Simulation and measurement of wheel on rail fatigue and wear

Babette Dirks

Doctoral Thesis in Vehicle and Maritime Engineering

KTH Royal Institute of Technology
School of Engineering Sciences
Dep. of Aeronautical and Vehicle Engineering
Teknikringen 8, SE-100 44 Stockholm, SWEDEN
Preface

The work presented in this thesis was carried out at the Department of Aeronautical and Vehicle Engineering at KTH Royal Institute of Technology in Stockholm, Sweden. It was part of the SWORD (Simulation of Wheel on Rail Deterioration Phenomena) project and was supported by the Swedish Transport Administration (Trafikverket), Stockholm Transport (SL), Bombardier Transportation, the Association of Swedish Train Operators (Tågoperatörerna) and Interfleet Technology.

I would like begin by thanking my main supervisor Prof. Mats Berg for his support and for giving me the opportunity to work as a PhD student at the Rail Vehicles division. A very special thank you goes to my supervisor Dr. Roger Enblom. It was really nice having you nearby and I don't think you have ever told me you didn't have time for me.

I would also like to thank Bombardier Transportation for their hospitality by taking me in for all these years in Västerås. Many thanks to all my "colleagues" there!

I would like to thank everybody at the Alstom depots in Bro, Älvsjö and Södertälje who made it possible for me to measure the trains. Also many thanks to you Adam Argulander (Alstom) for making the final measurements possible.

I would also like to thank Hans Hedström (InfraNord) for guiding me through those dark nights on the track. I would definitely have got lost or been run over by a train if you hadn't been there. I would also like to thank Stephen Löwencrona (InfraNord) for making all the arrangements.

I am also really grateful for all the help I received from Andreas Böttcher (Alstom). You have always been so enthusiastic and involved in this project.

I appreciate all the advice and help I got from all participants at the reference group meetings. Thank you Ulf Bik (SL), Roger Deuce (Bombardier), Mattias Eriksson (SL), Lars-Ove Jönsson (Interfleet), Ulf Olofsson (KTH) and Pär Söderström (SJ). Three people from this group I would like to thank extra. Thank you so much Rickard Nilsson (SL) for helping me out on those two cold nights, I really couldn't have done it without you! I would also like to thank Anders Ekberg (Chalmers) for his support; you always made time for me and I really appreciated that. Thank you Martin Li for all the track data.

Stefan Lundström Sveder (Trafikverket), I also want to thank you for all your support.

Many thanks also to Ingemar Persson at AB DEsolver for helping me out with my gensys simulations.

I also appreciate the help I got from my former colleagues at DeltaRail (DEKRA Rail) Martin Hiensch, Pier Wiersma, Mark Linders, Leendert Vermeulen and Geert-Jaap Weeda. Thank you for letting me use the many measurements you did and for answering all my questions.
Thanks to my fellow PhD students (fellow sufferers), Alireza, Matin, Saeed, Tomas, Yuyi and Zhendong, and to my other colleagues Carlos, Evert and Sebastian. I also want to thank my former colleagues Anneli, Dirk, Rickard, Piotr and David. You all made me feel welcome.

Very special thanks to my family! Thank you zuzzel for being my sister and best friend. Many thanks to my parents for supporting me in everything I do.

I also want to thank my two bundles of joy Dax and Silas for their unconditional love and for putting a smile on my face. You made me see, Dax, that on the front cover of Johnson's Contact Mechanics book there are not two round black objects in contact, but that it is actually Mickey Mouse.

Finally, I also want to thank Jeroen, especially for the last year. You are the light in my deepest darkest hour and you are my saviour when I fall.

Stockholm, April 2015

Babette Dirks
Abstract

The life of railway wheels and rails has been decreasing in recent years. This is mainly caused by more traffic and running at higher vehicle speed. A higher speed usually generates higher forces, unless compensated by improved track and vehicle designs, in the wheel-rail contact, resulting in more wear and rolling contact fatigue (RCF) damage to the wheels and rails. As recently as 15 years ago, RCF was not recognised as a serious problem. Nowadays it is a serious problem in many countries and "artificial wear" is being used to control the growth of cracks by preventive re-profiling and grinding of, respectively, the wheels and rails. This can be used because a competition exists between wear and surface initiated RCF: At a high wear rate, RCF does not have the opportunity to develop further. Initiated cracks are in this case worn off and will not be able to propagate deep beneath the surface of the rail or wheel.

When wheel-rail damage in terms of wear and RCF can be predicted, measures can be taken to decrease it. For example, the combination of wheel and rail profiles, or the combination of vehicle and track, can be optimised to control the damage. Not only can this lead to lower maintenance costs, but also to a safer system since high potential risks can be detected in advance.

This thesis describes the development of a wheel-rail life prediction tool with regard to both wear and surface-initiated RCF. The main goal of this PhD work was to develop such a tool where vehicle-track dynamics simulations are implemented. This way, many different wheel-rail contact conditions which a wheel or a rail will encounter in reality can be taken into account.

The wear prediction part of the tool had already been successfully developed by others to be used in combination with multibody simulations. The crack prediction part, however, was more difficult to be used in combination with multibody simulations since crack propagation models are time-consuming. Therefore, more concessions had to be made in the crack propagation part of the tool, since time-consuming detailed modelling of the crack, for example in Finite Elements models, was not an option. The use of simple and fast, but less accurate, crack propagation models is the first step in the development of a wheel-rail life prediction model.

Another goal of this work was to verify the wheel-rail prediction tool against measurements of profile and crack development. For this purpose, the wheel profiles of trains running on the Stockholm commuter network have been measured together with the crack development on these wheels. Three train units were selected and their wheels have been measured over a period of more than a year. The maximum running distance for these wheels was 230,000 km. A chosen fatigue model was calibrated against crack and wear measurements of rails to determine two unknown parameters. The verification of the prediction tool against the wheel measurements, however, showed that one of the calibrated parameters was not valid to predict
RCF on wheels. It could be concluded that wheels experience relatively less RCF damage than rails. Once the two parameters were calibrated against the wheel measurements, the prediction tool showed promising results for predicting both wear and RCF and their trade-off. The predicted position of the damage on the tread of the wheel also agreed well with the position found in the measurements.

**Keywords:** multibody simulations, prediction, wear, RCF, wheel, rail, cracks, measurements.
Sammanfattning


Den här uppsatsen beskriver utvecklingen av ett verktyg för uppskattning av hjul-rällivslängd med hänsyn till både slitage och ytinitierad RCF. Huvudmålet med underliggande doktorsarbete var att utveckla ett verktyg i vilket samverkan fordon-spår är implementerad i form av dynamiska simuleringar. På sådant sätt kan man ta hänsyn till många olika kontakt situationer, som ett hjul eller en räl kommer att uppleva i verkligheten.

Slitagedelen av verktyget var redan framgångsrikt utvecklat av andra, för att användas i kombination med flerkroppssimuleringar. Sprickbildningsdelen läggde empi vetiska data till grunden för sprickbildningsmodellen, däremot, var mer komplicerad att användas i kombination med flerkroppssimuleringar, därför att sprickbildningsmodeller normalt är mycket tidskrävande. Därför behövdes fler förenklingar göras i sprickbildningsdelen av verktyget. Detaljerade och tidskrävande modeller av sprickorna med till exempel finita elementmetoden (FEM) var därför inte genomförbara. Användningen av enkla och snabba, men mindre noggranna sprickbildningsmodeller är första steget i utvecklingen av en modell för beräkning av livslängden hos hjul och räl.

Ett ytterligare mål i detta arbete var att verifiera beräkningsmodellen för hjul och räl mot mätningar av profiler och sprickbildning. Därför har hjulprofiler och sprickbildning i hjulen hos Stockholms pendeltåg mätts upp. Tre tåg valdes ut och deras hjuls mättes upp under mer än ett år. Den längsta körstreckten för hjulen var 230,000 km.

Den valda utmattningsmodellen kalibrerades mot mätningar av sprickor och slitage av rälen för att bestämma två okända parametrar. Verifieringen av den utvecklade modellen mot hjulumätningarna, däremot, visade att en av de kalibrerade parameterna inte var giltig för att

**Keywords:** flerkroppssimuleringar, prediktering, slitage, RCF, hjul, räl, sprickor, mätningar.
Dissertation

This thesis consists of an introduction to the area of research, a summary of the present work and the following appended papers:

Paper A


All multibody simulations were carried out by Dirks. The paper was written by Dirks under the supervision of Enblom.

Paper B


The multibody dynamics model of "vehicle B" was constructed by Dirks. All simulations were carried out by Dirks. The paper was written by Dirks under the supervision of Enblom.

Paper C


All measurements were performed by DeltaRail. The vehicle models of the two passenger trains were constructed by Dirks. All multibody simulations were carried out by Dirks. The paper was written by Dirks under the supervision of Enblom and Berg and in discussion with Ekberg.

Paper D


Planning and execution of the measurements were performed by Dirks. All multibody simulations were carried out by Dirks. The paper was written by Dirks under the supervision of Enblom and Berg.
Publication not included in the thesis

Part of this thesis work was presented at a conference:

Thesis contribution

This thesis presents the development of a wheel-rail life prediction model with regard to wear and rolling contact fatigue.

This thesis contributes to the present research field as follows:

- A computational tool was developed to predict RCF damage in terms of crack size (surface length and depth), in combination with wear of railway wheels and rails. The tool can predict a surface crack length or crack depth where the effect of wear is included. Thus, how the crack would be in reality.

- Existing RCF prediction models were extended to be used in the prediction tool. By calculating the shear stresses locally for each cell element in the contact patch, the effect of partial slip could be better taken into account. By including spin creepage, the prediction of RCF life for the outer rail in a curve is more realistic.

- A multibody dynamics vehicle model of the commuter train in question was developed and verified against measurements. The vehicle model consists of three different bogies: a standard motor bogie, a Jacobs motor bogie and a Jacobs trailing bogie.

- Both braking and traction were included in the simulations to make the wheel-rail contact conditions more realistic and to model the difference between the motor and trailing bogies.

- The effect of on-board lubrication and track-side lubrication on the wear rate of the flange was studied.

- Wheel profile measurements and RCF inspections on the wheels of the Stockholm commuter train were performed together with RCF inspections on the rails of the Stockholm commuter network.

- A correlation was found between the orientation of the cracks on wheels and the direction of the responsible forces.

- The wheel-rail life prediction tool was verified against wheel profile and crack measurements.
# Contents

Preface ................................................................................................................................ iii
Abstract ...................................................................................................................................... v
Sammanfattning ....................................................................................................................... vii
Dissertation ................................................................................................................................ ix
Thesis contribution .................................................................................................................. xi
1 Introduction ........................................................................................................................... 1
2 Vehicle-track interaction ...................................................................................................... 3
3 Wheel-rail wear .................................................................................................................... 7
   3.1 INTRODUCTION ............................................................................................................ 7
   3.2 RAIL WEAR ................................................................................................................ 8
   3.3 WHEEL WEAR .......................................................................................................... 9
   3.4 WEAR PREDICTION MODELS ................................................................................ 10
      3.4.1 Archard's model ................................................................................................. 10
      3.4.2 The energy dissipation model .......................................................................... 13
      3.4.3 The ‘brick’ model .............................................................................................. 14
   3.5 WEAR PREDICTION MODELS – CONCLUSIONS ................................................. 14
4 Rolling contact fatigue (RCF) ............................................................................................ 17
   4.1 INTRODUCTION ........................................................................................................ 17
      4.1.1 Contact in full slip ............................................................................................. 18
      4.1.2 Contact in partial slip ....................................................................................... 23
   4.2 RCF ON RAILS ......................................................................................................... 27
      4.2.1 Head checks ...................................................................................................... 27
      4.2.2 Squats ............................................................................................................... 31
   4.3 RCF ON WHEELS .................................................................................................... 32
   4.4 IMPORTANCE OF FLUID ENTRAPMENT ................................................................ 34
   4.5 SURFACE RCF PREDICTION MODELS ............................................................... 35
      4.5.1 Shear forces ...................................................................................................... 35
      4.5.2 Energy dissipation ............................................................................................ 36
      4.5.3 Crack growth modelling .................................................................................. 38
   4.6 SUBSURFACE RCF PREDICTION MODEL ......................................................... 42
   4.7 RCF PREDICTION MODELS – CONCLUSIONS ................................................. 43
5 Interaction of wear and RCF .............................................................................................. 47
6 The present work: a wheel-rail life prediction tool .......................................................... 49
   6.1 WHEEL–RAIL LIFE PREDICTION TOOL .............................................................. 49
   6.2 VERIFICATION WHEEL–RAIL LIFE PREDICTION TOOL .................................... 51
      6.2.1 Wheel measurements ......................................................................................... 52
      6.2.2 Rail measurements ............................................................................................ 55
6.3 Verification of vehicle model ................................................................. 57
7 Summary of appended papers........................................................................ 59
  7.1 Paper A .................................................................................................. 59
  7.2 Paper B .................................................................................................. 59
  7.3 Paper C .................................................................................................. 60
  7.4 Paper D .................................................................................................. 61
8 Conclusions and future work.......................................................................... 63
  8.1 Conclusions ......................................................................................... 63
  8.2 Future work ......................................................................................... 65
References ........................................................................................................ 67
Appended papers .............................................................................................. 73
1 Introduction

The maintenance costs of rails and wheels are mainly influenced by wear and rolling contact fatigue (RCF). A competition exists between surface-initiated RCF and wear: at a high wear rate, RCF does not have the opportunity to develop further. Therefore, one measure to control RCF is to grind the track and turn the wheels. This way, artificial wear wins over RCF. In order to increase the service life of rails or wheels, there is an optimal amount of metal to be removed both by natural wear and grinding or turning. This optimum rate of wear is often called "the magic wear rate" [1]. An example of the magic wear rate for a rail in a curve which is achieved by grinding is shown in Figure 1.1.

