceptable limits.

3 WHEEL-RAIL PROFILE EVOLUTION

The focus of this review is prediction of profile development by numerical simulations. Chudzikiewicz has given a review of historical and current research from a tribological point of view [4]. An introduction to wear mechanisms applicable to rolling-sliding contact is given followed by a summary of recent prediction models. The concept of different wear regimes, Archard’s wear equation, and the energy dissipation approach are addressed. The different approaches are compared and it is concluded that a key issue is the determination of wear coefficients.

The issue of specifying rail network and vehicle operations in general for a mixed service is addressed by Szabó and Zobory [54]. The wheel profile evolves in interaction with the actual rail profile which may show a wide range of worn shapes. The similar situation is true for the rail profile evolution with respect to possible wheel profiles running on the track. In addition relevant combinations of both vehicle and track design parameters have to be considered. To structure this combinatorial problem a hierarchy of probabilities based on known statistics of the complete system is defined.

3.1 Prediction of profile evolution

Often profile measurements and monitoring of field conditions shape the basis for wear prediction, in particular for maintenance planning purposes. Such measurement data may also serve as validation of simulation models and determination of wear coefficients.

Extensive measurements of wheel and rail wear during a service period of three and a half years of the Stockholm commuter traffic have been reported by Nilsson [34], [35], [36] and Olofsson and Nilsson [38]. Profile wear, hardness, and geometry (figure 7) have been monitored and evaluated with respect to different conditions.

Two types of vehicles operate on this network, the older X1 electric multiple unit with stiff primary suspension and the more recent design X10 with softer suspension and better self-steering capability. Wheel wear has been measured on both types.
Figure 7: Rail profile development during 24 months in a non-lubricated 303 m curve for a new and a worn-in high rail [35].

Rail wear rates in different curves have been compared to rail wear on tangent track showing substantial influence of the curve radii. Taking the tangent track as unity, the worn-off area increases to about 20 for a 800 m curve and 160 for a 300 m curve.

Weather data from nearby located meteorological stations has been related to measured wear rates. The effect of trackside lubrication and efficiency of related devices have also been investigated. Finally the susceptibility to wear for different steel grades and track ages was evaluated (figure 8).

Figure 8: Rail wear rate for the high rail in a non-lubricated 303 m curve, comparing steel grades UIC900A and UIC1100 [35].

Upon significant increase of freight tonnage and axle loads McIlveen and Roney have evaluated the economic trade-off of different policies for track maintenance [33]. The conclusion is that improvements in track quality and maintenance procedures do pay off in extended rail life.

One approach to wear prediction is to collect measurement data and identify critical operating conditions. Clayton [5] uses a combination of laboratory tests and field measurements to predict rail wear at high axle loads. The paper includes a review of two
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decades of research in that direction and aims at validating laboratory test results against field measurements. In the field study wear properties of four different rail materials were investigated in a 350 m curve. Similar conditions were created in the laboratory by testing the same materials in a twin-disc machine. Reasonably good agreement between field and laboratory results was achieved. In addition the wear rate was evaluated with respect to material hardness and microstructure as well as contact pressure.

One conclusion is that existing general wear models are not suitable for practical use. A more successful approach would be models with very restricted application to reduce the otherwise vast number of parameters. With respect to wear control parameters it is concluded that the energy concept can only be useful for a given wear mechanism and for specific materials.

Strategies to control wheel wear and understanding of the underlying control parameters are addressed by Fröhling [9]. The objective of the paper is to investigate practical strategies, based on experience and scientific analysis, to control and limit wheel profile wear due to dynamic wheel-rail interaction. Several wheel and rail sparing strategies are listed. Simulations have been used to confirm the relevance of some of those actions in terms of comparative wear distributions (figure 9).

Figure 9: Predicted wheel wear distributions [9], S1: Influence of track layout, S2: Influence of primary suspension stiffness, S3: Influence of nominal track gauge, S4: Influence of curving speed.

Once the mathematical modelling of vehicle dynamics gained pace during the eighties, several attempts to simulate wear were initiated. Recent advances in contact mechanics facilitated some progress, reported in the early nineties.

