Lubricants Influence on Wear in Sharp Rail Curves

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Abstract
Rail lubrication in curves was widely introduced in Sweden during the 1970’s due to the high wear rate of gauge faces in sharp curves. The first tests performed in Sweden showed that lubrication decreased wear rates by roughly 10 times. Based upon these results, the Swedish track system was equipped with wayside lubricators called Clicomatic.

Aside from wear, rolling contact fatigue (RCF) is another limiting factor in railway infrastructure productivity. Head checking is one of the fatigue forms that arises in curves of heavily loaded tracks as well as fast train tracks. Head checkings grow at the rail head, close to the gauge corner. High pressure and a sliding motion lead to plastic strain of the material from the rail ball towards the gauge face. There is a coupling between wear rates and RCF damages, high wear rates may lead to reduced RCF damages and vice versa. Hence, frequent application of lubricants to the rail does extend the rail life by several times but may, on the other hand, lead to fatigue problems. Preventive grinding in combination with lubrication has been the most efficient way to avoid fast degrading of curved rail track.

This thesis concerns the progress of rail lubrication in terms of how environmentally adapted lubricants function both in wayside lubricators and at the track.

Another aspect of the thesis is to find a lubricant able to reduce the wear rate at the gauge face while simultaneously decreasing the crack growth at the rail head. One thought was to use controlled wear to reduce rolling contact fatigue. Controlled wear in this situation means that the usage of a lubricant should give a sufficiently high wear rate to continuously wear off the surface layers. Another idea along the same lines was that of low contact friction. Low friction in the contact gives less plastic strain and subsequently less head checking. Rapeseed oil had shown low contact friction in earlier testing, therefore, synthetic esters were of interest to examine in this application.

From both field tests and laboratory tests it was concluded that the wear protection for environmentally adapted lubricants was comparable to that of previously used lubricants if they were applied by trackside lubricators. Although they were known to be more sensitive to low temperatures, testing showed how to overcome this issue.

Wear and friction tests were performed in the laboratory to evaluate lubricating blends. The tests evaluated rolling / sliding as well as pure sliding contacts.

The research performed in this thesis yielded the following results. Pure synthetic ester (TMP-Oleat and TMP-C8-C10) in a sliding contact resulted in high wear rates and a lot of abrasive marks at the surface. By adding friction modifiers, the friction as well as the wear rate decreased considerably in the sliding contact.

In the rolling sliding contact, synthetic ester alone and synthetic ester with the addition of friction modifiers both showed similarly low friction and wear rates. PAO gave higher friction, but at the same time, the surfaces had significantly fewer cracks as compared to synthetic ester formulations.

One observation was that surfaces lubricated by synthetic esters became less hard and the hardness interval was wider than those lubricated with PAO. One explanation is that even if the overall friction is low, local spots of high friction between asperities can lead to localized strain and crack initiation.

The conclusions are that Rapeseed oil is the recommended base for rail lubricants. The tested synthetic esters are a more expensive alternative useful if excellent low temperature properties are needed. Graphite is not necessary in the lubricant if it is applied continuously by a wayside lubricator. Twin-disc tests showed that low contact friction does not necessarily mean that crack initiations are avoided in the surface. In conclusion, the base fluid and the additives influenced not only the wear rate but also the ability for cracks to grow.
Acknowledgments

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The project was financed through JVTC, the railroad research centre located at the university. It was funded by several member companies including Banverket, who took an active roll in my research.

When I started my Ph.D. studies in 1998, I felt young and enthusiastic. I was able to dedicate a lot of time to my study of surfaces and wear. The environment at the university, with so much equipment waiting to be explored, was perfect for an enthusiastic soul. But many winds have blown since I began this project. I met my wife in 2000 and we had 2 daughters. But the past 2 years have been very hard for me and my family. My oldest daughter, Matilda, was found to be seriously sick and she died at the age of 3 in May 2005. I thank my wife, Anneli, for her support during this time.

Dr. P-O Larsson-Kräik has been my co-supervisor during my Ph.D. studies. I actually met P-O in 1990. He was commanding officer in the army during the summer of my service. Seven years later, it was he who suggested that I start my studies as a Ph.D. student. He has been a great source of knowledge about tribology and all sorts of practical matters. But P-O also has a humanistic and social side that I appreciate very much.

Furthermore, I would like to thank my supervisor, Professor Erik Höglund, as well as Professor Braham Prakash and Professor Tomas Norrby; all of whom have helped me with my research.

In all, my time at the university has been fantastic. I have enjoyed spending time with my colleagues at work and in my spare time and I thank them all for their help and support. It has been an incredible time and I believe we will always keep in touch!

Time has passed and it is now 2005. The enthusiasm is still here but I have definitely grown a few years older.

Luleå, October 2005

Patric Waara
Dedicated to the memory of my loved daughter Matilda
Appended papers:

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F  P. Waara, P-O Larsson Kräik, A. Kapoor, Rail/Wheel steel lubricated by ester based model lubricants in a rolling/sliding contact, Submitted for publication.
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1 Introduction

Rail lubrication in curves was introduced worldwide during the 1970s partly due to increased traffic volume and higher axle loads that led to rapid track degradation. Another reason was that the rail had actually been well lubricated since the very beginning. Much oil had leaked from older locomotives until they became better sealed. When the older locomotives became replaced by new, the rails dried up completely and severe wear occurred at railroad curves. The increase in traffic in Sweden during the last century is shown in Table 1. Notice how the total track length is the same as 100 years ago while freight traffic has increased by a factor ~8, SJ [1]. This increase in traffic volume requires more efficient maintenance.

<table>
<thead>
<tr>
<th>Year</th>
<th>Passenger traffic [Million passenger km]</th>
<th>Freight [Million gross tonnes-km]</th>
<th>Track length [km]</th>
</tr>
</thead>
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<td>1198</td>
<td>13088</td>
</tr>
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<td>2959</td>
<td>16087</td>
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<tr>
<td>2002</td>
<td>19000</td>
<td>9100</td>
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1.1 The wheel/rail interface

In Fig. 1, a cross section of the rail/wheel interface helps to identify names of parts. The wheel consists of a flange and a wheel tread with the wheel root in the middle. The wheel and rail field side are located to the left in Fig. 1. The rail consists of a foot and a rail head connected by the web of the rail. The only visible part in Fig. 1 is the rail head where the top of the rail head is called the rail ball. Most rolling contact between the rail and wheel is located on the rail ball. The gauge corner is located between the rail ball and the rail gauge face.

![Fig. 1. Schematics of rail and wheel cross sections which identifies the parts, Esveld [2].](image)

The wheel flanges are in contact with the gauge face during train operation, in particular when a train has cant deficiency. The rail has canting in curves, i.e. the outer rail is situated higher than the inner rail. Therefore, the outer rail is identified as the “high rail” and the inner rail as “low rail”. When a train is operating along a tangential track and approaches a curve, wheel conicity, canting and train speed contribute to steer the train set through the curve. Ideal cant applies to one speed; if freight and passenger trains traffic the same track a compromise has to be made. The track is

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1 The metric system is default in this thesis and 1 ton =1000kg. In USA short ton (2000 pounds) is normally used which correspond to 907.2kg.
normally designed to give fast trains a certain degree of cant deficiency. The contact between the wheel flange and the gauge face is the major reason to wear of rails and wheels. Flange wear of the wheels leads to expensive maintenance because much material has to be removed to restore the wheel geometry.

The contact between the rail ball and the wheel tread causes minor wear of the rails. The wheels can on the other hand get significant wear at the wheel tread. Tread wear on wheels will continuously change the ability to steer when the wheel conicity becomes misshapen.

The rail/wheel contact on track curves displays a one or two-point contact situation. A single point contact between the wheel tread and rail ball can be identified as a rolling/sliding contact displaying areas of stick and slip inside the contact. Two point contacts also include contact between the wheel flange and rail gauge face in addition to the rolling contact. In curves the rolling/sliding contact at the steering wheel will move from the rail ball towards the gauge corner. This implies that the curvature radii at both rail and wheels become smaller, which lead to higher contact pressure. High pressure combined with creep will lead to plastic deformation of material in the area between the gauge corner and the gauge face.

Friction in the wheel/rail contact is an important factor. A certain level of traction to haul and brake the train is needed. A closer look at the contact spot between the wheel tread and rail ball shows that the contact can be divided into areas of stick and slip. The slip in the contact spot can normally be characterized as micro slip. The rail ball surface normally becomes very smooth. Low friction is desired when the contact between wheel flange and gauge face is a pure sliding contact. The surface at the gauge face normally becomes rougher than at the rail ball, depending on the severity level of the contact.

1.2 Tribology

A general introduction to the field of lubrication begins by dividing the lubricated contact into three regimes, i.e. full-film, mixed and boundary lubrication, see Hamrock [3]. The Stribeck diagram [4] describes the lubrication regimes by measuring the friction as a function of viscosity, sliding speed and load, see Fig. 2. Full-film lubrication means that the surfaces in contact are totally separated by a lubricant, including two main types, viz. hydrodynamic lubrication (HL) and elastohydrodynamic lubrication (EHL). Film thickness is generally a function of the load, relative rolling/sliding velocity, viscosity (pressure and temperature dependent), geometry and elastic properties of the bodies. HL gives a relatively thick film that normally exceeds 10\(\mu\)m. The pressure in HL is usually too low to cause significant deformation of the surfaces in contact. Characteristic of EHL are increased pressure and elastic deformation of the surfaces, which become significant when the film thickness is below 1\(\mu\)m. The contact becomes mixed lubricated (ML) when the lubricating film is not sufficiently thick compared to the surface’s roughness. The opposing solid surface asperities are partly in contact. In the ML regime the friction increases when more asperities get in contact. As the asperities get in contact, wear also becomes significant. Lubricants aimed for these types of contacts are normally added with extreme pressure (EP) and anti wear (AW) additives. If the film thickness is so thin that only the surface asperities carry the entire load, the contact is in the boundary lubrication (BL) regime. The friction stabilises at a high level, though the wear may be severe.

In general the rail/wheel contact due to low relative sliding speed and relatively rough surfaces is located in the BL regime. The contact is also starved, i.e. the supply of lubricant is too small to build a sufficient lubricating film. The main objective with lubrication in this application is to reduce both wheel and rail wear, save energy and, in some cases, reduce noise.
The lubrication regime in a lubricated contact determines which type of additive is needed. When the contact pressure is low and the film thickness relatively thick, friction modifiers are suitable additives for the base fluid, and can be of the type fatty acids or synthetic esters. Fatty acids and synthetic esters are polar CH-chains, though fatty acids are much more polar compared to synthetic esters. Fatty acids physisorb to the surfaces; the closer the molecules are packed the better the surface is protected. When the contact pressure increases, antiwear (AW) additives are better suited to protect the surfaces. AW additives consist of polar CH-chains that include sulphur or phosphor. The polar part of the molecule chain is attracted to the surface and is tied by a covalent binding. The covalent binding is stronger than the physisorbed molecules which are typically of the type Van der Waal’s binding. Very high contact pressure renders extreme pressure (EP) additives suitable to protect the surfaces. EP additives attach to the surface by chemireaction and can build up multilayer structures much thicker than AW additives and friction modifiers that normally only create a mono layer. Suitable chemical substances to create extreme pressure protection are sulphur, phosphor, chloride and boron. Several of these chemicals are toxic and not biodegradable.

1.3 Lubricant effect on the track

Lubricants are generally excellent to reduce wear in severe contacts, but other aspects must also be considered before deciding to lubricate thousands of kilometres of rail track.

As early as 1954-1960, Dearden [5,6] evaluated materials as well as lubrication and suggested the development of better rail materials in Great Britain. The recommendation was based on the absence of many sharp curves on domestic rail track. When the choice was made between rail lubrication and improved materials, stronger materials had the advantage due to the higher costs of maintaining a lubrication system.

Lubrication was of great interest during the late 70s and early 80s, especially in the USA and Canada. A rapid investment in equipment for rail lubrication started without any research evidence of the effectiveness or negative effects of lubrication. Examples of negative effects are that the lubricant can immigrate to the rail ball and give low friction, as well as the use of liquids can cause crack tip pressurisation. However such fast change into new technology is unusual in this type of conservative business, see Welty [7].

During the 70s the effect of lubrication was also examined in Sweden by measuring wear at rail gauge face and wheel flanges. A good lubrication strategy was considered to result in great savings for infrastructure players. The results from one of the tested tracks, shown in Fig. 3, led to investment in lubricant equipment. The wear on the rail gauge face at dry conditions was 0.37 mm/MGT (million gross tonnes), while lubrication led to ~9 times lower wear (0.042 mm/MGT)
From these results the Swedish track was equipped with wayside lubricators of the type Clicomatic and a recent update showed ~2200 wayside lubricators are currently in service in curves with radii <800 m.

![Gauge face wear reduction achieved by introducing lubrication tested in Sweden during the late 70s. The curve radii was 455m, rail steel 900 and traffic volume 8 MGT/year (SJ [8]).](image)

The retentivity and spreadability of lubricants applied from wayside lubricators are closely related. The amount of applied lubricant is one important factor to control wear. Lubricant type and the addition of solid lubricants are also influencing factors. The lubricant type and effects from solid lubricants were examined in several independent tests by Clayton et al [9, 10] and Sato et al [11] as laboratory test. A field test was also carried out at FAST (Facility for Accelerated Service Testing in Pueblo, Colorado) by Reiff [12]. These tests basically aimed to find out if different types of grease and the added quantity of solid lubricants affected retentivity and spreadability. In Reiff [12] the wheel forces of a former locomotive were measured, showing that MoS\textsubscript{2} gave the best effect on retentivity while graphite greases did not reveal any clear evidence about spreadability or retentivity. MoS\textsubscript{2} gave low wear rates in a twin-disc test in Clayton et al [10], while graphite added to lubricants did not indicate any opportunities according to wear.

A laboratory test by McEwen and Harvey [13] with a full-scale wheel/rail test machine showed that the durability of lubricants at the rail could be best improved by using more viscous lubricants. Adding solid lubricants also improved durability.

Spreadability was examined in a Swedish field study by Nilsson [14]. The test was carried out at the commuter train track in Stockholm. The examined curve had one-way traffic and radii ~600 m. The worn off volume was 0.5 mm\textsuperscript{2}/MGT at a distance of 50 to 150 m while the wear rate was twice as high (1.1 mm\textsuperscript{2}/MGT) at the distance 250 m from the lubricator. A similar curve without lubrication resulted in much higher wear (2.4 mm\textsuperscript{2}/MGT).

During the 1970s, an early effort to understand the influence of water on friction was conducted by a British rail researcher. This extensive study showed that water seems to be the main factor influencing the friction in the wheel/ rail interface (Beagley and Pritchard [15]). Moisture often causes very low friction, especially in small quantities. A critical mixture of debris and water can give enough slurry viscous to diminish friction coefficients down to 0.015. When raining, the friction coefficient becomes almost 0.2. Some rail tracks suffer badly from autumn leaves that diminish the friction precipitously causing large problems to gain enough traction.

Rail lubrication was initially intended to only reduce wear, though as the rail curves became equipped with wayside lubricators, the energy savings became a new area of interest. Tests at FAST by Reiff [16] in 1983-1984 showed large energy savings where fuel savings of up to 30% were found. The ability of different greases to spread and reduce friction was investigated. Grease was applied to the rail flange using a wayside lubricator while the test train spread the lubricant
along the track. During the test the longitudinal wheel force was measured at the leading wheel set on a standard track. The tests showed grease containing graphite to be spread relatively fast on dry track and give a low constant level of wheel force. Another observation was that even if the grease was properly applied at the rail flange, grease migrates to the rail head and causes slip.

In other tests, tracks with many sharp curves saved fuel in the range 25-30%, see Samuels and Tharp [17]. The possibility to save energy by lubricating the rail was further strengthened. However, its benefit on straight track was not of the same magnitude as the earlier tests, fuel savings of about 5%, see Dahlman and Stehly [18].

The present work does not involve any studies concerning fuel or energy savings due to wheel flange lubrication, though it is important to understand that the technique is useful and an easy way to save energy as wear on both wheels and rails will be reduced.

Lubrication can also result in unwanted effects in this complex system. An accident during a lubricated test at AAR (Association of American Railroads) highlighted steering problems that follow by controlling the friction. Later research at AAR by Mace et al [19] found the occurrence of reduced, unstable steering at a three piece boogie. Unstable dynamics possibly leading to derailment were found if five conditions were fulfilled, i.e. strong two point contact, hollow wear, heavy gauge corner grinding, high gauge face lubrication and dry rail heads. Other unwanted effects related to crack propagation have also been found. This type of problem is discussed later in this thesis.

The lubricant can be applied at either the rail ball or gauge face. Applying the lubricant at the rail ball resulted in traction problems. High gauge face lubrication and dry rail heads can lead to derailment. The necessary quantities of lubricant to be applied need careful adjustment.

Finally, rail lubrication does influence the environment, especially as this type of lubrication is a so called “total loss application”.

1.4 Application technology for rail lubrication

There are three methods to lubricate the rail/wheel contact. One way is to locate wayside lubricators in sharp curves, reducing wear in the contact between wheel flange and rail gauge face. The other ways are to mount onboard systems on a track inspector vehicle or at a locomotive. Onboard systems mounted on locomotives are normally intended to reduce energy consumption for the operator as well as protect the wheel flanges mainly at the locomotive. Systems mounted on a track inspector vehicle and wayside lubricators are primarily expected to reduce wear at the rail gauge face. Whatever its purpose and how it is applied, wheel and rail wear will be reduced.

Sharp curves require continuous lubrication; therefore, wayside lubricators are recommended at tracks with many sharp curves. Tracks with light-tonnage and few trains have been suggested to replace wayside lubricators by onboard systems, but for most situations complementary usage is preferred, Kramer [20].

As mentioned earlier, wayside lubricators are the base in the Swedish rail lubricating system. In addition some of the railway operators use onboard systems at the locomotives. Today, a total of ~100 locomotives have onboard systems where the majority apply the lubricant at the rail ball. Generally, the onboard systems do not operate during the winter season, though throughout the rest of the year, grease consumption for Swedish onboard systems due to travel distance is roughly 5 g/km.

The maintenance of wayside lubricators is always planned, and unexpected stops normally imply a malfunction until the next maintenance inspection. A disadvantage with the equipment is that it does not work in the arctic climate of northern Sweden during the winter season.

1.5 The environment

The total quantity of sold lubricants in Sweden was 163 500 m³ (2004). Lubricants can be divided into two types, i.e. lubricants with the ability to be recycled or disposed, or as “total loss applications” like chain saw oil, two stroke oil, mobile hydraulic liquids and lubricants suitable for
rail/wheel interface. Wayside lubricators in Sweden use about 20 ton whereas onboard systems consume 11-21 ton. Rail lubrication corresponds to about 1.0% of the total grease consumption or ~0.02% of the total Swedish lubricant consumption.

The railroad in northern Sweden passes through areas that are environmentally sensitive to pollution; hence, the need to find environmentally adapted lubricants is high. The National Swedish Rail Administration (Banverket) is continually looking for better environmentally adapted lubricants. A wayside lubricator such as those used in Sweden can pollute the local environment with 50 to 150 kg of lubricants annually for 10 consecutive years. This will lead to unacceptable environmental damage if biodegradable and non-toxic lubricants are not used. Caring for the environment in this application is so far driven by “free will” and not legislation. The need to find and develop new lubricants with a low environmental impact is a big issue, as the need for environmentally adopted rail lubricants is expected to increase in the future.

Generally lubricants aimed to reduce the environmental burden should not be described as environmentally “friendly” or “compatible”, because they are not. From that point of view the word environmentally “adapted” makes more sense. Environmentally adapted lubricants (EAL) can constitute six typical selection criteria, viz. biodegradability, renewability, toxicity, bioaccumulability, life cycle assessment and fuel economy; see Norrby [21].

The performance additives added to EALs should fulfil toxicity criteria. Among environmentally adapted base fluids, soybean oil, jojoba oil, corn oil, rapeseed oil and castor oil are all vegetable oils and examples of lubricants that fulfil EAL criteria. A few examples of synthetic esters that also fulfil the environmental criteria are TMP-Oleate, TMP-C₈-C₁₀ and Pentaerythritol. Compared to mineral oil, vegetable oils cost about 2-3 times more and synthetic esters cost about 5-10 times more.

Several environmentally adapted lubricants have become available on the market during the last 7-8 years. Base fluids as rapeseed oil, soybean oil and synthetic esters are normally used as bases in greases aimed for the rail/wheel contact. Grease based upon high-oleic, genetically-enhanced soybean was developed to lubricate the rail track interface, see Honary [22]. This soybean grease was introduced during 2001. Environmentally adapted lubricants evaluated for hydraulic systems showed that “conventional” mineral oil gave 3-6 times less wear volume compared to a mixture of synthetic ester/rapeseed oil, Rieglert and Kassfeldt [23]. For this reason it is important to investigate if EALs give higher wear rates in rail/wheel contacts to be aware of the consequences.

1.6 Crack propagation

Highly loaded wheel/rail contacts are subjected to cyclic loading that leads to rolling contact fatigue. Cracks propagate from the surface down into the material. In Kapoor [24], crack growth is explained where the crack growth rate \( \frac{da}{dn} \) is a function of the number of load cycles \( n \) and the crack length \( a \), see Fig. 4. Depending on the crack depth, different mechanisms determine the crack growth rate. During a crack’s early life the growth through the surface layer can be estimated by using ratcheting models (curve R). The depth of this layer depends on wheel load, yield strength of the rail and its ductility. When the crack grows underneath the elastically-deformed layer it can be modeled by using linear elastic fracture mechanics. Here, the crack is driven by fracture mechanism Mode I (open the crack) and Mode II (sliding). The crack propagation amplifies when the crack grows toward the region with high stress intensity (curve S₁). Subsequently, the propagation rate decreases when the crack tip moves away from the high stress region (curve S₂). Finally, when the crack tip is deep down in the material the effect of contact stresses diminish. Bending stresses due to the unsupported length between the ties and residual stresses will suddenly and rapidly increase the crack growth.
Crack tip pressurisation is an influencing factor on crack growth, as first described by Way [25]. Another theory is when reduced internal friction inside the crack caused by lubrication increases the crack growth due to Mode II (the sliding motion), see Fletcher and Beynon [26]. Fletcher and Beynon tested the effect from intermittent lubrication in a laboratory, and found that intermittent lubrication probably shortens rail life extensively. High load and high friction initiate rolling contact fatigue (RCF). The friction then decreases and the cracks still continue to grow.

1.7 The relation between wear and rolling contact fatigue (RCF)

Since the early 1970s, the rail track connecting Edmonton and Vancouver in Canada went through several changes, described by Worth et al [27]. On this track, curves with a radius around 218 m were worn out after only 18 months. Several years later, the rail life was extended to around 7 years with better lubrication, rail grinding, continuously welded rail and substitution of concrete ties for wood ties. Rail material made of chromium alloys was tested around 1978. Lubrication had proven to reduce corrugation growth of standard carbon rail, but did not have the same effect on chromium rail. Rail made of chromium alloy suffered badly from rail corrugation and without grinding the rail had to be replaced just by the corrugation itself. Similar observations were made a few years later by Daniels and Blume [28]. Steel alloys with high chromium content did not fulfil the expedient. The chromium rail was replaced by head hardened rail in 1981 and after 40-50 MGT (Amer. short ton) of traffic it was found that severe wear was replaced by fatigue damages. The profile of the rail was unsatisfying and needed improvement to unload the gauge corner. An asymmetric rail profile was suggested and an intensive grinding program began.

Rail grinding has become an efficient way to remove decarbonised layers on new rail, plastically deformed material, short and long pitch corrugations, head checking, pitting and spalling. In the late 80s the rail profile was adjusted to reduce contact stresses, Kalousek et al [29].

Head checks have become a major problem in many heavily loaded or high speed tracks, see Fig. 5. Head checks normally appear at the high rail in curves with radii of 400 to 800 meters. When cracks start to grow into the material the rail needs to be grinded. Otherwise, pieces of rail material will spall off.
To extend the components life due to fatigue and wear, Kalousek and Magel [30] introduced the idea of a “magic” wear rate. If the actual wear rate is less than the magic wear rate, material has to be removed by corrective grinding. If the actual wear rate is higher than the magic wear rate the metal is wasted from too much dry wear. The idea is to find a balance between preventive grinding and lubrication, see Fig. 6. Kalousek also suggested moving the wheel/rail contact distribution by grinding alternate asymmetric rail profiles. By altering the contact over the rail head the material should last longer due to fatigue.

The relationship between wear and fatigue was also described by Kapoor [24], see Fig. 7. The maximum rail life was found where the curves intersect. Safe operation requires the rail to fail by wear. Grinding becomes an artificial way to get sufficiently high wear rates.

In Sweden maintenance grinding on a small scale has been performed for a long time, but the first large grinding campaign started in 1997 at the ore line “Malmbanan” in the north, see Grassie et al [31]. Head checks had been a problem for many years and international experts participated in a pre-study to upgrade the ore line from 25 to 30 metric tonnes axle load. To reduce the need for rail renewal it was suggested to introduce a grinding strategy for the heavy haul line. The first grinding campaign was corrective grinding to achieve the target profile. When the target profile was achieved preventive grinding was introduced. In total the grinding campaign reduced the need for rail renewal by ten times. The yearly rail degradation based on measurements of wear and rail grinding is a good aid when predicting future rail renewal.
2 Outline of the thesis

This thesis may contribute to make railway transportation more efficient by progress of lubrication technology aimed for the wheel/rail interface.

