On wear in rolling/sliding contacts

RICKARD NILSSON

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Department of Machine Design
Royal Institute of Technology
SE-100 44 Stockholm

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Academic thesis, which with the approval of Kungliga Tekniska Högskolan, will be presented for public review in fulfilment of the requirements for a Doctorate of Engineering in Machine Design. The public review is held at Kungliga Tekniska Högskolan, Valhallavägen 79, Kollegiesalen at 09:30/2005-02-11.
ABSTRACT

The aim of this thesis is to increase the understanding of wear in rolling/sliding contacts such as the wheel-rail contact for railroads and the roller-washer contact for roller bearings.

The Stockholm commuter train network has been the subject of papers A and B in this thesis in which the wear and surface cracks on rails has been observed for a period of three years. By comparing the wear depth with the crack length, equilibrium between these two damage mechanisms was found for a lubricated rail. By using a lubricant with friction modifiers the stresses was low enough to prevent crack propagation; at the same time, the rail was hard enough to reduce the wear rate. This is probably the most favourable state in terms of rail maintenance cost.

Roller bearings subjected to lubricant borne particles have been the subject of papers C, D and E in this thesis. Particles in the lubricating oil can have a significant impact on the wear in lubricated contacts. Even at low concentration levels can self-generated particles cause significant wear. The here presented results shows that filtration during run-in can significantly reduce both the mass loss and the number of self generated particles. A series of experiments has been carried out to study the wear of roller bearings by ingested lubricant borne hard particles. The form of the worn profile and the length of wear scratches correspond closely to the sliding within the contact. A count of the number of wear scratches on the rolling element surface indicates that the contact concentrates particles. A novel wear model based on the observation of a single point on the contacting surface when a concentration of particles passes through it has been developed and the necessary data for the model has been determined from the experiments. Comparison of the simulation results with the experimental results shows good qualitative agreement for the form change of the washer surfaces.

**Keywords:** wear, rolling, sliding, contact, environment, wheel, rail, bearing.
PREFACE

This thesis was written at the Division of Machine Elements, Department of Machine Design, Royal Institute of Technology (KTH), Stockholm, Sweden.

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Stockholm, January 2005

Rickard Nilsson
APPENDED PAPERS AND DIVISION OF WORK

Paper A

Paper B

The work on surface cracks was mainly performed by Ulf Olofsson and the work on wear was mainly performed by Rickard Nilsson. Most of the writing was done by Ulf Olofsson, but Rickard Nilsson also contributed to the writing and editing of the text.

Paper C

Experimental work and writing equally divided between Rickard Nilsson and Ulf Olofsson. Krister Sundvall assisted during the preparation of the experimental work.

Paper D

Experimental work performed by Rickard Nilsson. Scanning electron microscopy and energy dispersive X-ray spectrometry analyzes equally performed by Rickard Nilsson and Fredrik Svahn. Most of the writing was done by Rickard Nilsson, Ulf Olofsson contributed.

Paper E
Nilsson Rickard, Dwyer-Joyce Rob and Olofsson Ulf, The abrasive wear of rolling bearings by lubricant borne particles. Submitted for publication.

Experimental work, analysis and evaluation performed by Rickard Nilsson. The abrasive wear model was developed by Rickard Nilsson. Most of the writing was done by Rickard Nilsson and Rob Dwyer-Joyce, Ulf Olofsson contributed.
CONTENTS

ABSTRACT ........................................................................................................ iii
PREFACE ............................................................................................................ v
APPENDED PAPERS AND DIVISION OF WORK ........................................ vii
1 INTRODUCTION ........................................................................................... 1
2 WEAR .............................................................................................................. 2
3 ROLLING/SLIDING CONTACTS ................................................................. 3
4 EXPERIMENTAL STUDIES ........................................................................ 5
  4.1 Wheel/rail .............................................................................................. 5
  4.2 Roller bearing ....................................................................................... 5
5 EXPERIMENTAL RESULTS ......................................................................... 6
  5.1 Contact situation ................................................................................... 6
  5.2 Environment .......................................................................................... 8
    5.2.1 Weather conditions ......................................................................... 8
    5.2.2 Lubrication .................................................................................... 9
    5.2.3 Contaminations ............................................................................. 9
  5.3 Material properties ............................................................................... 12
    5.3.1 Steel grade ..................................................................................... 12
    5.3.2 Surface hardness .......................................................................... 13
    5.3.3 Coating .......................................................................................... 14
  5.4 Surface cracks ....................................................................................... 15
6 ABRASIVE WEAR MODEL ......................................................................... 16
7 CONCLUDING REMARKS .......................................................................... 16
REFERENCES .................................................................................................. 19