Kalker presented a method [16] for simulation of wheel tread wear achieving reasonable results with respect to measurements on metro operation. After an opening discussion about contact modelling issues, the frictional work traversing one contact patch is formulated. The Fastsim algorithm is used to calculate the frictional work and
corresponding wear for a number of wear steps. Instead of calculating each contact occurrence it is possible to use the probability of the contact being at a certain lateral coordinate. The wear is assumed to be proportional to the frictional work and the rate of wear was derived from field studies.

An early approach to simulate wheel profile evolution is presented by Pearce and Sherratt [40]. The basis is repeated time domain simulations of one or more journeys with profile updating in-between. The amount of wear is assumed to be proportional to the dissipated energy in the contact zone, taken as product of creep force and creep ratio. The relationship varies depending on whether the wear is in the mild or severe wear regime or in the transition between them. The worn-off profile area is given a parabolic distribution laterally across the contact patch. Simulated results have been compared to measurements in terms of development of equivalent conicity over running distance.

An investigation of the vehicle model sophistication required for wear calculations is reported by Liebig et al. [29]. The vehicle dynamic response used as input to the wear prediction was calculated by (i) a two-dimensional lateral model and (ii) a full three-dimensional model. The material removal was assumed to be proportional to the frictional work. For the two-dimensional case the wear was rather evenly distributed over both tread and flange. For the three-dimensional cases, however, the tread wear was dominating. It was concluded that improvements of the vehicle model are necessary to be able to predict for instance life cycle costs.

An attempt to apply a non-iterative, non-elliptic contact model to wheel wear simulation is reported by Linder and Brauchli [30]. The shape of the contact area is taken as the rigid body penetration area scaled down by a factor 0.55 – 0.65. The obtained contact patch is subdivided into strips in the rolling direction and the contact quantities are calculated still using Hertz’ theory for a substituting ellipse adapted to each individual strip. The wear calculation procedure follows the concept of wear step, simulating a certain distance with constant profiles and then update. The wear coefficients are taken from literature and the wear is assumed to be proportional to the frictional power at the two levels mild oxide wear and severe metallic wear.

Wear calculation by combining complete contact theory and full scale testing is initiated by Braghnin, Bruni, and Resta [3]. The approach is to evaluate the simulated vehicle dynamic response with Kalker’s code Contact [15] obtaining traction and slip distributions over the contact area. The material removal by wear is assumed to be proportional to the frictional work. The wear rate was determined by tests carried out on a full-scale wheelset in a roller rig. Comparisons between calculated and measured wear corresponding to a running distance of up to 10 Mm showed good agreement for the tread. The flange wear could not be captured by the calculations since the wear coefficient was determined for the mild wear regime only.

Non-Hertzian multi-point and conformal contact models have been developed by Li [28]. Application to wear simulation is described by Li et al. [26], [27]. The wear volume is assumed to be proportional to the frictional work. The selection of appropriate wear step length as well as the influence of worn profiles on the dynamic response of the vehicle are briefly addressed.

A comprehensive description of a wear simulation approach covering the whole process from specification of operating conditions to material loss and profile updating is given
by Zobory [67]. The focus is on wheel and rail profile evolution while running on arbitrary networks and appropriate wear models, formulated in the framework of stochastic processes. First some modelling and discretisation issues are addressed. Lumped mass vehicle models, discrete track and profile descriptions, track irregularity spectral density functions, and creep force laws are discussed. Further the wheel and rail wear process is described in terms of contact occurrence frequency and debris mass flow. The contact frequency is proportional to the number of wheel revolutions or wheel passages respectively and may be seen as stochastic realisation functions.

The material loss due to wear is again treated stochastically as a debris mass flow process. Since the vehicle dynamic response depends on the wheel-rail contact geometry, the profile evolution provides a very slow feedback to the system, offering the possibility of discretisation by piece wise constant profiles. The value of the debris mass flow is considered function of the energy dissipation in the contacts, distributed over the contact area in proportion to the normal pressure. The wear coefficient is determined empirically and takes different values for mild and severe wear as well as flange wear in pure sliding.

The application of this method to simulation of wheel and rail profile evolution is addressed by Szabó and Zobory [57].

**Figure 10:** Flow chart of the proposed wheel profile prediction tool [14].
An extensive procedure for simulation of wheel profile evolution through wear has been developed by Jendel [11], [12], [13], [14] (figure 10). The approach is to study a single vehicle with a given initial wheel profile. A load collective is designed consisting of a limited set of dynamic time domain simulations representing the actual operating conditions. In the load collective the network is primarily discretized based on the curve radius distribution. Other key parameters are rail profiles, track irregularities and wheel-rail coefficient of friction.