Rail wheel lubrication normally deals with the contact between gauge face and wheel flange, which is examined in the first part of the thesis.

Environmentally adapted lubricants are required in applications such as wayside lubricators, since they are located in sensitive environments and contribute to high concentrations of unwanted grease. The effect on the wear rates is unknown if using environmental lubricants. Therefore, tests were conducted on five greases, four being approved as environmentally adapted, while one was based on mineral oil (paper A). The mineral oil based grease as well as one of the environmentally adapted greases contained graphite.

A field test was carried out to investigate two greases (from paper A) in their right environment. One of the two greases was environmentally adapted. The field test was conducted in two curved sections equipped with wayside lubricators with similar properties. The wear was verified to find differences between the lubricants, i.e. if environmentally adapted greases are suitable for wayside lubricators.

The second part of the thesis deals with lubrication at the rail ball close to the gauge corner. This contact requires different solutions and the idea is to find a lubricant that compromises and fulfils both contact types here discussed.

Based upon Kapoor’s [24] crack growth theory, an idea was formed where a lubricant could reduce crack initiation at the surface. In Fig. 8, \( \frac{da}{dn} \) describes the crack growth speed and wear speed perpendicular down in the material. Heavily loaded contacts without any lubrication lead to high wear rates and extensive plastic deformation where ratchetting drives the cracks; see the continuous horizontal wear line in Fig. 8 and the continuous line showing the crack growth driven by ratchetting.

In the wheel/rail contact a shear stress leads to plastic deformation of the rail material. Each load cycle contributes to a small plastic deformation in the same direction. These accumulated deformations can be characterized as material flow. When the plastic strain exceeds a certain limit, cracks can be initiated at the surface. Unlubricated contacts give a relatively fast material flow, which also lead to a fast crack growth. The initial crack growth rate close to the surface is shown in Fig. 8.

By introducing lubrication, the shear stress will decrease and lead to less plastic deformation and slower crack initiation (the dotted inclined line). As well, the wear rate will simultaneously decrease (see the big grey arrow). The idea in this thesis is to find a lubricant that sufficiently increases the wear rate to wear off crack initiations (follow the grey small arrow), while the crack growth rate stayed unchanged.

\[ \frac{da}{dn} \]

\[ \text{Crack driven by ratchetting} \]

\[ \text{Reduced crack growth rate by lubrication} \]

\[ \text{Wear rate in unlubricated conditions} \]

\[ \text{Wear rate in tribochemically lubricated conditions} \]

\[ \text{Wear rate in lubricated conditions} \]

Fig. 8. The tribochemical idea.
Grinding is considered as artificial wear and a way to remove cracks from the rail track. Grinding and wear contribute together to track degradation. The idea here is to increase wear by using a tribochemical lubricant, and thereby possibly result in less frequent grinding activities at the track. One step is to find the relation between wear and grinding. The highest potential is if the grinding part is overridden compared to the wear part. This relation was investigated in paper C.

The next step was to find the right wear rate. The objective of the following papers was mainly to identify lubricant compositions that fulfill the wear rates described in Fig. 8. The demands on a lubricated contact led to tailored lubricant solutions. The idea is to attain low friction able to reduce crack initiation and growth. The lubricant base oil and suitable additives are the variables to get a sufficiently high wear rate, able to continuously shear off thin material layers without damaging the surface.

In paper D additives aimed for higher loads were examined in boundary and mixed lubricated contacts. In this examination AW and EP additives together with a copper passivator were added to mineral oil and a synthetic ester. The blends had different levels of additive concentrations to scan a wide range of combinations so as to find antagonistic or synergistic effects and relative wear rates. These effects are well known in mineral oil blends, but not in synthetic esters. The lubricant blends were tested in a pure sliding contact and a factorial design was used to evaluate the effects achieved by the additives. This work contributed to a better understanding of how synthetic esters respond to different additive combinations.

Fatty acids are well known as friction modifiers when used in mineral oil suited for lightly loaded contacts and are a milder type of additives compared to the AW and EP additives tested in paper D. In paper E these milder blends were tested together with specimens made from rail material. The wear rates and surface morphology were investigated for a pure sliding contact, which illustrates the gauge face/wheel flange best. However, the potential in the contact between the rail ball close to the gauge face and the wheel was also considered.

The most promising and worst blends chosen from paper E were further examined in a rolling/sliding contact in paper F. The rolling/sliding contact was evaluated in a twin disc machine that best describes the contact spot on the rail ball close to the gauge corner and the wheel. As described earlier, surface cracks such as head checks normally appear at the rail in the area between the gauge corner and rail ball. Head checks can be produced in the twin-disc machine. The outcome of these experiments shows if fatty acids in synthetic esters reduce the initiation of cracks. The specimens for this test were manufactured from rail material 900A, the most commonly used in Sweden today, and the standard wheel material “Blue light”, which is most likely a future standard for ore cars in northern Sweden. If the wear rate is low and not many cracks are found at the surface, then it is probably a good lubricant.

3 Experimental methods and results

3.1 Laboratory wear tests on environmentally adapted greases (paper A)

EALs are important in total loss applications. Wayside lubricators may be located in sensitive environments and the waste of toxic and slowly degrading lubricants may risk polluting the ground water. Evaluating EALs aimed for rail/wheel lubrication is important due to the wear performance. Solid lubricants such as graphite are normally added to grease aimed for boundary lubricated contacts to improve the wear performance. Pressurized lubricating systems do risk clogging if the lubricant contains graphite. This influences the reliability of wayside lubricators such as Clicomatic.

3.1.1 The Plint & Partner high frequency machine

In paper A the contact between the gauge face and the wheel flange was simulated, and the wear performance of environmentally adapted lubricants with and without solid lubricants was investigated. The earlier used standard graphite grease was not environmentally adapted. In this test the standard grease serves as a reference when evaluating wear rates. As mentioned in the introduction rail/wheel contacts operate in the boundary lubricated regime. There is still no theoretical way to predict wear when considering important factors such as material hardness, texture of the surface, material
hardening during run-in, sliding speed, sliding directions etc. The contact pressure was held at the levels 521 and 737 MPa, representing a medium high contact pressure due to the somewhat conformal contact between wheel flange and rail gauge face. The choice of apparatus was between a pin-on-disc machine and a reciprocating apparatus that both simulate pure sliding. The pin-on-disc machine simulates wear in one direction and is able to give continuous sliding speed. The reciprocating motion in the machine simulates the reverse direction of the sliding motion at the gauge face that occurs on a track having traffic in both directions. A Plint & Partner high frequency reciprocating machine (Plint & Partner TE 77B) was chosen to evaluate the wear performance. The contact geometry employed was a cylinder-on-flat line contact. In this test the specimens were hardened steel rollers designed for roller bearings. The hardness of these specimens was 850 HV. Measurements on the rail gauge face surface after “work hardening” shows the hardness to remain roughly in the range 350-400 HV, depending on the type of material. The usage of such hard and smooth specimens (Rq less than 0.1 μm) limits this test to evaluating the lubricant in the mild wear region. The larger roller (Ø 10 mm) standing with the flat end upward gives a flat contact surface, while the cylinder specimen is a smaller roller (Ø 6 mm) lying on top of that flat surface. The actual length of the contact was limited by a chamfer on the specimen’s edge and measured 4.5 mm. Under controlled loading, a drive arm reciprocates the cylinder specimen over the flat specimen, see Fig. 9. The specimens were held in contact in a temperature controlled lubricant container. The surface roughness Rq of the upper specimen was 0.05 μm and 0.08 μm for the supporting specimen. The frequency of the reciprocating motion was 11.7 Hz and the amplitude 2.3 mm.

![Schematics of the flat line contact in the Plint & Partner high frequency reciprocating machine.](image)

3.1.2 Lubricants and setup conditions

Five different commercial greases were tested based upon mineral oil, synthetic ester and rapeseed oil, of which four were approved as environmentally adapted. The mineral oil based grease and one of the environmentally adapted greases contained graphite. The composition of the greases is presented in Table 2.

<table>
<thead>
<tr>
<th>Grease</th>
<th>Base oil</th>
<th>Soap (wt%)</th>
<th>Base oil viscosity (cSt)</th>
<th>Graphite</th>
<th>NLGI grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Synthetic ester</td>
<td>No soap</td>
<td>A mix of different viscosities</td>
<td>No</td>
<td>00-000</td>
</tr>
<tr>
<td>2</td>
<td>Rapeseed oil and ester</td>
<td>Lithium complex (6-8)</td>
<td>40 cSt</td>
<td>Yes</td>
<td>00</td>
</tr>
<tr>
<td>3</td>
<td>Rapeseed oil</td>
<td>Silica gel</td>
<td>38 cSt</td>
<td>No</td>
<td>00</td>
</tr>
<tr>
<td>4</td>
<td>Mineral oil</td>
<td>Lithium complex (6)</td>
<td>15 cSt</td>
<td>Yes</td>
<td>000</td>
</tr>
<tr>
<td>5</td>
<td>Synthetic ester</td>
<td>Lithium/ Calcium (5-15)</td>
<td>40 cSt</td>
<td>No</td>
<td>000</td>
</tr>
</tbody>
</table>
The operating parameters for the test are summarized in Table 3. Rail temperatures in Sweden normally range between $-45^\circ C$ and $+55^\circ C$. During the test the grease bulk temperature of $+36^\circ C$ softened the grease, allowing it to return more easily to the contact during the test. Each set up with operating parameters was tested five times to yield valid results.

Table 3. Operating parameters in the Plint & Partner high frequency friction apparatus

<table>
<thead>
<tr>
<th>Sliding length [m]</th>
<th>Temp. [(^\circ C)]</th>
<th>Max sliding speed [m/s]</th>
<th>Hardness HV 1000g</th>
<th>Load [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>31.4</td>
<td>36 ±1</td>
<td>0.17</td>
<td>850</td>
<td>100/200</td>
</tr>
</tbody>
</table>

The width of the wear scar of the specimen on top (the roller with \(\varnothing 6\) mm) was evaluated in an optical microscope and calculated into a wear volume \(V\).

Archard's Law Eq. (1) is the most common "dry contact" model using sliding distance \(S\), applied load \(F_N\), the softer surface hardness \(H\) and worn off volume \(V\) giving the dimensionless wear coefficient.

\[
K = \frac{V \cdot H}{S \cdot F_N}
\]  

3.1.3 Results of laboratory wear tests

Laboratory test results showed some differences between the lubricants. A box-plot in Fig. 10 shows the wear coefficient for the greases. Each grease type had a total of 10 measurements and each box include 50\% of the measurements. Inside the whiskers all measurements are found except for outliers. The load factor did not significantly affect the test results; therefore, all wear measurements were presented together independent of load applied during the test. The standard grease showed the least wear with very little scatter. Grease 2 showed the highest wear rate despite the graphite. The environmentally adapted greases yielded some varying results, where wear rates of about 2-4 times higher should be assumed (from ref [23]). In this laboratory test graphite proved to not significantly reduce the wear rates, and lubricants without graphite may be recommended. In addition to this laboratory test a field test was carried out to examine how Greases 3 and 4 reduce wear in sharp curves. This field test is described in paragraph 3.2.

![Fig. 10. The wear coefficient measured for commercial rail greases in a pure sliding laboratory contact.](image)
The conclusion from this paper was that the mineral oil based grease (4) gave the lowest wear result. Among the environmentally adapted greases number 3 gave lowest wear and is interesting to evaluate in a field test.

3.2 Field wear test on environmentally adapted greases (paper B)

A field test was conducted to evaluate wear performance of rail greases on the rail track. Even if wear differences were found in laboratory tests, the significance of the found differences in a real rail contact need to be evaluated. Environmentally adapted grease not containing any graphite was tested against mineral based grease with graphite (grease 3 and 4 were tested in paper A). This was an excellent opportunity to evaluate both EAL and graphite performance. Other purposes were to learn more about track lubrication and not only find out wear rates given from proper lubrication, but also when the wayside lubricators were not in service, e.g. during winter due to the arctic climate.

3.2.1 Field test arrangement

The test was initiated during the summer of 1997, the test sight consisted of six curves distributed over two sections. Each section contained three curves, see Table 4. All curves were equipped with wayside lubricators of the type Clicomatic mounted at the middle of each curve. Curves 1 and 3 in section A had similar radii, super elevation and train speed, whereas all three curves in section B were more or less identical. All the test sections were single track with traffic in both directions. The rail profile was measured at 6 fixed points on the high rail in each curve. The fixed point was located 5, 15 and 50 m on each side of the wayside lubricator. Grease based on rapeseed oil without graphite and mineral oil with added graphite were used in the wayside lubricators, see Table 2. According to the results from Rieglert and Kassfeldt [23] as well as Fig. 10, the wear rate for blends of rapeseed oil was expected to be higher for the synthetic ester compared to mineral oil.

Table 4. Specification of rail sections.

<table>
<thead>
<tr>
<th>Section</th>
<th>Curve</th>
<th>Radius [m]</th>
<th>Speed [km/h]</th>
<th>Super elevation [mm]</th>
<th>Steel*</th>
<th>Grease (see Table 2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A:1</td>
<td>581</td>
<td>100</td>
<td>117</td>
<td>900A</td>
<td>Rapeseed oil (3)</td>
<td></td>
</tr>
<tr>
<td>A:2</td>
<td>293</td>
<td>70</td>
<td>150</td>
<td>1100</td>
<td>Mineral oil + graphite (4)</td>
<td></td>
</tr>
<tr>
<td>A:3</td>
<td>590</td>
<td>90</td>
<td>110</td>
<td>900A</td>
<td>Mineral oil + graphite (4)</td>
<td></td>
</tr>
<tr>
<td>B:1</td>
<td>385</td>
<td>85</td>
<td>140</td>
<td>900A</td>
<td>Mineral oil + graphite (4)</td>
<td></td>
</tr>
<tr>
<td>B:2</td>
<td>354</td>
<td>85</td>
<td>150</td>
<td>900A</td>
<td>Rapeseed oil (3)</td>
<td></td>
</tr>
<tr>
<td>B:3</td>
<td>350</td>
<td>85</td>
<td>150</td>
<td>900A</td>
<td>Mineral oil + graphite (4)</td>
<td></td>
</tr>
</tbody>
</table>

*Steel quality refers to the tensile strength [MPa].

3.2.2 The Miniprof measuring device

A measuring device of the type Miniprof measured the rail profiles; see the schematic in Fig. 11. The device (4) is mounted on a telescopic stick (1) and consists of two bars, (3) and (5), connected by a joint (2). The outer bar (5) has a small magnetic wheel (6) that follows the rail profile by rolling over it. This device is described in Esveld and Gronskov [32] as well as in paper B. The rail wear can easily be evaluated for the rail head h, the gauge face s or by measuring the worn off area, see Fig. 12.
3.2.3 Results from field test

Wear at the rail gauge face is here expressed in terms of mm/10 MGT (Million Gross Tonnes). The wayside lubricators are located in an arctic climate and are therefore turned on only during the summer months (May-October). The rail profiles were not exactly measured when the wayside lubricators were turned on or off. Thus, the time periods between the profile measurements include both lubricated and non-lubricated periods.

No significant difference was found of the rail flange wear between the environmentally adapted grease (3) and mineral based grease (4). The mineral based grease contained graphite whilst the rapeseed based grease did not, indicating graphite to not greatly affect wear reduction in this type of application.

The main wear difference was found between summer and winter. For the summer period the curves were well lubricated, but during winter the wayside lubricators are turned off. The results of the measurements are summarized in Fig. 13. The left diagram shows the results from curve A:2 (radius 293 m) while the right diagram shows the results from all three curves in section B (radii 354-385 m). Generally the gauge face wear (s) was low during the lubricated summer session. Even negative wear rates due to the plastic material flow from the rail head combined with very low wear at the gauge face were found. By recalculating the diagram in section B to yearly wear with the present lubricating situation, the average gauge face wear became 0.81-0.86 mm/10MGT. The current rail life of the curves in section B is ~13 years or ~130 MGT.

From these measurements, the achieved wear rate for a well lubricated curve is much lower than for the non-lubricated winter season. If year-round lubrication is introduced the theoretical rail life at such a curve could be estimated to ~60 years or ~650 MGT. Other factors such as fatigue or rail breakage will have an increased influence on the rail life. From this point of view the wear reducing effect from the lubricant is not the only factor to consider. Hence, the usage of environmentally adapted lubricants is recommended in wayside lubricators of the tested type. Note that the results reflect wayside lubricators, whereas mobile lubrication systems differ and will most likely yield other results.
Section A Curve 1-3

<table>
<thead>
<tr>
<th>Time period</th>
<th>Gauge face wear [mm/10MGT]</th>
</tr>
</thead>
<tbody>
<tr>
<td>970501-980702</td>
<td>-0.3</td>
</tr>
<tr>
<td>980703-981020</td>
<td>0.83</td>
</tr>
<tr>
<td>981021-990630</td>
<td>0.70</td>
</tr>
<tr>
<td>990701-991208</td>
<td>0.61</td>
</tr>
</tbody>
</table>

Fig. 13. The diagrams show the difference in gauge face wear (s) between the non-lubricated winter periods compared to lubricated summer periods. The dates are describe according to yy-mm-dd. The left diagram shows rail steel 1100 in a 293 m radius. The right diagram shows rail steel 900A with radii in the range 350–385 m.

The conclusion from this paper is that the environmentally adapted grease number 3 is recommended to be used in the wayside lubricator Clicomatic. Lubrication during the winter period can reduce the wear further.

3.3 The influence of grinding and wear on rail life (paper C)

The “state of the art” in rail maintenance is a combination of lubrication and continuous grinding. A grinding program was initiated in 1997 by The Swedish National Rail Administration (Banverket) at Malmbanan, the Swedish ore track between Kiruna and Riksgränsen (127 km). The track was lubricated during the summer by locomotives applying grease to the rail gauge face. Grinding is mainly done to remove head checks, which have become an increasing problem in recent years. Rail renewal diminished completely when the grinding campaign started, as described by Grassie et al. [2]. The focus of paper C was to determine the actual rail life as well as evaluate how much material was removed by wear and grinding during one year. The data used in this paper was collected by Banverket.

3.3.1 Evaluation method

The Miniprof rail system described earlier was used to measure the profiles with Banverket performing all the measurements. Before the grinding program started the track was prepared with 60 measurement points in the curves and 10 points on the tangential track. The profiles were measured shortly before and after the grinding of the rails. An asymmetric profile was introduced to give a wider contact than previous profiles, delaying the onset of head checks. Without detailing all tolerances for the grounded profile, the main idea is to grind a certain profile known as MB1 with a minimum metal removal of at least 0.2 mm with 23 MGT intervals.

The track was divided into three categories: track curves with a radius below 800 m, track curves with a radius over 800 m and tangential track, see Table 5.
Table 5. Grouping of the track and planed grinding activities at Malmbanan.

<table>
<thead>
<tr>
<th>Length [m]</th>
<th>Amount of grinded track/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius&lt;800 m</td>
<td>51 791</td>
</tr>
<tr>
<td>Radius&gt;800 m</td>
<td>30 526</td>
</tr>
<tr>
<td>Tangential track</td>
<td>48 220</td>
</tr>
</tbody>
</table>

The grinding was evaluated by measuring how much material was removed and by calculating the theoretical rail life due to the removed material. Two methods were used.

Method 1:
The removed material was evaluated as a residual between a new and an old rail profile (reference). The residual was measured perpendicular to the reference profile. The divergence angle between the reference profile and a horizontal line indicate the position at the rail head.

Method 2:
The removed material was measured according to the Swedish standard for railhead wear measurement; see Fig. 12. The yearly vertical wear $\Delta h$ and the yearly gauge face wear $\Delta s$, see Eq. (2).

$$\Delta H = \Delta h + \frac{\Delta s}{2}$$  \hspace{1cm} (2)

The yearly removed material $\Delta H$ indicates the proportion of consumed rail life according to Banverket regulations (BVF 524.1 [33]). According to the traffic at this track, the safety wear limit $H_{\text{limit}}$ is set to 11 mm for the 50 kg/m BV50-rail profile used. The average rail life $n$ years is then calculated as the ratio between the safety limit $H_{\text{limit}}$ and the measured $\Delta H$ value of the studied section (Eq. 3).

$$n = \frac{H_{\text{limit}}}{\Delta H}$$ \hspace{1cm} (3)

3.3.2 Rail life based upon wear and grinding

The material removed by grinding was evaluated for year 2000; the material removed by wear throughout was during 10 months in 1999/2000. The wear evaluated over 10 months was later compensated to correspond to wear per annum. The rail gauge face was lubricated by locomotives during the summer. The results for high rail in curves less than 800 m are summarized in Fig. 14. The mean material loss by wear on the top of the railhead was 0.17 mm, plus an additional 0.47 mm by one grinding campaign. At the gauge corner ($\beta=40^\circ$), grinding off the crack initiations is essential. The wear rate at the gauge corner was 0.24 mm. Higher wear from traffic at the gauge corner is reflected by an increased grinding at the rail ball, if the profile shape should be retained.
Fig. 14. The average wear for one year and the amount of removed material during one grinding campaign according to method 1.

The calculated rail life presented in Table 6 can be used at the actual track to roughly estimate the average length of rail that need to be replaced annually. These calculations suggest the average yearly rail renewal to be 12 000 m ± 1 900 m. A more exact analysis starts with the present track status from where an investment plan for rail renewal will be settled for each year. This analysis was not made here. However, grinding prevents the rail from crack damages and avoids extensive rail renewal.

<table>
<thead>
<tr>
<th>Curve radius &lt;800</th>
<th>Curve radius &gt;800</th>
<th>Tangential track</th>
</tr>
</thead>
<tbody>
<tr>
<td>High rail</td>
<td>Low rail</td>
<td>High rail</td>
</tr>
<tr>
<td>13.6</td>
<td>17.7</td>
<td>19.6</td>
</tr>
</tbody>
</table>

The conclusion from this paper is that roughly one-third of the removed material is related to wear, while two-thirds is related to grinding activities. This provides the potential to use a lubricant that gives tribochemical wear. An increase in wear rate in the range 0.2 mm/year (at the present track) might lead to grinding only every second year. The amount of removed rail material is the same, but the cost from grinding will be just 50%.

3.4 EP and AW additives influence on wear using synthetic esters (paper D)

The objective of paper D was to identify lubricant compositions fulfilling the wear rates described in Fig. 8. The demand on a lubricated contact led to tailored lubricant solutions. In paper D additives aimed at higher loads were examined in boundary and mixed lubricated contacts. Factorial design was used to determine how additives influence the wear rate depending on if they were solved in synthetic ester or mineral oil. The performance of additive packages suited for mineral base oils has been known for a long time. When the base oil is replaced by synthetic esters the additive performance is less known. The combination of additive packages and synthetic ester need to be tested.

3.4.1 Lubricant blends for the test matrix

A $2^3$ factorial design matrix was used as an efficient way to optimize the test series, Box et al. [34]. This method measures the effect of one or more variables on the response(s) of interest. Each variable is held at two levels, a high (+) and a low level (−).
Two base fluids were chosen that had the same dynamic viscosity at 90 °C, i.e. a synthetic polyol ester (TMP-Oleat) and a mineral base fluid. The additive combinations were blended with the base fluids. The additive components are fully described in paper D. The combinations had three different additive types: extreme pressure (EP), anti wear (AW) and copper passivator (Cu-passivator). The EP additive containing phosphor and nitrogen (P, N); the AW additive were of two different types. Sulphur and phosphor (S, P) is one type and the other consist of sulphur and nitrogen (S, N). Depending on the concentration and AW additive, four additive combinations were formed. Combinations 1 and 2 were used in a first test sequence and combinations 3 and 4 in a second. The additive combination is explained below;

- Additive combinations 1 and 3 contain the same AW and EP additives, but at different concentration levels. In combination 3 the Cu-passivator was excluded. The EP additive was an amine neutralised phosphoric acid ester. The AW additive was triphenyl phosphothionate (TPPT) and the Cu-passivator was a thiadiazole deriviate.

- Additive combinations 2 and 4 also contain the same AW and EP additives, but with different concentration levels. The EP additive was an amine neutralised phosphoric acid ester. The AW additive was a methylene bis (dithiocarbamate) and the Cu-passivator was of the type toly thiadiazole deriviate. The concentration levels can be seen in Table 7.