APPENDED PAPERS:

Paper A  Rail wear development – measurements and evaluation
Paper B  Surface cracks and wear of rail: a full-scale test on a commuter train track
Paper C  Filtration and coating effects on self-generated particle wear in boundary lubricated roller bearings
Paper D  Relating contact conditions to abrasive wear
Paper E  The abrasive wear of rolling bearings by lubricant borne particles
1 INTRODUCTION

Tribology, the science and technology of friction, wear and lubrication, is an interdisciplinary subject. It can therefore be addressed from several different viewpoints. This thesis focuses on the wear of rolling/sliding contacts such as the wheel-rail contact for railroads and the roller-washer contact for roller bearings from a mechanical engineer’s viewpoint.

A comprehensive overview of the science of tribology is presented in the ASM handbook [1], while a closer examination of the material science field is given by Hutchings [2]. The mathematical modelling aspects of tribology, i.e. contact mechanics and fluid film lubrication, are presented by Johnson [3] and Dowson and Higginson [4]. An excellent historical overview of the field is presented in Dowson [5].

When two surfaces under load move relative to each other, wear will occur. Wear is often defined as damage to one or both surfaces, involving loss of material. Aspects of wear and other surface damage mechanisms in rolling/sliding contacts are presented in papers A to E in this thesis. In papers A and B the wear of the wheel-rail contact are studied and in papers C and D the wear of the washer-roller contact caused by lubricant borne particles are studied. In paper E a novel model is used to simulate the abrasive wear caused by lubricant borne particles.

Lubricant applications to the rolling/sliding contacts as well as surface coatings are used to reduce friction and damage due to wear. How applied lubrication influences the damage mechanisms as such wear and surface cracks is studied in papers A and B in this thesis. Also natural lubrication (humidity, rain etc.) influences the wear of rolling/sliding contacts as shown in paper A. How coatings can reduce wear in rolling/sliding contacts is presented in papers C and D.

The friction force can be defined as the resistance encountered by one body moving over another body. This definition covers both sliding and rolling bodies. Note that even pure rolling nearly always involves some sliding and that the two classes of motion are not mutually exclusive. Any substance between the contacting surfaces may affect the friction force. The contact conditions may cause the substance to be wiped away quickly and its effect will be minimal. On the other hand, surface films formed between interposed substances have a major effect on the frictional behaviour.

In contrast to other well-investigated machinery, such as roller bearings, the wheel-rail contact is an open system. It is exposed to dirt and particles and natural lubrication, such as high humidity, rain and leaves, all of which can seriously affect the contact conditions and the forces transmitted through the contact. In contrast, in roller bearings the roller-washer contacts are sealed. The steel rail meets a population of steel wheels from a number of different vehicles and the form of both the wheels and the rail can change due to wear. In contrast, a roller bearing meets the same rollers without any significant form change of the contacting bodies. However, a roller bearing can be more sensitive to small form changes and a drastic life reduction can be noted for a form change of 10 µm due to wear, see [6]. In contrast, for the wheel-rail contact a form change in the scale of centimetres can be measured (see paper A) and this wear-in of the wheel and rail contact is often considered to be positive for the life of the contact. If this
form change is too large the rail and the wheels have to be replaced or grind due to safety reason.

What one always should bear in mind when studying and using tribological data is that friction and wear are system parameters and not material parameters like modulus of elasticity or fracture toughness. This means that frictional and wear data taken from one system, like a roller bearing, cannot be directly applied to another system like the wheel-rail contact. This also highlights the need for special studies of the tribology of the wheel-rail contact and the tribology of roller bearings.

2 WEAR

Wear is the loss or displacement of material from a contacting surface. Material loss may be in the form of debris. Material displacement may occur by transfer of material from one surface to another by adhesion or by local plastic deformation. There are many different wear mechanisms that can occur between contacting bodies each of them producing different wear rates. The simplest classification of the different types of wear that produce different wear rates is “mild wear” and “severe wear”. Mild wear results in a smooth surface that often is smoother than the original surface. On the other hand, severe wear results in a rough surface that often is rougher than the original surface [7]. Mild wear is a form of wear characterised by the removal of materials in very small fragments. Mild wear is favourable in many cases for the wear life of the contact as it causes a smooth run-in of the contacting surfaces. However, in some cases it has been observed that it worsens the contact condition and the mild wear can change the form of the contacting surfaces in an unfavourable way [8]. Another wear process that results in a smooth surface is the oxidative wear processes characterised by the removal of the oxide layer on the contacting surfaces. In this case the contact temperature and asperity level influence the wear rate [9].