The vehicle-track interaction simulations are based on multibody models of the vehicle and track and, in particular, proper modelling of the wheel-rail contact. As output the simulations give contact forces and motions at certain time instants. These quantities are input to the wear modelling and calculations. The calculated wear distribution is smoothed and the wheel profile is updated according to the calculated material removal.

Since the actual wheel profile affects the contact response, the successive change of wheel profile is discretized into several wear steps, each starting with an updated wheel profile.

The normal and tangential contact problems are solved with Hertz’ theory [10] and Kalker’s simplified theory respectively. The numerical implementation Fastsim [17] by Kalker is employed. The use of elliptical contact formulation is adequate for the tread contact but may be questioned for the flange/gauge-corner contact. Since the wear calculation, however, includes thousands of contact problems to be solved an averaging effect will compensate for that deficiency and the advantage of fast execution can be maintained.

Figure 11: Wear chart for wheel and rail steels. Dry, room tempered conditions [14].
The wear modelling is based on Archard’s wear equation used together with experimentally determined wear charts and the model is applied locally in each element of discrete representation of the contact area.

The wear coefficient is generally a function of sliding velocity, contact pressure, temperature, contact environment etc. Laboratory tests have been performed using wheel and rail steels to determine the wear coefficient. To cover a wide range of sliding velocities both twin-disc and pin-on-disc machines were used. The result was compiled in wear charts showing the wear coefficient as function of sliding velocity and contact pressure (figure 11).

The wheel wear simulation tool has been verified by comparing simulated wheel wear on a commuter vehicle operating the Stockholm commuter network (figure 12). The network comprises about 200 km track with many curves with radii in the range 300 – 2000 m.

Figure 12: Simulated and measured wheel profile development after 200 Mm [14]. (a) Wheel profile (b) Wheel radius change. Solid line – simulation, dashed line – measurement, dash-dotted line – initial profile.

A further application, investigated by Dirks [6], is the Swedish high speed train service between Stockholm and Göteborg. Simulations up to 350 Mm running distance have been performed. The results were compared to available wheel profile measurements with some focus on scalar wear measurements (figure 13).
Simulation of Wheel and Rail Profile Evolution - Wear Modelling and Validation

3.2 Profile design

Several attempts to modify standard wheel or rail profiles to adapt to certain operating conditions and reduce wear and fatigue have been reported. Often the goals are single point contact and increased conformity of profiles.

Leary et al. report a case study on development of a wear optimised wheel profile for freight cars [24]. The intention was to design and evaluate a reasonable number of candidate profiles providing (i) reduced wheel and rail wear, (ii) increased derailment safety, (iii) stable running performance, and (iv) reasonable contact stresses. In addition the chosen profile should be geometrically stable throughout its life. Two approaches were used, one based on the average of measured worn wheels and the other based on expansion of rail shapes to ensure single-point contact. The candidate profiles were experimentally evaluated with respect to rolling resistance, curving, and stability characteristics. The contact stresses were determined by calculations. The most promising candidate became the profile based on measured wheel shapes. This profile was further refined, basically by reducing the equivalent conicity, and put into revenue wear test. The flange wear turned out to be about one third compared to the old 1:20 conical profile.

Piotrowski et al. [42] have developed a modified wheel profile for low speed industrial locomotives. The aim was reduction of wear by ensuring one point contact. Wear simulations were carried out using a non-Hertzian contact model by Kik and Piotrowski.
Quasistatic curving was used including non-linear characteristics of axle box guidance, wheel loads, and traction or braking moments. The design criterion was the lateral distribution of the frictional work over the profile. Field measurements on locomotives with the new profile showed an increase in operating period between reprofiling of about 20%.

Yamada et al. report on wheel profile design to improve running stability while maintaining curving performance for vehicles running on worn narrow-gauge lines [64]. The proposed profile is characterised by low conicity and good conformance at the tread and high conicity towards the flange. The new profile is used on commercial lines and the interval of wheel turning is extended more than twice compared to the original 1:20 conical profile.