<table>
<thead>
<tr>
<th>Base fluid (additive comb.)</th>
<th>EP Chemistry</th>
<th>AW Chemistry</th>
<th>Cu-passivator Chemistry</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester (1) P, N 0.20% 0.70%</td>
<td>S, P 0.30% 1.00%</td>
<td>S, S 0.02% 0.50%</td>
<td></td>
</tr>
<tr>
<td>Ester (2) P, N 0.20% 0.70%</td>
<td>S, N 0.30% 1.00%</td>
<td>N 0.02% 0.50%</td>
<td></td>
</tr>
<tr>
<td>Ester (3) P, N 0.20% 0.70%</td>
<td>S, P 0.00% 1.00%</td>
<td>N, S 0% 0%</td>
<td></td>
</tr>
<tr>
<td>Ester (4) P, N 0.20% 0.70%</td>
<td>S, N 0.00% 1.00%</td>
<td>N 0% 0%</td>
<td></td>
</tr>
<tr>
<td>Mineral oil (2) P, N 0.20% 0.70%</td>
<td>S, N 0.30% 1.00%</td>
<td>N 0.02% 0.50%</td>
<td></td>
</tr>
</tbody>
</table>

3.4.2 Operating parameters for the Plint & Partner High Frequency Friction apparatus

The Plint & Partner reciprocating high frequency friction apparatus described earlier was used to evaluate the wear properties (see paragraph 3.1). Two different operating cases were tested, i.e. Case I representing a severe contact and Case II a milder contact. The milder contact in Case II was given by a higher maximum sliding speed that separate the surfaces better. Because of this milder contact, the sliding distance in Case II had to be extended by a factor 3 to get wear scars large enough to be evaluated. The dimensionless film thickness for a rectangular conjunction was calculated according to Pan and Hamrock [35];

\[
\tilde{H} = 1.714(W^{0.938}U^{0.484}G^{0.584})
\]  

(4)

where \(W\) is the dimensionless load parameter, \(U\) is the dimensionless speed parameter and \(G\) is the dimensionless material parameter. The film thickness parameter \(\tilde{H}\) was used to calculate the dimensionless film parameter \(\Lambda\) (Eq. (5)). The radius of the specimen cylinder (ø 6 mm) described in paragraph 3.1 was used to determine \(R_x\). The roughness of the surfaces in contact a and b are described by \(R_{ax}\) and \(R_{bx}\) with equal values to the test in paragraph 3.1. The operating properties are summarized in Table 8. The wear rate was evaluated by using the well-known Archard's law, Eq 1, described earlier.
\[ \Lambda = \frac{\bar{H} - R_s}{\left( R_{sat}^2 + R_{ps}^2 \right)^{1/2}} \]  \hspace{1cm} (5)

Table 8. Operating parameters.

<table>
<thead>
<tr>
<th>Case</th>
<th>Load [N]</th>
<th>Pressure [MPa]</th>
<th>Temperature [°C]</th>
<th>Maximum sliding speed [m/s]</th>
<th>Sliding distance [m]</th>
<th>( \Lambda )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case I</td>
<td>50</td>
<td>367*</td>
<td>90</td>
<td>0.056</td>
<td>21.6</td>
<td>0.5</td>
</tr>
<tr>
<td>Case II</td>
<td>50</td>
<td>367*</td>
<td>90</td>
<td>0.34</td>
<td>65.4</td>
<td>1.5</td>
</tr>
</tbody>
</table>

*The width of the specimens was estimated to be 4.5 mm caused by chamfered edges of the specimens.

3.4.3 Synergies and antagonists in the lubricant blends.

The mineral based reference oil was tested under both operating conditions in Cases I and II. In Case I it was impossible to evaluate wear when mineral oil was used, because the wear scars were too small to be accurately measured. Additive combinations affecting wear at 95% confidence level are summarised in Table 9. The optimum additive mix is the solution with minimum wear. An additive mix such as EP+AW+Cu− means that EP and AW are blended at a high concentration, while Cu is blended at a low concentration, according to the blend levels in Table 7. The wear rate for the mineral oil in Table 9 was influenced by EP, AW and the interaction of EP*Cu. Since the wear rates were dependent on the test conditions (Cases I, II), the wear coefficients must not be quantitatively compared between the test conditions.

Table 9. Additive combinations affecting wear at 95% confidence level are marked with “X”.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester (1), Case I</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu−</td>
<td>1.39</td>
</tr>
<tr>
<td>Ester (2), Case I</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu−</td>
<td>0.560</td>
</tr>
<tr>
<td>Ester (1), Case II</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu−</td>
<td>2.38</td>
</tr>
<tr>
<td>Ester (2), Case II</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu−</td>
<td>1.87</td>
</tr>
<tr>
<td>Mineral (2), Case II</td>
<td>X</td>
<td>X</td>
<td></td>
<td>EP+AW−Cu−</td>
<td></td>
<td></td>
<td>EP+AW−Cu−</td>
<td>0.417</td>
</tr>
</tbody>
</table>

The results from Table 9 show that AW or Cu does not influence wear when using ester based oil. The additive concentration was changed and the Cu-passivator was completely removed. The new ester blends were identified as esters (3) and (4), see Table 7. This test showed that the AW used in ester (3) did not give any effect, whereas ester (4) gave effect on wear depending on EP, AW and the interaction between them, see Table 10. The absence of a Cu-passivator could explain the apparent “turn-on” of the AW effect at the higher concentration level, if an antagonism exists between the AW additive and the Cu-passivator. AW gave effect on wear in the range 0-0.3%. Adding more AW (0.3-0.7%) gave no additional effect.
Table 10. Additive combinations affecting wear at 95% confidence level are marked with “X”.

<table>
<thead>
<tr>
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<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester (3), Case I</td>
<td>X</td>
<td></td>
<td>EP+</td>
<td>EP+ AW-*</td>
<td>2.08</td>
</tr>
<tr>
<td>Ester (4), Case I</td>
<td>X</td>
<td>X</td>
<td></td>
<td>EP+ AW-*</td>
<td>1.91</td>
</tr>
</tbody>
</table>

*Note: No Cu-passivator was present in these lubricants. The AW additive at “low” level was held at 0%.

As an example of different wear coefficients some of the results are summarised in Fig. 15. The wear reducing effect from the EP additive in Case I is seen in the figure. Case I represents the severe contact. In Case II the EP additive made a difference, but not to the same degree as in Case I. The mineral oil gave approximately four times lower wear coefficients than the ester based lubricants. None of the tested additives could make TMP-Oleate comparable with mineral oil due to wear rates in the tested sliding contact. This is in agreement with the results obtained by Rieglert and Kassfeldt [23].

Fig. 15. Wear coefficient for Case I (left) and Case II (right).

The examined surfaces in Case I revealed that ester based lubricants with a low concentration of EP generally gave very mild abrasive wear (the other additives had no influence). The surface layer was mildly sheared off in a tribochemical way, resulting in a smooth and shiny surface without any marks, see Fig. 16a.

The high concentration of EP resulted in a smooth surface with a protective coloured layer consisting of products of phosphorus, nitrogen and iron, see Fig. 16b.

The surfaces in Case II were not greatly influenced by the EP concentration. The base fluid gave surfaces similar to the surface described in Fig. 16a.

The most efficient combination to reduce wear in TMP-Oleate was found when an amine neutralised phosphoric acid ester (EP) was blended with triphenyl phosphothionate (TPPT)(AW) and thiadiazole derivate (Cu-passivator). The combination of amine neutralised phosphoric acid ester as an EP additive and methylene bis (dithiocarbamate) as an AW additive with or without tolyl thiadiazole derivate (as Cu-passivator) proved to be less efficient in reducing wear using TMP-Oleate.
Fig. 16. From Case I the wear scars on the upper cylindrical specimen were photographed in a Scanning electron microscope. The mid area in the micrographs is the worn surface where the sliding direction is horizontal. At each side of the wear scar the original grinding marks can be seen. Phosphorus reacts with the surface and creates a protective dark-coloured layer when the “EP” content is high, Figure b. When the “EP” content is low the phosphorus does not form any protective layer, Figure a. Neither surface suffers from severe damage.

Using TMP-Oleate gave up to 12 times higher wear compared to the tested mineral oil, the highest wear rate corresponding to a removal of 65 nm/sliding meter. It is clear that the EP additive reduced the wear rate in both mineral and synthetic formulations. In the rail application here discussed higher wear rates were of interest.

A rough calculation on a rail track exposed to 1.4 million load cycles, a rolling slip estimated to be 5 ‰ and a 10 mm diameter contact spot gives about 3.4 $\mu$m/sliding meter (wear rates from paper C).

It is concluded from this paper that synthetic base fluids without EP additives gave interesting wear rates. EP additives reduce the wear rates and protect the surfaces, though they are not recommended in this application. The addition of AW additives needs to be further evaluated. It is recommended to investigate milder additives or pure synthetic base fluids to get higher wear rates without the risk of seizure or severe damage on the surface.

3.5 Tribochemical wear at the gauge face and gauge corner

The next step was to evaluate synthetic esters with friction modifiers. The friction modifiers here tested were different types of fatty acids and could also be characterized as mild AW additives. The main objective of paper E was to obtain higher wear rates without damaging the surface. A pure sliding contact between specimens made from steel was evaluated.

3.5.1 Controlled wear evaluated in a reciprocating machine aimed for the gauge face (paper E)

The contact geometry employed was a cylinder-on-flat line contact, simulating the sliding between gauge face and wheel flange. The radii in a typical gauge face wheel flange contact are smaller compared to the contact at the rail head. The transversal forces are definitely smaller at the gauge face compared to the normal load but the sliding distance is much higher at the gauge face. The contact pressure in this test was about 316 MPa. The reciprocating motion in the test apparatus is somewhat similar to the sliding motion in the reverse direction at the gauge face that occurs on a track having traffic in both directions. The specimens are manufactured from rail steel of type 900A and 1100 with similar geometries as described in paragraph 3.1. The test parameters are summarized in Table 11.
### Table 11. Test parameters.

<table>
<thead>
<tr>
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<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>316</td>
<td>0,11</td>
<td>265</td>
<td>90</td>
<td>0,31 / 0,86</td>
<td>900A vs. 900A</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1100 vs. 1100</td>
</tr>
</tbody>
</table>

Two synthetic esters were used as base fluids, i.e. a TMP-Oleate (TMP-Oleate was also used in *paper D*) and a TMP-C$_8$-C$_{10}$. TMP is trimethylol propane (the polyol alcohol part of the ester), oleate is the monounsaturated C$_{18:1}$ fatty acid, and C$_8$-C$_{10}$ is the mixed fraction Octanoic-Decanoic (Caprylic-Capric) saturated fatty acids (making up the acid part of the ester).

The choice of esters in this study was partly based on wear results from the experiments made in *paper D*. Two synthetic esters were chosen to examine if a non polarity index NPI gave any synergism or antagonism between the different free fatty acids and the ester base fluid. TMP-Oleate has a high NPI of about 185 whereas the TMP-C$_8$-C$_{10}$ is significantly more polar, with an NPI of about 65. The properties of the two ester based fluids are summarized in Table 12.

### Table 12. Base fluids and their physical and chemical properties.

<table>
<thead>
<tr>
<th>Synthetic ester</th>
<th>NPI</th>
<th>Formula</th>
<th>$\rho$ @ 15 °C [kg/m$^3$]</th>
<th>η @ 90 °C [cP]</th>
<th>λ</th>
<th>h (mm) j [μm] 90 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>TMP-Oleate</td>
<td>185</td>
<td>C$<em>{60}$H$</em>{110}$O$_6$</td>
<td>920</td>
<td>16.454</td>
<td>0.016</td>
<td>14.6</td>
</tr>
<tr>
<td>TMP-C$<em>8$-C$</em>{10}$</td>
<td>65</td>
<td>C$<em>{64}$H$</em>{64}$O$_6$</td>
<td>945</td>
<td>7.016</td>
<td>0.008</td>
<td>7.5</td>
</tr>
</tbody>
</table>

Six free fatty acids were chosen as performance additives. Fatty acids such as Oleic and Stearic acids have been used as friction modifiers in lubricants for a long time. The attraction to the surface is stronger for acids with short chain lengths. The function of fatty acids in mineral oils is known, though their functionality in synthetic esters is not well explored. Three of the fatty acids were saturated mono-acids with different, straight carbon chain lengths (Stearic acid C$_{18}$, Decanoic (Capric) acid C$_{10}$ and Octanoic (Caprylic) acid C$_{8}$), i.e. one was a mono-unsaturated straight chain fatty acid (Oleic acid C$_{18:1}$) while the other two were dibasic acids with intermediate carbon chain length (Sebacic acid, C$_{10}$ and Azaleic acid, C$_9$), see Table 13. Each fatty acid was blended at 2 wt-% with one ester at a time, giving 12 synthetic ester formulations in all. A blend level at 2% was assumed to give a significant effect, see Weller and Perez [36, 37].

### Table 13. Properties of the free fatty acids.

<table>
<thead>
<tr>
<th>Chemical name</th>
<th>Formula</th>
<th>Melting point [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oleic acid</td>
<td>C$<em>{18}$H$</em>{34}$O$_2$</td>
<td>13.4</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>C$<em>{18}$H$</em>{36}$O$_2$</td>
<td>69.3</td>
</tr>
<tr>
<td>Decanoic acid (Capric)</td>
<td>C$<em>{10}$H$</em>{20}$O$_2$</td>
<td>31.5</td>
</tr>
<tr>
<td>Octanoic acid (Caprylic)</td>
<td>C$<em>{8}$H$</em>{16}$O$_2$</td>
<td>16.3</td>
</tr>
<tr>
<td>Sebacic acid</td>
<td>C$<em>{10}$H$</em>{18}$O$_4$</td>
<td>134.5</td>
</tr>
<tr>
<td>Azaleic acid</td>
<td>C$<em>{9}$H$</em>{16}$O$_4$</td>
<td>100</td>
</tr>
</tbody>
</table>

The present experiments were performed at a bulk temperature of +90 °C. In rail/wheel contacts the rail temperature will normally range from −45 °C to +55 °C. An earlier study by Bowden and Tabor [38] showed a transient point for fatty acids blended in mineral oil. The surface layer undergoes thermal decomposition and the wear protection will be reduced above the transient point, normally ~50 °C above the fatty acid’s melting temperature. Below this temperature the fatty acids will work satisfactorily. Fatty acids are attracted by chemical bonding and not chemical reaction and need therefore no additional energy in the form of any bulk temperature or asperity flash temperature. At 90
°C, most fatty acids were solved in the solutions without exceeding the transient temperatures too much. Sebacic and Azaleic acid had melting points above 90 °C. These solutions were preheated up to 140 °C to solve the fatty acids into the esters. Although Sebacic and Azaleic acid solidified slightly when the temperature fell below to 90 °C, they were mainly dissolved.

3.5.2 Results for the gauge face

Wear rates and the morphology of worn surfaces were two main aspects considered in this study. When wear rates were evaluated in the Plint & Partner machine the cylinder specimen that reciprocates upon the flat specimen due to the stroke length and area in the contact got more concentrated wear. Wear was evaluated by a multifactor ANOVA analysis. Three components were considered in this analysis: material type (composition and hardness), base fluid type and free fatty acid type.

A box plot presents the results in Fig. 17 and Fig. 18. The box includes 50% of the measurements (this is called the inter quartile). The median is marked with a horizontal line in the box; the mean value is marked with a +. The whiskers include the lowest and the highest values if they are at most 1.5 inter quartile from the box. Outliers are located 3 inter quartiles from the box. The figure shows that pure esters give high wear rates. No difference was detected between the ester types. The addition of free fatty acid gave a significant reduction in wear. When the wear differences between the synthetic ester with fatty acids were analyzed, pure synthetic ester was excluded from the analysis. Esters with addition of Oleic (with long carbon chain length) and Azaleic acids (di-basic acid) resulted in the lowest wear rates. Octanoic acid (mono-acid with shortest carbon chain length) resulted in the highest wear except for pure ester.

Fig. 17. The box plot shows how the wear rates are affected by adding fatty acids.

Fig. 18. The box plot shows how the wear rates are affected by different fatty acids.
Scanning electron microscope (SEM) examination of the worn surfaces showed that pure synthetic ester gave generally mild abrasive wear (Fig. 19a). Fatty acids added to the synthetic ester then made the surfaces smoother. Synthetic esters with Oleic acid gave a surface covered by some fine pits and micro-crack-like surface flaws; see Fig. 19b). Blends with Azaleic or Stearic acid resulted in very smooth surfaces, see Fig. 19c (Note that micrograph 19c has a different magnification). The observations from the microscopic examination of worn surfaces are summarized in Table 14.

![Micrographs of the worn surfaces with vertical sliding direction lubricated by; a) Pure TMP-C8-C10, b) TMP-C8-C10 and Oleic acid c) TMP-Oleate and Stearic.](image)

**Fig. 19.** Micrographs of the worn surfaces with vertical sliding direction lubricated by: a) Pure TMP-C8-C10, b) TMP-C8-C10 and Oleic acid c) TMP-Oleate and Stearic.

**Table 14. Surface characteristics.**

<table>
<thead>
<tr>
<th>Fatty acid</th>
<th>900A</th>
<th>1100</th>
<th>900A</th>
<th>1100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
</tr>
<tr>
<td>Oleic</td>
<td>Surface partly covered by thin layer</td>
<td>Uniform surface covered by thin layer</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
</tr>
<tr>
<td></td>
<td>Tribochemical wear, some scoring marks and micro pits</td>
<td>Tribochemical wear, some scoring marks and micro pits</td>
<td>Tribochemical wear, some scoring marks and micro pits</td>
<td>Tribochemical wear, some scoring marks and micro pits</td>
</tr>
<tr>
<td>Stearic</td>
<td>Very smooth surface</td>
<td>Very smooth surface</td>
<td>Very smooth surface</td>
<td>Very smooth surface</td>
</tr>
<tr>
<td></td>
<td>Tribochemical wear</td>
<td>Tribochemical wear</td>
<td>Tribochemical wear</td>
<td>Tribochemical wear</td>
</tr>
<tr>
<td></td>
<td>And some scoring marks</td>
<td>And some scoring marks</td>
<td>And some scoring marks</td>
<td>And some scoring marks</td>
</tr>
<tr>
<td>Decanoic</td>
<td>Surface covered by thin layer</td>
<td>Tribochemical wear and scoring marks</td>
<td>Tribochemical wear and scoring marks</td>
<td>Tribochemical wear and scoring marks</td>
</tr>
<tr>
<td></td>
<td>Tribochemical wear and some scoring marks</td>
<td>Tribochemical wear and scoring marks</td>
<td>Tribochemical wear and scoring marks</td>
<td>Tribochemical wear and scoring marks</td>
</tr>
<tr>
<td></td>
<td>Octanoic</td>
<td>Surface partly covered by thin layer</td>
<td>Tribochemical wear and some scoring marks</td>
<td>Tribochemical wear and some scoring marks</td>
</tr>
<tr>
<td></td>
<td>Sebacic</td>
<td>Surface partly covered by thin layer</td>
<td>Tribochemical wear and some scoring marks</td>
<td>Tribochemical wear and some scoring marks</td>
</tr>
<tr>
<td></td>
<td>Azaleic</td>
<td>Tribochemical wear and some scoring marks</td>
<td>Tribochemical wear and some scoring marks</td>
<td>Tribochemical wear and some scoring marks</td>
</tr>
<tr>
<td></td>
<td>Tribochemical wear and smooth surface covered by surface layer</td>
<td>Tribochemical wear and very smooth surface</td>
<td>Tribochemical wear and very smooth surface</td>
<td>Tribochemical wear, quite smooth surface partly covered by thin layer</td>
</tr>
</tbody>
</table>

Here, pure synthetic ester resulted in mild abrasive wear and a high wear rate corresponding to ~120 nm/ sliding meter. The use of fatty acids at the rail gauge face is interesting because the surfaces became very smooth and the wear rate was low. At the gauge corner the ability of fatty acids to hold the surface smooth is good. The blend with Octanoic acid gave the highest wear rate among the fatty acids, though the surface did show some marks. Blends with Azaleic and Stearic acid gave very smooth surfaces indicating mild wear, with wear rates of 11 nm/sliding meter for Azaleic and 20 nm/sliding meter for Stearic.

The function is hereby examined in a laboratory pure sliding contact that simulates the gauge face/wheel flange contact. These lubricants were aimed to work at the gauge face as well as the rail ball; therefore, the wear behaviour had to be examined in a highly loaded rolling sliding contact. Pure
TMP-Oleate and blends with Stearic and Azaleic were evaluated in the following rolling/sliding test. The surface, lubricated by TMP-C_{8}C_{10} with oleic acid, showed some micro pits. To see if this blend gave more fatigue in a rolling/sliding contact is why it was also chosen for the following test.

It is concluded that pure ester did not give a smooth surface. Using a pure synthetic ester can be a risk at the rail gauge face and rail ball. Adding fatty acids is one solution to keep the surfaces smooth and avoid accelerated wear in a pure sliding contact.

3.5.3 Controlled wear evaluated in a Twin-disc machine aimed for the gauge corner (paper F)

The above described synthetic formulations were reformulated with a lower proportion of fatty acid (0.3wt-%). Poly-alpha-olefin (PAO) was chosen as the free-from-additives reference oil that represents the base fluids ability to build up lubricating film. The idea was to simulate the rolling/sliding contact between the rail gauge corner and the wheel root to find out the result of wear rates in such contact. Fatigue might appear during the test; if the wear rates became sufficiently high the idea was that no fatigue cracks would appear.

The test apparatus used in this investigation was a twin-disc machine known as the SUROS machine (Sheffield University Rolling Sliding machine). Fletcher and Beynon [39] offer an extensive description of the machine, but a brief description is made in paper F, see Fig. 20.

![Fig. 20. Schematics of the Twin-disc machine 1) lathe, 2) frequency controlled AC motor, 3) torque transducer, 4) hydraulic cylinder, 5) rail disc, 6) wheel disc, 7) gear, 8) bearing assemblies, from Fletcher and Beynon [39].](image)

The operating parameters chosen for this twin-disc test are summarized in Table 15. The maximum Hertzian pressure $p_{0}$ in the rolling/sliding line contact was calculated by the Timoshenko and Goodier’s formula [40].

$$p_{0} = 0,418 \left(\frac{WE}{BR}\right)^{0.5}$$

(6)

where $W$ is the load, $B$ is the line contact length, $E$ is the Young’s modulus and $R$ the disc radius. The disc surface temperature $T$ is measured with an infrared temperature probe. $S_{c}$ is the cumulative slip chosen for the sliding part in this contact, and $N_{rail}$ is the rolling speed for the rail disc. The wheel disc was manufactured from a rail car wheel of the type “Blue light” and the rail disc was manufactured from rail steel type 900A. Both discs had widths of 10 mm and diameters of $41.97 \pm 0.01$ mm. The bulk hardness measured in $HV_{100g}$ is presented along with the other operating properties.
Table 15. Operating parameters.

<table>
<thead>
<tr>
<th>Po [MPa]</th>
<th>T [°C]</th>
<th>S_i [%]</th>
<th>N_m [rpm]</th>
<th>Bulk hardness HV_{1000g}</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>32±2</td>
<td>-5.0 %</td>
<td>398</td>
<td>Rail disc (900A)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Wheel disc (Blue light)</td>
</tr>
</tbody>
</table>

The initial surface roughness for the disc specimens was Rq 0.8-1.2 μm. The lubricating regime was determined to be in the boundary region and the dimensionless film parameter Λ was determined as close as possible to the pure sliding contact in paper E. The lubricant blends and running conditions used in this test are specified in Table 16, with the base fluid viscosity, the calculated minimum film thickness and the dimensionless film parameter. Observe that the dimensionless film parameter changed during the test when the surfaces became smoother (see Λ after 100 000 cycles in Table 16).

Table 16. Lubricating fluids and test data.

| Label | Base fluids | Fatty acids [0.3wt%] | η (cP) at 32 °C | h_{min} [μm] | Λ
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Initial</td>
<td>100 000 cycles</td>
<td></td>
</tr>
<tr>
<td>S1</td>
<td>TMP-Oleate</td>
<td>Stearic acid</td>
<td>64.13</td>
<td>0.156</td>
<td>0.11</td>
</tr>
<tr>
<td>S4</td>
<td>TMP-C_{8-10}</td>
<td>Oleic</td>
<td>27.28</td>
<td>0.0830</td>
<td>0.056</td>
</tr>
<tr>
<td>PAO</td>
<td>PAO</td>
<td></td>
<td>53.14</td>
<td>0.147</td>
<td>0.099</td>
</tr>
</tbody>
</table>

Before and after a test, the discs were cleaned and then separately weighed on a high-resolution scale (0.0001 g). The disc surfaces were examined with an optical microscope and a scanning electron microscope (SEM). This investigation indicated the wear type and the amount of cracks. To evaluate the influence from friction and plastic deformation of the surfaces, cross sections from the rail discs were cut out. The surface roughness Rq was measured perpendicular to the sliding direction before and after running the tests. The friction was monitored during all tests. The hardness of the rail discs was evaluated at the surface and subsurface with a micro Vickers hardness indenter. Each test condition was tested once except for a couple of conditions that were repeated.

3.5.4 Result of controlled wear at the gauge corner

The wear rates from the twin-disc tests summarised in Fig. 21 separate the rail and wheel disc wear for different lubricating fluids. Material transfer between the discs is not considered. This test showed that by using synthetic formulations S3 and S4 the wear rate was of the same magnitude as PAO, 0.003-0.009 g (or 0.012-0.035 nm/cycle) in the 25 000 cycle tests. With these lubricants, wear rates gradually decreased due to "run-in" with the 100 000 cycle tests giving wear rates in the range ~0.01 nm/cycle. The wear rate 0.035 nm/cycle corresponds to removing 2134 nm/sliding meter in the 25 000 cycle test. The results from the disc tests lubricated with S1 and S2 showed partly high wear rates, but with large differences. The explanation is that the protecting film is close to rupture for these lubricants. A small variation in surface roughness may cause this.
The coefficient of friction depended on two mechanisms. The surface roughness initially gave relatively high friction until the asperities were flattened down. After the “run in” phase the influence of the asperities became weaker with the lubricants’ properties influencing the coefficient of friction, see Fig. 22. Compared to pure esters the addition of fatty acids did not influence friction. A significant difference was found between PAO and the ester formulations. When the same lubricant solution was tested for 25 000 and 100 000 cycles, it was expected that the coefficient of friction would not diverge much. However, the friction results indicated a significant divergence that was suspected to be caused by the discs surface roughness. A linear regression analysis was made to find out if there was any correlation between the initial surface roughness and the friction coefficient. In Fig. 23 a-d, the monitored coefficients of friction after 800, 3500, 25 000 and 100 000 cycles were plotted against the initial surface roughness (wheel disc) of each individual disc couple. The fitted black line is drawn by least square analysis. Any slope on the fitted line indicates a relation between surface roughness and friction. The analysis showed a moderately strong relation between the initial surface roughness (rms) and the coefficient of friction at 800, 3500 and 25 000 cycles, but not after 100 000.