An observation that can be made on sliding wear is that an increase of the severity of loading (normal load, sliding velocity, or bulk temperature) leads at some stage to a sudden change in the wear rate (volume loss per sliding distance). The severe wear form is often associated with seizure. The transfer from mild acceptable wear to severe wear depends strongly on the surface topography. The loading capability of a sliding contact may be increased considerably by smoothing the surface [10]. Chemi-reacted boundary layers imposed by additives in the lubricant can improve the properties of lubricated contacting surfaces and reduce the risk of seizure [11].

In the wheel-rail contact three wear regimes are usually identified; mild, severe and catastrophic, see [12]. The wear regime named catastrophic wear leads to an extremely large wear rate and this wear regime is often considered unacceptable for the wheel-rail contact.

In roller bearings as well as other rolling/sliding contacts can lubricant borne particle contamination seriously reduce the life of the contact. The contamination can result from either generated or ingested particles. Generated particles are produced during the operation of the system. Ingested particles enter the system from the outside environment. During the run-in period of rolling and sliding components, particle generation can be very high even for a clean system with clean components, see Olofsson and Svedberg [13]. It is therefore important that filtration be very efficient
during this period. Particles can be divided into those that are harder than the contact surfaces and those that are not. Particles that are harder than the contact surfaces may cause wear, while those softer than the contact surfaces may cause indentation, see Dwyer-Joyce [14]. Abrasive wear is usually divided into two types: two-body and three-body abrasion. In two-body abrasion the particle embeds into the softer surface and scratches the harder surface. Compared to two-body abrasion, three-body abrasion is much more common and more complicated: see Kusano and Hutchings [15] and Fang et al. [16]. There are two movement patterns for the particles in three-body abrasion, rolling and sliding. When the particle rolls or tumbles through the contact a series of indentations is found, resulting in little wear damage. On the other hand, the particle can indent both surfaces, rotate until it reaches a force equilibrium condition, and then scratch both surfaces. However, in an experimental study by Fang et al. [17], the situation where both surfaces suffered scratching could not be detected experimentally.

Mild wear of rolling/sliding contacts has previously been modelled and simulated using the single point observation method and a generalised Archad’s wear law. This model was first proposed by Andersson and Eriksson [18] for gears and has later been successfully used to simulate the mild wear of other rolling and sliding contacts such as the cam-follower contact Hognell and Andersson [19], the roller-washer contact Olofsson et al. [20] and the wheel-rail contact Telliskivi and Olofsson [21].

3 ROLLING/SLIDING CONTACTS

The contact area between surfaces that moves relative each other can be divided in to zones with stick respective slip. Stick is the situation when there is no relative motion between the opposing surfaces. Slip, on the contrary, occurs when the surfaces moves relative each other. If only stick occurs in the entire contact area the whole contact is subjected to pure rolling. If only slip occurs the whole contact area is subjected to sliding. If both stick and slip occurs in the contact the contact is called a rolling/sliding contact.

As shown in Figure 1, the contact area between a wheel and rail can be divided into stick (no slip) and slip regions. Longitudinal creep and tangential (tractive) forces arise due to the slip that occurs in the trailing region of the contact patch. With increasing tractive force, the slip region increases and the stick region decreases, resulting in a rolling and sliding contact. When the tractive force reaches its saturation value, the stick region disappears, and the entire contact area is in a state of pure sliding. The maximum level of tractive force depends on the capability of the contact patch to absorb traction. This is expressed in the form of the friction coefficient, \( \mu \) (ratio of tractive force to normal load). Normally wheel/rail traction reaches a maximum at creep levels of 0.01 to 0.02.
Figure 1. Relationship between traction and creep in the wheel/rail contact, from Olofsson and Sundvall [22].

The traction/creep curve can be dramatically affected by the presence of a third body layer in the wheel/rail contact. This could be formed either by a substance applied to increase/decrease friction (friction modifier or lubricant) or by a naturally occurring substance acting to decrease friction (water or leaves etc.).

In wheel-rail contact, both rolling and sliding also occur in the contact zone due to the geometrical fit between wheel and rail. On straight track, the wheel tread is in contact with the rail head, but in curves the wheel flange may be in contact with the gauge corner of the rail. Due to the conicity of the wheel profile, flanging results in a large creep and also a large sliding motion in the contact.