Further experimentally based wheel profile improvements are reported by Sasaki et al. [44] and Krettek [19]. There are also proposals on how to adapt the rail profiles to the system requirements, mainly to reduce contact stresses and the likelihood of fatigue damage. Sato has studied fatigue damage on Shinkansen lines and proposed a strategy for adapted rail grinding [45]. The profile pairing is unusual in that the rail head radius is as much as 600 mm and the wheel tread a 1:40 cone. The proposed grinding pattern distinguishes between tangent track, large radius curves and tight curves. Governing parameters are the contact position and the lateral deformation of the rail. Practical experience or field test of the new profiles are not reported.

Smallwood et al. [49] apply a different approach to reduce the problem of rail fatigue. The work is based on theoretical analysis aiming at the modification of the rail head geometry to reduce contact stresses. An extended Hertzian theory has been used allowing for other shapes of the contact. Flange contact has not been considered. As a starting point the contact stresses were evaluated for several measured worn profiles and the lower stresses were found to be associated with a flattening of the rail head. The new geometry was generated by optimising the radii in each of a chain of sections and combine with the original gauge face geometry. The resulting profile, however, needed further modification to reduce grinding depth as well as equivalent conicity. The proposed profile reduced the contact stress by up to 45% for curve radii above 1000 m.

Smith and Kalousek describe a design technique enabling wheel and rail profiles to be developed having a limited range of conicity throughout their life, generate little noise, and greatly reduce corrugation [50]. When used on steered axle vehicles, together with the described techniques of trade geometry control, the obtained profiles can be virtually self-perpetuating. Implementation of the designed rail profile requires grinding. The successful prototype testing indicates that there is a potential for system-specific wheel-rail profile design.

Zacharov and Zharov [66] propose a procedure for determination of optimal conform profiles. The optimum profile pair is selected from a family of conform profiles relevant for selected operating criteria and providing minimum mutual wear. Total flange/gauge-face wear is used as objective. The wear rate was determined by twin-disc testing using normal load and creep as parameters.
3.3 Vehicle design

In the eighties, research on freight car running gear with focus on steering ability and wear has been reported. Schwier [47] accounts for a research project aiming at cost comparison of different running gear concepts. A cost estimation model was used including fuel consumption, car maintenance, and track maintenance. With respect to wheel and rail wear, radial steering is superior to stiff suspension designs. In general it is concluded that steering running gears are economically justified only at high combined levels of annual utilisation and curvature.

Based on roller rig testing, Specht has developed a model for estimation of material loss through wear [51]. The worn-off mass is taken to be proportional to the frictional work in the contact zone. Mild and severe wear is recognised. The transition between the two regimes is determined by the frictional power. The material loss distribution over the wheel profile is determined by discretisation of the surface.

The method is applied to three freight car bogies (figure 14) being (i) the stiff suspension design Y25, (ii), the DB steering design 665, and (iii) the South-African Scheffel design. The two latter designs are approximately comparable with respect to wear whereas the stiff design shows about 30 times higher material loss, disregarding the effect of tread braking. A rough estimate indicates that this contribution is almost similar to the stiff bogie curving wear.

![Figure 14: Comparison of material loss through creep and tread braking for three running gear designs [51].](image)

Blokhin et al. [2] and Ushkalov et al. [61] are concerned by the trend towards increased flange and gauge corner wear mainly in freight operations. Both research groups have undertaken parameter studies with respect to frictional work or some wear index as function of running gear design parameters and geometrical tolerances. Ushkalov also includes riding comfort in passenger services. Blokhin includes a semi-empirical verification of the wear model while Ushkalov also develops an improved wheel profile
Szabó and Zobory have two contributions with relevance to vehicle design. The first paper [55] deals with maximising of the mileage performance of metro trains with respect to wear. The second [56] is a more general study on wear simulation when running on a specified network, including the significance of the axle-box guidance stiffness. In order to be able to perform an adequate investigation of the intended metro operation, extensive pre-calculations were carried out to determine realistic worn profiles. Firstly, a set of worn wheel profiles was calculated using a representative curve distribution and nominal rail geometry. Secondly, worn rail profiles were generated using the obtained wheel profiles. Finally the wheel profile evolution could be calculated for three metro vehicles under relevant operating conditions and with nominal running gear parameters. Mileage sensitivities with respect to longitudinal and lateral axle-box guiding stiffness rates for three selected critical wear measures were calculated. In addition mileage performance of two different initial profiles were compared with respect to flange thickness development.