Fig. 22. In the left diagram the coefficient of friction for the 25 000 cycle test is presented. In the right diagram the coefficient of friction was monitored for the 100 000 cycle test.
Fig. 23. The correlation between the surface roughness and the monitored coefficient of friction was analysed using linear regression. The test showed a moderately strong relation in Figures a-c while Figure d showed almost no correlation.

The coefficient of friction is initially at its highest level and decreases during the “run in” phase. If the harder disc had a rough surface the friction decreased slowly, whereas the friction coefficient diminished rapidly if the harder surface was initially smooth. The synthetic formulations S1 that ran for 25 000 cycles and S2 for 100 000 cycles did “run out” instead of being “run in”. “Run out” means that the surfaces are full of cracks combined with high wear rates. The discs that “run out” had higher initial friction compared to ester based lubricants. PAO showed the highest long term friction, though these rail discs did not “run out”. The result is that “run out” of surfaces appears when the harder surface is rougher compared to the softer surface. High local friction gives local plastic deformation at the surface and causes cracks.

When examining the surfaces through the optical microscope and the scanning electron microscope (SEM), it was found that the surfaces could be divided into two groups, since PAO differed from synthetic ester formulations. SEM micrographs show the general differences between the surfaces. In Fig. 24a, the surface lubricated by S1 for 100 000 cycles was covered by cracks and was representative of all surfaces lubricated by synthetic ester formulations tested here. Fig. 24b shows a surface lubricated by PAO after 100 000 cycles with far less cracks than any surface lubricated by synthetic ester formulations.
Fig. 24. SEM picture a) shows a surface lubricated by S1 during 100,000 cycles. The surface is covered by cracks. SEM picture b) shows a surface lubricated by PAO during 100,000 cycles. In general, surfaces lubricated by PAO in the present test restrain far less cracks than surfaces lubricated by any of the synthetic ester formulations tested.

The hardness of the subsurface is plotted vs. the depth (Fig. 25). Polynomials were fitted to the subsurface hardness to evaluate the hardness distribution, indicating a local hardness maximum slightly lower down in the material, which declined as the measurements approached the surface. The coefficients of friction were the main differences between the lubricants. PAO showed an overall higher friction coefficient ($\mu_{\text{average}}=0.050$) whereas S2 showed a lower friction coefficient ($\mu_{\text{average}}=0.034$). The surface hardness was measured and showed that rail discs lubricated by PAO resulted in a harder surface compared to rail discs lubricated by synthetic formulations of the type S1 and S2 (see Fig. 26). The box plot is explained in paragraph 3.5.2. From Fig. 26 it was shown that the surface lubricated by S1 and S2 had differing hardness while PAO had a more even and higher hardness at the surface. The discs lubricated by S1 and S2 achieved probably differing hardness through local plastic deformation.
A rail disc cross section lubricated by PAO was examined in an SEM; see Fig. 27. The micrograph shows how the laminar pearlitic structure below the white dotted line at depth ~8 \( \mu \text{m} \) seems unaffected. The hardness measurements shown in Fig. 25 decline as the measurements approach the surface (hardness was not measured closer than 23 \( \mu \text{m} \) from the surface). Above the white dotted line the laminar pearlitic structure has shifted plastically in the sliding direction. The hardness measurements from the rail disc surfaces show this surface layer to be work hardened.

![Fig. 27. Above the white dotted line the pearlitic microstructure was deformed plastically in the sliding direction while the structure below this line was not.](image)

The wear rates in this rolling/sliding test normally remained under ~2.2 \( \mu \text{m} / \text{sliding meter} \) or 0.035 nm/load cycle (except for the surfaces that “run out”). The examination made by Fletcher and Beynon [26] showed crack growth of 0.3 nm/load cycle or ~7 \( \mu \text{m} / \text{sliding meter} \) for the lubricated conditions, while intermittent lubrication gave 52 nm/load cycle or 2 mm/sliding meter. Crack growth speed was slightly faster compared to the wear rate. However, if intermittent lubrication occurs the crack speed will greatly increase; hence, continuous lubrication is recommend.

Conclusions from this rolling sliding test showed that formulations based on synthetic ester could be placed into the group with generally low friction and surfaces somewhat covered by cracks. Addition of fatty acids showed no significant effect on friction or wear. PAO, on the other hand, gave higher friction and surfaces with significantly fewer surface cracks.

The low friction of the synthetic ester was believed to result in less plastic surface deformation. Hardness measurements at the surface indicated that the hardness fluctuated compared to surfaces lubricated with PAO. It seems that some spots at the surface have been work hardened while other spots are not. In addition the Scanning electron microscope showed significantly more surface cracks. It is reasonable that hard asperities result in spots exposed to severe conditions even if the contact friction is low. Variations of strain at the surface lead to crack initiations.

4 Discussion

Environmentally adapted greases aimed for the gauge face were tested in papers A and B. The environmentally adapted lubricant did not give higher wear rates compared to the standard grease in the field test. From the laboratory test the standard grease showed slightly lower wear. The non-lubricated winter period resulted in the majority of the wear, while wear from the lubricated summer period could be considered negligible. In paper B, year-around lubrication was found to have the highest potential to further extend rail life. From the tests it was possible to calculate a theoretical increase of the rail life from ~130 MGT to the range 450-800 MGT for sharp curves (radius 350 m).

No reliable method is so far known of how to apply the lubricant at the rail track when the temperature is low. By using onboard systems the equipment will be located in warm surrounding and could be one possible alternative. An onboard system has continuous power supply and is easier to maintain compared to wayside lubricators.
The main objective of rail lubrication is to reduce wear at the gauge face and the wheel flange. Another view was considered in this thesis. The idea was to find a lubricant that also protects the surface from head checks. The first part of the thesis deals with the gauge face, while the other deals with the rail ball. These demands can contradict each other. Low wear is best at the gauge face, while higher wear rates can be a way to reduce head checks at the rail ball.

In a pure sliding contact in paper D with hard steel specimens, pure synthetic esters with the tested AW and EP additives gave smooth surfaces. Blends with only AW additives (triphenyl phosphothionate or methylene bis (dithiocarbamate)) gave wear rates around 12 times higher than the tested mineral based lubricants, with the surfaces still remaining smooth. When softer steel specimens (rail steel) were used pure TMP-Oleate and TMP-C6-C10 resulted in abrasive wear with an accelerated wear rate. The addition of fatty acids gave smooth surfaces for both the synthetic esters and remarkably lower wear rates. In the rolling/sliding contact pure synthetic ester and blends with fatty acid were tested. The pure TMP-Oleate did not result in abrasive wear as in the sliding test. TMP-Oleate gave here a surface structure similar to the surface given from TMP-C6-C10 with oleic acid. In the rolling/sliding test PAO gave the best surface with the least crack initiations. TMP-Oleate with Stearic or Azaleic acid resulted in more crack initiations.

There is obviously a difference between the rolling/sliding contact and the sliding contact. The rolling sliding contact had a far higher contact pressure and the plastic deformation resulted in work hardening of the material. In the pure sliding contact the tested rail steel never work hardened. Despite the higher contact pressure in the rolling/sliding contact TMP Oleate protected the surfaces. In the sliding contact TMP-Oleate did not protect the surfaces. The AW additives from paper D or Oleic acid in TMP-Oleate could be the combination that fulfils the demands at the gauge face and at the rail ball. The AW additives were unfortunately not tested at the rail material and therefore require further tests.

Table 17 provides a rough overview of the results. The wear was recalculated to represent the removed depth perpendicular to the contact surface. It is possible to compare this kind of wear rate with crack growth results from other researchers.

Lubricated tests in a twin-disc machine made by Fletcher and Beynon [26] showed a lubricated rolling sliding contact that gave very low crack growth (0.3 nm/load cycle). A much higher crack growth was achieved when the contact was intermittently lubricated, as seen in Table 17. The wear rates achieved in the twin-disc test (paper F) were too low to remove the crack initiations. A wear rate of 0.035 nm/load cycle has been recalculated to 2134 nm/sliding meter; see Table 17. The contact pressure of 1.5 GPa in the twin-disk test did influence the hardness of the surface. The initial hardness of the rail disc was HV 270-290, but increased to HV 450-550 after 25 000 cycles. The pure sliding test in paper E had lower contact pressure (316 MPa), with wear rates ranging from 15-120 nm/m and an initial hardness similar to the twin-disc test. The pure sliding contact had lower contact pressure, similar dimensionless film parameter, and a softer surface that did not work harden during the test. The addition of fatty acids gave a significant difference in wear rates. The fatty acids formed a film on the surface and did not wear off.

In the twin-disc test conducted in paper F, the addition of fatty acid did not show any influence on the wear rate. However, using ester resulted in low friction. The wear rates found in the twin-disc machine remained at 2134 nm/m. As a comparison the wear rate between the rail ball and the gauge corner in paper C was recalculated with some general assumptions and remained at ~3400 nm/m. The wear rates were of the same magnitude.
### Table 17. Wear rates versus crack growth.

<table>
<thead>
<tr>
<th>Properties/Test type</th>
<th>Amount of lubricant</th>
<th>Type of lubricant</th>
<th>Who made the test</th>
<th>Crack depth perpendicular to surface</th>
<th>Crack growth</th>
<th>Wear [nm/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crack growth</td>
<td>Twin-disc</td>
<td>Flooded lubricated</td>
<td>Mineral oil</td>
<td>Fletcher and Beynon [26]</td>
<td>7 μm/m</td>
<td>0.3 nm/cycle</td>
</tr>
<tr>
<td>Crack growth</td>
<td>Twin-disc</td>
<td>Intermittent lubricated</td>
<td>Fletcher and Beynon [26]</td>
<td></td>
<td>2000 μm/m</td>
<td>52 nm/cycle</td>
</tr>
<tr>
<td>Wear rate</td>
<td>Twin disc</td>
<td>Flooded lubricated</td>
<td>Synthetic ester, PAO</td>
<td>Paper F</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>Wear rate</td>
<td>Plint &amp; Partner</td>
<td>Flooded lubricated</td>
<td>Pure ester</td>
<td>Paper E</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>Wear rate</td>
<td>Plint &amp; Partner</td>
<td>Flooded lubricated</td>
<td>Synthetic ester + stearic acid</td>
<td>Paper E</td>
<td>——</td>
<td>——</td>
</tr>
<tr>
<td>Wear at gauge corner</td>
<td>Malmbanan lubricated by onboard system</td>
<td>Paper C</td>
<td></td>
<td>——</td>
<td>——</td>
<td>3400</td>
</tr>
</tbody>
</table>

In the rolling/sliding test in paper F synthetic ester formulations caused more rolling contact fatigue than PAO, probably due to the work hardening of the material. PAO seems to strain harden the material in this specific test; therefore, material lubricated by PAO seems to withstand fatigue better.

The price ratio between mineral oil, natural esters (as rapeseed oil) and synthetic lubricants are roughly 1:2:5. The use of natural esters instead of mineral oil is better for the environment, which motivates the higher price. The use of synthetic lubricants in the rail/wheel contact has not been clearly motivated. Synthetic lubricants have better low temperature properties compared to natural esters and may thus be considered in arctic conditions.

### 5 Conclusive summary

The thesis has treated rail friction and wear phenomena using fully formulated greases and base fluids with different additives.

The laboratory tests showed that mineral oil based greases gave the lowest wear rate. In the field test, no difference between the environmentally adapted grease free from graphite and the earlier used mineral based grease with graphite could be detected when the wear rates at the rail gauge face were evaluated. The reason was that the lubricated period gave very little wear and to distinguishing the lubricants was therefore not possible. The wear in the field test occurred mainly during the non-lubricated winter period whereas the lubricated summer period contributed with very little wear. Mineral oil based greases have proven to give better wear performance in many laboratory tests and it can also be expected that the wear rates will be lower in rail/wheel applications. Grease without graphite avoids the risk of clogging the trackside lubricators. Hence, lubricants free from graphite increase the reliability of the equipment. It is recommended to use the environmentally adapted rapeseed grease tested here in trackside lubricators.

Belief from the field without proof stated that lubrication is not necessary during the arctic winter conditions. This is partly true because water and snow do reduce the wear rates. However, the wear rates measured here were several times higher during the winter, allowing for greater potential to extend rail life. Great savings could be made if adequate year-around lubrication is introduced. Such extended life may however lead to RCF problems such as head checking. If year-around lubrication
reduces wear rates, inspections for head checks become important. It is also recommended to plan for future grinding activities and introduce proper year-around lubrication.

For the conceptual base fluid studies the idea of this thesis was to find a lubricant that protects the gauge face and avoids crack initiation at the rail ball.

In a pure sliding contact the conditions are rather though and the base fluids need some type of additive to give good protection to the surface. Pure synthetic ester gave high wear rates and the surfaces showed mild abrasive wear. The tested EP additive (amine neutralised phosphoric acid ester) but also some of the fatty acids (Azaleic and Stearic acid) gave smooth surfaces and low tribochemical wear.

In a rolling sliding contact formulations based on synthetic ester can be placed into a group with generally low friction and surfaces somewhat covered by cracks. In this contact the fatty acids showed no significant effect on friction or wear. PAO on the other hand gave higher friction and surfaces with significantly fewest surface cracks.

Surface roughness influenced the friction at the beginning of each test. A rough surface of the harder disc resulted in high friction that slowly decreased with running distance and also an increased amount of surface cracks in the softer material. A smoother surface of the harder disc resulted in a rapidly decreasing friction. The surface roughness of the harder material is an important factor in rolling sliding contacts.

The Twin-disc tests showed that low contact friction does not necessarily mean that crack initiations are avoided in the surface. Lubrication by PAO gave discs with work hardened surfaces while the esters resulted in varying hardness over the surface. The results indicate that some base fluids are better to protect the surface from crack initiations.

In conclusion, the base fluid and the additives influenced not only the wear rate but also the ability for cracks to grow.

6 Future work

Different researchers have conducted many wear measurements of the rail. These measurements should be the base for a model describing rail degradation. This kind of model can be used at the real track to evaluate the economic effects of lubrication and provide recommendations.

There is a potential to lubricate year-round. Techniques to achieve longer life are needed. Lubrication from a maintenance vehicle or through the use of locomotives is interesting to develop. Another solution is to develop wayside equipment able to lubricate the rail flanges year-round.

The tests showed that base oils such as PAO and the tested synthetic esters resulted in different growth of the RCF cracks. Not surprisingly, differences between the base fluids exist. With the increase of head checking at the track, findings to reduce these kinds of damages are of course interesting. Evaluating different base oils should be done to better understand how the best rail lubricant should be mixed. Different additive solutions and their influence on head checks should also be tested.

Surface roughness seems to affect the friction in a rolling/sliding contact, as well as introduce surface damages. In a wheel/rail contact with insufficient or no-lubrication the rail gauge becomes rough; this effect will probably be similar at the wheels. Variations in surface roughness of both the wheels and rails are an interesting issue, since today’s rails become grinded to a higher extent.
References


Grease behaviour in a rail lubricating system exposed to arctic conditions

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ABSTRACT

An experimental and numerical approach to generate an insight into the behaviour of lubricating systems for flange lubrication on railroad curves is presented. The study investigated several greases (NLGI grade 00-000) developed specifically to lubricate a rail gauge face, including low environmental impact greases. The base oil viscosity ranged from 15 to 40 cSt at 40°C except for one lubricant that was a “bounded soap from synthetic ester”. Wear and friction were studied in a Cameron & Plint apparatus. To understand the grease’s behaviour at the nozzle outlet over a range of temperatures, a high-speed video camera was used to investigate a full-scale rail gauge face lubricator (Clicomatic type). Also the flow of grease in the fed pipe was measured and numerically modelled using a power-law. These experiments were carried out in a cold chamber. The results showed that it is possible to use environmentally adapted grease and it is also possible to use grease without graphite. The study showed that grease behaviour in a rail lubricating apparatus could be predicted through tests in a Bohlin CVO rheometer and a Cameron & Plint apparatus.

Introduction

In railroad applications, maintenance costs for wheels and rails subjected to heavy traffic, such as ore lines, can be up to fifty per cent of the total maintenance cost. In such systems, lubricants can be applied to the side of the rails to reduce friction and wear that occur between the wheel flange and the gauge side of the rail. With proper lubrication it is possible to reduce the rate of rail/wheel wear by a factor of 20 which leads to significant cost savings for both the vehicle and infrastructure owner, see Elkins et al [1].

Wear is most significant in curved track sections and curves with a radius of less than 600 m are often lubricated with stationary lubrication equipment (3000 pieces in Sweden). The design of the lubricator and it’s reliability is clearly important, however, maintenance of such equipment, often spread over great geographical distance and used under arctic condition, is also necessary. During 1996 the Swedish Railroad Association established that only 25% of the track lubricating equipment worked satisfactorily, see Hedlund and Hellström [2].

Two types of lubricating equipment are currently in production, both consist of a grease reservoir and a firing nozzle connected by a five-meter hydraulic feed tube. The difference between the two designs is how they are powered; one uses compressed gas whilst the other is powered electrically via a piston pump and a solenoid. The lubricating equipment used in the experiments was powered electrically. The equipment signals a passing train by detecting vibrations with an accelerometer, the firing nozzle is then activated and a jet of grease shot against the rail gauge face. It is recommended that the firing nozzle is located ca. 100 – 300 mm from the rail gauge face, although in practice most of the nozzles are located at a distance of 300 mm.

The main task of the grease is to reduce wear, which requires it to be effective in the small contact area between the wheel flange and the rail. It is also important that the grease is applied accurately. The grease used in such equipment must be designed for a wide range of different parameters. For example, in the northern part of Sweden the temperature can vary from −30°C up to +30°C depending on the season.

Another problem is oil separation during storage in the lubricator. The most widely used rail lubricant used in Sweden today is grease based on mineral oil with a graphite additive. One of the main problems with this lubricant is separation of the oil / graphite mix during storage under pressure due to
oil leaking through small gaps in the system whilst the graphite particles remain and clog the reservoir and the firing nozzle.

Method

Five different lubricant characteristics must be optimised to achieve satisfactory performance.

1) Good adhesion and low friction with minimal wear in mixed or boundary lubrication.
2) Environmentally friendly.
3) Resistance to oil separation during storage under pressure.
4) Fluid at low temperatures.
5) Easy to disperse using the firing nozzle.

To evaluate the lubricating apparatus three tests were performed. The first test was to evaluate what kind of grease works best in the small contact spot between wheel flange and rail. The tests evaluated wear and friction for mixed or boundary lubrication using a Plint & Partner high frequency apparatus. The test also investigated the effect of graphite in the grease because of the separation and clogging problem described earlier. The performance of environmentally adapted greases was also examined as the lubricant is often used in environmentally sensitive areas. The laboratory tests were complemented with a field test, which studied the effect of gauge face lubrication and the differences between lubricants concerning wear.

The second test considers effectiveness of the grease pump depends on working pressure, length and inner radius of feed tube and grease characteristics. Low temperatures can be a problem, especially for newly developed environmentally adapted lubricants. To investigate their behaviour under arctic conditions, the flow profile in a circular tube was evaluated in a cold chamber. Rheometer tests were also performed and the results compared with the measured flow profile. A power-law model was used as the basis for numerical computer simulations of the flow profile distribution and compared with experimentally determined profiles.

The third test studied the grease just outside the firing nozzle. The firing nozzle is typically located 300 mm from the rail and when a train passes a jet of grease is fired against the rail to lubricate the contact. To understand more about the behaviour of the firing nozzle and grease impact on the flange, a high-speed video camera was used to observe the firing and ballistic motion of the grease jet. These tests were also performed in a cold chamber to recreate realistic operating temperatures.

Tested lubricants

Five different greases, based upon mineral, ester and rape-seed oil, were used in the experiments. The base oil viscosities were in the range 15-40 [cSt] at +40 °C, Table 1.

<table>
<thead>
<tr>
<th>Grease</th>
<th>Base oil</th>
<th>Soap (wt%)</th>
<th>Base oil viscosity @ 40°C</th>
<th>Graphite</th>
<th>NLGI grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Synthetic ester</td>
<td>Not known</td>
<td>Not known</td>
<td>No</td>
<td>00-000</td>
</tr>
<tr>
<td>2</td>
<td>Rape seed oil and ester</td>
<td>Lithium complex (6-8)</td>
<td>40 cSt</td>
<td>Yes</td>
<td>000</td>
</tr>
<tr>
<td>3</td>
<td>Rape seed oil</td>
<td>Silica gel</td>
<td>38 cSt</td>
<td>No</td>
<td>00</td>
</tr>
<tr>
<td>4</td>
<td>Mineral oil</td>
<td>Lithium complex (6)</td>
<td>15 cSt</td>
<td>Yes</td>
<td>000</td>
</tr>
<tr>
<td>5</td>
<td>Synthetic ester</td>
<td>Lithium/ Calcium (5-15)</td>
<td>40 cSt</td>
<td>No</td>
<td>000</td>
</tr>
</tbody>
</table>
Wear and friction

At the wheel / rail contact point the contact pressure is of the order 0.8 to 2 GPa but the surfaces have a low relative velocity which implies a boundary or mixed lubrication regime. To protect the surface from severe wear, additives such as molybdenum disulphide or graphite are used. Due to the problems described earlier related to environmental concerns and separation problems, both laboratory and field tests were performed. The tests helped to find out whether it is possible to use environmentally adapted greases and greases without graphite. The laboratory tests used to study wear and friction were performed in a Plint & Partner high frequency apparatus 77B. This equipment slides two test pieces against each other in an oscillating motion. In between the test pieces at thin film of grease protects the surfaces which are held in contact by an applied force. The friction force was measured and an average friction coefficient determined as the mean value of the last 25 per cent of the friction values obtained during a test.

Properties under the tests:

1. Five greases was tested.
2. Applied force: 100 - 200 N.
3. Temperature: 36 ±1°C.
4. Max sliding speed: 0.17 m/s.
5. Oscillating amplitude: 2.3 mm.
7. Hardness of the test pieces: 850 HV.
8. Each setup repeated 5 times.

To complement the laboratory tests, a field test was also performed to study wear at gauge face in lubricated railroad curves. The method used was to measure the profile of the rail sections at the same point twice a year. From these measurements the wear / month could be determined. Two sections, each consisting of three railroad curves, were studied. The curves were equipped with railroad lubrication equipment and the curve radii and speed and volume of traffic were equal. At the first section the curves were lubricated with grease 3 and 4 and the second section was lubricated with grease 2 and 4. A Miniprof, a commercially device frequently used by railroad owners to measure rail profile or measure wear, was used to determine the wear rate. The Miniprof uses an optical method to detect the position of measuring arms 1 and 2 as the small wheel rolls over the railhead, see Fig. 1.

![Miniprof rail profile measuring device](image)

*Figure 1 Miniprof rail profile measuring device.*
Flow measurements

As discussed earlier, the effectiveness of the grease pump depends on working pressure, length and inner radius of the delivery tube and the grease characteristics. The grease flow profile for grease 3 in a circular tube has been evaluated in a cold chamber and compared with the rheological measurements from a Bohlin viscometer. The temperature varied in the range +25 °C to -20 °C.

Bohlin viscometer

An understanding of rheology is vitally important in the design of pump systems and for the prediction of lubricant flow using computer simulations. The most well behaved liquids are known as Newtonian liquids, and are characterised by the linear relation between shear stress ($\tau$) and shear strain rate ($\frac{\partial u}{\partial y}$), and the viscosity ($\eta$) is the constant of proportionately between them, see equation 1.

$$\tau = \eta \frac{\partial u}{\partial y}$$ (1)

Greases, however, display non-Newtonian behaviour i.e. they exhibit a non-linear relationship between the applied deformation rate and the corresponding stresses. Viscoelastic materials such as greases will react differently depending on how the load is applied over time. Using a commercial Bohlin CVO rheometer, a simple power-law model was developed to characterise the investigated grease. The power-law model has a non-linear relationship between the shear stress and the strain rate, see equation 2. One problem with this formula is when then the strain rate is small the shear stress is below the yield value. This behaviour can be neglected when the flow profile is calculated because of the velocity profile is integrated. All measurements were carried out using a rotating motion with cone-on-plate geometry.

$$\tau = K \left( \frac{\partial u}{\partial y} \right)^n$$ (2)

where $K$ is a constant depending on type of grease and has the units Pa.s$^n$ and $n$ is the power-law index.