In ball bearings as well as spherical roller bearings there is always sliding present in the contact due to the curved contact conditions (Halling [23] and Kellström [24]). In Figure 2 the curved contact is presented for a spherical roller thrust bearing. For an unskewed roller there will be at most two points along each contact where the sliding velocity is zero. At all other points along the contact, sliding is present. Under boundary lubricated conditions, such bearings can break down due to mild wear, see Olofsson [25].

Figure 2. Curved contact between roller and washer in a spherical roller thrust bearing.
4 EXPERIMENTAL STUDIES

Studies of wear and different aspects influencing the wear have been performed on two different systems; the wheel-rail contact and the washer-roller contact in spherical roller thrust bearings. For both systems the profiles of the contacting surfaces were measured, surface damages were studied and operational conditions monitored.

4.1 Wheel/rail

The Stockholm local network had been the subject of this study in which the wear of rail and wheel profiles had been studied, see [26], papers A and B. In this test case both lubricated and non-lubricated rails as well as seasonal variations were studied. In addition two different rail hardness’ (standard UIC 900A grade rail steel and the harder UIC 1100 grade rail steel) were studied in the same test curves. For the wheel-rail system the profiles of the contacting surfaces were measured and the area worn off could be calculated, see Figure 3. Also observations of surface cracks in the headcheck zone and surface hardness have been done in the Stockholm test case.

![Figure 3. Schematic description of wheel and rail wear.](image)

4.2 Roller bearing

Spherical roller thrust bearings, see Figure 4, have been used to study abrasive wear, see papers C, D and E. Both standard bearings, SKF 29412 E, and bearings with coated rollers, SKF 29412 EL5DA (NoWear™), were used.

![Figure 4. A spherical roller thrust bearing and schematic description of load and contact situation between washers and rollers.](image)
To study different aspects of abrasive wear both self generated and ingested abrasive particles have been used. During the tests with self generated particles different filtering strategies were used. The self generated particles were counted with an on-line particle counter. Samples of the contaminated oil which circulated through the contacting surfaces were also taken. These oil samples were later examined in a ferrographic analyser equipment.

The macro scale wear depth across the contacting surfaces was measured with a 2-dimensional profile measuring device. Mass loss for the different components was measured with a balance scale. To study abrasive wear on a micro scale scanning electron microscope (SEM), energy dispersive X-ray spectrometry (EDS) and an atomic force microscope (AFM) were used.

5 EXPERIMENTAL RESULTS

From the performed studies results regarding how wear in rolling/sliding contacts are influenced by the contact situation, the contact environment and material properties have been achieved. Also the interaction between wear and surface cracks of rails has been studied for different steel grades.

5.1 Contact situation

The contact situation in terms of pressure and sliding between surfaces strongly influences the wear. When the surfaces are worn the contact situation changes due to changed geometries. The changed geometries can lead to altered conditions regarding sliding and pressure distribution between the surfaces. This can affect the wear acting on the surfaces. For a railway wheel this can be seen in Figure 5. During the first 100,000 km of driving the wear rate at the flanges was rather high for both the powered and the trailing unit. After that the wheel profiles had been worn to such a shape that the flange wear almost stopped, the wear had led to a different contact situation.

![Figure 5. Wheel profile area worn off (from Nilsson [26]).](image-url)
It can also be seen that the tread wear for the powered unit is higher than for the trailing unit. This can be explained by the fact that the powered unit transmits tangential (tractive) forces and also is heavier, and hence generate higher contact pressure for a given contact situation, than a trailing unit.

The curve radius of the track has a strong influence on rail wear. The influence depends also strongly on the vehicles and their behaviour. In the Stockholm test case all vehicles were of the same type and passed over all the test sites with the same frequency. In this case the influence of curve radius can be clearly seen when comparing rail wear rate as function of curve radius.

![Wear rate versus curve radius](image)

**Figure 6. Wear rate for high rail as function of curve radius in the Stockholm test case (from paper A). MGT = mega gross tonne traffic.**

The rail wear rate seems to increase exponentially for decreasing curve radius, as shown in Figure 6. This is a function of increased creep due to geometrical reasons; the sharper curve the larger flange-contact area will be obtained. Sharper curves also lead to increased track guiding forces acting on the wheels, which lead to increased creep and hence increased wear. Higher track guiding forces also leads to higher contact pressure, resulting in higher wear rate.