In the second referenced paper [56] the aim is to formulate a formal optimisation problem including the stochastic nature of the operating conditions. When the railway operation is considered as a stochastic process, the mileage performance also becomes a stochastic field defined on the axle-box guidance stiffness parameters. The mileage objective function to be maximised then becomes the expected value function of that field.

4 PRESENT RESULTS IN SUMMARY

Recalling the objective of the wear simulation research at the Royal Institute of Technology (KTH), the recent results are summarised. Future directions are indicated in the next section.

The major goals are:

- Quantitative prediction of wheel and rail profile evolution with sufficient accuracy for use in vehicle dynamics simulations.
- Application of state-of-the-art models for both vehicle dynamics simulation and wear calculation and their interaction.
- Proposed system improvements for optimised wear performance without jeopardizing other performance requirements.

The methods applied to achieve those goals are:

- Systematic selection of operational conditions.
- Numerical simulation of vehicle-track interaction.
- Relevant wear models.
- Validation through measurements.
The following three subsections are devoted to the new achievements in terms of contact modelling, improved wheel wear simulation, and rail profile evolution.

### 4.1 Contact modelling

The wear model employed relies on relative sliding velocities for determination of wear coefficients. In **Paper A** it is shown that the influence of the tangential elastic deformation of the contacting surfaces may not be neglected, in particular under partial slip conditions. Further, equations for calculation of relative sliding velocities from quantities available in Fastsim are derived and implemented in the wear calculation step:

\[
\begin{align*}
    v_x^{(k)}(x, y) &= -\frac{v_r}{d_x} \cdot L_x \cdot p_x^{(k)}(x, y) \cdot \left(1 - \frac{1}{q^{(k)}(x, y)}\right) \\
    v_y^{(k)}(x, y) &= -\frac{v_r}{d_y} \cdot L_y \cdot p_y^{(k)}(x, y) \cdot \left(1 - \frac{1}{q^{(k)}(x, y)}\right)
\end{align*}
\]

where

- \( v_i \) = sliding velocity component; \( i = x, y \).
- \( v_r \) = running speed.
- \( p_i \) = tangential stress component; \( i = x, y \).
- \( L_j \) = tangential flexibility parameters; \( j = x, y, \varphi \).
- \( k \) = integration step index.
- \( x, y \) = contact patch co-ordinates; origin at the centre.

**Figure 15:** Radial wheel wear distribution considering tangential surface flexibility, friction 0.30, natural lubrication. Wheel flange towards positive lateral co-ordinate.

(a) Tangent track, (b) Radius 1500 m.

Bold line - elastic contribution to sliding velocity included, thin line - not included.
Section 4 - Present Results in Summary

The influence on the wear distribution is demonstrated by examples (figure 15). A total running distance of 6000 km is chosen in order to accommodate a few profile updates although each case simulates a single curving situation only. The track is modelled as symmetric with respect to left and right curves with the consequence that both wheels in a wheelset experience the same wear. Results are shown for the leading axle.

The model has been applied to the Stockholm commuter service simulation accounted for in Paper B together with the other model improvements. Taking the surface elasticity into consideration has been shown to improve the accuracy of tread wear prediction.

4.2 Improved wheel wear simulation

In Paper A also the issue of braking simulation is addressed with the aim to replace the previously applied empirical scaling. Creep and creep forces due to braking are calculated by dynamic simulation of the complete vehicle. The retardation moment is smoothly applied to all axles, representative for disc brake operation. Tread braking is not considered. Example wear distributions over the wheel profile are calculated for tangent track (figure 16) as well as curves with radii 1500 m and 600 m. The running distance is 6000 km and comparison is made to the case of constant running at the initial speed 120 km/h. Coefficient of friction and lubrication conditions are varied as well.

![Figure 16: Radial wheel wear distribution on tangent track. Sliding velocity due to rigid creep only. (a) Friction 0.15, natural lubrication, (b) Friction 0.30, natural lubrication, (c) Friction 0.60, dry. Bold line - braking, thin line - no braking.](image)
For wheel wear simulation the simulation set has been extended by the quantities related to traction and braking simulation.

In the initial version, the following parameters define the simulation set:

- Curve radius type class, including transition curves and cant.
- Vehicle speed, if applicable related to the appropriate cant deficiency.
- Selection of rail counterpart profiles.
- Coefficient of friction.
- Specified or randomly selected track irregularities.