![Figure 1.1 The "magic wear rate" for a rail in a curve achieved by grinding [2].](image)

An example of the predicted interaction between wear and RCF for a wheel of the Stockholm commuter train is shown in Figure 1.2. The actual crack length is determined by subtracting the wear development from the crack development. Figure 1.2 shows that no RCF damage occurs when the wear is higher than the crack growth rate. But, RCF damage does occur and the cracks are growing when the crack growth rate increases above the wear rate.

Simulation models that can predict wheel/rail wear and RCF can be used to understand the causes of high wear rate and/or RCF damage and to develop cost-effective measures, for example:

- Wheel/rail optimization. This way, different wheel/rail profile combinations can be tested in order to reduce the amount of wear and/or RCF.
- Vehicle/track optimization. This way, the influence of certain vehicle and track parameters (primary stiffness, axle load, track geometry, etc.) can be tested.
Preventive rail grinding/wheel turning programmes. How long a train can run or how much tonnage a rail can take before the RCF damage on a wheel or rail becomes critical in terms of high crack propagation rates, can be tested.

Each vehicle and each track curve is unique, so detailed information about the network on which a vehicle is running and about the vehicles which are running through a specific curve is necessary input for such a prediction tool.

Figure 1.2 The predicted development of the surface crack length on a wheel of the Stockholm commuter train in case the effect of wear is excluded and included.

The main goal of this PhD thesis is to develop a model which can predict the total expected life of railway wheels and rails. The first objective was to gain insight into the causes of wear and RCF damage and to obtain an overview of the existing wear and RCF damage models. An introduction to vehicle-track interaction is therefore presented in Chapter 2 and an overview of the existing wear and RCF models is discussed in Chapters 3 and 4 respectively. Chapter 5 describes how wear and RCF interact.

Another goal of this thesis is to verify the prediction tool against wear and crack measurements and a number of reference vehicles and curves were therefore selected for measurements. Chapter 6 discusses the present methodology for the prediction tool and also describes the wheel and rail measurements. Chapter 7 gives a summary of the appended papers and finally Chapter 8 presents the main conclusions of this PhD work together with some suggestions for future research.
2 Vehicle-track interaction

In order to understand how damage on wheels and rails is caused, it is important to describe how the forces and sliding motions in the wheel-rail contact are. This chapter briefly discusses vehicle-track interaction.

In curves, the outer rail is longer than the inner rail. When a wheelset moves laterally towards the outside of a curve, the rolling radius of the outer wheel will increase and the rolling radius of the inner wheel will decrease due to the conical shape of the wheels [3], [4]. Therefore, a difference in rolling radius occurs. Since the circumference of the outer wheel is larger, it will try to roll further than the inner wheel for a given and common rotational speed. If the wheelset moves sufficiently far laterally towards the outside of the curve, the rolling radius difference will be enough to compensate for the difference in rail lengths. In this case, the single wheelset achieves "pure rolling" without any wheel-rail friction forces being generated. However, in most cases, the wheelset is not able to position itself perfectly radial in a curve. For example, in Figure 2.1 the wheelset is yawed in a direction relative to the track, which is called an under-radial position. Wheelset yaw relative to the track, as shown in Figure 2.1, is often called "angle of attack" (ψ). When the contact surface of the wheel moves relative to the rail, sliding motions occur which are known as creepage. Creepage is defined as the quotient of sliding velocity (v) and vehicle speed (V) and can be divided into three components: 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 when divided by the vehicle speed [3], [4]:

\[
\begin{align*}
\text{Longitudinal creep:} & \quad v_x = \frac{v_x}{V} \\
\text{Lateral creep:} & \quad v_y = \frac{v_y}{V} \\
\text{Spin creep:} & \quad \varphi = \frac{\omega}{V}. 
\end{align*}
\]

(2.1)

Longitudinal creep mainly depends on the lateral shift of the wheelset and the wheels' conicity. Lateral creep is essentially proportional to the angle of attack. As indicated above, spin creepage is a relative angular velocity between the wheel and the rail. It consists of two components. The first component is the yaw velocity of the wheelset and the second component is the rotational speed. When the contact plane is not parallel to the rotational axis of the wheel, the rotational speed can be divided into a speed parallel to the contact plane, which is pure rolling, and a speed perpendicular to the contact plane. This is illustrated in Figure 2.2.
Due to these creepages in the wheel-rail contact, creep forces (friction forces) are generated on both wheel and rail. These creep forces act in longitudinal and lateral direction and are in the opposite direction to the relative motion between wheel and rail, see Figure 2.3 and Figure 2.4. The relationship between creepage and creep force is in general nonlinear. For small creepages, however, the relationship can be considered to be almost linear.

In some parts of the contact patch, the surfaces of wheel and rail move relative to each other, so that they slide or slip. If the parts of a contact are in slip, the tangential stress will be equal to the coefficient of friction times the contact pressure, whereas for the parts which are not in slip, the tangential stress is lower.

The tangential stress is lower at the leading edge of the contact and higher at the trailing edge [5]. Slip will, therefore, first occur at the rear side of the contact where a sliding area is established. In the rest of the contact area, the adhesion area, no slip occurs. The total transmitted friction force increases with the sliding area of the contact. When the entire contact area is in sliding, the total friction force is equal to the coefficient of friction times the normal force in the contact.
Figure 2.3 Radial position of a wheelset in a curve with in (a) the longitudinal creepages and (b) the longitudinal creep forces. Forces on the rails act in opposite directions.

Figure 2.4 Under-radial position of a wheelset in a curve with in (a) the lateral creepages and (b) the lateral creep forces. Forces on the rails act in opposite directions.

Due to the longitudinal creep forces on the outer and inner wheels in a curve, a steering yaw moment initiates which forces the wheelset to yaw, resulting in a smaller yaw angle for an under-radial wheelset. The longitudinal creep forces improve the position of the wheelset in the curve towards a more radial position. The lateral creep force on the outer and inner wheels, however, points outwards in the curve in case of under-radial steering, which is in the same direction as the centrifugal forces. The lateral component of the normal force on the wheels has to compensate for these forces.

For a wheelset to be able to steer radially in a curve, the yaw moment, caused by the longitudinal creep forces, has to be higher than the resisting forces of the primary suspension. A more radial position of a wheelset in a curve can, therefore, be accomplished by making the primary suspension more 'soft'. A softer primary suspension, however, will give lower stability on tangent track at higher speeds.

If a bogie with a very rigid primary suspension and wheelbase $2\alpha$ negotiates a curve of radius $R$, the angle of attack ($\psi$) can be defined as:
\[ \psi = \frac{a}{R} \]  

Bogies with a larger wheelbase therefore experience higher creep forces in curves. This can result in more damage to the wheels and rails in terms of wear and/or rolling contact fatigue (RCF) which is shown in the present work [6].
3 Wheel-rail wear

This chapter describes what wear on wheels and rails looks like and what methods exist to predict wheel-rail wear. The location of wear on a rail or wheel depends on where the contact position is. The amount of wear will be much less when the vehicle is running on straight track since the wheels are mainly rolling with very little sliding.

3.1 Introduction

Many wear mechanisms exist and there are several that are dominant in the wheel-rail contact. Some mechanisms will be briefly discussed here [7].

Oxidative wear occurs under mild contact conditions, when the forces and sliding velocity are low. Under the influence of water and oxygen, oxides form on the surface and will eventually break away in the wheel-rail contact.

Adhesive wear occurs when the adhesion forces in a sliding contact are high; shear takes place in the weakest material instead of at the surface interface. This may result in detachment of fragments from one surface and attachment to the other surface. The surface often looks quite smooth when adhesive wear takes place.

Abrasive wear occurs when a hard surface cuts material away from a softer surface. This hard surface can also be wear debris. The surface generally looks quite rough when abrasive wear takes place.

Fatigue wear takes place due to repeated loading and unloading. Cracks occur after a while at the surface or underneath (subsurface). The cracks propagate and after a time particles from the surface may break out. A competition can exist between wear and fatigue: when the wear rate is high, fatigue cracks have no time to grow and simply wear away. This effect was also seen with head-hardened rails. Due to the increased hardness of the rails, the amount of adhesive and abrasive wear was reduced, resulting in more fatigue damage.

Plastic deformation is not a wear mechanism, but it is considered to be surface damage without loss of material. The shape of a rail or wheel profile in this case is changed due to transfer of material to a different location.

Different wear regimes can be defined depending on the wear rate. Three regimes can often be found in the literature: Type I (mild wear), Type II (severe wear) and Type III (catastrophic wear) [8]. The jump from mild to severe and catastrophic wear depends on the combination of sliding velocity and contact pressure.

The main reason why wheel/rail wear is so important is safety against derailment. When the shape of the rail and wheel profiles is changed due to wear, a train could usually derail more easily. Another factor is that the profile shape influences the dynamic behaviour of the train,
which affects safety and ride comfort. But also, when a wheel/rail profile has changed due to wear, the wheel-rail forces could increase, resulting in more damage to vehicle and track.

The wheel profile development of a vehicle running on the Stockholm commuter network is shown in Figure 3.1 as an example of wheel wear.

3.2 Rail wear

How the rails are affected by wear will be discussed first. The location of the highest wear on rails depends on whether the rail is located in a curve or on straight track. Two examples of what worn rails can look like are shown in Figure 3.2. In a curve, the rail is exposed to higher creep forces and creepages and will show more wear. Since the wheel-rail contact on the outer (high) rail in a curve is located at the gauge corner, the highest wear will take place here as well. For the inner rail in a curve, the highest wear will be located on top of the rail. For rail profiles on straight track, the wear will be more evenly distributed on the top and at the gauge corner, mainly due to irregularities in the track. The wear rate can be reduced by applying track-side lubrication in sharp curves. The influence of track-side lubrication has been measured in [9]. It was found that lubrication is quite effective to reduce the wear rate in a curve up to a distance of 200 m after the lubrication device. And also after the 200 m, the rails were still affected significantly by the lubricant.
3.3 Wheel wear

The location of the highest wear on a wheel profile depends on the type of railway network the train is running on. The wear on the flange will be highest when the network has many sharp curves. The smaller the curve radius, the more wear due to a larger angle of attack of the wheelset [2]. When the network consists mainly of straight track, the wheel profiles will show high tread wear, see Figure 3.3. This high tread wear is often called ‘hollow’ wear and can give ride instability problems since the conicity is increased. To reduce the amount of flange wear, on-board or track-side lubrication can be applied.

Figure 3.3 Worn (dashed lines) and new wheel profiles, (a) evenly distributed wear on the tread and flange, (b) mainly tread wear and (c) mainly flange wear.

In dry contact conditions, the wear rate will in general also be high. However, studies have also shown that a relatively high coefficient of friction can improve the wheelsets’ radial steering ability, resulting in less wear which is shown in [10] and in the present work [6].

The wear rate also increases with a stiffer wheelset suspension and a longer wheelset base, due to a larger angle of attack [2].

Wheel profile measurements showed that powered wheels experience more tread wear than trailing wheels [9].

To determine the worn status of a wheel profile, the following quantities can be used:

- Flange thickness, $t_f$
- Flange height, $h_f$
- Flange inclination, $q_f$.
Flange wear results in reduced flange thickness \( (t_f) \) and increased flange inclination \( (q_r) \) decreases) [2]. Whereas, wheel tread wear results in increased flange thickness \( (t_f) \) and increased flange height \( (h_f) \).

![Running circle](image)

Figure 3.4 Definition of wheel profile scalar measures: Flange thickness \( (t_f) \), flange height \( (h_f) \) and flange inclination \( (q_r) \). Units are in mm [10].

### 3.4 Wear prediction models

Wear models can be divided into two principal modelling methods: frictional (energy) work models, where the wear rate is related to the work done in the wheel-rail contact [11], [12], [13], and sliding models according to Archard [14], where the wear rate is related to the sliding distance, normal force and hardness of the material. Both models will be discussed briefly here.

#### 3.4.1 Archard's model

According to Archard’s wear model [14], the wear volume \( V_{\text{wear}} \) \( [\text{m}^3] \) can be calculated with the equation:

\[
V_{\text{wear}} = k_i \frac{S \cdot N}{H},
\]

where \( k_i \) is the wear coefficient \([-]\), \( S \) the sliding distance \([\text{m}]\), \( N \) the normal force \([\text{N}]\) and \( H \) the hardness of the material \([\text{N/m}^2]\).

In [15], the wear has been calculated in detail by discretizing the contact patch with a grid of elements. By replacing the normal force \( N \) in Eq. (4.1) with the product of the contact pressure \( p \) \([\text{N/m}^2]\) and the area of an element \( \Delta x \Delta y \), the wear depth for each cell element \( \Delta z \) \([\text{m}]\) could be calculated:

\[
\Delta z = k_i \frac{S \cdot p}{H}.
\]

One important feature of Archard’s wear model is that there will be no wear in the adhesion zone of the contact since the sliding distance is zero (zero slip velocity).

In Archard’s wear model, the wear distribution is calculated for each wheel revolution \( (dz \) \([\text{m}]\)) by adding the wear depth along the longitudinal direction for each cell element.
The wear coefficient $k_i$ can be determined by laboratory measurements and expressed in a wear map depending on the sliding velocity and contact pressure. A wear map is shown in Figure 3.5. It can be concluded from this example that the sliding velocity has a substantial impact on the wear coefficient. It was concluded in the present work [6] that the wear map is sensitive to small changes in the sliding velocity.