Wear rate ratio for high rail:
(worn off area for site with new high rail/worn off area for site with worn high rail)⁻¹

![Wear rate ratio](image)

**Figure 7. Influence of initially new high rail profile compared to worn high rail profile on high rail wear (from paper A).**
The influence on rail wear from the contact geometry can also be seen in Figure 7. That figure shows that new rails have a higher wear rate than old rails that already have been run-in. In the actual test case was the wear rate approximately four times higher for new rails compared to rails that already had been worn, for UIC 900A grade rails. In this case is it obvious that the contact geometry had a significant influence on rail wear rate.

5.2 Environment

Environmental parameters surrounding the rolling/sliding contact can have a significant impact on the behaviour of the wear of the contacting surfaces. Some of these parameters are given and can not be influenced, like the different weather conditions a railway track is subjected to. Other parameters can be controlled from outside the rolling/sliding contact during operation of the system containing the rolling/sliding contact, for example supplied lubricant to the contact. Even other parameters, like the creation and behaviour of self generated wear debris depend mainly on the situation inside the contact.

5.2.1 Weather conditions

An analysis of the relationship between weather conditions and measured rail wear shows that the precipitation has a significant effect on rail wear, see Figure 8. Precipitation creates a natural lubricating layer of water in the wheel/rail contact. The water in the wheel/rail contact may also absorb energy, when transformed from solid or liquid phase to gaseous phase, and hence reduce the energy build-up in the contact surfaces, which could lead to higher wear rate.

![Figure 8. Influence on rail wear from average daily precipitation for a specific track site during different measuring periods (from paper A).](image)

In Figure 8 it can also be seen that low temperatures, below 0°C, also seem to contribute to decreased wear rate. This may be because condensation takes place on the rail surface, generating a thin film of water on the surface. Another reason may be that lower bulk temperature reduce the temperature in the contact and hence reduce the wear rate.
5.2.2 Lubrication
Track-side lubrication reduced the wear significantly, see Figure 9, and a lubrication benefit factor 9 for small radius curves (300 m) was measured. For 600 - 800 m radius curves the lubrication benefit factor varied between 2 to 4.

![Wear rate graph](image)

*Figure 9. Influence on rail wear from track side lubrication (from paper A).*

The effect of lubricating the rail varies at different times of the year. The lubricant is affected by weather conditions such as air temperature and solar radiation. An increase in the temperature of the rail, or rather the lubricant, seems to increase the rate of wear, see paper A. This may be because high temperatures cause the lubricant to become more liquefied and thus resulting in that it vanishes more easily from the wheel-rail contact. It can also be that the oil in the grease evaporates, which results in reduced effect of the lubricant.

5.2.3 Contaminations
Contaminations entering rolling/sliding contacts can severely damage the contact surfaces. Example of form change on housing washer surfaces caused by self-generated particle wear is shown in Figure 10. In that figure is the results of measured wear presented for three different tests; one test with standard rollers and continues filtering with a 3μm filter (test 1), one with standard rollers and no filtering (test 2) and one with coated rollers and no filtering (test 3). It is hard to see any significant wear on the washer surface from the bearing with coated rollers. On the other hand for the bearings with standard rollers and no filtering the wear is clear and significant. The wear is low or negligible at the zero sliding points, and high at the zones where sliding occurs. The results from the test with continuous filtering show that a small amount of wear could be measured; this was probably due to a mild wear caused by the rolling sliding contact. The shaft washers show results similar to the housing washers. For the rollers no form changes could be measured for standard rollers as well as coated rollers. It was found that the number of generated wear particles in the contact was significantly less for a bearing with coated rollers compared to a standard bearing.
Once the particle has entered the contact it can behave in different ways. Figure 11 shows SEM images of some embedded ingested diamond particles. Particles were found permanently embedded in the washer surfaces but not in the rollers. In Figure 11a there are two small embedded particles; they are probably either some small fragments in the abrasive mix, or some parts which have been fractured from a larger particle. The mechanism by which a particle remains embedded in the surface after it has ploughed a groove is not entirely clear. Adhesive forces are likely to be much lower than the tractive forces on a particle; but perhaps some of the ploughed material is mechanically locking the particle in place. Figure 11b shows the tip of a particle which is locked beneath the surface. Figure 11c shows an image of a particle at the end of the groove that it has just scratched.