Flow measurement, experimental set-up

To establish the flow profile in a tube for the rail lubricating systems, an experimental test rig was developed consisting of a hand pump, pressurised grease container, tube system, firing nozzle, camera and a linear scale, see Fig 2. The equipment was placed in a cold chamber when the tests were performed. The pressure container included a plastic bag containing the test grease. The bag was submerged in hydraulic oil pressurised by a hand pump. A one-metre thermoplastic PFA tube with an inner diameter of 7 mm was used to connect the pressure container and the firing nozzle. A section of the PFA tube was replaced by a glass tube with length 100 mm which was filled with grease and prepared with a coloured grease string markers. In Fig. 2 the flow profile is illustrated in the initial condition. After the glass tube was connected to the system the pressure was increased to 0.5 MPa and a picture of the initial position of the coloured marker strings was taken. The nozzle was then fired twice and a picture of the new marker string position was taken. The firing procedure was then repeated.
Evaluation of flow measurements

All pictures were digitalized using a scanner and stored in a computer for evaluation. The motion of the grease string was determined using an image processing system. A total of 14 coordinates were measured to model the curvature of each grease string. Four of these coordinates were fixed markings on the outside of the glass tube and were used as a reference. The other ten coordinates were taken at the front edge of the coloured grease strings and used to mark the flow profile, see Fig. 2. True position and form of the string was assured by using compensation for refraction. From the 14 coordinates it was possible to fit a polynomial that represents the shape of the string and hence tube flow profile using the commercial program Matlab.

A numerical method to establish the flow in a tube for a non-Newtonian liquid was considered and the following assumptions were made in the analysis:

1. Inertia and body force terms are neglected.
2. No pressure variations across the fluid.
3. The derivative of \( u \) with respect to \( r \) is much larger than other derivatives of velocity components.
4. The finite elements radial surface separated by \( dr \) is assumed to be \( (r+dr)dpdx = rd\phi dx \).

From those assumptions the Navier-Stokes equation can be reduced to:

\[
\frac{\partial p}{\partial x} = K \cdot \frac{\partial}{\partial r} \left( \frac{\partial u}{\partial r} \right)^n
\]  

Equation (3) can be modified to

\[
\left( \frac{\partial u}{\partial r} \right)^n = \frac{r}{K} \frac{\partial p}{\partial x} + c_1 \frac{K}{r}
\]  

By symmetry \( \frac{\partial u}{\partial r} = 0 \) when \( r=0 \), thus the constant \( c_1 = 0 \) and the velocity can be described as:

\[
u(r) = \frac{r^{\frac{1}{n}}}{\left( \frac{1}{K} \right)^\frac{1}{n} + \frac{1}{n} \frac{\partial p}{\partial x} + c_2 + n \cdot c_2}
\]

The boundary condition \( u=0 \) when \( r=R \) gives:
From Barnes et al [4] the relation between pressure gradient and flow-rate for a power-law model is given by:

$$\frac{\partial p}{\partial x} = \frac{2 \cdot K}{R} \left( \frac{(3n+1) \cdot Q}{\pi \cdot n \cdot R^3} \right)^n$$

(7)

The final velocity is given by:

$$u(r) = \left[ \frac{n}{n+1} \left( \frac{1}{R} \right)^\frac{1}{n} \left[ \frac{(3n+1)Q}{\pi n R^3} \right] \left[ \frac{n+1}{2} - \frac{n+1}{2} \right] \right] \left[ \frac{n+1}{2} \right]$$

(8)

If the velocity $u(r)$ and the flow $Q$ in equation (8) is multiplied by time $t$ and the result is length $s(r)$ and volume $V$. The volume obtained from the Matlab calculations allowed a power-law index to be developed from the experiments.

Flow through the firing nozzle and sureness of aim

The behaviour of the grease leaving the firing nozzle is separated into two parts; exit and ballistic trajectory. The jet of grease was examined using a high-speed camera just as it left the firing nozzle during a grease shot. Conventional railroad lubricating equipment consisting of a grease reservoir connected to a piston pump, a 5 metre hydraulic feed tube and a solenoid activated firing nozzle was used. The nozzle consisted of four drilled holes, each with diameter of 0.5 mm. With the high speed camera it was possible to determine the speed and the duration of the ejection, see Fig. 3.

The second part concerns the ballistic trajectory of the jet of grease. When the grease is fired against the gauge face it is important to hit the flange with some accuracy. The differences in the ballistic curve at different temperatures give some measure of the sureness of the aim. When the temperature is high the viscosity is low and the jet of grease has a high speed on leaving the firing nozzle and the ballistic curve tends to be flat. When the temperature is low the viscosity is higher with a lower exit speed from the nozzle and the grease spot then hits the gauge face somewhat lower. The firing nozzles are typically located at a distance of 300 mm from the gauge face. The test was therefore performed with a distance of 250 to 575 mm between the firing nozzle and the gauge face. A plastic board was used to mark how the grease jet hits at the varying distances. The temperature range for the tests was +20 to -20 °C.
Results

Wear and friction

From the measurements in the Plint & Partner apparatus it was determined that grease 1 gave the lowest average friction and grease 4 the highest, see Fig. 4 (b). The average friction coefficient for grease 2, 3, 4 and 5 were in the range 0.070 to 0.113 except for grease 1 whose result lay in the range 0.009-0.038. Grease 4 gave the lowest wear coefficient and grease 2 the highest, see Fig. 4 (a). The second best lubricant to reduce wear was grease 3. This is graphite free grease made of rape seed oil.

<table>
<thead>
<tr>
<th>Grease</th>
<th>Coefficient of friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0123456</td>
</tr>
<tr>
<td>2</td>
<td>0.0012345</td>
</tr>
<tr>
<td>3</td>
<td>0.0345678</td>
</tr>
<tr>
<td>4</td>
<td>0.0876543</td>
</tr>
<tr>
<td>5</td>
<td>0.1234567</td>
</tr>
</tbody>
</table>

Figure 4(a) Wear coefficient     (b) Coefficient of friction

The wear measurements from the field tests are presented in Fig. 5 (a) and (b). The results are from one railroad section consisting of three curves. The wear measurements in Fig. 5 (a) were separated into three time periods. The first period covered 96 month from 980701 when the rail was new and ended on 970813. The second period lasted for 10.5 month and ran from 970814 to 980702 and the third period lasted for 3.6 months from 980703 to 981020. The second period during the winter was unlubricated or lubricated with water/snow. The third period was completely lubricated. A number of methods to establish the rail wear are available. In this investigation the wear during each period was determined from three measurements. From two measured profiles the difference in three directions, w1, w2 and w3 were determined, see Fig. 5 (b). The vertical wear at the mid point of the rail is the w1 measurement. The horizontal wear 14 mm down from the top of the reference railhead is the w2 measurement and the difference at an angle of 45° is the w3 measure. The evaluated wear measurements from each time period was shown in Fig 5 (a). The wear rate on the flange (w2) was during the first period 0.082 mm/month, for the second period 0.045 mm/month and for the third period the wear rate was negative. A negative wear rate is the result of plastic deformation when the railhead material rolls down from the gauge corner down to the rail gauge face. There was no significant difference to the wear w1 between the time periods. From the field tests it was not possible to determine any significant difference between the performances of the different greases under test. Some differences between the greases tested in the Plint &Partner were found, but in the field test the lubricated period was too short to see any significant wear difference between the greases, at the gauge face.
Average wear/month

Figure 5(a) Wear rates from the field test. (b) Rail profiles.

Flow model

The measurements from the Bohlin CVO rheometer test were fitted to the power-law model in Fig. 6. From the curve fitting, the power-law index $n$ and the power-law constant $K$ are shown. The match of the curve fitting is given by the coefficient of determination or $R^2$. The power-law index was also determined from the tube test in the cold chamber. The results are presented in Table 2.

Figure 6. Bohlin rheometer test for grease 3.

The power-law index is about 30 % lower for the measurements in the tube compared with those given from equation (8).
Flow through the firing nozzle and sureness of aim

The speed on the jet of grease just outside the firing nozzle and the firing time were determined from the high speed video and are summarised in Table 3. The mean volume is the volume of one grease shot.

Table 2 Bohlin measurement results for grease 3.

<table>
<thead>
<tr>
<th>Temperature [°C]</th>
<th>K (Bohlin)</th>
<th>n (Bohlin)</th>
<th>R’ (Bohlin)</th>
<th>n (Tube)</th>
<th>n(Tube) – n(Bohlin)</th>
<th>n(Bohlin)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-17,4</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0,07</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>-10</td>
<td>81,8079</td>
<td>0,1778</td>
<td>0,9747</td>
<td>0,12</td>
<td>-32,5</td>
<td>0,15</td>
</tr>
<tr>
<td>0</td>
<td>61,6266</td>
<td>0,2055</td>
<td>0,9863</td>
<td>0,15</td>
<td>-27,0</td>
<td>0,15</td>
</tr>
<tr>
<td>+20</td>
<td>41,7022</td>
<td>0,2349</td>
<td>0,9926</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 3 Behaviour just outside the firing nozzle for grease 3.

<table>
<thead>
<tr>
<th>Temperature [°C]</th>
<th>Mean volume [mm³]</th>
<th>Firing time [ms]</th>
<th>Measured exit speed [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>+20</td>
<td>356</td>
<td>23</td>
<td>16,7</td>
</tr>
<tr>
<td>+10</td>
<td>328</td>
<td>18</td>
<td>15,8</td>
</tr>
<tr>
<td>0</td>
<td>367</td>
<td>21</td>
<td>17,1</td>
</tr>
<tr>
<td>-10</td>
<td>371</td>
<td>36</td>
<td>8,0</td>
</tr>
<tr>
<td>-21,6</td>
<td>171</td>
<td>71</td>
<td>8,7</td>
</tr>
</tbody>
</table>

From the high-speed video film it was found that the firing time is about 20 ms. At some temperature below 0°C the force in the solenoid cannot overcome the liquid’s viscous behaviour. Therefore the firing time increases incrementally as the firing speed decrease by some 50 per cent. The lower exit speed from the firing nozzle at low temperatures increases the uncertainty in hitting the target accurately.

A typical phenomenon when the temperature was below – 8°C can be seen in Fig. 7. Outside the nozzle, the jet of grease starts to build a drop at the front of the jet. The air resistance slows down the lump and the jet of grease behind passes the low speed lump and a new grease lump begins to grow at the front. The result of this behaviour is that the grease is spread mostly below the target. At higher temperatures, air resistance has a smaller influence since the grease does not form lump as easily.

![Figure 7 Grease jet leaves the firing nozzle at low temperature.](image-url)
represents the distance. The grease spot’s vertical position was determined at temperatures in the range +22.5 to -13.6 °C. Two trajectory curves, not taking air resistance into account, were fitted to the measured curves. The ideal (theoretical) ballistic curves are named BC in the figure and the measured curves are named MC. The initial speed (8 and 16 m/s) chosen for the ballistic curves were taken from Table 3.

![Figure 8 Bc (trajectory curve) and Mc (measured curve) show how the grease jet leave the firing nozzle and move toward the gauge face.](image)

An acceptable accuracy with which the jet of grease hits the gauge face is vertically about 20 mm. From the figure, the maximum length between the nozzle and the gauge face is 174 mm when the operating temperature varies over the range –13.6 to +22.5 °C.

**Discussion**

Wear and friction of environmentally adapted hydraulic oils were studied by Kassfeldt and Rieglert [3]. The result from that study showed that the friction was lower for an ester than for a mineral oil. However the wear was higher for the ester than the mineral oil. These results correspond to the present test results.

In Sweden, and especially in the northern parts, the lubrication equipment is turned off from November to April due to the cold conditions encountered. When the contact spot is lubricated by grease, the wear rate is 100 times less than for a dry contact spot. Whilst it is possible to perform laboratory tests to determine the wear rate associated with different lubricants, the difference in the field are often not measurable. This implies that in the field the most important thing is to apply grease as frequent as possible all the year round.

The non-Newtonian behaviour of greases leads to a flow profile with a central plug. Equation (8) can probably be used to predict the flow in a tube when the flow profile is completely developed. Because of the short firing time flow profile can never be developed in the studied equipment. Therefore the power-law index $n$ is about 30 per cent lower than in the rheometer tests.

When a centralised grease lubricating system is designed, equation (9) can be used to dimension the minimum radius $r_o$ depending on the tube length $l$, pressure $P$ and the yield value $\tau_c$, see Barnes et al [4].

$$r_o = \frac{\tau_c \cdot 2L}{P} \quad (9)$$

If the yield value is the same as the value $K$ in Table 2, then the pressure over a 5 meter tube with a radius of 3.5 mm needs to be 2.34 bar (-10.0° C), 1.76 bar (0.0° C) and 1.20 bar (+20.0° C) for the studied grease. The pressure in the actual tube should be 9 bar (From the piston pump). Whilst this should be sufficient, the mean value measurements in the equipment differed by up to 50 per cent
when the temperature was -20° C, see Table 3. The reason for this is probably that the yield value has increased and is too high. The yield value for grease 3 is difficult to determine when the temperature is below -10° C. Another reason for the decreased mean volume is that a rape seed oil solidifies at low temperatures.

The initial speed of the jet of grease is about 16 m/s in the temperature range +20.0 to 0.0° C and the measurements correspond well to the ballistic curve presented in Fig. 8. At a temperature of -10.0 to -20° C the exit speed was measured at about 8 m/s as at some temperature below 0° C the force generated by the solenoid is insufficient to overcome the liquid’s viscose behaviour.

**Conclusion**

From the results presented and discussed the following conclusions can be drawn:

1. It is possible to use environmentally adapted lubricants and graphite free lubricants without risk to increase wear.
2. The field tests showed that the wear could be reduced by a factor of 3 - 6 if it was possible to lubricate all year round.
3. The Plint and Partner apparatus could be used to evaluate friction and wear of greases intended for rail lubrication.
4. It is possible to evaluate grease in the Bohlin rheometer and, with the results in this study, predict the behaviour in the lubricating apparatus.

**References**

Features

Lubricant influence on flange wear in sharp railroad curves

Patric Waara

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>α</td>
<td>Angle</td>
<td>–</td>
</tr>
<tr>
<td>a₀</td>
<td>Acceleration</td>
<td>m/s²</td>
</tr>
<tr>
<td>b₀</td>
<td>Track gauge</td>
<td>m</td>
</tr>
<tr>
<td>Fₙ</td>
<td>Lateral force</td>
<td>N</td>
</tr>
<tr>
<td>g</td>
<td>Gravitation</td>
<td>m/s²</td>
</tr>
<tr>
<td>hₐ</td>
<td>Track displacement</td>
<td>m</td>
</tr>
<tr>
<td>H</td>
<td>Hardness</td>
<td>MPa</td>
</tr>
<tr>
<td>K</td>
<td>Dimensionless wear coefficient</td>
<td>–</td>
</tr>
<tr>
<td>M</td>
<td>Vertical mass</td>
<td>kg</td>
</tr>
<tr>
<td>R</td>
<td>Curve radius</td>
<td>m</td>
</tr>
<tr>
<td>S</td>
<td>Sliding length</td>
<td>m</td>
</tr>
<tr>
<td>v</td>
<td>Speed</td>
<td>m/s</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
<td>m³</td>
</tr>
</tbody>
</table>

Introduction

Rail and rail wheel flange wear occurring in sharp rail track curves has been the focus of much attention over the last 30 years. As this kind of wear affects both wheel and rail flanges it is a problem for both the train operating company as well as for the infrastructure owner. The rate of flange wear depends on many things including, for example, the number of wheel sets, axle load, speed and curve radius. Axle loads of 22.5-25 metric tons are now common and loads of 30-35 metric tons are expected in the next few years. Rail flange lubrication has been used since the middle of the 1970s to reduce wear, and a number of application techniques have been developed including the use of grease, oil, aerosols and solid lubricants.

This paper presents an evaluation of the efficiency of grease lubrication of sharp railroad curves with a trackside lubricating device known as the Clicomatic. In Sweden, about 3,000 such trackside lubricators are installed, a total investment of c. Mkr75 (US$7.9), excluding annual running and maintenance costs. The efficiency of these lubricators is essential in reducing wheel and rail wear.

The author would like to thank Professor Erik Hoglund and Dr P.-O. Larsson for advice and support. He would also like to thank the staff of the Swedish National Rail Administration Region North for their skilful assistance in carrying out the field measurements. The financial support of the Swedish National Rail Administration, MTAB, Inexa and Durco is also gratefully acknowledged.
A Clicomatic consists of a grease reservoir connected to a feed tube. At the end of the feed tube a firing nozzle is connected. The firing nozzle consists of four outlet holes for the grease. An accelerometer "feels" the vibrations when a train approaches and activates the trackside lubricator which starts to shoot grease from the firing nozzle toward the rail flange. The trackside lubricator simultaneously shoots four grease lumps which hit the rail flange within 0.75m in the length direction.

The trackside lubricators cannot work in the winter climate of northern Sweden because of the low average temperature, which significantly increases the lubricant viscosity. In addition, build-up of snow can prevent the lubricant being applied to the rail flange. For these reasons, the trackside lubricators are not used from around the end of October until the beginning of May. During the period between November until the end of April no grease lubrication takes place; however, the rail will be lubricated to some extent by water from loose snow blown up by the passing train. From around the end of March the snow generally starts to melt and will form an ice crust over any loose snow so it cannot be blown onto the track, which will therefore be completely dry. This situation continues until the trackside lubrication is turned on again at the beginning of May.

Rail wear is a well studied phenomenon. The wear mechanisms associated with unlubricated rails were simulated and investigated in laboratory tests performed by Danks and Clayton (1997). By using an Amsler twin-disk machine they found that three different types of wear occurred. Wear on the top of the rail and wear on the gauge face were found to be separate issues. Reiff (1985) investigated the effect of different levels of lubrication on wear rate. Hannafious (1995) investigated how the hardness of the rail steel influences rail flange wear. A field test performed by Rippeth et al. (1996) showed that the life of track sections originally worn out after 18 months could be extended up to four to five years through proper lubrication and re-finishing by grinding. Another investigation to establish the effectiveness of lubrication was performed by Elkins et al. (1984). They showed that even moderate levels of lubrication on standard carbon steel rail had a relative average improvement factor of 17 compared with dry rail, while low levels of lubrication gave a relative improvement factor of five.

The input to a wear model usually comes from laboratory or field tests. To obtain data as a basis for wear predicting models, a number of experiments have been carried out using different contact material combinations with varying levels of lubricant in between. The level of wear depends on parameters such as temperature, geometry, applied force, type of material or reacted material layers. Experimental data can thus be used to predict wear if the above mentioned parameters do not differ too much between the laboratory and the field. Laboratory machines such as the Amsler twin-disk machine, Pin-on-disk and oscillating apparatuses from Plint and Partner have been used to evaluate numerous different combinations of parameters under different operating conditions. The results of such tests can be used in formulae like Archard’s (1953) to classify the contact and help to predict wear.

Predicting wear in railroad applications needs some parameters to be taken into account. These include, for example, traction, angle of attack and load, all of which influence wear (see Kumar et al. (1995)). Angle of attack appears to have the greatest effect on flange wear and depends mainly on curve radius and wagon characteristics. The angular difference between the wheels’ rolling direction and the tangent of the rail road curve in the contact point is called “angle of attack”. Freight cars often steer badly in curves and curved track resulting in “angle of attack”. Bad steer characteristics imply a possible difference in patterns of wear between the transition curve and the midpoint of the curve. This possible wear difference is of interest. Knowledge of the wear characteristics of a track section under given operating conditions can be used to better assess when rails need to be replaced. Parameters such as adhesion and axle load also affect the wear rate, but not to the same extent as the “angle of attack”. Sliding surfaces in contact are often lubricated to reduce friction forces and wear. Mineral oil is a well known lubricant, and experience and research have lead to oils with properties especially suited to particular applications. Mineral oils are, unfortunately, more toxic and less biodegradable than, for example, rapeseed oil (Kassfeldt and Dave,
1987). A significant problem, therefore, when lubricating rail flanges is that the lubricant is spread along the track and eventually ends up in the ground. In Sweden, the yearly track grease consumption is about 25 metric tons. In comparison, the quantity used to lubricate rail in Canada is around 500 tons (see Goyan et al., 1997). The anti-wear effect of lubricants has been tested at AAR (1988) and showed no universal agreement on the effectiveness of different greases in the track tests. It was also shown that greases without solid additives performed as well as graphite greases, although unfortunately this test did not include any environmentally adapted greases. Environmentally adapted greases, with and without graphite, were compared to mineral-based grease with graphite by Waara and Larsson (1999), and the difference between anti-wear properties was found to be insignificant. Furthermore, Waara and Larsson showed that wear does not increase even if the graphite powder is excluded, which agrees with the results from AAR (1988).

The questions addressed in this paper can be summarised as:

- What is the effect of rail flange lubrication?
- Is there a wear model that can translate the result from laboratory tests to the field?
- Is it possible to use environmentally adapted grease based on rapeseed oil instead of mineral oil without increased track and wheel flange wear?
- Is it possible to exclude solid lubricants (such as graphite) from the grease without the risk of increasing the level of wear?
- How far away from the lubricator is the grease still effective in reducing wear?

Method

The material presented in this paper is based on a field test where the rail profiles in sharp curves were measured. In addition to the field measurements, laboratory tests were also carried out and the results evaluated and compared.

Field measurements

A field test was initiated during the summer of 1997 aimed at examining flange wear in sharp railroad curves. The field test covered six curves – two sections each divided into three, (see Table 1). The section A curves 1 and 3 had almost the same curve radius, super elevation and train speed. In section B all three curves were more or less identical in terms of geometry and operating conditions. All the test sections are single track with traffic in both directions.

The six curves were equipped with Clicomatic trackside lubricators mounted at the middle of the curve. The high rail of each curve was measured at six fixed points, 5, 15 and 50m on either side of the trackside lubricator. A grease based on rapeseed oil and a grease based on mineral oil with added graphite were used in the trackside lubricators (see Table 1). The rapeseed oil based grease has less environmental impact, but it is known from experience concerning oils that wear rates can be three to six times higher for blends of rapeseed oil and synthetic ester compared to mineral oil (Riegler and Kassfeldt, 1996). In view of the fact that the rapeseed oil based grease had no added graphite, this grease was expected to increase the flange wear even more. The aim of the test was thus to determine the performance differences between these two greases.

Rails wear differently depending on specific local conditions and can acquire very different profiles, which makes it difficult to evaluate wear exactly. There are, however, at least four different methods that can be used:

1. Comparing the difference in worn area between two rail profiles.
3. Comparing the horizontal wear at a vertical distance, h, from the rail head – W2. In Figure 1 it is shown that this measurement is taken right below the gauge corner. This measurement may be stable if the rail is worn in.
4. Comparing the wear measured at some angle, α, on the rail or gauge corner between two profiles – W3.

These different evaluation methods are illustrated in Figure 1. The area worn away is defined as the area between the two measured profiles and is calculated from where the profiles intersect on the rail head to the intersection point with the lower inner flange. Material plastically displaced from the rail head can result in a “bump” on the side of the
Table 1 Specification of the rail sections

<table>
<thead>
<tr>
<th>Section: curve</th>
<th>Radius (m)</th>
<th>Speed (km/h)</th>
<th>Super elevation (mm)*</th>
<th>Steel quality**</th>
<th>Grease based on</th>
</tr>
</thead>
<tbody>
<tr>
<td>A: 1</td>
<td>581</td>
<td>100</td>
<td>117</td>
<td>900</td>
<td>Rapeseed oil</td>
</tr>
<tr>
<td>A: 2</td>
<td>293</td>
<td>70</td>
<td>150</td>
<td>1,100</td>
<td>Mineral oil + graphite</td>
</tr>
<tr>
<td>A: 3</td>
<td>590</td>
<td>90</td>
<td>110</td>
<td>900</td>
<td>Mineral oil + graphite</td>
</tr>
<tr>
<td>B: 1</td>
<td>385</td>
<td>85</td>
<td>140</td>
<td>900</td>
<td>Rapeseed oil</td>
</tr>
<tr>
<td>B: 2</td>
<td>354</td>
<td>85</td>
<td>150</td>
<td>900</td>
<td>Mineral oil + graphite</td>
</tr>
<tr>
<td>B: 3</td>
<td>350</td>
<td>85</td>
<td>150</td>
<td>900</td>
<td>Mineral oil + graphite</td>
</tr>
</tbody>
</table>

*Super elevation refers to the height difference between the inner and the outer rail of a curve. The outer rail refers to high rail and the inner rail refers to low rail; **Steel quality refers to the yield stress (MPa)

Figure 1 A worn rail profile is evaluated using four measurements

Note: W1 is the rail head wear, W2 is the horizontal rail flange wear. An angular measurement on the gauge corner, W3, may be of interest, especially when the rail is new. The area worn away is shown shaded.

rail. This bump is outside the contact region and is not considered in the area calculations. In this work the rail profiles were measured using a Miniproof which is described in Waara and Larsson (1999). One feature of the Miniproof is the possibility to include the rail corners in the profile which can be used as reference points, which makes it possible to determine how much material has been worn away over a period of time. The absolute accuracy of the equipment is typically 0.02mm and relative error between a number of profiles measured independently is typically less than 0.01mm. Obviously, measurements should be taken exactly at the same place – to achieve this the measurement location is marked on the rail and the measurement of the profile made 1mm from this mark. It is also important to measure perpendicular to the rail.

Laboratory wear measurements

Different greases suited for use with trackside lubrication have been examined earlier in a laboratory test (Waara and Larsson, 1999).