![Figure 3.5 Wear map for the wear coefficient $k_i$ under dry conditions [15].](image)

The wear map above was obtained under dry contact conditions. The wheel-rail contact, however, is often lubricated either naturally (rain, snow, etc.) or artificially using a lubrication device (track-side or on-board). The amount of calculated wear under dry conditions therefore has to be downscaled for lubricated conditions.

Pin-on-disc wear tests were performed in [16], [17] to determine the wear rate under dry and grease lubricated conditions. Both tests were conducted at a humidity of 30%. The effect of lubrication can be seen in Figure 3.6, which shows the results (wear coefficients) of these pin-on-disc tests. The tests were made for five different sliding velocities (0.05, 0.25, 0.6, 1.2 and 1.8 m/s) and three different contact pressures (0.68, 1.28 and 2.19 GPa). The wear coefficients in Figure 3.6 were derived from the measured wear rate $W$ [kg/m] from the tests according to:

$$k_i = \frac{(W/\rho) \cdot H}{F},$$

(3.3)

where $k_i$ is the wear coefficient [-], $\rho$ is the density of steel (7600 [kg/m$^3$]), $F$ [N] is the applied load and $H$ is the hardness of steel (2.94x10$^9$ [N/m$^2$]). It could be concluded from the tests that the influence of lubrication is significant.
Figure 3.6 Wear map with wear coefficients for (a) non-lubricated and (b) lubricated contact. Derived from pin-on-disc testing [17]. The numbers inside the wear map show the wear coefficient for different regions of contact pressure and slip velocity, $x10^{-4}$ for non-lubricated and $x10^{-7}$ for lubricated contact.

The influence of natural lubrication, for example water, on the wear rate was not tested. In [8] a comparison was made between the wear coefficient determined from full-scale field tests and laboratory pin-on-disc tests. The pin-on-disc tests were carried out on specimens cut out from a rail section where field tests were performed. These rail specimens came from a 300 m radius curve on the Stockholm commuter network. Results from this study show that the wear coefficient (Archard) for the full-scale field tests was about 4 times lower than for the pin-on-disc tests. This was probably caused by natural lubrication in the full-scale field tests since the pin-on-disc tests were performed under dry conditions. So the effect of a natural lubricant on the wear rate seems to be much lower than the effect of an "artificial" lubricant.
Archard’s wear model has already been successfully used for predicting wheel wear [15], [18], [19]. The model was also applied to predict rail wear in [20]. The study showed good results for the shape of the predicted rail profiles, but the amount of wear was too high compared to measurements.

### 3.4.2 The energy dissipation model

Pearce and Sherratt developed a method to calculate wear based on the energy dissipation in the contact [11]. This relationship between wear and energy dissipation depends on in which regime the wear is:

\[
T_Y < 100 \text{ N}: A_w = 0.25 \cdot \frac{T_Y}{D} \quad \text{(mild regime),}
\]

\[
100 \leq T_Y < 200 \text{ N}: A_w = \frac{25.0}{D} \quad \text{(transition regime),}
\]

\[
T_Y \geq 200 \text{ N}: A_w = \frac{1.19 \cdot T_Y - 154}{D} \quad \text{(severe regime),}
\]

where \(T\) is the total creep force [N], \(\gamma\) the creepage [-], \(D\) the wheel diameter [mm] and \(A_w\) [mm\(^2\)] the worn-off area per travelled kilometre. The worn-off area for one wheel revolution (independent of wheel diameter) is shown in Figure 3.7. The calculated worn area is assumed to be distributed parabolically across the width of the contact patch. A scaling factor can be applied to take natural lubrication into account.

![Figure 3.7 Wear rate according to Pearce and Sherratt [11] for one wheel revolution.](image)

Another wear model which is based on the energy dissipation in the wheel-rail contact has been developed by using twin-disc wear results [21], [12], [13]. An equation was derived from these tests, which is similar to the model developed by Pearce and Sherratt:

\[
\text{Wear rate} = K \frac{T_Y}{A},
\]

where the Wear rate is expressed as the weight of lost material [\(\mu\)g], per distance travelled [m], per contact area \(A\) [mm\(^2\)]. This model also has three different wear regimes: mild, severe
and catastrophic. The tests were carried out under dry contact conditions. The wear can be calculated locally by dividing the contact area into elements.

### 3.4.3 The ‘brick’ model

The ‘brick’ model is a tool to predict the wear rate and the potential for crack initiation [22], [23]. In this model, a cross-section through the rail (parallel to the direction of traction) is modelled as a mesh of elements, or ‘bricks’. Each element is assigned the material properties initial shear yield stress \( k_0 \) and critical shear strain for failure \( \gamma_c \). The elements are able to harden depending on the amount of strain they have accumulated. This results in an effective shear yield stress \( k_{\text{eff}} \):

\[
k_{\text{eff}} = k_0 \max \{1, \beta \sqrt{1 - e^{-\alpha \gamma}}\},
\]

where \( \gamma \) is the total accumulated shear strain in an element and the material parameter \( \alpha \) is a measure of how fast a material hardens, while \( \beta \) is a measure of how much it hardens.

If the maximum shear stress \( \tau_{\text{zx}}(\text{max}) \) in an element in row \( j \) (depth) exceeds the effective shear yield stress \( k_{\text{eff}} \), the element in row \( j \) and column \( i \) has an increment of plastic shear strain:

\[
\Delta \gamma_{ij} = \frac{\tau_{\text{zx}}(\text{max})}{k_{\text{eff}}_{ij}},
\]

where \( C \) is an estimated constant of 0.00237 for BS 11 rail steel. A brick fails if

\[
\sum_N \Delta \gamma_{[ij]}^{[ij]} = \gamma_{c[ij]}^{[ij]} \geq \gamma_{c[ij]}^{[ij]},
\]

where \( N \) is the number of cycles. When a brick fails, it is marked as weak. When a weak brick, which is exposed to the surface, is unsupported by neighbouring bricks, it detaches and is considered to be wear debris. When there are clusters of failed bricks beneath the surface, it is likely that crack initiation and propagation will occur at this location.

### 3.5 Wear prediction models – conclusions

Some of the wear models discussed above were investigated in several studies [24], [25], [26] and in the present work [6]. All these studies used the output from multi body dynamics simulations as input for the different wear models. Some of the results will be discussed here.

Mainly three different wear models have been compared in these studies (see Section 3.4). The first model is based on Archard’s wear model and was developed by KTH and will be called AR here. The second and third models are based on the energy dissipation according to Pearce and Sherratt, PSH here, and Ward, WD here.

In the first study [24], the wheel wear was determined for a vehicle running between two cities (96 km) on the Italian railway network. The track is considered rather curved, since
61% of the curves have a curve radius below 450 m. The track irregularities are not included in this study and only dry adhesion conditions are applied here. The results of the three different wear models (AR, PSH and WD) were compared for a total travelled distance of 5,000 km. The results show a good agreement between the three different wear models. The calculated wear on the flange is around 20-30% higher for the PSH model compared to the AR and WD models. For the calculated wear on the tread, the AR model shows slightly more wear compared to the PSH and WD model.

In the second study [25], the wheel wear was determined for two different cases: a curve with a 1200 m curve radius with a coefficient of friction of 0.3, and an R300 m curve with a coefficient of friction of 0.5. Different adhesion conditions for each curve were also studied. The wear was determined for a vehicle running on the Stockholm commuter network. The running distance for the 300 m curve was 1,000 km and for the 1200 m curve 6,000 km. Track irregularities were included in this study. The results show that for the 1200 m curve (mainly tread contact), the AR model predicts much more wear than the PSH and WD model. they also show that the AR model calculates more wear for poor adhesion conditions, due to more sliding, and that the WD model calculates more wear at the flange for dry adhesion conditions. The results for the 300 m curve (flange contact) show much more wear for the WD model compared to the AR and PSH model. The conclusions with regard to the adhesion conditions are the same.

In the third study [26], two different cases were investigated with the same vehicle running on the Stockholm commuter network as in study 2: a curve with a 1350 m radius and a corresponding running distance of 6,000 km and an R = 400 m curve with a 1,000 km running distance. Updating of the wheel profile and different adhesion conditions was applied. The results show that the WD model calculates less wear on the tread and more wear on the flange compared to the AR and PSH model. The WD model shows here as well that for dry contacts the wear at the flange increases. The AR model also shows here that the wear increases in poorer adhesion conditions.

A summary of all the above studies is shown in Table 1 for the highest calculated wear for each wear model.

<table>
<thead>
<tr>
<th>Study</th>
<th>PSH</th>
<th>AR</th>
<th>WD</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>flange</td>
<td>tread</td>
<td>flange</td>
</tr>
<tr>
<td>1 [24]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 [25]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3 [26]</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

It can be concluded that according to these studies, the wear on the tread is highest for the AR model. For the wear on the flange, however, study 1 calculated more wear for the PSH model and study 2 and 3 calculated more wear for the WD model. This difference is mainly because
study 1 did not investigate the influence of different adhesion conditions for the wear models whereas studies 2 and 3 showed that the WD model reacts quite strongly for flange wear in dryer contact conditions.

It can thus be concluded from these studies that it is not really clear which wear model would be the best choice to use in a prediction tool, in particular because none of the studies compared the results of the different wear models with real wear measurements of wheel profiles or rail profiles. Good wear predictions were obtained with the different wear models, for example in [15], but it is not certain if these results would have been the same or perhaps even better with a different wear model.

The brick model has too much detail to be used in combination with vehicle dynamics simulations.

Since there is already a great deal of good experience at KTH of using Archard's wear model, this model was selected for use in the life prediction tool in this PhD work (Chapter 6).
4 Rolling contact fatigue (RCF)

Like wear, rolling contact fatigue (RCF) is also a deterioration phenomenon and will be discussed in this chapter. Some information will be given about when, where and how RCF damage occurs. Some of the already existing RCF prediction models will also be discussed.

4.1 Introduction

RCF can be divided into Head Checks, Squats [27] and Tache Ovales. The first two initiate at the surface and are caused by a combination of high normal and tangential stresses between wheel and rail. Squats only occur in the rail, whereas head checks can also occur in railway wheels. Squats are caused by local irregularities in the track surface which lead to high dynamic contact stresses.

The initiated cracks grow inside the rail at a shallow angle to the rail surface until a depth of a few millimeters. This often leads to "spalling" of the material from the surface of the rail. Some cracks, however, can turn down into the rail, which could cause the rail to break. Rail breakage can lead to catastrophic accidents like the one in Hatfield in the UK in 2000 [28].

Tache ovales are commonly considered to be defects which develop inside the railhead due to longitudinal cavities caused by the presence of hydrogen [27]. Thanks to improved steel production techniques, tache ovales rarely occur in the base rail material, but still sometimes occur at welds [29].

The measures which can be taken to control RCF are:

- Grind/turn the cracks away on the rails or the wheels so that the initiated cracks are stopped [27].
- Grind the high rails of a curve in a special profile shape which prohibits contact in the RCF sensitive area of the rail (gauge corner).
- Lubrication in curves, which reduces the friction forces in the wheel-rail contact [30].
- Maintenance of the track and vehicles. For example, vertical and lateral track irregularities can give high dynamic forces. The dynamic forces also depend on the amount of wear and the structure underneath the track.
- Use of head-hardened rails in curves, since they have a greater resistance to RCF and wear [31], [32]. Head checks will develop more slowly.
- Use of two-material rails, where a coating is applied on the railhead [33]. This coating improves resistance to RCF damage.
- Use of active steering systems for wheelset/bogie yaw moment control and wheelset/track lateral position control [34].
- Reducing the primary yaw stiffness in order to reduce the friction forces in the wheel-rail contact [35].

The response of a material due to cyclic loading in a rolling contact may be one of four kinds [36],[37]: A perfectly elastic response if the maximum stress does not exceed the yield stress.
of the materials in contact (see Figure 4.1a). When at first the elastic limit is exceeded, but due to residual stresses and strain hardening, the response is after some load cycles elastic again; this is called *elastic shakedown* (Figure 4.1b). The response is called *plastic shakedown* (Figure 4.1c) when a closed cycle of plastic stress-strain loop occurs without any accumulated plastic deformation. When plastic strain accumulation takes place (open cycle of plastic stress-strain) for each load cycle, this is called *ratchetting* (Figure 4.1d).

Plastic deformation at the beginning of the cycles can influence the shakedown in the steady cyclic state. There are three possible mechanisms [36]:

- Residual stresses, which are protective and make further plastic deformation less likely.
- Strain hardening of the material.
- Geometry changes.

4.1.1 **Contact in full slip**

As mentioned earlier, the wheel-rail contact can be in full slip (sliding). This often occurs for contacts located on the flange of an outer wheel and on the gauge corner of an outer rail in curves.