![SEM images of embedded ingested diamond particles on the components in a spherical roller thrust bearing](image)

(a) (b) (c)

*Figure 11. SEM images of embedded ingested diamond particles on the components in a spherical roller thrust bearing. (a) on the house washer, scale bar 2 μm. (b) on the axle washer, scale bar 1 μm. (c) on the axle washer, scale bar 1 μm. From paper E.*

The scratch length is influenced by the sliding in the contact. The measured length of scratches created by ingested diamond particles is shown in Figure 12 for the axle washer, roller and house washer. One can clearly see that the length of the scratches reflect the contact situation. The length of the scratches continuously increases when moving from the two points of pure rolling to the inner respectively the outer side of
each component. At the pure rolling points there are no scratches or just indentations. Between the pure rolling points the length of the scratches increases again, and a local maximum is obtained approximately in the middle. It can also be seen that scratches created by the ingested diamond particles occur on all three bearing elements. It is therefore not simply the case that the particles stick on one surface and scratch the other surface.

![Graph showing length of scratches by component](image)

**Figure 12.** Measured length of scratches created by ingested diamond particles on the components in a spherical roller thrust bearing (from paper E).

Examples of scratches created by ingested diamond particles on a house washer surface and measured with an AFM can be seen in Figure 13.

![Three images showing scratches](image)

**Figure 13.** Scratches on a house washer surface created by ingested diamond particles. The distance between horizontal lines is 50 nm for the left and the middle image profile. For the right image profile is the distance between horizontal lines 200 nm. From paper E.
The cross sections of all the scratches are approximately V-shaped. Compared to the size of the diamond particles it was also found that the scratches were very shallow compared to the theoretically expected abrasive scratch depth, see paper E. By comparing the shape of the scratches on the house washer to the shape of the scratches on the roller, see paper E, it was also found that the ingested diamond particles were first pressed down in to the house washer by the roller. When pressed down deep enough, so that tractive forces from the roller can be resisted, the particle starts to scratch the roller, which is harder than the house washer.

5.3 Material properties

The properties of the contacting materials in terms of steel grade and surface treatment (for example coating) can have a significant impact on the behaviour of the wear of the contacting surfaces. Some material properties, like surface hardness, which may influence the wear rate, can also be affected by the contact situation.

5.3.1 Steel grade

For a given situation a higher steel grade usually reduces rail wear. This effect is shown in Figure 14 for two different high rails with steel grade UIC 900A respectively UIC 1100 within the same lubricated as well as a parallel non-lubricated 300 m radius curve. For the non-lubricated curve the ratio between rail wear rate for the 900A grade rail compared to that of the 1100 grade rail is approximately 2. This can be compared with the lubricant benefit factor that was approximately 9 in this curve, as can be seen when comparing the non-lubricated and lubricated cases. However, when Lewis and Olofsson [27] compared rail steel wear coefficients taken from laboratory tests run on twin disc and pin-on-disc machines as well as those derived from measurements taken in the field, they found that the introduction of more modern rail materials had reduced wear rates by up to an order of magnitude in the last 20 years.

![Figure 14. Rail area worn off for the high rail at four sites with and without track-side lubrication, and rails of both UIC 900A and UIC 1100 rail steel grade (from paper A).](image)
5.3.2 Surface hardness

When new the surface hardness of both wheels and rails can increase due to work hardening. The increase in surface hardness for wheels at a powered vehicle unit is shown in Figure 15.

![Graph showing wheel surface hardness vs. driven distance][1]

*Figure 15. Measured wheel surface hardness, powered X10 vehicle (from Nilsson [26]).*

The surface hardness of the wheels on the trailing unit did not increase in the same way, but tended to remain constant, see [26]. This is probably because the axle load and longitudinal forces were lower than for the powered unit.

Rails that had been in use for a longer period before the measurement started and hence had accumulated a substantial traffic load and adopted a worn rail profile showed no change in surface hardness; see [26]. The surface hardness of new rails, however, can change. Rails of UIC 900A grade steel in particular become harder at the surface. UIC 1100 grade rail steel does not change surface hardness significantly, see Figure 16.

![Graph showing surface hardness ratio for top of high rail][2]

*Figure 16. Surface hardness ratio for new rails compared to original worn high rails for top of high rail (from paper A).*
5.3.3 Coating

Coatings can significantly reduce the abrasive wear of roller bearings caused by self generated particles, see Figure 10. This may partly be because the coating prevents the creation of self generated particles. In paper C there were found that twice as many self-generated particles were generated when using a standard bearing compared with a coated bearing.