Inclusion of braking simulation divides those cases where speed changes occur in two. The additional parameters are:

- Amount of retardation for each relevant curve class.
- Fraction of the curve class length where braking occurs.

The application of the extended simulation set to the Stockholm commuter operation is accounted for in Paper B.

The service braking of the X10 commuter vehicle is investigated by performance simulation assuming a retardation rate of 0.8 m/s². Vehicle acceleration and speed are visualized over the curve distribution to determine at which radii braking occurs (figure 17).

**Figure 17:** Vehicle operation example. Portion close to Stockholm C of the Märsta – Södertälje line. From top: Acceleration [m/s²], speed [km/h] (thin line = speed limit), curve radius [m].

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The simulation results are compared to measurements after 200 000 km running distance (figure 18). Surface elasticity as well as braking simulations are included and good agreement with the measurements is achieved. A selection of four different wheels are shown to demonstrate typical scatter in the measurements.

Figure 18: Results with extended simulation set compared to measured profiles. 200 000 km running distance. Surface elasticity included. Outer axles 5 and 8 and inner axles 6 and 7.

Figure 19: Results with extended simulation set compared to measured profiles. 200 000 km running distance. Updated wear map. Surface elasticity included. Outer axles 5 and 8 and inner axles 6 and 7.
The final topic in both Paper A and Paper B is the striving to improve the description of the wear coefficients and compile them into wear maps for dry contact, environmental influence, and lubricated conditions. Based on recent pin-on-disc test results a simulation with an updated wear map for natural lubrication combined with some further down scaling for deliberately lubricated curves has been carried out (figure 19).

Very good agreement between simulation and measurement is achieved for the tread wear while the flange wear is somewhat underestimated. The generality of the simulations is however improved since the wear map for natural lubrication is based on laboratory tests and the lubrication factor on field measurements. No attempt to tune the model to the specific application has been made.

4.3 Rail profile evolution

It is of interest to apply the wear simulation procedure to rail profile evolution as well. In Paper C a structure for a corresponding simulation process is outlined and some tentative calculations reported. The basic building blocks are the same as for wheel wear simulation, namely simulation set, dynamic analysis of the track-vehicle interaction, wear depth calculation and profile updating.

In this case, however, each application focus on a particular location along the track, typically a circular curve loaded by a selection of different vehicles. The governing parameters are thus axle passages and accumulated traffic load. The appropriate simulation set addresses variations in vehicle design, operating conditions, and contact environment. The rail wear simulation set is defined by the following parameters:

- Selection of vehicles to be simulated with corresponding occurrence probability.
- Vehicle speed and braking effort if applicable.
- Selection of wheel counterpart profiles.
- Coefficients of friction.
- Appropriate wear map selection.
- Scaling of Kalker coefficients.
- Specified or randomly selected track irregularities.

A fundamental difference between rail wear simulation and the corresponding wheel wear calculations relates to the variety of contact conditions. A vehicle traversing a large network experiences a wide range of curving conditions, track alignment variations, and friction levels. This has an averaging effect on the wear distribution and makes the procedure less sensitive to occasionally poor contact conditions, which is not the case for the rail. Despite possible problems due to poor contact modelling at the gauge corner, the same methods as used for wheel wear calculation are applied as a starting point. The tentative simulations (figure 20) generally shows higher wear rates than available field measurements.

It is concluded that a larger simulation set accounting for variations in friction and environmental conditions would improve results. Furthermore the limited applicability of traditional elliptic contact modelling calls for improved methods able to handle the actual profile geometry and non-elliptic contact patch shape. Finally continued
development of wear maps directing wear coefficients relevant to the actual wear mechanisms, transitions, and influence by environmental factors seems to be adequate.

![Figure 20: Vertical wear distribution at selected non-lubricated curves of different radii. Note the different vertical scales.](image)

### 5 FUTURE RESEARCH DIRECTIONS

The final goal of this research field at KTH is to provide tools able to contribute to the improvement of the rail-vehicle system performance. In engineering work a useful tool should provide functionality for both fast investigation of simple models and aid to carry out complete simulations with adequate accuracy. The existing procedure is proposed to be amended in both directions.

For tentative investigations the inclusion of quasi-static curving analysis, scalar wear measures, and comparisons with frictional work related quantities may be useful.