Assume a cylinder rolling on a half-space under the action of a normal load $P$ and a tangential force $Q$, see Figure 4.2. The stresses at the contact surface are dependent on both the pressure and the friction forces.
According to the Tresca yield criterion, material will flow plastically when the principal shear stress $\tau_1$ exceeds the shear yield strength $k$ [38]:

$$\tau_1 = \frac{\sigma_1 - \sigma_2}{2} = k,$$

(4.1)

where $\sigma_1$ and $\sigma_2$ are the principal stresses [39]:

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_z}{2} \pm \sqrt{\frac{(\sigma_x - \sigma_z)^2}{2} + \tau_{xz}^2}.$$

(4.2)

The principal shear stress $\tau_1$ is therefore:

$$\tau_1 = \frac{1}{2} \{(\sigma_x - \sigma_z)^2 + 4\tau_{xz}^2\}^{1/2}.$$

(4.3)

The normal pressure distribution for a line contact is according to Hertz [38]:

$$p(x) = p_0 \{1 - x^2/a^2\}^{1/2},$$

(4.4)

where $p_0$ is the maximum contact pressure, $a$ is the half-width of the contact and $x$ is the longitudinal position. The tangential stress is:

$$q(x) = \mu p_0 \{1 - x^2/a^2\}^{1/2},$$

(4.5)

where $\mu$ is the coefficient of friction.

If there is no friction, the maximum value of the principal shear stress is $0.30p_0$ at a depth of $0.78a \ (p_0/k=3.33)$ [40]. When there is tangential traction as well, the maximum value of the principal shear stress occurs closer to the surface. For high values of the coefficient of friction ($\mu > 0.367$ [41]), yield first occurs at the contact surface.

For line contacts with full slip, the stresses at the contact surface are [40]:

Figure 4.2 Rolling/sliding contact of a cylinder [38].
\[ \sigma_x = -p_0 \left\{ (1 - x^2/a^2)^2 + 2\mu x/a \right\}, \]  
\[ \sigma_z = -p_0 (1 - x^2/a^2)^{1/2}, \]  
\[ \tau_{xz} = -\mu p_0 (1 - x^2/a^2)^{1/2}. \]  
(4.6)  
(4.7)  
(4.8)  

The principal shear stress is therefore:

\[ \tau_1 = \frac{1}{2} \left\{ (\sigma_x - \sigma_z)^2 + 4\tau_{xz}^2 \right\}^{1/2} = \mu p_0. \]  
(4.9)  

This equation shows that the material will reach yield when:

\[ \mu p_0 = k \implies \frac{p_0}{k} = \frac{1}{\mu}. \]  
(4.10)  

As mentioned earlier, shakedown under repeated loading occurs when the initial plastic deformation results in residual stresses which make the response elastic again. These residual stresses may remain in the solid after the load has passed. Melan’s theorem can be used to determine the elastic-perfect plastic shakedown limit [40].

The principal stresses with the residual stresses included are then:

\[ \sigma_1 = \frac{1}{2} \left\{ \sigma_x + (\sigma_x)_r + \sigma_z \right\} + \frac{1}{2} \left\{ (\sigma_x + (\sigma_x)_r - \sigma_z)^2 + 4\tau_{xz}^2 \right\}^{1/2}. \]  
(4.11)  

\[ \sigma_2 = \frac{1}{2} \left\{ \sigma_x + (\sigma_x)_r + \sigma_z \right\} - \frac{1}{2} \left\{ (\sigma_x + (\sigma_x)_r - \sigma_z)^2 + 4\tau_{xz}^2 \right\}^{1/2}. \]  
(4.12)  

According to the Tresca criterion:

\[ \frac{1}{4} (\sigma_x + (\sigma_x)_r - \sigma_z)^2 + \tau_{xz}^2 \leq k^2. \]  
(4.13)  

This cannot be satisfied if \( \tau_{xz} \) exceeds \( k \), but it can be satisfied when it equals \( k \). In this case therefore:

\[ (\sigma_x)_r = \sigma_z - \sigma_x. \]  
(4.14)  

It can therefore be concluded that the limiting condition for shakedown occurs when anywhere in the solid:

\[ \tau_{xz} = k. \]  
(4.15)  

The maximum value of \( \tau_{xz} \) in case of no friction is \( 0.25p_0 \) (for \( z = 0.5a \) and \( x = 0.87a \) [40]), so shakedown occurs when \( p_0 < 4k \) (so \( p_0/k = 4 \)).

Melan’s theorem has been extended by Ponter for a kinematically hardening material. The principal stress in Equations 3-9 can be rewritten:

\[ \tau_1 = \left( \frac{\sigma_x - \sigma_z}{2p_0} \right)^2 + \left( \frac{\tau_{xz}}{p_0} \right)^2 = \left( \frac{k}{p_0} \right)^2. \]  
(4.16)  

The stress trajectories of \( \tau_{xz}/p_0 \) can be plotted against \( (\sigma_x-\sigma_z)/2p_0 \) for several different depths as the rolling load passes over, see Figure 4.3 for \( \mu = 0.2 \) [38], [41], [42]. The yield limit then
equals the radius of the circle which circumscribes the stress trajectories. In order to find the shakedown limit for a perfectly plastic material, the centre of the circle is displaced horizontally by freely chosen residual stresses $\rho_x$. The radius of the smallest circle which circumscribes all the stress trajectories represents the perfect plastic shakedown limit. For a material which strain hardens kinematically, the centre of the circle is free to move in stress space so it can be displaced in a vertical direction as well with the value $\rho_{xz}$. The radius of this circle is even smaller and therefore has a higher shakedown limit. The yield limit and the different shakedown limits are shown in Figure 4.5 for different values of the coefficient of friction.

![Figure 4.3 Stress trajectories in rolling and sliding, $\mu = 0.2$ [38].](image)

In case of high friction, the stress trajectory for $z = 0$ has the form of a semi-circle centered at the origin and with a radius $\mu$, see Figure 4.4. The smallest circle which circumscribes this trajectory therefore has the same radius. There are no residual stresses which can displace the centre of the circle. If $\mu > 0.435$, the elastic limit and the shakedown limits are all equal [41]. It can therefore be concluded that protective residual stresses cannot be introduced into the surface layer and that the shakedown limit is not improved by hardening.

$$p_0^\gamma = p_0^\delta = k/\mu.$$  \hspace{1cm} (4.17)
Figure 4.4 Stress trajectory in rolling and sliding, $\mu = 0.4$.

Figure 4.5 The contact pressure for first yield and shakedown for increasing values of the coefficient of friction for a cylinder on a surface (line contact) [40], [41]. Dashed line – line contact, first yield (Tresca). Chain line – line contact first yield (von Mises). Solid line – line contact, shakedown (Tresca). Since the contact is in full slip, the traction coefficient equals the coefficient of friction.
In the previous section, the wheel-rail contact was assumed to be in full slip. The wheel-rail contact, however, is often in partial slip instead of full slip. This means that one part of the contact is in slip (rear) and the other part is in ‘stick’ (front). For low friction, the results between full slip and partial slip are not so different. This is because the subsurface stresses are not influenced so much by the distribution of the traction.

In Figure 4.6, the contact of two similar cylinders is shown. In case of partial slip, the combination of two tangential traction distributions works in the contact [40]:

\[
q'(x) = \mu p_0 \left(1 - \frac{x^2}{a^2}\right)^{1/2}.
\]  
(4.18)

\[
q''(x) = -\frac{c}{a} \mu p_0 \left\{1 - \left(\frac{x + d}{c^2}\right)^2\right\}^{1/2}.
\]  
(4.19)

Combining these two tractions gives the resultant traction \( q(x) \) in the contact, which is also shown in Figure 4.6 (\( \tau_{xx} = q(x) \)).

The stress parallel to the surface, due to traction force \( q'(x) \), is:

\[
(\sigma_x)_{q'} = -2 \mu p_0 \left(\frac{x}{a}\right).
\]  
(4.20)

And the stress due to traction force \( q''(x) \) inside the ‘stick area’ (-\( a \leq x \leq c-d \)) is:

\[
(\sigma_x)_{q''} = 2 \mu p_0 \left(\frac{c}{a}\right) \left(\frac{x + d}{c}\right).
\]  
(4.21)

Outside the stick area due to traction force \( q''(x) \), the stress is:

\[
(\sigma_x)_{q''} = 2 \mu p_0 \left(\frac{c}{a}\right) \left\{\frac{x + d}{c^2} - \left(\frac{x + d}{c^2}\right)^2 - 1\right\}^{1/2}.
\]  
(4.22)

These stress distributions are shown in Figure 4.7 together with the total stress distribution \((\sigma_x)_q\) for \( Q = 0.25P \) and \( \mu = 0.3 \).

The size of the stick region can be determined by the total tangential force \( Q \):

\[
Q = \int_{-a}^{a} q(x)dx = \int_{-a}^{a} q'(x)dx + \int_{-c}^{c} q''(x)dx = \mu P - \frac{c^2}{a^2} \mu P,
\]  
(4.23)

so that

\[
\frac{c}{a} = \left(1 - \frac{Q}{\mu P}\right)^{1/2}.
\]  
(4.24)
Figure 4.6 Distribution of tangential tractions in rolling contact of similar cylinders [40].

Figure 4.7 Stresses at contact of cylinders rolling with tangential traction $Q = 0.25P$ and $\mu = 0.3$. (a) Tangential surface traction ($q(x)$); (b) Surface stresses $\sigma_x$ for $\mu = 0.3$.

Now that $\sigma_x$ and $\tau_{xz}$ are known in case of partial slip, the principal shear stress can be calculated. Since the stresses $(\sigma_x)$ and $(\sigma_z)$ due to the normal pressure are equal, they can be ignored and only stresses due to traction $q(x)$ are used to calculate the principal shear stress $\tau_1$. This is shown in Figure 4.8 for $\mu = 0.3$ and $Q_x = 0.25P$. In case of full slip, the principal shear stress would be $\tau_1 = \mu p_0$. Thus, for the same traction ($Q_x = 0.25P$):
\[
\frac{\tau_1}{p_0} = \mu = 0.25.
\]  
(4.25)

It can be concluded from Figure 4.8 and Eq. (3.25) that the maximum principal shear stress for partial slip is higher than for full slip.

In order to determine the shakedown limits in case of partial slip, the stress trajectories can be plotted again. For example, the surface stresses for \( \mu = 0.5 \) and \( Q = 0.2P \) are shown in Figure 4.9 together with the stress trajectory.

The principal shear stress with the residual stresses and hardening included will change according to Figure 4.10. It can be concluded that the maximum value of the principal shear stress \( \tau_1 \) is quite similar to the maximum value of the surface shear stress \( \tau_{xx} \). The maximum value of the principal shear stress according to shakedown is much lower. The shakedown limits for partial slip are shown in Figure 4.11 for different values of \( \mu \) and the traction coefficient \( Q/P \). It can be concluded from Figure 4.11 that for a fixed value of \( Q/P \), the contact in partial slip will cause more damage than the contact in full slip.
Figure 4.10 Principal shear stresses ($\tau$ standard and shakedown) and shear stress in case of partial slip ($\tau_{xz}$). $\mu = 0.5$ and $Q = 0.2P$.

Figure 4.11 Shakedown maps under conditions of partial slip: (a) Elastic-perfect-plastic; (b) Kinematic hardening [38].

The previous results are, however, only valid for line contacts and it is more realistic for the shape of the wheel-rail contact to be elliptical (or circular) due to a point contact. Determining the shakedown limits for point contacts, however, is much more difficult than for line contacts. The shakedown limit in case of a low coefficient of friction is again governed by the maximum value of $\tau_{xz}$ [40]. The maximum shear stress is $(\tau_{xz})_{\text{max}} = 0.21p_0$, so the shakedown limit is

$$p_0 \leq 4.7k.$$  \hspace{1cm} (4.26)

In case of a high coefficient of friction, the shakedown limit for full slip is represented by the curve:
\[ \frac{p_0}{k} = \frac{1}{\mu} \]  

(4.27)

The shakedown map for point contact is shown in Figure 4.21 and is often used to predict RCF on wheels and rails.

4.2 RCF on rails

4.2.1 Head checks

Sometimes, a distinction is made between "head checking" (HC) and "gauge corner cracking" (GCC), depending on where the cracks on the rails are located [43], see Figure 4.12. When cracks occur on the rail up to 10 mm from the gauge face, this is GCC and cracks further towards the rail crown are considered to be HC. Often the term head checking is used while it is officially gauge corner cracking. There are three stages in the life of head checks: crack initiation, shallow angle growth and transverse branching. When the crack depth is between 5 mm and 10 mm, the cracks can begin to ‘‘turndown’’ [44]. This often occurs when the crack length at the surface is about 30 mm. The risk of a rail fracture occurring is in this case high.

For small cracks (<20-30 mm), a relationship exists between the length of the crack visible at the rail surface and the depth of penetration [45]; the longer the visible crack at the surface, the deeper the crack is beneath the surface.

Figure 4.12 Distinction between "head checking" (HC) and "gauge corner cracking" (GCC) [43].

Three different surface RCF initiation modes on the rails can be distinguished [46]:

- Mode 0 (steady state curving), which occurs in sharp curves due to the high wheel-rail forces. Here the curve radius itself and the horizontal stiffness of the bogie's primary suspension are the main factors that initiate RCF.
• Mode 1, when bi-stable (jumping) contact occurs in mid-range curves due to the conformity of the wheel and rail shapes. Mode 1 RCF is therefore heavily influenced by changes in the track geometry, especially in the lateral direction, and changes in rail profile.

• Mode 2 (convergent motion) occurs in shallow curves and on straight track if the lateral alignment of the track causes sudden changes in contact position on the wheel, which generates high longitudinal creep forces due to the rolling radius difference. The main factor in this mode is therefore lateral track alignment.

It can therefore be concluded that several parameters are important to the initiation of RCF: Wheel and rail profiles, track irregularities and vehicle properties.