Also the abrasive wear created by hard ingested particles can be reduced by using bearings with coated rollers. The wear distribution across the components of a standard bearing tested with ingested diamond particles is shown in Figure 17.

![Graph of wear depth (μm) vs position on component (mm) for axle washer, roller, and house washer.]

Figure 17. Wear depth on the axle washer, roller and house washer in a standard bearing tested with ingested diamond particles (from paper D).

For the same test conditions for a coated bearing, the wear depth on the axle and the house washer was minimal across their entire profiles. The wear depth on the roller was, however, measurable and is shown in Figure 18.

![Graph of wear depth (μm) vs position on roller (mm).]

Figure 18. Wear depth of a coated roller in a bearing tested with ingested diamond particles (from paper D).
5.4 Surface cracks

Examination of cracks in the rail indicated that UIC 900A rail steel grade seems to be as sensitive to crack initiation as the UIC 1100 rail steel grade. The UIC 1100 grade rail is, however, more sensitive to crack propagation, see paper B.

The 90th percentile of the crack length per accumulated traffic load was compared with the wear depth in the headcheck zone per accumulated traffic load, see Figure 19. The results indicate that there can be equilibrium between crack length propagation and propagation of wear depth. For the lubricated UIC 1100 grade rail (site B) there was equilibrium between wear depth and crack length for the new rail. For the five-year-old rail, crack length per million-gross-tones (MGT) had decreased more than wear depth per MGT. Thus, site B had crack initiation but no crack propagation or crack propagation that was less than the propagation of wear.

By contrast, for the curve with un lubricated UIC 1100 grade rail (site D), older rail showed a longer crack length per MGT than new rail while the wear depth per MGT decreased. The stresses on the rail were high enough to cause crack propagation, and the wear rate was not high enough to wear the surface cracks away.

Comparing the two sites with un lubricated UIC 900A grade rail (sites C and E), one can observe a difference in both wear and crack propagation behaviour. At site E with radius 611 m, wear depth was smaller than crack length per MGT. For the old rail at site C with radius 303 m, the opposite was true. The wear depth here was greater than the crack length per MGT. At both test sites the rail profile had changed significantly due to wear in the rolling-sliding contact. It is possible that at site C the rail profile had been worn to a more favourable shape that was better adapted to the wheel profiles. This would lead to lower stresses and a lower risk of crack propagation, and might account for the difference in crack length per MGT at sites C and E.

![Figure 19. Wear depth and the 90th percentile of the crack length per MGT for different test sites (from paper B).](image-url)
6 ABRASIVE WEAR MODEL

Based on the experimental observations of abrasive wear, a model of the abrasive wear process has been developed. The model considers a single point on the contacting surface as a concentration of particles passes through it, see Figure 20. Parameters required to predict the material removal, such as particle entrainment, particle motion, and scratch geometry, have been determined by a combination of analytical and empirical approaches. The result from such simulation can be seen in Figure 21. Comparison of the simulation results with the experimental results shows good qualitative agreement for the form change of the washer surfaces.

\[ \text{time=0} \quad \rightarrow \quad \text{time=tc} \]

\[ \begin{array}{c}
\text{P} \quad v_x = \frac{v_1}{2ae} \\
\text{x=ae} \\
\end{array} \]

Figure 20. Relative sliding of two surfaces with a trapped particle. The xyz-coordinate system is fixed to the contact. The Point P moves with the velocity \( v_2 \) from the location \( x = ae \), at \( \text{time}=0 \), to the location \( x = -ae \), at \( \text{time}=tc \), during one contact occasion for a trapped particle. The opposite surface moves with the velocity \( v_1 \). The creep for point P is equal to \( \xi \) relative to the opposite surface. From paper E.

![Wear depth graph]

Figure 21. Theoretical wear for a house washer (from paper E).

7 CONCLUDING REMARKS

This thesis comprises five papers that aim to increase the understanding of wear in rolling/sliding contacts.

The Stockholm commuter train network has been the subject of papers A and B in this thesis in which the wear and surface cracks has been observed for a period of three years. The data from the Stockholm test case has so far been used for the validation of different wear models in two PhD theses, Telliskivi [28] and Alwadhi [29], and one
licentiate thesis, Jendel [30], and also validated a surface crack model presented in the PhD thesis by Ringsberg [31].