These tests used a Plint and Partner high frequency apparatus to determine the anti-wear properties of the tested greases. In the test apparatus, a steel cylinder is rubbed against a flat steel surface with an oscillating motion. An applied load is used to hold the two specimens in contact, creating a lubricating film of the grease separating the specimen surfaces. After the test is complete, the wear mark, and hence wear volume V, on the steel cylinder is measured using an optical microscope. The field measurements were evaluated by using Archard’s law:

$$K = \frac{V \cdot H}{S \cdot F_N}$$  \hspace{1cm} (1)

K is the dimensionless wear coefficient and parameters such as specimen hardness H, sliding length during test S and applied force F_N are used to calculate K. In the present work, a laboratory test was also run under dry and water-lubricated conditions. These additional tests used the same applied force F_N, sliding length S and specimen hardness as the previous test.
Wear calculations
The field measurements were compared with the laboratory measurements using Archard’s law (equation (1)). The worn area is taken from rail profile measurements and the wear volume \( V \) is calculated over a length of 1m. The material hardness \( H \) is the hardness on the rail flange. The sliding length \( S \) was determined from the number of axles passing the measurement points over a 14-month interval, along with assumptions as to how and where the contact spot occurs (see Berghvud, 1999). The force \( F_N \) is the lateral force exerted by the wheel flange on the rail and can be calculated dynamically or statically. In this work the lateral force was assumed to be static and only depend on the curve radius. The dynamic forces can be neglected since the curve radius was constant and no track displacement existed. To calculate the lateral force the acceleration must first be estimated:

\[
a_y \approx \frac{v^2}{R} - \frac{g}{2} \cdot \frac{h_s}{h_s} \tag{2}
\]

where \( v \) is the train speed, \( R \) the curve radius of the track, \( h_s \) is the super elevation and \( h_s \) is the track gauge. The lateral force \( F_N \) is calculated as:

\[
F_N \approx M \cdot a_y \tag{3}
\]

where \( M \) is the vertical load. For a radius \( R \) of 350m, a train speed \( v \) of 85km/h, a track super elevation \( h_s \) of 0.150m and \( M = 222.5 \) metric tonnes, the force is 27.36kN. This force is assumed to be applied on the leading wheel.

Result and discussion
The results are divided into two parts, the first part dealing with variation in wear depending on seasonal conditions and the second concerning the wear as a function of distance from the trackside lubricator.

Flange wear variation depending on seasonable lubrication conditions
Flange wear is stated in terms of mm/10MGT (million gross tonnes) and thus the number of wheel sets during a given time period influences the wear rate. The rail profiles were not measured exactly when the trackside lubricators were turned off or on. Thus the time periods between the profile measurements include both lubricated and unlubricated periods. When evaluating the profiles W2 was used (see Figure 1). The reason for using W2 is that the measurement is taken on a planar surface compared to W3, which is measured on the gauge corner.

The results of the measurements are summarised in Figure 2. The left diagram shows the result for the sharpest curve (radius 293m) in section A. During the lubricated period (3 July 1998-20 October 1998) the flange mean wear rate was 0.015mm/10MGT. The same conditions were valid from 1 July 1999-8 December 1999, which had a flange mean wear rate of 0.082mm/10MGT. Both wear rates can be considered as very small. The reason for the higher value during the latter period (1 July 1999-8 December 1999) is almost certainly the unlubricated period during November and eight days in December. For the first unlubricated period (1 May 1997-2 July 1998) the mean wear rate was 1.78mm/10MGT, and for the second period (21 October 1998-30 June 1999) 1.28mm/10MGT.

The three curve radii in section B (see Table I) were more or less the same and therefore the measurements from these three curves are presented in the same diagram (a total of 18 points for each time period). The mean wear rate in section B was found to be 0.79mm/10MGT for the unlubricated period (13 August 1997-2 July 1998) and 1.28mm/10MGT for the period 21 October 1998-30 June 1999. The lubricated period 3 July 1998-20 October 1998 gave a negative average flange wear rate of -0.0064mm/10MGT, while the period 1 July 1999-8 December 1999 also gave a negative mean flange wear rate of -0.018mm/10MGT. For the curves in section B it was not possible to calculate the relative effect due to low or negative wear, and negative wear in this case is almost certainly due to plastic material flow from the rail head into the rail flange (see for instance Grassie and Kalousek, 1997).

The difference between the wear rate for unlubricated and lubricated conditions in rail section A in the 293m curve was in the range 16-21 times. This is in accordance with the results of Elkins et al. (1984) who found a reduction of between five and 17 times depending on whether the rails were dry or lubricated.

Compared to the unlubricated conditions, the wear of the lubricated rails can be considered almost insignificant. As
mentioned earlier, the points in time when profile measurements were made did not coincide exactly with the start or end of lubricated/unlubricated conditions. For this reason, it seems inappropriate to include traffic volume that actually took place under lubricated conditions when calculating a wear rate for the unlubricated track. This will, in fact, reduce the volume of traffic, contributing to unlubricated track wear, and result in even higher actual wear rates. Therefore the wear during the unlubricated conditions presented above is already recalculated.

No significant differences in flange wear were observed between curves lubricated with rapeseed based grease and those lubricated with grease based on mineral oil. The mineral based grease contained graphite while the rapeseed based grease did not, which indicates that the addition of graphite makes little or no difference in the wear reduction properties of a grease. This is in accordance with earlier results (AAR, 1988; Waara and Larsson, 1999).

In Figure 3 the dimensionless wear coefficient from the field test under winter conditions (no grease lubrication) calculated according to equation (1) are presented along with the results from laboratory tests under a range of conditions. The figure shows that wear in the field during winter conditions is slightly higher than water lubricated conditions in the laboratory, but much lower than for dry conditions. This confirms the assumption that some lubrication occurs from snow blown onto the track by the passing train being melted in the high pressure contact between the track and rail.

The laboratory results in Figure 3 for the lubricated condition contains both environmentally adapted greases as well as mineral oil based grease with and without graphite. This shows that the relative difference between grease and water-lubricated conditions tested in the laboratory are relatively large compared to the difference between individual greases. From this it is clear that it is important to lubricate the rail flanges in curves for as large a part of the year as possible; especially when “dry” conditions can be expected during the late winter. It is also clear that it should be possible to use graphite-free, environmentally adapted grease without any increase in rail wear rates.

**Grease transport along the track**

The transport of grease along the track and associated protection of the surfaces in the rail/wheel contact were investigated. At rail section B the rail profile was measured in each curve at a distance ±5m and ±50m from the trackside lubricator. The profile measurements taken on 13 August 1997 and 20 October 1998 were used to determine the amount of material worn away. When the W2 measurements were used there were no distinct differences between the wear close to the trackside lubricator compared to 50m from it. However, when the worn-off areas were calculated it was possible to see the difference. Using the worn-off area may be a method with better “resolution” than the W2 measurement. In Figure 4, measurements from all three curves are presented (a total of 12 measurements). The measurements indicate that the amount of wear increases with distance from the trackside lubricator.
Figure 3 Dimensionless wear coefficient K (see equation (1)) calculated from the field test and compared to laboratory tests

Note: It was found that the field test concerning winter conditions gave slightly higher wear than the laboratory test using water as lubricant and lubricated conditions from the field test were comparable with lubricated laboratory experiments.


Figure 4 The area of track profile worn away plotted against the distance from the trackside lubricator

Note: Measurements made at 5 and 50 m from the lubricator

and that the wear is approximately double at 50m from the trackside lubricator compared with the wear 5m from the lubricator.

The increased wear rate depending on distance to the trackside lubricator was unexpected, as most of the wear occurs during un lubricated conditions and this result shows a significant difference depending on the distance to the trackside lubricator. A speculative explanation is that the rail is exposed to severe wear during a few weeks and when the lubricator is turned on in the beginning of May it may take some time before the rail flange 50m from the trackside lubricator is sufficiently lubricated.

Conclusions

- From the field test it was found that wear under grease lubricated conditions are 16-21 times less than wear under winter condition, which consist of a mixture of water lubrication and dry conditions.
- It was not possible in the field test to see any difference in wear performance between the different lubricants used and the effect, if any, from the addition of graphite is insignificant. It is clear that environmentally adapted grease without graphite can be used without fear of increased wear rates occurring.
- Comparison between the wear results from the winter field measurements agreed well with water lubricated laboratory conditions, which supports the assumption that water lubrication occurs during at least some of the winter months.
- The amount of wear was approximately double 50m from the lubricator compared to 5m from the trackside lubricator.
The wear characteristics of the rail profile vary as the rail wears. This indicates that methods are required to be able to compare the wear between track of different ages. Measuring the worn off areas may give a better resolution than the W2 measurement.

References

Paper C
Technical and economic evaluation of maintenance for rail and wheels on Malmbanan.

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Summary: This paper provides an overview of how maintenance costs for rails and wheels are distributed between infrastructure owner and rolling stock owner on the basis of a study performed in close co-operation with MTAB (Malmötrafik i Kiruna AB, iron ore transportation company), Banverket and Duroc Rail AB. This paper presents a technical and economic correlation between maintenance activities and decisions performed by MTAB, Banverket and Jernbaneverket. Technical aspects are generated by controlling rolling contact fatigue (RCF) failures and wear in combination with grinding. RCF damages such as head checks are a common problem on high rails gage corner, causing the main replacement cost for rails. To reduce replacement of rails caused by RCF a grinding program started in 1997. The balance between controlled wear, with or without lubrication and grinding, are very important tools to ensure long life and effective maintenance operation for both the infrastructure and rolling stock. Rail profile measurement since 1997 gives an indication of parameters that have to be taken into account when choosing grinding strategy at Malmbanan.

Economic aspects are generated from different maintenance activities such as grinding and re-profiling wheel sets. The results give an indication of how different parameters affect maintenance performance. Aspects such as wheel/rail interface and car steering ability affect maintenance costs over the studied period.

Index Terms: Grinding, Wear, Costs, Wheel, Rail, Maintenance.

1. INTRODUCTION

In railroad heavy haul applications, grinding and lubrication are routinely used as a maintenance tool in curved track sections in order to reduce friction, wear and rolling contact fatigue (RCF). The cost of maintenance of wheel and rail for heavy traffic such as ore lines can be up to fifty per cent of the total maintenance cost of the rail/wheel system. For instance, proper lubrication can reduce wear rates by a factor of 20, Elkins et al. [1] and Waara [2]. Grinding programs can also produce significant cost savings for both the wheel and infrastructure owner, see Grassie et al. [3].

During the past 30 years axle loads have increased from 25 tons to 32.5 tons per axle, Allen [4]. This has been possible due to improvements in metallurgy, larger rail cross sections standardization of heat-treated wheels, development of new rail/wheel profiles and the extensive use of track lubrication and grinding. Wear of rails and wheels has long been identified as a major reason for the high maintenance costs of rail infrastructure. A second major cause of the degradation of rail is rolling contact fatigue (RCF). As the wear rates of rail have decreased due to the factors described above and the axle loads have increased, RCF has become a more important problem. If the wear rate drops below a certain value the rail surface reaches its fatigue limit before it can be worn away. Preventive grinding programs are designed to grind away a thin layer of material from the rail surface before RCF cracks can develop. The depth to which RCF cracks penetrate determines the amount of material that has to be removed. Lubrication can slow the initiation of surface and contact fatigue cracks, but it might accelerate crack propagation.

To optimize the life of railway infrastructure, wear rates, the development of RCF cracks and grinding have to be carefully balanced. The rate of wear is larger if softer rail steel is used and any damage or cracks are worn away before the critical deformation is reached. However, while no cracks are observed in the softer rails, they are consumed earlier by wear, see Pointner and Frank [5]. A hard rail will not wear as much, but after a certain time it will suffer from RCF and may need replacement having suffered only a small amount of wear. Also, lubrication, which reduces wear, shifts the failure mode from wear to crack formation.
A systematic experimental test program of wheel/rail adhesion and wear was undertaken using the Illinois Institute of Technology's 1/4 scale wheel rail simulation facility, see Kumar et al.[6]. The tests determined the effects of axle load, adhesion coefficient, angle of attack, class of wheels and mode of operation. Wear was measured by overlaying profiles of the wheel/rail surface at different stages of wear and measuring the change in cross-section area. The hierarchy of influencing parameters for wheel/rail wear in order of priority was given as:

1) rail curvature or angle of attack
2) adhesion coefficient
3) axle loads

Lubrication improves interactions at both micro and macro level of the wheel/rail interface. However, today no general model or strategy that describes the interactions between wear, lubrication and grinding at the wheel/rail interface exists, see Clayton [7] and fundamental studies have only been carried out during the last thirty years. Models that have been developed to simulate real field phenomena are often designed for specific problems and cannot be extrapolated to the general situation.

More studies have to be completed before a general model can be developed to treat combination of technological-economic maintenance activities such as grinding and lubrication correlated with stringent cost efficiency evaluation. The challenge is in combining the outcomes of technical maintenance activities with an economic analysis.

2. WHEEL/RAIL INTERACTION

The ore car is a four-axle heavy-haul freight wagon specifically designed for the transportation of ore. This so-called BoBo vehicle contains a car body sitting on two two-axle bogies. A side view of an ore wagon is shown in Figure 1.

![Figure 1. Four-axle ore wagon with three-piece bogies.](image)

The bogies are so-called three-piece bogies, where the bogie frame consists of two side frames and a bolster coupled together into a bogie frame.

In service, the wheel and rail profiles change substantially due to wear and the fleet of trains passing over a section of rail will have a range of wheel profiles at any one time, ranging from new to worn. Simulations have been performed in order to investigate the wear rate sensitivity as function of the wheel/rail profiles. In this investigation steady-state curving of a Malmbanan ore car was investigated using a multi body dynamics model and the numerical algorithms implemented in the commercial software GENSYS [8]. The model was validated using field measurements, see Berghuvud [9]. Seven different wheel profiles, including a nominal S1002, were used to compare the influence of wear rate as function of wheel profile status. The vehicle is run on designed track with new (nominal) UIC60 rail profile, see Figure 2.

![Figure 2. Measured in service measured wheel profiles (dark) and a new nominal (gray) wheel profile S1002. The scales is in mm.](image)

2.1 Simulation results

Changes in wheel profile have only a small influence on the contact forces, contact dimensions and positions on the high rail for the trailing wheel set. The contact conditions for the other contacts are significantly influenced by the change of wheel profile. Calculated energy dissipation is used as an indication of the amount of expected relative change of wear for the different profiles, [9]. High difference in creepage and tangential forces gives high difference in energy dissipation at the wheel/rail interface. Some combinations of profiles produce more severe wear than with a single contact patch, see Table 1.

<table>
<thead>
<tr>
<th>Wheel Profile Status</th>
<th>High Rail (J/m)</th>
<th>Low Rail (J/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leading wheel set</td>
<td>180-245</td>
<td>90-135</td>
</tr>
<tr>
<td>Trailing wheel set</td>
<td>45-55</td>
<td>45-130</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Wheel Profile Status</th>
<th>High Rail (J/m)</th>
<th>Low Rail (J/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leading wheel set</td>
<td>50-250</td>
<td>75-175</td>
</tr>
<tr>
<td>Trailing wheel set</td>
<td>10-50</td>
<td>10-75</td>
</tr>
</tbody>
</table>

The results presented here should be viewed upon as examples of what types of results the developed model can support used in maintenance planning.
3. EVALUATION OF WEAR AND GRINDING

Transverse profile measurements of outer and inner rails were performed at 60 positions along the rail located on Malmbanan. Banverket used the MiniProf Rail system to measure the profiles. The profiles were measured shortly before and after the grinding of the rails. Results of these measurements are presented in different ways and the MiniProf Rail system provides an easy, portable tool to monitor and evaluate rail profiles. The system is a useful aid for rail grinding operations, providing instant information on profile and metal removal. MINIPROF is a standard system for the determination of rail profiles in the field. The sensing element consisted of a magnetic wheel 12 mm in diameter attached to two joint extensions. When the magnetic wheel was moved manually over the rail surface, two angles were measured and stored in a computer. The profile was then transformed to Cartesian co-ordinates. Marks on the edge of the rail were used to ensure that the measurements were performed at the same location each time. Further information on the MINIPROF system can be found in Esveld and Gronskov [10]. The accuracy of the MINIPROF system is of the order ± 0.015 mm for similar profiles.

A grinding program was initiated by The Swedish National Rail Administration (Banverket or BV) 1997 at Malmbanan, the Swedish ore track between Kiruna and Riksgränsen. The grinding is primarily to remove RCF, which has been an increasing problem in recent years. In the seventies, the axle load was increased on this track to 25 tonne and again in 2001 a new ore care was introduced giving 30 tonne axel loads. This rise in axle load increases the susceptibility of the rail to RCF. In 1997 a new asymmetric profile MB1 was introduced. As the contact path in this profile is wider than previous profiles, the onset of RCF should be delayed. Before the grinding program started the curved track was prepared with 60 measurement points and 10 points on the tangential track. BV has carried out rail profile measurements before and after the grinding activities every year since 1997. The grinding campaign is here evaluated at the ore track between Kiruna and Riksgränsen with a total length of 127 km.

To evaluate the combined effects of the track wear and the grinding, the track is divided into three categories: track curves with radius under 800 m, track curves with radius over 800 m and tangential track, see Table 2. In this paper, the yearly wear and the amount of material removed due to grinding is evaluated at these points. The yearly wear and grinding indicate the proportion of the life of the rail consumed according to BV’s regulations (BVF 524.1 [11]), which indicate when the rail should be replaced.

Table 2.

<table>
<thead>
<tr>
<th>Grouping</th>
<th>Length [m]</th>
<th>Grinded/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius&lt;800</td>
<td>57 791</td>
<td>100%</td>
</tr>
<tr>
<td>Radius&gt;800</td>
<td>30 526</td>
<td>100%</td>
</tr>
<tr>
<td>Tangential track</td>
<td>48 220</td>
<td>33%</td>
</tr>
</tbody>
</table>

The tolerances for the ground profile compared to the reference profile have been changed during the period from 1997 to 2001 and nowadays the high rail diverge should be within +0.3/-1.0 mm on the railhead between -20/50 Miniprof degrees. A minimum metal removal is set to be of at least 0.2 mm in the area between 0° and 45° every 23 MGT.

3.1 Results of Wear and grinding

The grinding results were evaluated over year 2000 and the wear over 10 months in 1999/2000. The wear results over 10 months are afterward compensated to correspond to wear per annum based on ore traffic volume.

The results of grinding and wear from traffic at the high rail for curves under 800 m is summarized in Figure 3. The mean yearly traffic wear was around 0,18 mm at β=10° (β is Miniprof degrees according to Figure 3). As the figure shows, the average material lost on the railhead to grinding per annum was 0.4 mm and when coupled with the wear, 0.6 mm was lost on the railhead per annum.

The measurements in Figure 3 also show that traffic wear and grinding wear is of the same order in the 45-50 degree region. Grinding wear (0.4 mm) is twice as much as traffic wear (0.18 mm) for the 10-40 degree region. The train/track simulations, Table 1, indicate that it could be possible to use an other portion of the existing worn wheel profile population distribution to increase the traffic wear to the same amount as the total wear presented in Figure 3. This might produce higher rail wear in the 10-40 degree region and hence save costs by prolong the grinding intervals.
The rail profile was evaluated due to the rail grinding campaign year 2000 as well as the wear for one year. This yearly rail profile evaluation will help to predict future renewal of rail. Around 20 profiles for high respective low rail and 10 profiles at tangential track were measured and used in this investigation. The profiles used here were measured before and just after a grinding campaign each year. The profiles were considered of measuring quality chosen among 60 profiles in curves and 20 profiles at tangential track. When evaluating the rail wear there are two measurement values to consider according to the Swedish standard for railhead wear measurement. The vertical wear on the railhead is determined by a rail classification system, the rail must be replaced. The rail life is separated into 5 different groups, high and low rail under 800 m curve radius, high and low rail over 800 m curve radius and tangential track. One third of the tangential track is ground each year. The yearly wear on the tangential track railhead is estimated to be 4 mm/year and the total flange wear (s mm/year) for the 50 kg/m BV50-rail profile used. The average rail life is then calculated as the ratio between the measured H value of the studied section. Using this approach it is also possible to estimate the number of meters of rail in each section, that needs to be replaced due to railhead material loss each year. These calculations suggest that 12000±1900 m rail need to be replaced each year average, which corresponds to a yearly investment of 9.6 million SEK (1.05 million USD$). The yearly cost of grinding this track should be around 4 million SEK, giving a total yearly maintenance cost of 13.6 million SEK.

4. Evaluation of Cost Drivers

An economic analysis of Malmbanan made 1995 indicates that about 30% of the total cost for maintenance and renewal was related to traffic and 50% was related to other factors such as signaling, electricity, snow-clearance etc. Costs for maintenance and renewal of rails on some lines account for more than 50% of the costs related to traffic. The results from this analysis have made it possible for the mining company LKAB to start up the 30 tons traffic with new wagons and locomotives on the Malmbanan line in year 2001. The focus in this economic evaluation is therefore to examine the costs generated by maintenance and the renewal of rails and wheels.

The approach is to combine the technical and economic modelling to determine how decisions based on technical activity affects the economy of the wheel/rail interface. In this case the technical activity of concern is rail grinding.

4.1 Data collection

Data were collected from a literature search, different databases at MTAB, Durroc Rail, BV and Norwegian Rail and from interviews with personnel at the different companies. It was found that primary data is located at different levels for each company, depending on internal needs and demands. It was also found that economic costs and results are presented and based on technical aggregated data, resulting in difficulties in comparing between different companies at primary information levels. Therefore it is not possible to determine how every technical detail affected the system.

4.2 Infra structure maintenance activities and costs

The rail-grinding project on the Malmbanan line between Kiruna and Riksgränsen reduced the requirement for rail replacement from approximately 25 000 meter to 5 000 meter annually, as shown in Figure 4. In this paper the rail life due to wear and grinding was evaluated on the basis of material loss rates and the mean renewal level is suggested to be 12000±1900 m over the same distance. In Figure 4 it is possible to see a very significant reduction of rail renewal during the first years after the grinding campaign started. However, it is likely that this effect is transient and is simply delaying the requirement for track replacement. After some years of grinding the track will become degraded and an accumulated volume of rail will need to be replaced. The track status is not considered in this paper and therefore it is not possible to make any more exact investment calculations. However it is likely that the cost level will be a bit over
13.3 million SEK. However this cost level is significantly reduced when compared with the years 93, 94, 95 and 96. The result so far in economic terms is a reduction in rail maintenance costs of approximately 50%.

![Figure 4. Quantity of annually rail renewals on the ore track between Kiruna and Riksgränsen.](image)

In 2000 Norwegian Rail changed their maintenance strategy, deciding that renewal of rails shall be done only on the basis of MINIPROF data or similar objective measures. This new strategy has halved rail renewal demands. So far, economic results indicate reduced maintenance costs, but no deeper analyses are made to confirm new cost levels.

**4.3 Ore car maintenance activities and costs**

MTAB’s maintenance program is based on experience and relates to driven distance for the ore car. In practice this means that the ore car wheel set is checked every 300,000 kilometer. Normally the wheel profile lasts for approximately 120,000 kilometer, after which it is necessary to re-profile the wheel and its geometry. Each wheel can be re-profiled 4 or 5 times. The type of wear/damage that dominates changes over time, but analyses shows that as one wear mode decreases another mode increases. More important is that the total number of wheel sets replaced per annum decreased, as shown in Figure 5. However, at this stage it is too early to link the reduction in wheel set replacements to the ongoing grinding program.

![Figure 5. Number of worked/shifted wheel sets](image)

The major problem for MTAB is that the cost of replacing wheels is at least ten times more than re-profiling an old wheel. During the last year, MTAB has introduced a new wheel material with better wear resistance. This could be one of the explanations why the turnover of wheel sets decreased. The distribution of wheel age in the population of wheels is not constant over the years and this will affect the cost levels between different years. Different wheel wear modes will require different re-profiling. If one wear mode is very fast for short periods this can result in high maintenance costs if that particular wear mode results in shifting to new tires. Therefore it is possible for MTAB to incur higher overall costs even if the total number of worked and shifted iron ore wheel sets decreases, which also happened in year 2001.

**4.4 Results from an economic perspective**

Analysis made from an economic overview perspective for the Malmbanan as a transporting system, shows that maintenance processes and strategies related to rail/wheel interaction do not seem to affect the total system in a negative way. However, once again the problem is that all comparisons have to be made at high aggregated data levels, which make it possible for subsystems to show different results.

**5. DISCUSSION**

Using train/track simulation in maintenance planning provides a powerful tool to predict changes in wear due to different rail/wheel profile strategies. Simulations and field observations clearly suggest that, by altering load traffic direction of cars would produce longer wheel life. The results in Table 1 indicate that for the same three-piece bogie a leading wheel set can have up to five times higher wear than a trailing wheel. Some years ago MTAB had problems with high leading wheel set wear. They introduced a logistic change of loaded car direction and hence prolonged the leading wheel life. This kind of train/track simulation is a powerful low cost tool for estimating and predicting traffic wheel and rail wear. It can be used in the optimization of wheel and rail profiles on a traffic routes such as Malmbanan. A technically optimized wheel/rail profile combination can be used in maintenance planning to decide on new strategies such as maintenance intervals for both rail and wheels.

![Diagram](image)

The grinding project was an important step to make it possible to increase the axle loads to 30 tonnes. The grinding program was also one of the main strategies to prolong the renewal of existing 50 kg rails. Even after only a short time, there are measurable costs savings for both infrastructure and traffic. However, this procedure may simply be delaying the costs and an accumulated rail renewal will appear in the future due to degradation of the rail. It is too early to suggest, that for the long time run, the introduction of grinding has led to decrease costs and an increase in rail and wheel life. Maybe future life cycle cost analysis can provide an answer. However, ultrasonic inspection is indicating that grinding has increased the number of rails replaced due to rolling contact fatigue.
This rail life investigation indicates that the maintenance program has delayed maintenance costs to the future and a suggested mean level of rail renewal is suggested. Optimizing the grinding process by for example using train/track simulations is one possible step to increase rail life, however it is uncertain if the economic potential is enough.