Cracks on the rails often appear on the gauge side of the high rail in a curve. This is due to the direction of the longitudinal creep force, which mostly faces backwards on the high rail, see Figure 4.13 (see also Section 4.4 about fluid entrapment). For both rails and wheels, the resulting creep force influences the orientation of cracks at the surface [47]. If the curve radius increases, the lateral creep force decreases, resulting in a larger angle $\alpha$ to the travel direction. This effect was also seen in rail crack measurements [48].

The cracks grow longer and deeper into the rails, which can result in pitting and shelling of the rail (see Figure 4.14).

![Figure 4.13 Direction of creep forces on the outer/high rail in a curve. The orientation of the cracks is perpendicular to the resulting creep force $F_R$.](image-url)
In the late 1990s, a large number of rail samples that had been cut through RCF cracks were examined [28]. From these samples, a rough correlation was found between the crack length at the surface and the depth of the crack into the rail. This correlation is illustrated in Figure 4.15. Here, four RCF categories were defined, depending on the visible crack length at the surface of the rail, see Figure 4.15.

It could be concluded from these samples that for cracks between 20 and 30 mm at the surface, there is a somewhat linear relationship between the visible crack length and the depth of the crack. But for cracks which are longer than 30 mm (severe region), there is no longer any relationship. The depth increases rapidly in the severe region; for visible cracks at the surface which are longer than 30 mm, nothing can be said about how deep the cracks are.
In [50], rails with RCF cracks were removed from service and cut into several slices, allowing the shape of the cracks to be visualized. How these cracks look in three dimensions is illustrated nicely in Figure 4.16. It was found from these cross-sections that when a crack is easily visible on the rail surface, it is around 3-5 mm deep. All the cracks found in that study fell within the region of expected crack depth and size covered by the chart in Figure 4.15.

RCF can be influenced by thermal stresses in the rail due to seasonal variations in temperature. When the rail is constrained longitudinally (no joints), temperatures above the stress-free (neutral) temperature result in compressive stresses in the rail. This has a positive
effect on RCF, since the crack faces are pressed together. However, when the rail is below the stress-free temperature, the tensile stress will open the cracks in the rail. The influence of thermal stresses on RCF is mostly not so great, but it may have a substantial influence in the final phase of cracking (see Section 4.5.3).

As mentioned earlier, one measure to control damage on the outer rails in curves due to head checks is to apply a so-called anti-head check rail profile. This rail profile looks similar to a worn rail profile on the outer rail in a curve. The shape of this rail profile avoids the contact being located inside the head check sensitive area (gauge corner) where the stresses are high.

4.2.2 Squats

Squats are small cracks initiated at the surface of a rail which can lead to spalling of the material. Like head checks, they can grow downwards in the rail which can result in a broken rail. The name derives from the shape of the squat; ‘'it looks as though a very heavy gnome has sat or squatted on the rail producing an indentation shape with two lobes of similar size’’ [49]. Squats appear as dark spots and often occur on straight track, very shallow curves and in switches.

According to [51], squats are mainly caused by rail surface irregularities like welds, indentations, wheel burns, corrugation, etc. A characteristic of a squat can be a wave pattern immediately behind it. A squat consists of two cracks growing in opposite directions at an (horizontal) angle of approximately 10-30° [52]. The trailing crack is much longer than the leading crack (in the direction of travel) [1]. After a while, the trailing crack may start to branch and grow more vertically. These dangerous cracks are often difficult to detect because they are shielded by the long horizontal crack. The two sub-surface cracks are responsible for the two lobes which can be seen on top of the rail.

Preventive grinding of the rails has been found to be an effective measure to control squats [52]. Studies in [53], however, show that many ‘squats’ found in previous studies (like the one in [51]) are actually a different type of defect, namely "studs". A stud is not in fact a rolling contact fatigue defect whereas a squat is. Studs are caused by thermal damage to the rail due to wheel slip. Wheel slip is caused by different friction conditions between the left and right rail. When one wheel slips due to low adhesion, the other wheel will slip as well because the wheels are interconnected. Thermal damage can therefore occur on the rail with higher friction. The main differences between squats and studs are [53]:

- Squats initiate on the gauge corner side of the running band whereas studs initiate in the middle of the running band.
- The presence of a hard white etching layer (WEL) is not a necessary condition for squats whereas a WEL is observed in all locations where studs have been found.
- Squats usually branch down and can form transverse defects whereas there is no evidence that studs can form transverse defects.
- Squats are found in locations with high driving traction whereas studs are found in locations with high driving and braking traction (approaches to signals).
- Squats develop after much more traffic (40 MGT) compared to studs (10 MGT).

### 4.3 RCF on wheels

"Head checks" can also occur on the wheels but are officially not called head checks since they are not located on a rail head. The RCF damage on the field side of the wheels is generally more severe than on the flange.

In comparison with rails, wheels experience much more variety in forces. They travel through different curves at different speeds, with different traction/braking and under different contact conditions. Rails are, on the other hand, installed in a curve with usually uniform conditions.

The highest creep forces on the wheels while curving often occur on the leading axle. Cracks therefore initiate/grow on the wheels of the first axle when the first axle is the leading axle and on the second axle when the second axle is the leading axle (thus different travel direction). According to the theory of fluid entrapment (see Section 4.4), the creep forces on the inner wheel in a curve will lead to most of the damage. This does not mean that damage on the flange does not occur. The damage is only less severe since the longitudinal creep force on the wheel is facing in the running direction so fluid does not become entrapped [54]. The most common direction of the creep forces on the leading inner wheels in a curve is according to Figure 4.17. The resulting creep force influences the orientation of cracks at the surface. The cracks are orientated perpendicular to the resulting creep force [55], [54].

**Figure 4.17 Creep forces on the wheels of a bogie at curving which are dominated by those occurring when the wheels are running as leading inner wheel. High resulting creep forces can cause cracks at the surface of the wheels in a perpendicular direction, from the present work [56].**

The orientation of the cracks on a wheel can thus say something about which creep force on the wheel mainly caused this damage (see Figure 4.18). Circumferential (longitudinal) cracks...
are for example caused by high lateral creep forces in sharp curves. Inclined cracks (angle around 45°) are caused by an equal share of longitudinal and lateral creep forces. Transverse cracks are caused by high longitudinal forces, for example due to braking or traction. These three different orientations of cracks are shown in Figure 4.19.

In [57], a difference was found between the RCF damage on powered and unpowered wheels. It was therefore concluded that traction and braking forces have a significant influence on wheel RCF. Circumferential cracks were found on the powered wheels only, so it was concluded that the traction forces (facing forward) in combination with high lateral creep forces due to curving in sharp curves were responsible for initiating these cracks. It is believed that the propagation of these cracks was caused by more frequently encountered forces in the moderate curves. It was also concluded from this study that the shear stresses in the wheel-rail contact were higher when running through switches, but that switches did not have a great influence on the observed RCF damage. The main damage would then have been closer to the flange of the wheel.

Figure 4.18 Common cracks on a wheel due to different creep forces in a longitudinal direction ($F_x$) and in a lateral direction ($F_y$). The orientation of the cracks is perpendicular to the resulting force ($F_R$) [1].
Figure 4.19 Cracks on the wheel surface dominated by: (a) longitudinal and lateral creep forces during curving, (b) longitudinal creep force at braking or traction on straight track and (c) high lateral creep forces in sharp curves, from the present work [56].

4.4 Importance of fluid entrapment

The presence of fluid in the wheel-rail interface plays an important role for the propagation rate of cracks. Twin-disc testing has been performed under dry and wet conditions [58] and showed that when water is present, the depth of the cracks increased drastically. Full-scale tests also showed that fluid was able to penetrate the cracks [59]. One explanation for this increased crack growth is that fluid can become entrapped in a crack when the crack's mouth closes, causing high pressure inside the crack under the wheel load (fluid entrapment mechanism). Due to the longitudinal creep forces on the inner wheel and outer rail in a curve, the cracks on wheel and rail are orientated in such a way that fluid can become entrapped inside the cracks, see Figure 4.20. This is because the crack opening enters the contact first, causing the crack to close. If the orientation of the cracks is in the opposite direction (due to creep forces on the outer wheel and inner rail), fluid will be squeezed out of the cracks instead.

Another effect is that the creep force on the inner wheel or on the outer rail causes the crack to open before the crack enters the contact and therefore fluid can easily enter.

When fluid can penetrate a crack, it can also lubricate the crack faces [61] and the crack can propagate easily in the shear mode (fluid lubrication mechanism). Studies show that decreasing the crack face coefficient of friction results in a dramatic increase in the predicted crack growth rate [62].

The fluid can also be forced into the crack by the load and pushes the crack faces apart [63]. This way, the contact pressure can be transmitted to the crack's mouth (hydraulic pressure mechanism).
The presence of fluid causes the friction in the wheel-rail contact to decrease. This can result in less wear and cracks can be freer to propagate.

Figure 4.20 Influence of fluid on crack propagation. (a) Longitudinal force on the inner wheel ($F_{x,wheel}$) causes fluid entrapment inside a crack on the wheel. (b) Longitudinal force on the outer rail ($F_{x,rail}$) causes fluid entrapment inside a crack on the rail, from the present work [60].

4.5 Surface RCF prediction models

4.5.1 Shear forces

A prediction model for surface-initiated RCF on railway wheels was developed in [64], [65] and is based on the shakedown theory [38], [42]. The probability of RCF depends upon the level of pressure and creep forces in the contact. A shakedown limit is defined which is a function of the maximum contact pressure ($p_0$ [N/m$^2$]) divided by the material yield stress in shear ($k$ [N/m$^2$]) and the utilized friction coefficient ($\mu$ [-]). If the combination of pressure and creep forces exceeds this limit, surface cracking will occur. The shakedown diagram is often used to compare the contact conditions with the shakedown limit, see Figure 4.21.

In [64], [65], the utilized friction coefficient was defined as the quotient between the tangential resultant creep force ($F_T$ [N]) and the normal force ($F_z$ [N]):

$$\mu = \frac{F_T}{F_z} = \frac{\sqrt{F_x^2 + F_y^2}}{F_z},$$

(4.28)

where $F_x$ and $F_y$ are respectively the longitudinal and lateral creep forces in [N]. The equation for the boundary curve (BC in Figure 4.21) was defined as:

$$\nu = \frac{p_0}{k} = \frac{1}{\mu},$$

(4.29)

A surface fatigue index ($FI_{surf}$ [-]) was defined which can be seen as an indication of RCF initiation:

$$FI_{surf} = \mu - \frac{1}{\nu} = \mu - k \cdot \frac{1}{p_0},$$

(4.30)
The fatigue index \( (FI) \) depends on to what extent the shakedown limit has been exceeded. The \( FI \) is the horizontal distance in the shakedown diagram between the working point of the contact conditions and the shakedown limit. Damage is assumed to occur only for \( FI_{\text{surf}} > 0 \).

One disadvantage of this RCF prediction model is that the applied shakedown diagram is derived under conditions of full slip. A contact on the flange of a wheel is often in full slip whereas a contact on the tread is often in partial slip. For these contact conditions, the model might therefore be invalid. It was shown in Section 0 that for line contacts and for a fixed utilized friction coefficient, the shakedown limit for contacts in partial slip is lower than for contacts in full slip.

Another disadvantage of this model, which is mentioned in [66], is that the shakedown diagram does not explicitly take the creepage in the wheel-rail contact into account, although creepage can be an important variable in determining fatigue life. In addition, the model does not take the possible influence of wear into account, which can also be seen as a disadvantage. The model, however, can be applied in combination with a separate wear model.

### 4.5.2 Energy dissipation

Another predictor to determine the risk of RCF is the Whole Life Rail Model (WLRM) based on the energy dissipation in the wheel-rail contact [67], [68]. This model was primarily developed to predict RCF on rails but has also been used to develop a model which can
predict wear and rolling contact fatigue initiation on wheels [69]. The model combines the longitudinal and lateral creep forces ($T_x$ and $T_y$ [N]) with the corresponding creep ($\gamma_x$ and $\gamma_y$ [-]) to calculate the "wear number" ($T\gamma$):

$$T\gamma = (T_x \cdot \gamma_x + T_y \cdot \gamma_y).$$  \hspace{1cm} (4.31)

The fatigue damage depends on the value of the wear number, $T\gamma$ [N], according to the damage function in Figure 4.22. The "damage index" ($DI$) is shown in Figure 4.22, which is defined here as the proportion of the fatigue life of the material for that specific contact condition. For each contact position, the damage index can be summed and fatigue will occur when unity is reached for the total $DI$.

![Figure 4.22 Rail RCF damage function [66].](image)

The model also considers the influence of wear (negative slope in Figure 4.22). For high $T\gamma$ values ($T\gamma > 65$ N), wear begins to dominate and for $T\gamma > 175$ N the wear rate is even higher than the crack growth, resulting in no RCF damage at all. This is the region where the severe wear regime is assumed to commence. The wear rate is expected to increase faster than the crack growth for $T\gamma > 65$ N. According to [68], however, the wear rate as a function of the wear number at three different test sites increased at a constant rate with increasing wear number, see Figure 4.23. This contradicts the expected behaviour of the damage function.