Trackside lubrication can reduce the rail wear substantially for curved track. Paper A found a lubrication benefit factor of 9 for small radius curves (300 m). A harder more wear resistant rail steel show a benefit factor of 2 compared with a standard rail steel. In paper A was also shown that the wear rates vary at different time of the year. This variation is probably due to natural lubrication in form of water. Also the temperature seems to influence the contact conditions and hence the wear. Precipitation especially correlates well with the wear rate: increased precipitation is associated with a decrease in the wear rate.

Damage mechanisms such as surface cracks and wear on a rail can reduce the service life of a railway track. The purpose of paper B was to study the development of these two damage mechanisms on rails in a commuter railway track. By comparing the wear depth in the headcheck zone with the crack length, equilibrium between these two damage mechanisms was found for a lubricated rail. Both the crack length and the wear depth showed low values. By using a lubricant with friction modifiers the stresses was low enough to prevent crack propagation; at the same time, the rail was hard enough to reduce the wear rate. This is probably the most favourable state in terms of rail maintenance cost.

Roller bearings subjected to lubricant borne particles have been the subject of papers C, D and E in this thesis. Particles in the lubricating oil can have a significant impact on the wear in lubricated contacts. Even at low concentration levels can self-generated particles cause significant wear. Most of the self-generated particles were generated during a short running-in period. It is particularly important that filtration be very efficient during this period to lengthen the life of dirt-sensitive components. The here, in paper C, presented experimental results shows that filtration during run-in for one hour with a 3 μm filter can reduce both the mass loss and the number of self generated particles by a factor ten.

Also the effect of how coated rollers affect the self generated wear was studied in paper C. During boundary lubricated conditions was the wear of the bearings with coated rollers so low that it was not measurable with the used instrumentation. At the same time the wear of the standard bearings was up to 10 μm. The number of particles generated in the contact was significantly less for the bearings with coated rollers. There were twice as many self-generated particles when using a standard bearing compared with a coated bearing.

A series of experiments has been carried out to study the wear of a spherical roller thrust bearing by ingested lubricant borne hard particles in papers D and E. The form of the worn profile and the length of wear scratches correspond closely to the sliding within the contact. Measurements of the geometry of individual scratches show that only a very small proportion of the material disturbed by an abrasive particle is removed as wear. A count of the number of wear scratches on the rolling element surface indicates that the contact concentrates particles. This occurs because as soon as a particle is trapped by the rotating elements the friction forces pull it into the contact. Therefore all the particles in an incident layer of oil, of similar thickness to the particle itself, are drawn into the contact. This abrasive wear process has been successfully simulated in paper E. A novel wear model based on the observation of a single point on the
contacting surface when a concentration of particles passes through it has been developed and the necessary data for the model has been determined from the experiments. Comparison of the simulation results with the experimental results shows good qualitative agreement for the form change of the washer surfaces.

Abrasive wear particles cause wear on the contacting bodies especially in zones with large amount of sliding. This changes the load distribution of the running tracks and the difference in load distribution increase with time. The zones with small amount of sliding will get small amount of wear and will be the highest loaded areas. Fatigue starts in these areas with zero sliding and a drastic reduction of bearing life can be expected, see Olofsson [6].

Some examples of future work for extension and improvement of the work are given below:

- The profile change of the rail and wheel profile due to wear can lead to a different load condition and a change in the stress level. If this new load condition is favourable this can lead to reduction of the wear rate and prevention of crack propagation. However, the new load condition may also lead to a worse or steady state load condition. Whether the change in load condition is favourable or not should be the next step in studying the wear of wheel and rail. A further step could also be to develop a rail profile that is adapted to actual vehicles and the load conditions which they are subjected to.

- It has been shown that the length of the wear scratches depends on the contact conditions existing between the contacting components. This is of great importance when building a simulation model of abrasive wear in a rolling/sliding contact, since the worn off volume of a single particle is nearly proportional to the length of the scratch produced by that particle. When comparing the wear graphs and the graphs of the length of the scratches it is, however, obvious that they are not directly proportional. Some other parameters must therefore also affect the wear. These parameters would mainly be the number of particles entering each part of the contact, and the way a particle behaves in the contact. Schematically these parameters have been examined in paper E but more work needs to be done on this. Especially what happens in the case with coated surfaces should be studied further. Also the scratching process and removal of material is important to know more about. The influence of fluid dynamics on entrapment of lubricant borne particles should also be studied to see whether the entrapment of particles can be prevented or reduced. Another interesting thing to study is whether some kind of surface treatment can prevent particles to embed in to contacting surfaces and hence reduce the amount of two-body abrasive wear.
REFERENCES