6. CONCLUSIONS
- The grinding campaign delays major replacement of rail to the future.
- The grinding program is evaluated to give a yearly cost of 13.6 million SEK included rail renewal and grinding campaign.
- Train/track simulations can be used as a tool in maintenance planning.
- Train/track simulation clearly suggests that by altering the load traffic direction of the cars it would be possible to obtain longer wheel life.
- Neither grinding campaigns on the Swedish side or objective measurements to increase rail life on the Norwegian side seems to affect the total system in a negative way.

7. ACKNOWLEDGEMENT
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8. REFERENCES
Paper D
Additive influence on wear and friction performance of environmentally adapted lubricants

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Abstract

In this paper, the influence of concentration level and chemical composition of three different additive types on friction and wear coefficient are presented for a synthetic ester base fluid and a mineral base oil. One extreme-pressure (EP), two antiwear (AW) and two yellow metal passivator (Cu-passivators) additives were used. Factorial experimental design was used as the basis for a systematic evaluation of wear rates under mixed and boundary lubrication conditions. A total of 33 different lubricant blends were tested in a Plint and Partner High Frequency Friction Machine. For the synthetic ester, the extreme-pressure (EP) additive, containing phosphorus and nitrogen, was found to be much more effective in reducing wear than either of the two antiwear (AW) additives tested. In fact, the AW and Cu-passivator additives made little or no contribution to the wear protection in most of the cases studied. A synergy effect between the three additive combinations was observed only for the reference mineral oil blend. A significant difference between the antiwear performance of the test lubricants was found. This study suggests that the traditional “AW” and “EP” labels associated with commonly used additives are poor aids when designing of ester based lubricants. © 2001 Published by Elsevier Science Ltd.

Keywords: Boundary lubrication; Mixed lubrication; Extreme pressure; Antiwear additives; Polyol ester; Synthetic ester

1. Introduction

Since the middle of the 1970s, the environmental impact of lubricants, both in their manufacture and use, has attracted increasing attention, Bartz [1]. Unfortunately, from an environmental point of view, mineral oil based lubricants are not readily biodegradable and contain many kinds of additives; many of which are known to be toxic or otherwise harmful to the environment, Hewstone [2]. Lubricants based on mineral base oils have been used in all kinds of applications since the beginning of industrialisation: industrial gears and hydraulic systems, automotive engines, transmission and hydraulic systems, as well as in metalworking applications to name but a few. The physical, and to some extent chemical, properties of mineral oil based lubricants have been studied for almost as long and lubricant manufacturers have, over the years, gathered know-how and the necessary technologies to blend lubricants to give the required performance. Different kinds of additives are used to improve the performance and longevity of lubricants. Depending on the specific demands and performance level requirements, several different classes of additives may be used. These include detergents, dispersants, extreme pressure (EP), antiwear (AW), viscosity index improvers (VII’s) and corrosion inhibitors; see for instance Rizvi [3].

One of the earliest uses of environmentally adapted lubricants (EAL’s) in Europe began after the great storm fellings in Germany’s Black Forest around 1985, in the form of readily biodegradable hydraulic fluids for mechanised forestry operations. Since then technological development in this field has been substantial, which is also seen in the growth of research literature. Some recent examples of research in the field of mobile hydraulics are Kassfeldt and Dave [4], Norrby and Kopp [5] and Goyan et al. [6]. When formulating EAL’s, it is also important to consider additive toxicity. Toxicity
limits associated with water-living organisms like daphnia and fish, as well as in mammals, are usually closely regulated via environmental legislation, industrial standards or Ecokilling. Clearly, some additives are better suited to EAL applications than others from an environmental and/or toxicological point-of-view. Examples include organic compounds containing oxygen, sulphur, phosphorus and nitrogen as active components. Compounds to be avoided in EAL’s are those based on lead, chlorine, barium, and zinc (as in ZDDPs, Zinc Dialkyldithiophosphates).

In addition to their use in mobile hydraulics, EAL’s are also found in many total-loss applications, such as railway track lubricants, saw chain and saw bar oils, concrete mould release agents, two-stroke engine oils, and greases. Other application areas are turbine oils, industrial hydraulic fluids, industrial gear oils, metal-working fluids and even some internal combustion engine oils.

Many studies have been undertaken in order to assess the working range and applicability of EAL’s; not all of which have been favourable. For example, a laboratory bench test study by Riegler and Kassfeldt [7] showed that a “conventional” mineral oil gave 3–6 times less wear (by wear volume) than a rapeseed oil/synthetic ester blend and a pure synthetic ester under boundary lubrication conditions. Wear and friction properties of EAL’s as a synthetic ester with additives have also been investigated by Ren et al. [8] who studied a Pentaerythritol ester with TCP, BMDEC, BMDBC and BMDOC additives over range of load and additive concentration levels. They found that the load-carrying capacities and antiwear properties decreased with the increasing alkyl chain length in the additive molecular structure. They also found that TCP showed inferior quality compared to the other tested additives as far as load carrying capacity, antiwear and friction reduction was concerned.

The interaction between the base fluid, the different additives and the metal surfaces of the machine elements being lubricated is complex and depends on a delicate balance between the different chemical and physical properties of all the components of the system. Many additive classes are polar compounds, which are attracted to the metal surfaces. This attraction process is influenced by all other molecular species present in the system and, of course, by base fluid properties such as polarity (or rather, non-polarity), dissolving power, viscosity etc. Van der Waal [9] developed a Non-Polarity Index (NPI) based on the chemical composition of the base fluid. Ester molecules are more polar (or rather, less non-polar) than typical mineral base oils and therefore experience stronger interactions with polar molecular species (such as most additives), as well as the metal surfaces of the machine elements themselves. In a recent study by Kassfeldt et al. [10], an ester-based and a PAO-based hydraulic fluid were compared in a friction and wear study similar to the present work. The wear characteristics of the ester-lubricated system were found to be substantially different from the PAO-lubricated system; although the NPI of the ester (TMP-Oleate, NPI 185) ranks it as an “oil-like” ester. It is therefore clear that polarity is only one of the factors contributing to the tribosystem. Kassfeldt’s study also indicated some inadequacies in the antiwear/pressure additive chemistry used to mitigate wear under boundary lubricated conditions.

Another important issue is additive synergies and antagonisms; i.e. how the different additive components interact in a way that enhances (synergy) or weakens (antagonism) their effect. These phenomena have been studied for many years in mineral oil systems and were reviewed by Papay in 1998 [11]. In another study, Barcroft and Park [12] investigated how ZDDP’s interacted with other highly polar species like dispersants and metal containing anti-rust additives. In order for the additives to give any antiwear and extreme-pressure protection, the film forming conditions must be right and the chemical reactivity of the additives must match the reaction conditions; i.e. temperature, pressure, other chemical species present on the metal surface etc. In a study of sulphur and phosphorus compounds as EP additives, Kawamura and Fujita [13] investigated the film forming ability and extreme-pressure (EP) protecting ability of organic sulphur and phosphorus carriers. The EP action was found to be initiated by the frictional heat generated in the contact surface film and was also very sensitive to the chemical reactivity of the additive species. Few tribochemical studies of synthetic or natural biodegradable base fluids commonly found in EAL’s have appeared to date. Some of these include studies of additive response in refined olive and soybean oil (Choi et al. [14]) rapeseed oil (Vizintin et al. [15]) and High Oleic Sunflower Oil (HOSO) (Minami et al. [16]), and wear studies in polyol ester by Han and Masuko ([17,18]). The latter studies are of interest for the present work; although the polyol esters investigated are of lower molecular weight and much more non-polar with van der Waal NPI’s in the range of 34–71. The NPI for TMP-Olate is 185, which definitely ranks it as a non-polar (or at least, low-polarity) fluid. Furthermore, the solvatisation effect of the polar additives in TMP-Olate are expected to contribute to the “deactivation” of the antiwear additives whilst the dielectric constant is likewise expected to have some influence on the additive interaction in non-polar fluids like TMP-Olate. The occurrence of an “optimum” concentration of antiwear additives also suggested in the present work, see Section 3, Table 4 below.

From the foregoing discussion it is clear that several properties of the tribosystem must be considered: polarity, surface activity, chemical reactivity and properties related to film forming ability. Amongst these are
film strength and load carrying capacity, heat transfer ability and, not least, the temperature–viscosity relationship since most esters have high inherent VI’s, Larsson et al. [19]. Thus, when comparing mineral oil and synthetic ester based lubricants, quite different “activation conditions” for the additives are in operation under mixed and boundary lubrication, where typical antitrust and extreme-pressure additives are used.

In this paper, the interactions between three types of additives; extreme-pressure (EP), antitrust (AW) and yellow metal passivator (Cu-passivator), in a synthetic polyol ester (TMP–Oleate) has been investigated. The aim of the study was to find evidence of synergism or antagonism between the different additives in relation to the wear coefficient and friction coefficient in mixed and boundary lubrication conditions.

2. Method

The purpose of an experiment is generally to measure the effect of one or more variables on the response(s) of interest to the experimenter. The use of factorial experimental design, see Box et al. [20], is an efficient way of optimising a test series. In factorial experimental design, all variables of interest are tested at a number of chosen levels in all possible combinations. The most commonly used factorial experiment looks at each variable or factor at two levels, a “high” and a “low” level. In this test, three additive types expected to influence friction and wear characteristics were evaluated. The additives were blended in the base fluids (synthetic ester and mineral oil) at two different concentration levels; one “high” and one “low”. The design matrix is described as a 2×3 matrix where the “2” represents two concentration levels, and the “3” represents three additive types. Thus, one experimentally designed test series used eight different lubricant blends.

2.1. Additives

The base fluids chosen for this test were a synthetic polyol ester (TMP–Oleate) and a mineral oil blended to the same dynamic viscosity at 90°C. Four different additive combinations in the synthetic polyol ester were chosen for this test; referred to as “additive combination 1–4”, see Table 1. These combinations used three different additive types: EP, AW and Cu-passivator. Combinations 1 and 2 were used in a first test series and combinations 3 and 4 in a second series developed based on the results from the first.

In “additive combination 1” the EP additive was an amine neutralised phosphoric acid ester; a common type of general purpose EP additive. The AW additive was triphenylphosphorothionate (TPPT). The Cu-passivator was a thiadiazole derivate. In “additive combination 2” the same EP additive was used. The AW additive was a methylene bis(dithiocarbamate), and the Cu-passivator was a tolyl thiadiazole derivate, see Fig. 1. “Additive combination 3” contained the same EP and AW additives as “additive combination 1”, but the AW additive had a lower “low” level. No Cu-passivator was used. “Additive combination 4” contained the same EP and AW additives as “additive combination 2”; again with the AW additive at a lower “low” level, and without any Cu-passivator.

2.2. Test apparatus

A Plint and Partner High Frequency Friction Machine (TE 77B) was used to evaluate the friction and wear properties of the lubricants by simulating a sliding motion between two specimens with a lubricating film in between. The operating parameters can be adjusted to simulate mixed and boundary lubrication regimes. The test apparatus, shown in Fig. 2, uses an electric motor to drive a specimen holder (1) with an oscillating movement with stroke amplitude of 4.6 mm. The upper test specimen (2) is held in contact with the supporting specimen (3) by an applied vertical force of 12 and 50 N. The specimens used were hardened steel rollers taken from a roller bearing. The surface roughness, Rₚ, of the upper specimen was 0.05 μm and 0.08 μm for the supporting specimen. The hardness of both specimens was 850 HV. The lubricant was held in a temperature-controlled container (4) at a constant bulk temperature of 90°C. The sliding motion combined with the applied force causes wear on the specimen. To evaluate the wear on the upper specimen, Archard’s law was used:

\[
K = \frac{V 
\cdot \frac{H}{F_n}}{S}
\]  

Where \(K\) is the dimensionless wear coefficient and \(V\) the volume of material removed by wear. To determine \(V\), an optical microscope was used to measure the dimensions of the wear marks on the upper specimen. From those measurements it was possible to calculate the worn volume. The wear coefficient \(K\) was then calculated using the sliding distance, \(S\), the hardness of the specimen, \(H\), and the applied force, \(F_n\).

The friction coefficient can also be measured in the apparatus. The exact friction coefficient varies throughout the cycle since it depends on the relative velocity of the specimen. Fig. 3 shows a schematic picture of how friction varies with sliding speed. At points A and B, the relative speed between the specimens is zero at which point the friction is purely static whilst between points A and B the friction is dynamic. The friction coefficient reported in this study is the mean friction.
Table 1
The base fluid, additive combination (add. comb.), chemical composition and concentration levels were the variables in the test lubricants

<table>
<thead>
<tr>
<th>Lubricant</th>
<th>EP Chemistry</th>
<th>Low</th>
<th>High</th>
<th>AW Chemistry</th>
<th>Low</th>
<th>High</th>
<th>Cu-passivator Chemistry</th>
<th>Low</th>
<th>High</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ester, add. comb. 1</td>
<td>P, N</td>
<td>0.20%</td>
<td>0.70%</td>
<td>S, P</td>
<td>0.30%</td>
<td>1.00%</td>
<td>N, S</td>
<td>0.02%</td>
<td>0.50%</td>
</tr>
<tr>
<td>Ester, add. comb. 2</td>
<td>P, N</td>
<td>0.20%</td>
<td>0.70%</td>
<td>S, N</td>
<td>0.30%</td>
<td>1.00%</td>
<td>N</td>
<td>0.02%</td>
<td>0.50%</td>
</tr>
<tr>
<td>Ester, add. comb. 3</td>
<td>P, N</td>
<td>0.20%</td>
<td>0.70%</td>
<td>S, P</td>
<td>0.00%</td>
<td>1.00%</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Ester, add. comb. 4</td>
<td>P, N</td>
<td>0.20%</td>
<td>0.70%</td>
<td>S, N</td>
<td>0.00%</td>
<td>1.00%</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Mineral, add. comb. 2</td>
<td>P, N</td>
<td>0.20%</td>
<td>0.70%</td>
<td>S, N</td>
<td>0.30%</td>
<td>1.00%</td>
<td>N</td>
<td>0.02%</td>
<td>0.50%</td>
</tr>
</tbody>
</table>

2.3. Operating parameters

The experiments were factorial designed to evaluate the lubricants under two lubrication regimes: mixed and boundary. It was therefore necessary to first determine which regime(s) were, in fact, present in the test system.

\[
\frac{H_e}{H_{11005}} = 1.714(W)^{-0.120}U^{0.694}G^{0.568} \quad (2)
\]

where \(W\) is the dimensionless load parameter, \(U\) is the dimensionless speed parameter and \(G\) is the dimensionless material parameter. The minimum film thick-
The dimensionless film parameter, $H_e$, is subsequently used to calculate the dimensionless film parameter $\Lambda$, Hamrock [23]:

$$\Lambda = \frac{\dot{H}_e}{(R_{e,a} R_{e,b})^{1/2}}$$

(3)

where $R_e$ is the effective radius and $R_{e,a}$ is the rms surface roughness for specimens a and b. Mixed lubrication is present when $1 < \Lambda < 5$ and boundary when $\Lambda < 1$. The mixed lubrication condition is only valid at sufficiently high sliding speeds and as long as the surface roughness is unaffected by wear. Test parameters concerning the specimen were kept constant through the tests. Other operating parameters are given in Table 2.

Case I represents boundary lubrication and was valid during the whole test cycle. In Case II, mixed lubrication was present near the centre of the oscillating movement, where maximum speed was achieved, whilst near the end points, boundary lubrication was present. The contact between two machine elements can be mild or severe. In most applications, an important task of the lubricant is to avoid severe wear, in order to achieve an acceptable working life. For Case I and II, severe wear was expected to occur especially in the beginning of the test, when the contact pressure was high. For this reason, it was also interesting to investigate a mixed lubrication regime with mild wear, Case III.

3. Results and discussion

The mineral oil based lubricant was used as reference material and was tested under both boundary conditions (Case I) and in mixed lubrication (Case II). In Case II, the sliding distance was much longer than in Case I, resulting in more wear. A summary of the additives that affect wear coefficients at a 95% confidence level are given in Table 3. Note that when mineral base oil was tested at Case I no results are reported, since it was impossible to evaluate the wear scars because they were too small. For the mineral oil based lubricant it can clearly be seen that the EP and AW additives have an impact on the wear coefficient. There was also a statistically significant interaction effect between the EP additive and the Cu-passivator. The “high” level of AW additive in the mineral oil resulted in increased wear. Possible explanations for this are competition between the AW and EP additives for surface sites or that the film forming process is upset by this particular combination of additives. According to Han and Masuko ([17,18]), excessive film formation due to very high levels of additive can lead to increased wear through a corrosive mechanism.

For both additive combinations in the synthetic ester based lubricants, and independent of lubrication regime, only the EP additive was found to have any significant effect on the wear coefficient. The wear coefficient was also reduced at the “high” EP concentration level. From the test series, it was possible to determine which additive combination gave the lowest wear and also to predict the minimum wear coefficient. These results are summarised in the two rightmost columns of Table 3. Since the wear coefficient is dependent on the test conditions (Case I or II), the wear coefficients should not be quantitatively compared between the cases.

As can be seen in Table 3, the wear coefficient for each Case differs a lot for the esters with additive combination 1 and 2. Surprisingly, although AW and Cu-passivator additives with different chemical composition and structure were used in additive combination 1 and 2, no significant effect on the wear coefficient was found. The expected additive response and interaction effects were observed for the mineral oil. One explanation for this could be that the AW effect is already “saturated” at the (arbitrarily) selected “low” concentration level, 0.30%, and thus no difference in the effect between these two levels was seen.

To verify whether AW additives actually gave any effect in the ester based lubricant blends, a new test series was run, see Table 1. In this test, additive combination 3 in the ester base fluid showed similar results as earlier i.e. only the EP additive had significant effect on the wear coefficient. For additive combination 4 in the ester base lubricant, EP, AW and the interaction between EP and AW had a significant effect on the wear coefficient, see Table 4. This indicates that the effect of AW chemistry in additive combination 4 is the same at all
Table 3
Additives affecting the wear coefficient at the 95% confidence level are marked with “X”. The table also shows the minimum wear coefficient and the optimised additive combination for each case. Only the EP additive showed any significant effect in the ester based lubricants. For the mineral oil based lubricant, EP, AW and the interaction effect between EP and Cu-passivator showed significant effects.

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</tr>
</thead>
<tbody>
<tr>
<td>Ester, add. comb. 1 Case I</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu-</td>
<td>1.39</td>
</tr>
<tr>
<td>Ester, add. comb. 2 Case I</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu+</td>
<td>0.560</td>
</tr>
<tr>
<td>Ester, add. comb. 1 Case II</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu-</td>
<td>2.38</td>
</tr>
<tr>
<td>Ester, add. comb. 2 Case II</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu-</td>
<td>1.87</td>
</tr>
<tr>
<td>Mineral, add. comb. 2 Case II</td>
<td>X</td>
<td>X</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>EP+AW+Cu-</td>
<td>0.417</td>
</tr>
</tbody>
</table>

Table 4
Additives affecting wear at the 95% confidence level are marked with “X”. The table also shows the minimum wear coefficient and the optimised additive combination for each Case. For additive combination 1, only the EP additive gave any significant effect. Additive combination 2 showed significant effect for EP, AW and the interaction between EP and AW additive.

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</thead>
<tbody>
<tr>
<td>Ester, add. comb. 3 Case I</td>
<td>X</td>
<td></td>
<td></td>
<td>EP+AW-</td>
<td>2.08</td>
</tr>
<tr>
<td>Ester, add. comb. 4 Case II</td>
<td>X</td>
<td>X</td>
<td></td>
<td>EP+AW+</td>
<td>1.91</td>
</tr>
</tbody>
</table>

* No Cu-passivator at all was present in these lubricants. The AW additive at “low” level was held at 0%.

concentration levels over 0.3%, but that a discernible effect is seen when the AW additive is present (remember that “low” in this test series is 0%). The wear also decreased with a “high” EP level. It should be noted that the difference between the wear coefficients (2.08×10^-6 and 1.91×10^-6) in the second test series is smaller than in the earlier test, (1.39×10^-6 and 0.560×10^-6; see Table 3). Also, the absence of the Cu-passivator in this second test series could explain the apparent “turn-on” of the AW effect at the higher concentration level, if an antagonism exists between the AW and Cu-passivator additives.

It was expected that a mild contact might still activate milder types of additives; which was the reason for the Case III test which resulted in mild wear under mixed lubrication. The only significant effect from this test was from the EP additive i.e. the same result as given by the synthetic ester in the first test series, Table 3. A “high” concentration level of EP additive resulted in practically no wear (minimum wear coefficient 0.011×10^-6), whilst the “low” level EP concentration gave approximately 40 times higher wear.

The wear coefficients for boundary lubrication (Case I) and mixed lubrication (Case II) are summarised in Fig. 4. The effect of the EP additive was most pronounced and reduced the wear most effectively. This was true for all lubricants and lubrication regimes tested. Under mixed lubricated conditions, Case II, the mineral oil based lubricant gave approximately four times lower wear coefficients than the ester based lubricants. This is in agreement with the results obtained by Rieglert and Kassfeldt [7]. The ester-based lubricant with additive combination 2 gave a lower wear coefficient than additive combination 1. It can also be seen that for practically all additive concentration levels and cases, additive combination 2 reduces the wear more effectively than additive combination 1.

All wear and friction measurements were done at an oil bulk temperature of 90°C. Some initial studies at lower bulk temperatures indicate that the AW additives tested may have a more significant influence on wear performance under these conditions. More work in this direction is needed and laboratory tests are proceeding to clarify those effects.

The mean friction coefficient measured during the tests, shows substantial differences between the ester-based lubricants and the mineral oil based reference lubricant, see Fig. 5. In Case I, the ester-based lubricants with a “high” level of EP additive resulted in a high friction coefficient. Disregarding the initial run-in period, Case II showed similar results. In both cases, the mineral oil based lubricant showed only a slight variation in the friction coefficient observed for the different additive combination. For the Case I lubrication regime, an additional experiment with pure ester base fluid without any additives gave the lowest friction. For Case II, the ester based lubricant blends with “low” levels of EP additives resulted in lower friction than those with
Fig. 4. Wear coefficient at boundary lubrication, Case I (left), and mixed lubrication, Case II (right). The ester-based lubricant with additive combination 2 gave lower wear coefficient than with additive combination 1. The mineral oil based lubricant gave approximately four times lower wear than the ester-based lubricant blends.

Fig. 5. Friction measurements from the Plint and Partner High Frequency Friction Machine. These show that the ester based lubricant with a "high" level of EP additive gave high friction (regardless of the differing chemical compositions and concentration levels of the AW and Cu-passivator) whilst a "low" level gave lower friction coefficients. The mineral oil based lubricant gave more uniform friction coefficients independent of the additive combinations.

"high" levels. In contrast to this, the mineral oil based lubricant gave slightly lower friction with a "high" EP level in Case I and Case II.

The different wear coefficients observed indicate that different wear mechanisms are acting in the contact. An optical microscope was used to examine the surfaces of the upper specimens following each test. A selection of micrographs of the worn surfaces is presented in Fig. 6. These show the flat worn surface where the sliding direction is horizontal. At the right hand side of each micrograph (and in some cases also on the left) the original ground roller surface can be seen.

For the ester based lubricant with "low" concentration level of EP, the surface appears to be smooth and shiny, see Fig. 6. micrographs a, b, e and f. For Case I when the EP level was "low", a rough calculation indicates that the average depth of material removed by each stroke is 0.2–0.3 nm. Typically, a cross section of the ground surface has a 0.3–3 nm thick physisorbed layer. The bond between this layer and the next layer are rather weak. When the physisorbed layer is sheared, the friction coefficient should therefore be very low. Beneath this first layer, a chemisorbed monolayer with a thickness of 0.3 nm occurs. This layer is covalently bonded to the metal surface and requires much higher shearing forces to yield. From the rough calculations and the friction measurements it appears that this second layer was also removed in all cases where the levels of EP additives were low.

In Fig. 6, micrographs c, d, g and h show the wear
Fig. 6. The worn contact surfaces of the upper cylindrical specimen photographed in an optical microscope. The micrographs show the flat worn surface where the sliding direction is horizontal. At the right hand side (and in some cases also on the left side) of each micrograph, the original grinding marks can be seen. Wear surfaces with synthetic ester blends with both "high" and "low" concentration levels of EP additive, for both Case I and Case II, are represented. At boundary lubrication (Case I), "high" EP, phosphorus builds a protective dark-coloured layer (micrographs c and d) while "low" levels of phosphorus does not seem to form any protective layer (micrographs a and b). For Case II, the film thickness was greater, which results in better surface separation. For this reason, the protective surface layer seen in micrographs c and d was not formed (micrographs g and h), although the levels of wear remained low. The surfaces in micrograph e and f appear to be similar to the surfaces in micrograph a and b, and all have suffered extensive wear (see Fig. 4).
with ester based lubricants containing “high” levels of EP additive. In Case I (micrographs c and d) the surface was covered with a coloured layer which is most likely a chemical reaction product of phosphorus, nitrogen and iron. Direct contact between asperities develops enough frictional heat to start these reactions. In micrographs c and d, adhesive wear also appears to be present, indicated by small areas of the coloured material being removed. When the lubrication film was thicker, as in Case II (micrographs g and h), the surfaces had greater separation and thus the contact temperature was presumably lower and the layer formed by chemical reaction not formed to the same extent.