The main disadvantage of this model is its empirical nature and need for recalibration for new applications. The damage function was tuned on the data from six sites. In addition, the contact pressure is not explicitly taken into account, which may influence the formation of RCF cracks [67]. Another disadvantage is that spin creepage is not explicitly included in the wear number, $T\gamma$. Especially for contact positions on the gauge corner of the rail, where spin creepage can be high and cracks often occur, it can make a difference.
4.5.3 **Crack growth modelling**

There are different phases in the life of a fatigue crack initiated at the surface, see Figure 4.24 [70], [71]. After the initiation of a crack, the crack growth is driven by ratchetting in the plastically deformed layer. As the crack becomes longer and deeper, the crack growth is driven by the stress field due to the repeated loading and is believed to be greatly influenced by fluid entering the crack. Ultimately, the crack might grow downwards and growth is then driven by bending stresses in the rail, producing tensile stresses in the railhead. Fast fracture can occur when the crack reaches a critical length. Not all cracks grow downwards into the rail; most cracks actually change their direction towards the rail surface. Whether a crack grows downwards or upwards is believed to be influenced by the tension residual stresses and the crack length [72]. If both are sufficient to overcome the mode I (tensile opening) threshold, the crack will probably grow downwards. This theory is supported by the fact that rail fracture often occurs during, or shortly after, the winter.

There are basically two different approaches to modelling crack growth due to surface initiated rolling contact fatigue. The first approach is a "fracture mechanics" approach where the growth of the crack is explicitly modelled. The second approach is a more "fatigue damage" approach where the cracks are not explicitly included. The fatigue is directly related to the operational loading. The fracture mechanics approach will be discussed first.
In fracture mechanics, three different modes of crack propagation (fracture) can be defined which are illustrated in Figure 4.25. Mode I is the tensile opening mode, where the applied loading pulls apart the crack faces. Mode II is the in-plane sliding mode (faces are sheared backwards and forwards) and mode III is the tearing/anti-plane shear mode (faces are sheared sideways) [73]. Since the load in a wheel-rail contact is high, the compressive forces will close the crack and mode I crack propagation would not be possible. Due to the fluid which enters the crack, however, mode I crack propagation does take place. During the passage of a wheel, a mode I crack opening failure mechanism will dominate at some stages, whereas mode II or III will dominate at other stages. Which mode is dominant also depends on the size of the crack.

A crack growing in the rails is assumed to undergo mode I (opening) and/or mode II (shearing), but not mode III (torsion) according to [61]. Cracks propagate mostly at an angle of 15-30° to the surface. When the crack reaches a critical length or depth, it may branch up...
towards the surface so that shelling of the material takes place. However, the crack may also branch down and propagate at a steeper angle to the surface, causing the rail to fracture.

According to Paris’s crack growth law, the growth of a crack under cyclic loading is as follows [73]:

\[
\frac{da}{dN} = C\Delta K^m,
\]

(4.32)

where \( da/dN \) is the change in the length of the fatigue crack per load cycle (\( a \) is the crack length and \( N \) is the number of fatigue cycles), \( C \) and \( m \) are material constants, and \( \Delta K \) is the stress intensity factor (SIF) range defined as:

\[
\Delta K = K_{max} - K_{min}.
\]

(4.33)

\( K_{max} \) and \( K_{min} \) are the maximum and minimum stress intensity factors, corresponding to the maximum and minimum load (or stress).

Stress intensity factors for a crack propagating in a rail have been determined by [62] and are used for a crack growth law. Since both mode I and II stress intensities are predicted to act on the crack during wheel passage of a contact, both must be considered in the crack growth law:

\[
\frac{da}{dN} = 0.000507(\Delta K_{eq}^{3.74} - 4^{3.74}),
\]

(4.34)

where

\[
\Delta K_{eq} = \sqrt{\Delta K_I^2 + \left[\frac{614}{507}\right]^{2.21} \Delta K_{II}^{3.74}}.
\]

(4.35)

The growth rate \( da/dN \) is given in nm per cycle, and the stress intensity factors are in MPa√m and are a function of the size of the crack, \( a \).

There are different input parameters to the crack growth rate prediction model: the total surface traction (coefficient of friction), the crack face coefficient of friction, the normal pressure in the contact and the angle of the resultant tractive force. The results showed that the normal pressure has the greatest influence on the crack growth rate, but also that the crack face coefficient of friction also has a great influence. The main conclusion from this study was that when plotting the crack growth rate versus the crack size, the plot first rises for small cracks and then falls as the crack grows. This is because when the size of the crack in a rail increases, the surface load is taken away from the crack (longer distance to the crack tip).

The model predicted crack development quite successfully for most severely cracked locations, but not for the locations where the damage was less severe [68]. The influence of fluid on the SIFs was also determined in [63] (BEM) and [74] (FEM).
The fatigue damage approach [75] is only valid for small cracks, since it ignores the fact that crack growth rate is influenced by crack size. In fatigue, a relationship (Basquin) exists between the applied stress amplitude $\sigma_a$ and the number of load reversals to failure $N_f$ [73]:

$$\sigma_a = \alpha (N_f)^\beta,$$

where $\alpha$ is the fatigue strength coefficient and $\beta$ is known as the fatigue strength exponent or Basquin exponent. Both $\alpha$ and $\beta$ are material parameters. "Failure" can be assumed to be a certain crack length (or crack initiation).

These material parameters ($\alpha$ and $\beta$) for crack initiation have been determined by curve fitting test results [76], where the stress magnitude is the $F I$:

$$FI = 1.78 N_f^{-0.25} \quad \forall FI > 0. \quad (4.37)$$

The fatigue damage for one single load cycle (wheel revolution $i$ or, for the rail, wheel passage $i$) for stress amplitude $\sigma_a$, is as follows:

$$D_i = \frac{1}{N_f} \quad (4.38)$$

The damage for one cycle can thus also be written as:

$$\sigma_a = \alpha (D_i)^{-\beta} \Rightarrow D_i = \left(\frac{\sigma_a}{\alpha}\right)^{-\frac{1}{\beta}} = \alpha' \sigma_a^{\beta'} = \frac{FI^4}{10}. \quad (4.39)$$

The magnitude of the stresses, however, varies between the wheel revolutions/passages, since the contact conditions are different (different wheel/rail shape, speed, adhesion conditions, etc.). A fatigue damage accumulation rule, the Palmgren-Miner rule is therefore needed:

$$D = \sum_{i=1}^{k} D_i, \quad (4.40)$$

which gives the total damage after $k$ load cycles. $D=1$ corresponds to "failure" and is assumed to be a certain crack length. In Equation 3-41 above, it was defined as visible macroscopic cracks, see [76].
Assuming that a certain magnitude of fatigue damage corresponds to a certain crack length, the crack length can then be predicted. It is assumed that there is a fixed ratio $r$ between the surface length of the crack, $\rho$, and the crack length, $L$, cf. Figure 4.27:

$$ r = \frac{\rho}{L} \quad (4.41) $$

So:

$$ \frac{\rho}{\rho_f} = \frac{D}{D_f} \quad (4.42) $$

where $\rho$ is an arbitrary crack length at the surface, $D$ the corresponding fatigue damage and $\rho_f$ and $D_f$ a predefined critical crack length and fatigue damage, respectively.

![Figure 4.27](Diagram of a surface-initiated RCF crack with mouth width $\rho$, length $L$, angle $\theta$ and depth $h$ [75].)

The actual crack depth below the surface ($h$) can now be calculated for a damage level $D$:

$$ h = L \cdot \sin \theta = \frac{\rho}{r} \sin \theta = \frac{D \rho_f}{D_f r} \cdot \sin \theta. \quad (4.43) $$

It should be noted that only the conditions where $FI > 0$ are considered. The yield limit has thus been exceeded for these stresses. Using low-cycle fatigue instead of high-cycle fatigue would therefore be more appropriate but on the other hand computationally much more demanding. The contribution of the lower stress levels is much smaller than the contribution of the stress levels above the shakedown limit ($FI > 0$). The lower stresses are therefore ignored in the calculation of the total fatigue damage $D$.

### 4.6 Subsurface RCF prediction model

The main focus of this PhD work is on surface-initiated cracks, but for the sake of completeness the subsurface-initiated cracks for wheels will be discussed briefly.

As mentioned before, fatigue cracks may initiate from the surface when the creep forces and the coefficient of friction are high. If there is no sign of macroscopic inclusions, fatigue may initiate at 3-5 mm below the surface due to high normal contact stresses. If there are macroscopic inclusions/defects present (about 1 mm in size), fatigue may initiate at the defect down to a depth of 10-25 mm. Stresses close to a defect are locally increased, which can result in local plastic deformations which give rise to residual stresses.

A subsurface material point on a wheel will be subjected to multiaxial loading during a revolution, which makes it complicated to analyze the subsurface fatigue. An equivalent stress measure has been used which represents the multiaxial stress field in such a way that material...
data from uniaxial testing can be used. This stress measure should therefore include the influence of the hydrostatic stress and the effect of the static shear stress should be eliminated. A stress cycle is naturally defined as one wheel revolution. In [77], the following criterion was used to determine if damage occurs at the material point considered:

$$\sigma_{eq} = \tau_a(t) + a_{Dv} \sigma_h(t) > \sigma_e,$$ (4.44)

where:

- $\sigma_{eq}$ is the equivalent stress,
- $\tau_a(t)$ is the time-dependent magnitude/amplitude of the local shear stress,
- $\sigma_h(t)$ is the hydrostatic stress (positive in tension),
- $\sigma_e$ is the fatigue limit of the material in pure shear,
- $a_{Dv}$ is a material parameter representing the influence of the hydrostatic stress.

In the criteria above, a shear stress "amplitude" is used in order to eliminate the influence of a mean shear stress. To determine the maximum equivalent stress is quite time-consuming, since it needs to be calculated for all possible shear planes containing the considered material point. In [65], an approximation of the subsurface equivalent stress has been made and a fatigue index has been determined for subsurface-initiated fatigue:

$$F_{I_{sub}} = \frac{F_z}{4\pi ab} \left(1 + \mu^2\right) + a_{Dv} \sigma_{h, res} \rho_{\sigma},$$ (4.45)

where $F_z$ is the normal force, $a$ and $b$ are the contact patch semi-axes, $\mu$ is the traction coefficient and $\sigma_{h, res}$ is the hydrostatic part of the residual stress. If the coefficient of friction becomes high (about 0.3), the location of maximum equivalent stress will shift to the surface. In order to take the risk of a material defect into account, the fatigue limit can be reduced using the following approximation [78]:

$$\frac{\sigma_w}{\sigma_e} = \left(\frac{d}{d_0}\right)^{-1/6},$$ (4.46)

where $\sigma_w$ is the reduced fatigue limit of the material which contains a defect with diameter $d$. Further, $d_0$ will be the maximum size of a defect for which the fatigue limit will not be affected.

### 4.7 RCF prediction models – conclusions

As mentioned earlier, all RCF prediction models have their advantages and disadvantages. The model based on the shear force ($F_I$ in the shakedown map), does not explicitly take the creepage into account whereas the model based on the energy dissipation ($F_I \gamma$) does. This difference between the two models can be seen when the adhesion conditions are poor (wet track). The energy dissipation model predicts a higher indication of RCF than the shear force.
model under poor adhesion conditions, which was shown in the present work [60]. This is due to the limitation of the creepage influence in the shear force model. Since the utilized friction coefficient ($\mu = F_r / F_z$) is limited by the coefficient of friction, the creepage does not have any influence on the amount of damage after reaching this limit (full slip). This effect is shown in Figure 4.28, where the creepage is plotted against the utilized friction coefficient for different contact conditions in simulations.

![Figure 4.28](image-url) Utilized friction coefficient against total creepage on inner wheels in curves, for all contact positions from all simulations. From the present work [60].

It can therefore be concluded that the shear force model might be limited, since the probability of fatigue becomes independent of the creepage after reaching the maximum utilized friction coefficient. The shear force model might underestimate the RCF prediction for high creepages. The effects of creepage on RCF are, however, not exactly known. Experiments in [79] showed that for small creepages and for water-lubricated contact, creepage did have effects on RCF life. For high creepages (> 5%), however, the RCF life was unaffected by creepage. The energy dissipation model might therefore overestimate the RCF prediction for high creepages. From development studies of the energy dissipation model [68], it was also concluded that the fatigue damage was reduced for higher contact patch energy values. This was not due to wear but due to another unknown mechanism.

The main disadvantage of the shear stress model, however, is that the wear is not included in the prediction of RCF. The shear stress model, on the other hand could be combined with a separate wear model (see Chapter 3 about wear models). Moreover, it is already necessary to have the wear prediction part separated from the RCF prediction part if, as here, the main goal is to predict the total life of a wheel and rail with regard to wear and RCF. The wear part is thus needed in any case. It can therefore be concluded that it is difficult to choose the best RCF model.

One major step for RCF prediction models would be to predict the actual crack size. Two different groups of models exist nowadays. The models in the first group were discussed above and are able to successfully predict an indication of RCF for many wheel-rail contact conditions. These models can easily be used in combination with vehicle dynamics simulations which give output (loads and creepages) for many different contact characteristics. For example, for a wheel which travels over a railway network containing
different curves, rail profiles and travelling at different speeds and under different adhesion conditions. These RCF prediction models, however, cannot predict the actual crack size. The RCF prediction models which are able to do so belong to the second group. In these models, the growth of a crack is explicitly modelled by FEM, but this is very time-consuming. These models are therefore limited to handling many different contact characteristics from vehicle dynamics simulations. Moreover, these models make use of fracture mechanics where an initial crack already exists.

A first step in predicting the lifetime of a wheel and rail with regard to wear and RCF could be to use an RCF prediction model from the first group in combination with a fatigue based model which relates the RCF damage parameters from the RCF model to an actual crack length or depth. This, however, would only be valid for the prediction of relatively small cracks. The RCF prediction tool will be less accurate but it will be fast, allowing it to be used in combination with vehicle dynamics simulations.