In optimising any system, it is of outmost importance with a relevant and realistic formulation of the objective and governing constraints. The ultimate goal of wear related system optimisation is to reduce maintenance and increase the operative life of wheels and rails. This by necessity also includes other deterioration mechanisms than profile related wear. A reasonable approach to a wear optimisation objective would be to minimise the wear depth in combination with evening out the wear distribution across the profile and consider the possible trade-off between wear and surface fatigue.

The principal parameters influencing these quantities are thought to be wheel suspension characteristics, wheel and rail profiles, and track stiffness properties. Furthermore the material properties and the presence of lubrication affect the results. Varying those parameters should allow for controlling the degree of profile conformity, curve negotiation capability, and wear distribution. Constraints to comply with are at least the traditional dynamic performance requirements being derailment safety, track forces, running stability, wheel unloading, and ride comfort.
With an adequate formulation along these lines it should be possible to apply numerical methods for optimisation and sensitivity analysis.

### 5.1 Contact model

Further development of the flange / gauge corner contact model is desirable, in particular for application to rail wear simulation in rather tight curves. Potential improvements are for instance non-elliptic contact and refined tangential stress distribution. Yet a larger step would be to include material plasticity by modelling strain hardening and material flow. Methods for including the plastic behaviour of the surfaces are however less mature and are suggested to be developed on a longer term basis. Inclusion of more sophisticated contact models may also make the computational effort a critical issue.

A reasonable alternative seems to be to replace Kalker’s simplified theory, realised by the Fastsim implementation, by a novel semi-Winkler model [58], [59] able to handle non-elliptical contact and coupled normal and tangential problems. This model may also serve as a basis for inclusion of material plasticity.

A more simple approach to handle non-elliptic contact is the procedure proposed by Linder and Brauchli [30] relying on a set of equivalent contact ellipses.

Tread braking is not at all considered in the current research although it is important for wheel wear in certain operations. To include this would however add a new wear mechanism entirely different from the wheel-rail contact, opening a new research area.

### 5.2 Contact environment

The basic tribology and contact mechanics research is carried out in parallel activities at the Machine Element division at KTH [25], [39], [60]. Extended contact modelling in the wear prediction will rely on these results.

Tests are being carried out to determine wear rates and regime transitions for a variety of contact conditions. It is proposed to use these results to formulate wear maps for the type conditions (i) dry contact, (ii) naturally lubricated (moist) contact, and (iii) deliberately lubricated contact. Adopting this approach may improve the model consistency in the following respects:

- Refined wear maps directing wear coefficients relevant to the actual wear mechanisms, transitions, and influence by environmental factors.
- Weighted selection of wear maps with respect to anticipated weather conditions and lubrication efficiency.
- Consistent set of wear map, coefficient of friction, and creep - creep force slope parameter.

### 5.3 Simulation set and validation

The selection of an adequate simulation set is a crucial interface between user and software and may be made more efficient by using formal numerical methods. Instead of relying on a series of manual parameter studies the simulation set is suggested to be defined by using design of experiments methodology. The selected parameter space
would be explored in a systematic way and as a spin-off sensitivities and coupling effects can be calculated.

It is believed that the results may be improved by using a somewhat larger simulation set accounting for additional variations in friction and environmental conditions. Both seasonal variations and differences between tread and flange could be taken into consideration.

For general purpose vehicles or track being used by a variety of different vehicles, it may be of advantage to select the likely operation by stochastic models as developed by TU Budapest [57].

The reliability of the wheel wear simulation facility should be further verified by application to other kinds of services and compared to related measurements. Recently the method has been successfully applied to the Swedish X2000 high speed train operation [6]. In that work some extended output has been generated related to scalar wear measures and single simulation contributions.

It is proposed to also simulate the Stockholm tram operation (Tvärbanan), where wheel and rail profile measurements are being carried out by Stockholm Transport (SL). Together with existing results the scope of validation would then cover the whole range from light rail over commuter service to high speed intercity operation. Application to freight traffic is more difficult since most freight cars are equipped with tread brakes. A possible application would thus be high speed light freight, as for instance post service, where disc braked cars are being used.

To validate the proposed rail profile evolution simulation procedure the extensive measurements available from the Älvsjö area in the Stockholm commuter network provide a valuable reference.

6 REFERENCES


Section 6 - References


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