The "fatigue approach" described in Section 4.5.3 and in [53] was therefore chosen for the RCF prediction tool and will be further discussed in Chapter 6.
5 Interaction of wear and RCF

Strong competition exists between wear and surface-initiated RCF: wear can worsen the contact geometry between wheel and rail, which may accelerate crack growth, but at higher wear rates, RCF does not have the opportunity to develop further. Cracks can initiate but will be worn off due to the high wear rate and will not be able to propagate far beneath the surface. Care must therefore be taken when optimizing to reduce the wear since RCF can in that case become the dominant problem.

As mentioned before, fluid can significantly influence the crack propagation rate. Not only because it directly influences the crack growth, but also because it decreases the coefficient of friction, resulting in less wear and therefore larger cracks [58]. RCF damage on the rails and wheels often becomes more severe due to the seasonal changes in the weather. When it is warm and dry, the creep forces are relatively high due to a high coefficient of friction. This can result in many crack initiations. When a dry season is followed by a wet season, the initiated cracks can grow rapidly due to the naturally lubricated tracks [80]. But also when a lubrication device has been switched off for a while and when it is switched on again, the cracks that could initiate during the ‘dry’ period can propagate rapidly with grease contamination.

Due to this competition between wear and RCF, the probability of RCF has a maximum for a certain curve radius, like in Figure 5.1 [46]. For fairly sharp curves (R < 1500 m), the wear rate is higher, resulting in a lower probability of RCF. For larger curves (R > 1500 m), the wear rate is lower, but due to lower creep forces the probability of RCF is also lower. Where this maximum is located depends on the types of vehicles. If there are, for example, many vehicles with a relatively stiff suspension, the maximum would shift more towards a larger curve radius. This is because a vehicle with a stiffer suspension generates higher creep forces in a curve with the same curve radius. RCF can therefore already become a problem in larger curves. This maximum also depends on whether it concerns the rails or the wheels. This is since RCF on the rails is mainly caused by high creep forces on the outer rail in a curve whereas RCF on the wheels is mainly caused by high creep forces on the inner wheels in a curve. This is probably due to the influence of fluid (see Section 4.4). Since the creep forces on the outer rail, where the contact is often on the flange of the wheel, are higher than on the inner rail, the curve radius for the highest risk of RCF is therefore larger for the rails than for the wheels, which is shown in the present work [6].
The optimal situation is when the amount of material removal, a combination of natural and artificial wear, is just sufficient to control crack initiation and propagation ("the magic wear rate") [1].

It can therefore be concluded that in order to predict RCF on rails or wheels, it is necessary to predict both wear and crack growth together.

Figure 5.1 Rough distribution of RCF on the rails in the UK as a function of curve radius [46].
The present work: a wheel-rail life prediction tool

The main goal of this thesis is to develop a tool which can predict the total expected life of wheels and rails. Two RCF models were selected to be used together with one wear model. This chapter briefly describes how the models were adjusted and implemented in the prediction tool. Another goal of this work is to verify the prediction tool with measurements of both wear and RCF. How this verification was done will also be discussed in this chapter.

6.1 Wheel-rail life prediction tool

For the wear prediction part of the tool, Archard's wear model was applied. For the RCF prediction part, two different RCF models were applied. The first model, the stress index (SI) model, is based on the shakedown theory and mainly takes into account only the shear stresses in the wheel-rail contact which exceed the yield limit in shear $k_y$, according to:

$$\tau_{zx}(x,y)^2 + \tau_{zy}(x,y)^2 = k_y,$$

where $\tau_{zx}(x,y)$ and $\tau_{zy}(x,y)$ are the shear stresses in each element of the contact ellipse.

The second RCF model, the energy index (EI) model, is based on the energy dissipation in the wheel-rail contact and in addition to the shear stresses also takes the creepage $\gamma$ and spin $\varphi$ into account:

$$EI(x,y) = \tau_{xx}(x,y)(\gamma_x - (\varphi \cdot y)) + \tau_{xy}(x,y) \cdot (\gamma_x + (\varphi \cdot x)).$$

Both RCF models have a threshold; only above this threshold does damage occur. For the SI model, damage only occurs for resulting shear stresses above the yield limit in shear. It was, however, seen in the present study [81] that the stresses below this limit also contributed to the damage. The EI model also has a threshold, which is determined by calculating the wear number $T\gamma$. Only if the wear number is above the threshold value of 15 N does damage occur. Compared to the threshold value of the SI model, it is much lower in the EI model.

Both RCF models were used to determine the stress magnitude in the fatigue model:

$$c_p = \sum_{i=1}^{N} \frac{1}{\alpha' (\sigma_a)^{\beta'}}$$

where $c_p$ is the predicted crack length or depth [mm], $i$ is a wheel passage (for rails) or wheel revolution (for wheels), $N$ is the number of load cycles and $\alpha'$ and $\beta'$ are material parameters.

The predicted crack size in Eq. (4.11), however, is the crack size without the influence of wear. Hence, to determine the actual size of the crack as it would appear in reality, the amount of wear needs to be subtracted:

$$c_{pi} = c_p - w, \quad (6.4)$$
where $c_{pi}$ is the crack size where the effect of wear $w$ is taken into account.

In the life prediction tool, the wear and RCF prediction models are used in combination with vehicle dynamics simulations. What the interaction between the vehicle dynamics simulations and the wear/RCF model looks like is shown in Figure 6.1. The rail life prediction model and the wheel life prediction model are regarded as separate tools and both consist of two parts: a pre-processor and a wear step loop. The pre-processor is needed to obtain a simulation set which represents the reality as efficiently as possible. As few as possible simulations in the simulation set therefore need to be determined. The wear part of the chosen methodology of the life prediction tool was developed in [15].

![Figure 6.1 Flow chart of the present wheel/rail life prediction tool.](image_url)

For the rail life prediction, a curve can be selected with initial new or already worn rail profiles. The simulation set must contain information about the geometry of the curve (radius, cant, transition curve, and length), the contact conditions (lubrication device), material parameters (hardness, elastic modulus and yield stress in shear), and also track irregularities (imperfections in track positioning). The track irregularities are really important since the difference in wear throughout a curve is often due to track irregularities. Moreover, a correlation may exist between track irregularities and the locations of severe RCF damage in a curve. The data of the vehicles running through the curve is also important. Information is
needed about the vehicle characteristics of each vehicle (passenger and freight), which wheel profiles they have (new and worn), at what speed they are running and how often.

For the **wheel life prediction**, a vehicle can be selected with initial new or worn wheel profiles. This vehicle is running on a specific network, so in the simulation set all the information about this network is needed; the geometry of all the curves the vehicle runs through (radius, cant, transition curve, length), the contact conditions (which curves are lubricated), material parameters (hardness, elastic modulus and yield stress in shear), and the track irregularities of all the curves. To limit the amount of simulations in the simulation set, only a few curves can be defined which represent the total network. Also, only a few worn rail profiles and a few track irregularity classes can be chosen to represent the total network. How fast the vehicle runs through each curve and how often should also be included in the simulation set.

Once the pre-processing is finished, the wear step loop will begin. It is called a wear step loop here, since the wheel/rail profile should be updated for each step based on the calculated wear. This is because the shape of a wheel/rail profile influences the vehicle-track simulations whereas cracks in a wheel/rail profile do not. Naturally, the crack length is needed to determine the crack growth in the next "wear" step, so the crack length from a wear step is used as input for the crack growth calculation. The crack growth rate, however, is considered to be constant and the size of the crack is therefore not taken into account here.

Wear of the wheel/rail profile influences the crack growth directly, which is why the calculated material removal is used as input to the crack growth calculation. If there are many cracks present in a wheel/rail and they are growing towards the surface, spalling of the material can occur which can be considered wear debris. This interaction, however, is not taken into account here.

The wear step loop continues until a certain distance travelled by a vehicle has been reached for the wheel or a certain number of axle passages or gross tons passages has been reached for the rail.

Comparison of the results with actual crack and wear measurements will show whether the proposed method is successful.

### 6.2 Verification wheel-rail life prediction tool

To verify the prediction models, wheel and rail profile measurements are needed to determine the wear rate of the profiles. The fatigue damage to the same wheel/rail profiles also needs to be monitored by visual inspections. For this purpose, the wheels of three units of the newest Stockholm commuter train and two units of the older train were selected here to be measured and inspected for damage; both wear and RCF. At the same time, different curves on the Stockholm commuter network were selected for damage inspections only, since the wear rate can be obtained from yearly rail profile measurements with a measuring train. It is mainly the commuter train which runs through the selected curves.
6.2.1 Wheel measurements

The older of the two vehicle types which run on the Stockholm commuter network was replaced during the measuring period by the newest. The running distance of the older vehicles was therefore insufficient to be used in this study and only the measurements of the newer vehicle units will therefore be discussed.

The vehicle was put into service in 2005. Each vehicle consists of six cars and seven bogies: two standard motor bogies (1 and 7), four Jacobs motor bogies (2, 4-6) and one Jacobs trailing bogie (3), see Figure 6.2. The axle load of an empty vehicle is approximately between 130 and 150 kN. The axle load is different depending on the axle position in the vehicle. When the measurements began in 2011, most of the axles of the three selected units had recently been renewed (including wheels).

The wheel profiles of all bogies in the three units were measured on each measuring occasion, except for the wheels of bogie 4. The RCF inspections were made only for the wheels of bogies 1, 2 and 3. An overview of the wheel profile measurements and RCF inspections for one unit is also shown in Figure 6.2. The wheels of the selected three units were measured 9, 12 and 6 times respectively within a period of 15 months (November 2011-February 2013), which corresponds to approximately every 22,500 km. The maximum total running distance was approximately 240,000 km. The wheels of two extra units were also measured, but only once just before re-profiling of the wheels. The wheels on these units had run approximately 134,000 and 110,000 km.

Figure 6.2 Overview of the wheel profile measurements and RCF inspections for one unit. For the green filled wheels, the profiles were measured and inspected for RCF, red dashed wheels were not measured/inspected and for the green solid lined wheels, the profiles were only measured. From the present study [56].

Wheel profile measurements

The wheel profile measurements were made with a MiniProf measuring device [82]. From the profile measurements it can be concluded that there is much scatter. This can be seen in Figure 6.3, where the scalar wear measures flange thickness ($t_f$), flange height ($h_f$), flange slope $q_r$ and worn-off area ($\Delta A$) are plotted against the running distance for all measured wheels of all three units. The wheels with the thickest flanges at around 200 kkm are typically connected to wheels with relatively thin flanges due to some asymmetry.
The main difference in wear between the wheels of the three bogie types was that the wheels of the trailing bogie showed less wear compared to the wheels of the motor bogies. This can be concluded from Figure 6.4, where the wear is plotted against the running distance for all three bogie types. It can also be concluded that there was an insignificant difference in wear between the wheels of the two motor bogies.
Figure 6.4 Development of the average worn-off area for all the measured wheels of all the three units. The wear is shown for each bogie type separately.

**RCF inspections**

For the RCF inspections, the length and orientation of the cracks on the wheel surface were measured. The distance from the back of the flange to the area where the cracks were located was also measured. A magnetic ruler was placed on the wheels and pictures were taken, see Figure 6.5.

From the measurements, no difference was found between the size and orientation of the cracks on the powered wheels or on the trailing wheels. Occasionally, however, the wheels showed longitudinally orientated cracks (in the rolling direction) due to high lateral creep forces and the wheels of the trailing bogies were more sensitive for this.

The measurements also showed that there was no consistent difference in RCF damage between the wheels within a bogie.

The area on the wheel where most of the RCF damage was found was between 80 and 115 mm from the back of the flange (10-45 mm from running circle towards treadside). Sometimes, the RCF damage also occurred closer to the flange, in the flange root between 40-60 mm from the back of the flange (10-30 mm from running circle).
Figure 6.5 Picture of the cracks on a wheel (axle 1, left side) with a running distance of approximately (a) 94,000 km and (b) 200,000 km. Flange side is on the right. Written on the wheel are the unit number, position of the wheel in the vehicle (1L) and the position of the cracks measured from the rear side of the flange.

6.2.2 Rail measurements

The curves selected for measurements on the Stockholm commuter network are all located in the track section between Flemingsberg and Södertälje. Six different locations were chosen and are shown in Table 2. For most of the locations, the curves in both directions (running north or south) were measured, allowing curves with about the same radius and environmental conditions to be compared. Each curve was marked with paint on the sleepers with approximately ten sections.

For the RCF inspections, the lengths of the cracks were measured together with the angles, lateral positions from the gauge corner and the bandwidths. For several sections in the curve, a picture was taken depending on if there was something of interest to see (Figure 4.14 (b)).

It could be concluded from the RCF inspections that for all curves, the cracks were distributed quite evenly throughout the curve. Cracks occurred on the high rail in all curves, also in the large radius curve $R = 2950$ m.

It was noticed during the inspections that the two curves at one selected location showed a large difference in RCF damage. This was in another track section on the commuter network.
Table 2 Overview of selected curves for RCF inspections between Flemingsberg-Södertälje.

<table>
<thead>
<tr>
<th>Location number</th>
<th>Curve number</th>
<th>Length [m]</th>
<th>Curve radius [m]</th>
<th>Track ID North/South</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>267</td>
<td>402</td>
<td>N</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>-</td>
<td>400</td>
<td>N</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>113</td>
<td>-577</td>
<td>S</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>104</td>
<td>-571</td>
<td>N</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>153</td>
<td>-888</td>
<td>S</td>
</tr>
<tr>
<td>3</td>
<td>6</td>
<td>148</td>
<td>-870</td>
<td>N</td>
</tr>
<tr>
<td>4</td>
<td>7</td>
<td>470</td>
<td>1578</td>
<td>S</td>
</tr>
<tr>
<td>4</td>
<td>8</td>
<td>470</td>
<td>-1575</td>
<td>N</td>
</tr>
<tr>
<td>5</td>
<td>9</td>
<td>200</td>
<td>2012</td>
<td>S</td>
</tr>
<tr>
<td>5</td>
<td>10</td>
<td>100</td>
<td>-1504</td>
<td>N</td>
</tr>
<tr>
<td>6</td>
<td>11</td>
<td>126</td>
<td>-2315</td>
<td>S</td>
</tr>
<tr>
<td>6</td>
<td>12</td>
<td>63</td>
<td>-2950</td>
<td>N</td>
</tr>
</tbody>
</table>

Both curves have a radius of around 600 m, so the conditions were relatively identical. The only difference was that the curve with the high RCF damage was lubricated and the other curve with less damage was non-lubricated. This difference in RCF damage is shown in Figure 6.6. The lubrication probably entered the cracks, causing the cracks to grow and pieces shell off due to fluid entrapment.

![Figure 6.6 RCF damage in two 600 m radius curves at the same location. The top picture shows the high rail in the lubricated curve and the bottom picture the high rail in the non-lubricated curve.](image)
6.3 Verification of vehicle model

A model of the commuter vehicle above, which was selected for the measurements, was needed for the vehicle dynamics simulations to enable the stresses and creepage to be calculated and used as input to the prediction tool. The model of the vehicle was built up in the multibody simulation (MBS) tool Gensys [83]. The model consists of only four bogies and three car bodies to save simulation time. All the different bogie types, the standard motor bogie, the Jacobs motor bogie and the Jacobs trailing bogie, were included in the vehicle model.

In order to ensure that the vehicle model in Gensys generates reliable results, the model needed to be verified. This was done by using on-train tests from 2004 performed by Interfleet Technology [84]. During these tests, the $Q$-forces (vertical), $Y$-forces (lateral), secondary spring deflections, acceleration in the car body and displacements of the axle guide bushing were measured. For the verification of the vehicle model, seven curves were chosen varying in curve radius from $R = 300$-1300 m. The track irregularities used for the simulations were from the same sections where the curves were located, but measured one year later (2005). The rail profiles were also from measurements made one year later. Only one measured rail profile was used per curve. The wheel profiles in the model were from the measured wheels from the test train. It could be concluded from the verification that there is a reasonably good agreement between the simulation results and the measurements. This can be seen in Figure 6.7 for the $Q$-forces and $Y$-forces for a track section with four following curves with a radius of: $R = -572$ m (left-hand), $R = 600$ m (right-hand), $R = -611$ m and $R = -572$ m. The speed was approximately 120 km/h.
Figure 6.7 Simulation (S) versus measurements (M) for (a) the $Q$-forces and (b) the $Y$-forces of axle 1 in four following (R=600 m) curves. 1= left side and 2= right side in direction of travelling.
7 Summary of appended papers

7.1 Paper A

In this paper, two existing RCF prediction models from the literature were extended. The first model is based on the shakedown theory and the fatigue damage was calculated locally by discretization of the contact patch in 50x50 elements.

The second model is based on the energy dissipation in the wheel-rail contact and wear is included in this model. As an extension, spin was included in this RCF model. Both models were compared in a parametric study and the results show that both models in general give similar results. Under poor adhesion conditions, however, the models behave differently. The energy dissipation model predicts more damage in this case than the shakedown model. This is because the creepage term is included in the energy dissipation model but not in the shakedown model. Since it is uncertain what the influence of creepage is on RCF life, it is not clear which model would be the better choice for the wheel-rail life prediction tool.

The simulation results also show that the adjustments in the extended models have a significant effect on the RCF damage. By calculating the fatigue damage in the shakedown model locally for each element in the contact area, instead of globally over the whole contact area, the damage increases. Including spin creepage in the energy dissipation model significantly influences the amount of RCF damage. But this is only the case for contact points on the flange of the wheel and the gauge corner of the rail, since spin is only high here. Spin is much lower on the inner wheel/rail in a curve.

7.2 Paper B

For the verification of the wheel-rail life prediction tool, two reference vehicles were measured. Both vehicles run on the Stockholm commuter network. The first vehicle is a traditional train with two two-axled bogies under each car body and is being replaced by the second vehicle, which has both traditional bogies and Jacobs bogies. Vehicle dynamics simulations were performed with both vehicle models to study their performance with respect to wear and RCF. Two different RCF prediction models were used, which were also used in Paper A. The RCF model based on the shakedown theory, however, was altered by taking only the shear stresses in the wheel-rail contact into account and excluding the contact pressure.

Two wear models were also selected for this study. The first model is based on Archard’s wear model and was locally applied to a discretized contact patch. The second model is based on the energy dissipation in the wheel-rail contact according to Pearce and Sherratt and was applied globally.
The effect of worn wheel and rail profiles, track irregularities and the coefficient of friction were studied in this paper.

The results show in general that both RCF prediction models predict less damage for the older vehicle due to the better steering performance of this vehicle.

It can also be concluded that both wear models predict more wear for the newer vehicle than for the older vehicle, but that this difference was small for the Pearce and Sherratt method and much larger for the Archard's method. This difference in wear is caused by the large influence of the sliding velocity on the wear coefficient in the wear map which is used in Archard's wear model.

Since the wheelbase of the Jacobs bogie is larger than that of the traditional bogie (in the newest vehicle), all damage models predict slightly more wear and RCF for the Jacobs bogies. Under poor adhesion conditions (lower coefficient of friction), the RCF models behave as shown in Paper A. Thus, the conclusion remains: the energy dissipation model predicts more damage under poor adhesion conditions.

7.3 Paper C

The RCF prediction models used in Paper A and B are only capable of predicting a certain indication of RCF, but not an actual crack length or depth. A fatigue approach (developed by others) was therefore used in combination with the two RCF models: the stress index ($SI$) model and the energy index ($EI$) model. The crack propagation model is a function of the stress magnitude in the wheel-rail contact, which is calculated by the two RCF models, and two unknown material parameters. The goal of the study in Paper C was therefore to calibrate these unknown parameters in the model against crack measurements. For this purpose, available crack depth and surface crack length measurements in a curve on the Dutch railways were used. Two different UIC54 rail types were installed in the curve: the softer 260Mn-grade rail and the head-hardened MHH-grade rail.

The energy dissipation RCF model was altered by disconnecting it from the wear function, so wear was no longer included. Also, the creep forces in the model were replaced by the shear stresses and the model was given a threshold which came from the original energy dissipation model.

Three different vehicle types were modelled for the vehicle dynamics simulations. These vehicles, two passenger trains and one freight train, accounted for 69% of the traffic through this curve.

It was found that also taking into account the shear stresses below the yield limit in shear (60%) improved the calibration of the parameters.

The unknown parameters were found successfully for both RCF models ($SI$ and $EI$), for both rail types (260Mn and MHH), and for the crack depth and surface crack length. A sensitivity
study, however, showed that one of the parameters was too sensitive and the model would therefore probably not work for other applications with this parameter.

It was concluded that both RCF models performed equally well. It should be kept in mind that the proposed prediction model is (probably) only valid for small cracks.

7.4 Paper D

The crack propagation model developed in Paper C in combination with a wear model needed to be verified. For the verification of the prediction tool, wheel measurements of the Stockholm commuter train were performed. Three units of the newest train were selected and both the wear and RCF development of the wheels were measured over a period of 15 months. The wheel measurements showed no significant difference in RCF damage between the bogies. The wear rate, however, was slightly lower on the wheels of the trailing bogie. All the measured wheels showed very little flange wear compared to the tread wear. The wheel profile measurements showed considerable scatter.

The vehicle had already been modelled in Paper B and could also be used in this study for the vehicle dynamics simulations needed as input to the wear and crack model. The wear was calculated using Archard's wear model, which was also studied in Paper B. The wear model was combined with the crack model from Paper C. The wear was considered to have a direct effect on the crack size. The wear was therefore subtracted from the "predicted" crack size to determine the actual size of the crack as it would appear in reality.

The results showed a reasonably good agreement between (some of) the measured and simulated wheel profiles. Unfortunately, it was concluded that one of the calibrated parameters from Paper C was not valid and the other parameter was not optimal for the current application. In comparison to the rails in Paper C, the wheels of the selected vehicles showed less damage. When the two parameters were calibrated against the application in this study, the crack model showed reasonably good results. Not only did the position of the predicted damage agree well with the measurements, the predicted crack development also showed good agreement with the measurements.

No significant difference was found between the two RCF models (SI and EI) for the prediction of damage to the tread. The EI model, however, seems to overestimate the damage to the flange.

To what extent the prediction tool is valid for other applications is unfortunately still questionable. The results, however, are promising and this can be considered a first step towards a wheel-rail life prediction tool.
8 Conclusions and future work

8.1 Conclusions

The main goal of this PhD study was to develop a tool for prediction of the total expected life of wheels and rails with respect to rolling contact fatigue (RCF) and wear since wear affects the crack growth. Initiated cracks may wear away and as a consequence stop growing. In addition, the wheel-rail profiles are altered by wear, resulting in changing contact conditions. RCF can also influence the wear rate, since small pieces of wheel or rail material can break out (spalling). This effect, however, was considered to be outside the scope of this study.

Another goal of this PhD study was to verify the proposed prediction tool against wear and crack measurements. For this purpose, extensive wheel measurements of the wear development and crack development were performed on Stockholm commuter trains. Also, some of the curves through which this train ran were selected and the high rails were inspected for crack development.

The main challenge for this PhD study was therefore to combine a wear prediction model with a crack prediction model. For this purpose, several already existing wear and RCF models were selected and studied. It could be concluded from these studies that spin creepage had an impact on RCF life and was therefore included in an extended version of an RCF model which is based on the energy dissipation. Including spin creepage is mainly of great importance for the RCF prediction of the outer rails in curves, since it can be high at this location.

It was shown in a study by Johnson [38] that for a fixed value of the traction coefficient \(Q/P\), a contact in partial slip will cause more damage than a contact in full slip. By calculating the shear stresses locally, in each element within the contact area, instead of globally over the whole contact area, this effect of partial slip would be taken into account. This method was therefore applied in the present study and found to give the expected result; the RCF life decreased.

It was also found from this study that the two selected RCF prediction models give mainly similar results as long as the adhesion conditions were "good" (sufficiently high coefficient of friction). The RCF model based on the energy dissipation, which has a strong dependence on creepage, showed much more damage in case of low friction.

It could be concluded from the wheel profile measurements that there was considerable scatter. This means that the contact conditions a wheel experienced varied greatly between wheels. This can be due for example to seasonal influences or asymmetry in the curves a wheel encounters. This makes it difficult to capture all the parameters that are needed as input to the vehicle dynamics simulations. The exact value of many of the parameters is often unknown, for example the coefficient of friction, the wear coefficient and the effect of fluid...
entrapment on the crack growth rate. It is therefore also difficult/impossible to predict the wheel shape or crack size exactly as it would be in reality.

It could be concluded from the wear simulations that due to the on-board flange lubrication, the wear coefficients for lubricated contact conditions had to be reduced by a factor between 1/10 and 1/800 compared to dry contact conditions. The predicted flange wear, however, was too low compared to the measurements, which means that this compensating factor would probably be lower. On the other hand, in the simulations the coefficient of friction and the scaling factor for lubricated contacts were only varied depending on if artificial lubrication was applied (track-side and on-board). The effect of seasonal variations was not taken into account.

For the crack prediction model, two unknown parameters needed to be calibrated. This calibration was done against crack measurements in rails. The results from that study showed that the crack prediction model has the potential to predict the actual crack size but that one of the calibrated parameters is probably too sensitive. This means that the accuracy of the model for other applications is uncertain.

Finally, the wheel life prediction tool was verified against the wheel measurements performed on the commuter train. From this verification, it could also be concluded that one calibrated parameter was not valid and that the second one was not optimal for this application. However, the tool again showed promising results once the parameters were calibrated for the wheel application. It seems that cracks in rails grow relatively faster than in wheels, which may explain why the calibrated parameters for the rail application do not match the parameters for wheels.

The verification results of the tool also showed that the RCF model based on the energy dissipation predicted too much damage to the flange where no damage was found in the measurements. It should, however, be mentioned that for the damage in the flange area, the negative effect of fluid entrapment is not present since the fluid is pressed out of initiated cracks due to the direction of the longitudinal creep force. The inspections of the wheels showed that occasionally, RCF damage did occur in the flange root of the wheel. Both RCF models, however, failed in predicting RCF damage in this area since the wear was higher than the crack length. The cracks in this area, however, did not occur on all measured wheels and the predicted wear in the flange root was relatively high compared to the measured wheels.

It can overall be concluded that the first step in developing a wheel-rail life prediction tool was made successfully.
8.2 Future work

In line with the conclusions, some recommendations for future work can be made:

- Since the model has only been verified for wheels so far, it should also be verified for rails.
- Other wear models can be applied and verified with measured wheels/rails.
- Determine wear maps for natural lubricated contact conditions (rain, snow, etc.)
- Include varying coefficients of friction and wear maps in the simulation set to take seasonal changes into account
- Since the present crack prediction tool is only valid to predict small cracks, how long the cracks can be, can be studied. And can the model be extended to model the crack growth for longer cracks which are in a different phase?
References


Appended papers