4. Conclusions

The use of the synthetic ester based lubricant, with all additive combinations, resulted in much higher surface wear than for the mineral oil based lubricant with the same additives. The extreme-pressure (EP) additive was much more effective than any of the antiwear (AW) additives in reducing wear in both base fluids. Under boundary lubrication conditions (Case I), the EP additive formed a chemically reacted sacrificial layer.

The additive combination 2, with an AW additive containing sulphur and nitrogen and a Cu-passivator containing nitrogen, showed better wear resistance than additive combination 1, with an AW additive containing sulphur and phosphorus and a Cu-passivator containing nitrogen and sulphur.

The antiwear additive gave no significant effect in any synthetic ester blends except in additive combination 4. This indicates that these AW additives, at concentration levels above 0.3%, have already passed their optimum effect concentration.

The synthetic ester based lubricant with a “high” level of EP additive gave low wear coefficients and high friction coefficients. The corresponding “low” level of EP resulted in high wear coefficients and low friction coefficients. These results were independent of the lubrication regime; boundary (Case I) or mixed (Case II).

The friction coefficient appears to be lower with the “high” level of EP additives in the mineral oil based lubricant. The friction coefficient showed only slight variations with the different additive combinations tested (1 or 2).

The commonly used “AW” and “EP” labels associated with an additive species are poor indicators of performance and hence unreliable aids when designing lubricants with good antiwear performance in situations where base fluids other than mineral oils are used. The effect of the traditional antiwear additives in synthetic ester based blends appears to be very weak or insignificant, whereas a significant amount of wear protection actually is derived from the influence of the EP additive alone.

Future work should be undertaken to investigate the effects of chemically modifying the antiwear additives, or selecting other chemical species which could serve as antiwear additives in ester based lubricants.

Acknowledgements

The authors would like to thank Prof. Erik Höglund for advice and valuable support and Mr. Markus Nordlund for skilful assistance with the experiments.

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Paper E
1. Introduction

Rail replacements along the track are a direct consequence of wear and fatigue events in the rail/wheel contacts. The rail life depends on rolling contact fatigue (RCF) on the one hand and gage-face wear on the other. The contact region between rail and wheel is normally located as a single point contact on the railhead and includes an adhesion zone and a slip zone. In sharp curves, two-point contact can occur depending on wheel/rail profiles and dynamics. The flange contact encounters pure slip, as compared to the railhead contact. The curve radius and track elevation give wear or RCF damage. In general, small curve radii result in flange wear while more moderate radii result in RCF damage.

The use of a lubricant on the rail fulfills several requirements. If the lubricant is applied on the gage face, the rail/wheel flange wear is reduced. Another beneficial effect of reducing friction on the flange is the prevention of wheel climbing. At the rail head, friction control is mainly required to get adequate tractive force for the locomotive. However, a dry rolling/sliding contact results in high stresses in the material that may initiate cracks known as headchecks.

It is important to define the wear type, or wear mechanism, while discussing wear rates. Generally speaking, adhesive, surface fatigue, abrasive and tribochemical wear are commonly encountered wear mechanisms in practice. In the field of rail-wear research, three different wear modes have been identified by Bolton and Clayton [1]. These are termed as type I (mild wear), type II (severe wear) and III (catastrophic wear). These wear types are identified under dry conditions in bench test machines. From a field test Danks and Clayton [2] showed that an unlubricated 5° curve charged by heavy axle loads encounters type III wear. Field studies by Waara [3] on rails of 350 m curve radius and 22.5-t axle load gave flange wear in the range 0.5–3.8 nm per axle passage, depending on lubricant quantity applied. The lower wear rate was obtained when 130 g of lubricant per 30,000 axles was applied, and the higher wear rate was obtained following application of 7 g of lubricant per 30,000 axles. When the wear rates are over 2 nm per passing axle, the surface becomes very rough, possibly owing to severe wear (type II). These curves do not suffer from headchecks, most likely because of high wear rates. Other field measurements from Malmbanan (a severely...
loaded Swedish ore track, carrying 24 million gross tonnes per year) showed wear rate from the railhead down to the gauge corner in the range 0.18–0.24 nm/loaded axle at the high rail in curves, see Åhrén et al. [4]. Headchecks appear frequently and to remove the cracks the rails are ground every year. In view of this, it is worth exploring the idea of using a lubricant to increase the wear rate from 0.18 to 0.36 nm/loaded axle on this specific track to slow down headchecks growth, and reduce the frequency of grinding of rails to every second year.

Wear rates obtained from laboratory tests using ball-bearing steel in a high-frequency friction and wear test machine (Plint & Partner model TE77B) were ~0.08–0.32 nm per stroke. The wear rate was measured perpendicular to surface and the contact was lubricated by trimethylol propane–oleate (TMP-oleate) containing ~ 0.2–0.7 wt.% extreme pressure (EP) additive (an amine-neutralised phosphoric acid half ester). A mineral oil with the same EP additive resulted in wear rates as low as 0.02 nm per stroke, Waara et al. [5].

Surface damage on rails occurring from RCF is a mounting problem in heavily loaded tracks, as well as in high-speed tracks [6]. In heavily loaded railroads, RCF damage such as headchecks frequently occurs, because the gauge corner suffers severe plastic deformation from frequent passages of highly loaded wheel axles. Headchecks are formed in the contact between rail and wheel, when the wheel encounters a sliding motion at the gauge corner, causing a plastic flow under high contact pressures. After repeated plastic deformation, cracks begin to grow along the flow lines at an angle \( \theta \) as shown in figure 1. Initially, the nascent cracks grow parallel to the surface and as the cracks grow, they move down into the material. After a certain amount of growth, these angular cracks either grow toward the surface, or move further down in the railhead. When the cracks move upward, a chunk of material from the gauge corner along the track is removed. This process is known as spalling.

The crack propagation rates depend on many factors and there are models that enable estimation of crack growth rate taking in to account bending stresses, residual stresses from heat treatment or plastic deformation. The crack growth rate is also a function of the crack length and at a critical crack length the crack propagates rapidly [7]. If headchecks frequently appear at a rail track, grinding will have to be carried out to remove them. High wear rates are believed to eliminate headcheck initiation, but the wear rates required are rather high.

Severity labels to describe the plastic material flow downwards through the railhead, which leads to RCF damages, one elastic shake-down, plastic shake-down and ratcheting. Plastic shake-down and elastic shake-down describe the material flow when the contact pressures exceed the material limits. Lubricants tested in a twin-disc machine by Fletcher and Beynon [8] showed that mineral oil containing molybdenum disulphide (MoS2) or graphite resulted in very low wear rates, but the overall life was unfortunately reduced by RCF damage. Solid lubricants gave very long RCF life, but resulted in significantly higher wear rates. Water with graphite gave high wear rates, and the RCF life was also reduced. The fatigue effect of intermittent lubrication in a rolling/sliding contact was investigated by Fletcher and Beynon [9]. These tests were initially carried out for a small number of revolutions without any lubricant, followed by application of a lubricant to the contact during continuous running. Tests showed that intermittent lubrication reduces the RCF life significantly. This behaviour is analogous to that in which surfaces are wetted by water and followed by dry periods in a repeated manner.

For total loss applications, such as rail lubrication, saw chain oil and mobile hydraulic systems, environmentally adapted lubricants (EALs) are currently the preferred solution in environmentally sensitive applications. Two main considerations when designing total loss EALs are toxicity and biodegradability for both base fluids as well as performance additives. The use of synthetic esters as lubricant base fluid is an attractive solution, since these esters are biodegradable, non-toxic, and have good lubricating properties.

The addition of a small amount of fatty acids to mineral base oil results in significant reduction of the friction. For this reason, fatty acids have been commonly utilized as friction modifiers (FM). It has been shown, using the Langmuir–Blodgett (LB) technique, that an increased thickness of the fatty acid layer improves both wear and friction performance [10]. Another observation made by Dacus [10] is that increasing the carbon chain length of the fatty acids making up the monolayers leads to enhanced durability of the contact. Increased thickness of the fatty acids of the reacted surface layers improved the “low friction” durability.

The tribochemical reaction between a fatty acid and the metal oxide layer results in the formation of a metal soap on the contact surface [10]. The metal soap reaction layer results in smooth sliding and low friction. The level of metal soap formation depends upon

---

**Figure 1. Crack growth along the deformation lines.**
the type of metal the fatty acid reacts with. Generally speaking, steel, magnesium, aluminium and chromium surfaces have low tribochemical reactivity, while copper, cadmium and zinc surfaces have high reactivity.

The temperature dependence of lubricating effectiveness of different fatty acids dissolved in mineral oil was investigated by Bowden and Tabor [10]. The lubricating effectiveness was good at the temperatures in the range from the melting point (of the pure acid) up to the transition point. The transition point is the temperature at which lubricating film breaks down due to friction and occurred at $\approx 50-70$ °C above the melting point. In another study, Owens [11] investigated fatty acids (mono acids) dissolved in mineral oil and deduced that the carbon chain length of the fatty acids influences the surface protection. Carbon chain lengths from 6 up to 18 were tested and the longest chain gave the best possible packing conditions, resulting in improved surface protection. Another aspect concerning the base fluids is the Non-polarity index (NPI) [12]. Synthetic esters have lower NPI index and are more polar fluids than mineral oil. Within the group of synthetic esters, a more “oil-like” fluid will have higher NPI. This NPI difference can have an antagonistic effect between base fluid and additives. However additives usually have much stronger attraction to the surface than base fluids.

In this work, six fatty acids (oleic, stearic, decanoic, octanoic, azaleic and sebacic) were blended in turn in two different synthetic ester base fluids, TMP-oleate and TMP-C$_8$–C$_{10}$. The aim is to identify a suitable EAL formulation which tribochemically removes adequate material from the gauge corner to prevent initiation of headchecks. Medium contact pressure and relatively high sliding speed have been utilized during the experiments to simulate the gage face/wheel flange contact.

2. Experimental work

2.1. Test apparatus

A high-frequency reciprocating machine (Plint and Partner TE 77B) was employed for evaluating the tribochemical performance of new lubricant formulations. The contact geometry employed was a cylinder-on-flat line contact. This contact geometry adequately simulates the gage face/wheel flange contact where the contact pressure is lower as compared to that at the rolling contact surface. Typical rolling contact pressure at the rail wheel interface is of the order of 1.5 GPa and the application of 316 MPa contact pressure employed in these tests is considered to correspond to gage face/wheel flange contact pressure. The reciprocating motion in the test machine is somewhat similar to reversal of direction of sliding motion at the gage face that occurs on a track having traffic in both directions. A reciprocating drive arm reciprocates the cylinder specimen under controlled loading over the flat specimen, see figure 2. The specimens are kept in a small, temperature-controlled lubricant container. The specimens are manufactured from rail steel of the type 900A and 1100. The chemical composition of the steels and their hardness values are given in table 1. The test parameters are summarized in table 2.

The minimum film thickness is calculated according to Pan and Hamrock’s [13] curve fitting formula for rectangular conjunctions. The minimum film thickness is calculated for the maximum sliding speed in the reciprocating cycle. The physical property measurements utilized to calculate the viscosity was made by Pettersson et al. [14], and to calculate the absolute viscosity at the operating temperature Roeland’s description of temperature-viscosity relationship [15] was used. The minimum film thickness together with the surface roughness $R_q$ gives the dimensionless film parameter $A$. These tests are designed to give boundary lubrication (BL). The minimum film thickness $h_{min}$ and dimensionless film parameter $A$ for both lubricant base fluids in this study are given in table 3.

2.2. Lubricants

Two synthetic esters were used as base fluids, a TMP–oleate and a TMP-C$_8$–C$_{10}$. TMP is trimethylol propane (the polyol alcohol part of the ester), oleate is the monounsaturated C$_{18:1}$ fatty acid, and C$_8$–C$_{10}$ is the mixed fraction octanoic–decanoic (caprylic–capric) saturated fatty acids (making up the acid part of the ester). The choice of esters in this study was mainly based on their different NPIs. TMP-oleate is “oil-like” with a high NPI of about 185 whereas the TMP-C$_8$–C$_{10}$ is significantly more polar, with a NPI of about 65. Using esters with different NPI is one approach to investigate any synergism or antagonism between the added free fatty acid and the ester base fluid. Typical structures of the two ester base fluids are shown in figure 3 and their properties are summarized in table 3.
Six free fatty acids were chosen as performance additives. Three of the fatty acids were saturated mono-acids with different, straight carbon chain lengths (stearic acid C18, decanoic (capric) acid C10 and octanoic (caprylic) acid C8), one was a mono-unsaturated straight-chain fatty acid (oleic acid C18:1) while the other two were dibasic acids with intermediate carbon-chain length (sebacic acid, C10 and azaleic acid, C9), see table 4 and figure 4. Each fatty acid was blended at 2 wt.% content with one ester at the time, which gave 12 synthetic ester formulations in all.

### 2.3. Wear measurements and examination of worn surfaces

Wear of metal in lubricated contacts is controlled by many factors. The wear process normally starts with a run-in phase, during which surface asperities are removed or flattened out. In this reciprocating machine, specimen geometries result in high initial wear until the contact pressure is reduced due to increased area coming into contact.

In this study, results from friction measurements, wear volume, and wear type have been evaluated. The worn surfaces were examined in an optical microscope under polarized light with a view to characterising the surface topography and wear mode. Some specimens were also examined by using scanning electron microscopy (SEM) with a view to understanding the nature of surface damage.

### 3. Results and discussion

Wear rates and the morphology of worn surfaces were two main aspects to be investigated during this study. The lower flat supporting surface was almost unaffected by the end of the tests, and the examination of worn surfaces indicated just flattening of original
grinding marks. Most wear occurred on the upper cylindrical surface, and the extent and character of this wear has been the basis of wear evaluation. The width of the upper specimen contact area compared to the stroke length makes the upper specimen 137 times more prone to wear as compared to lower supporting specimen. The contact situation is comparable to a piston ring and a cylinder linear.

A multifactor ANOVA analysis was made to evaluate the wear. Three components were considered in this analysis: material type (composition and hardness), base fluid type, and free fatty acid type.

In these tests, the material type was found to have some influence on the wear characteristics and, contrary to what is normally observed in dry conditions, the harder material (steel grade 1100) suffered higher wear, as can be seen in figure 5. The wear rate seems to be governed by the specimen’s surface roughness and hardness. When the lower supporting specimen is hard and rough, the grinding marks on its surface are unaffected and lead to higher wear of the upper specimen. The supporting specimen surface roughness has considerable influence on the wear rate, which is analogous to severe conditions encountered in marginally lubricated contacts. If the rail track flange becomes rough, it then encounters rapid increase in wear.

Wear rate dependence on different lubricants was analysed, and the tests showed that the wear rate was significantly lower if the base fluid contained 2% added free fatty acid, see figure 6. Pure esters were not able to form any lubricating film in these tests. The type of base fluids did not indicate any significant influence on the wear rate, see figure 7. The differences between these base fluids were mainly in their viscosities and the NPI indexes, but none of these properties had any significant influence on wear. The NPI index was however not expected to give any effect, as the fatty acids attraction to the surface is much stronger compared to that of the base fluids. The viscosity also was not expected to have any influence on wear under boundary lubrication conditions. When the differences between synthetic ester with fatty acids were analysed statistically, pure synthetic ester was excluded from the analysis. Oleic and azelieic acid gave the lowest wear, but it was not possible to statistically distinguish them. The wear rates obtained by using stearic, decanoic, and sebacic acids were significantly higher and it was possible to statistically distinguish the results. Octanoic

<table>
<thead>
<tr>
<th>Chemical name</th>
<th>Formula</th>
<th>m_w (g/mol)</th>
<th>Melting point (°C)</th>
<th>Boiling point (°C)</th>
<th>Density* (g/ml)</th>
<th>Flash point (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oleic acid</td>
<td>C_{18}H_{34}O_{2}</td>
<td>282.46</td>
<td>13.4</td>
<td>286</td>
<td>0.891</td>
<td>189</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>C_{18}H_{36}O_{2}</td>
<td>284.48</td>
<td>69.3</td>
<td>361</td>
<td>0.847</td>
<td>196</td>
</tr>
<tr>
<td>Decanoic acid</td>
<td>C_{10}H_{20}O_{2}</td>
<td>172.27</td>
<td>31.5</td>
<td>266-270</td>
<td>0.901</td>
<td>112</td>
</tr>
<tr>
<td>Octanoic acid (caprylic)</td>
<td>C_{8}H_{16}O_{2}</td>
<td>144.21</td>
<td>16.3</td>
<td>239.7</td>
<td>0.911</td>
<td>130</td>
</tr>
<tr>
<td>Sebacic acid</td>
<td>C_{10}H_{18}O_{4}</td>
<td>202.25</td>
<td>134.5</td>
<td>294.4</td>
<td>0.911</td>
<td>220</td>
</tr>
<tr>
<td>Azelieic acid</td>
<td>C_{9}H_{16}O_{4}</td>
<td>188.22</td>
<td>100</td>
<td>286</td>
<td>0.911</td>
<td>210</td>
</tr>
</tbody>
</table>

*Density of melts.

Figure 4. Molecular structures of the free fatty acids studied: (a) Oleic, (b) Stearic, (c) Decanoic, (d) Octanoic, (e) Sebacic and (f) Azelieic.

Figure 5. Mean wear results and 70% confidence interval for steel with different hardness.
acid gave the highest wear rate, considerably higher than that encountered with other free fatty acids, see figure 8.

Another aspect of the wear problem is the surface damage. The evaluation of the worn surfaces admittedly is somewhat subjective, but there are differences to emphasize. Scanning electron microscope examination of worn surfaces from tests using pure esters has indicated surface features quite similar to those seen in fine two and three-body abrasive wear, see figure 9(a). TMP-C₈–C₁₀ with oleic acid resulted in a smooth surface but the surface was covered by some fine pits and micro-crack-like surface flaws, see figure 9(b).

Oleic acid has the lowest melting point of all tested fatty acids, and it is possible that the metal soap undergoes thermal decomposition or degradation. The
worn surfaces for both steels from tests using TMP-oleate with stearic acid are relatively smooth as can be seen figure 9(c). Use of TMP-C<sub>8</sub>–C<sub>10</sub> with either dibasic acid (sebacic or azaleic acid) also resulted in smooth surfaces in both steels similar to that shown in figure 9(c).

Table 5. Surface characteristics.

<table>
<thead>
<tr>
<th></th>
<th>900A</th>
<th>1100</th>
<th>900A</th>
<th>1100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without fatty acid</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
<td>Mild abrasive wear</td>
</tr>
<tr>
<td>Oleic acid</td>
<td>Surface partly covered by thin layer</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear some scoring marks</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>Tribochemical wear very smooth surface</td>
<td>Tribochemical wear very smooth surface</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear scoring marks</td>
</tr>
<tr>
<td>Decanoic acid</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear scoring marks</td>
<td>Tribochemical wear scoring marks</td>
<td>Tribochemical wear scoring marks</td>
</tr>
<tr>
<td>Octanoic acid</td>
<td>Surface covered by thin layer</td>
<td>Tribochemical wear scoring marks</td>
<td>Tribochemical wear mild abrasive wear</td>
<td>Mild abrasive wear</td>
</tr>
<tr>
<td>Sebacic acid</td>
<td>Surface partly covered by thin layer</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear some scoring marks</td>
</tr>
<tr>
<td>Azaleic acid</td>
<td>Tribochemical wear some scoring marks</td>
<td>Tribochemical wear very smooth surface</td>
<td>Tribochemical wear very smooth surface</td>
<td>Tribochemical wear quite smooth surface</td>
</tr>
</tbody>
</table>

Table 6. Element analysis in SEM/EDS at a worn surface.

<table>
<thead>
<tr>
<th>Elements</th>
<th>Without visible layer</th>
<th>With a visible layer</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>wt.%</td>
<td>wt.% std. error</td>
</tr>
<tr>
<td>C</td>
<td>7.98</td>
<td>± 10.84</td>
</tr>
</tbody>
</table>

The detailed microscopic examination results of worn surfaces have been summarised in table 5. Dibasic acids blended in synthetic acid seem to form a layer on the contacting surface. This layer showed higher carbon content in SEM/EDS analysis indicating the formation of fatty acid metal soap film on the surface, see table 6.

An earlier study by Waara et al. [5] aimed at evaluation of the effect of environmentally adapted AW and EP additive added to synthetic esters showed that TMP-oleate with low additive concentration or without additives resulted in tribochemical wear. These experiments were made on ball bearing steel (850 HV), which is about 2.5 times harder as compared to 900A steel. In this study, free fatty acids were added in an effort to control the wear type, as well as the wear rate. This present study showed that softer steel lubricated by pure TMP-oleate or TMP-C<sub>8</sub>–C<sub>10</sub> will suffer mild abrasive wear. Tribochemical wear occurred only when free fatty acid was added to the synthetic ester. Some of the fatty acids in listed in table 5 can indeed induce tribochemical wear. The earlier results were different from the results of this study, mainly owing to the use of softer rail steel materials. The present experiments were performed at a bulk temperature of 90 °C.

The transition temperature for oleic acid dissolved in TMP-C<sub>8</sub>–C<sub>10</sub> was obtained by measuring the friction in the Cameron & Plint apparatus, while the temperature was increased slowly from 30 to 95 °C, see figure 10. This test showed that the coefficient of friction increased linearly as the temperature rose to 77 °C and it stabilised above this temperature. When the temperature was lowered, the coefficient of friction started to fluctuate at 77 °C. This experiment indicates
that the oleic acid (with melting point 13.4°C) has some sort of transient temperature about 64°C above the melting point of the pure acid. Good polishing quality can probably be obtained at lower temperatures for fatty acids with low melting points such as oleic, decanoic and octanoic acid.

In these experiments, different frictional characteristics were obtained for different ester and fatty acid blends, see figure 11. As mentioned in the section 1, the wear process normally starts with a run-in period, followed by a steady-state period. The absolute coefficient of friction is not related to the wear rate, but the relative differences in the coefficients of friction can indicate changes at the interface such as surface topography and the formation of reaction layers, etc. These experiments show that a pure ester base fluid had a relatively high initial coefficient of friction, followed by gradual reduction in friction when the asperities are flattened out. Ester with added fatty mono-acids showed similar friction characteristics as the pure esters, but the friction level is lower. The fatty dibasic acids blended with TMP-C8–C10 are different from the others, with a rapid friction reduction to very low levels initially, followed by increasing friction. The coefficient of friction increase up to 0.185 during the tests and after that the coefficient of friction slowly decreases. Observations from the oscilloscope showed that the friction at low sliding speed at reversal points (in the reciprocating apparatus) was considerably higher than at higher sliding speeds.

4. Conclusions
- Stearic and azaleic acid dissolved in ester base fluids, such as TMP-oleate or TMP-C8–C10, reduce wear rates for both rail materials. Ester base fluids without free fatty acids result in severe wear.
- Stearic acid and azaleic acid result in very smooth surfaces with a significant difference in wear. Stearic acid dissolved in ester resulted in wear volumes of \(33 \times 10^{-5}\) mm\(^3\)/m while Azaleic acid dissolved in gave \(12 \times 10^{-5}\) mm\(^3\)/m. The resultant smooth surfaces, and the difference in their wear rates, make these two free fatty acids useful in obtaining tribochemical wear in sliding contact.

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References
Controlled tribochemical wear by using synthetic ester model lubricant in rolling, sliding rail/wheel contacts

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Keywords: rail, wheel, wear, head checking, RCF, synthetic, ester, fatty acid.

Abstract

Two factors limiting railway infrastructure productivity are rolling contact fatigue (RCF) and wear. The idea in this paper is that the use of synthetic esters will give sufficient wear to remove crack initiations and prolong rail life at heavy loaded tracks. Fatty acids were added to synthetic ester formulations aimed at increasing wear rates. A twin-disc machine was used to evaluate the lubricant’s effect between rail/wheel discs under high pressure. The results showed the need for increased tribochemical wear to attain sufficiently high wear rates to remove cracks. Using a lubricant that produces relatively high wear rates to avoid RCF damage is in an early stage, and many solutions to achieve the desired results probably exist. Hence, more research is needed in this area.

1 Introduction

The rail/wheel contact on track curves displays a one or two point contact situation. A single point contact between wheel and rail head can be identified as a rolling/sliding contact, sometimes displaying stick and slip areas inside the contact. A two point contact includes contact between wheel flange and rail gauge face in addition to the one on top of the rail. In curves the rolling/sliding contact spot moves from centre of the rail head towards the gauge corner. When the contact spot moves, the profile radii decrease for both rail and wheel, leading to decreased contact area. High pressure in combination with creep forces leads to plastic material flow from the gauge corner towards the gauge face.

RCF and wear are two major limitations of railway infrastructure productivity. Besides RCF, wear of railhead and wheel flange is a central maintenance issue for railway operators. Lubricants are routinely applied to the sides of the rails in curved track sections to reduce friction and wear occurring between the wheel flange and the gauge face of the rail.

1.1 Wear in lubricated contacts

Lubricated conditions in laboratory tests typically provide very low friction coefficients beneath 0.1. Such low friction introduces a wear behaviour described as “filmy wear” by Akagaki and Kato [1], who conducted an experiment with a hard slider in contact with a machined softer surface. Very thin slivers <0.1µm thick were extruded from the edges of the asperities on the softer surface. Soft transversely orientated asperities (from machining the surface) extrude material from the trailing edge. If the soft asperities are orientated parallel to the sliding direction, the slivers extrude from the asperities laterally. Kapoor and Johnson [2] modelled this plastic flow with the use of a kinematical shakedown theorem. The mechanism of plastic ratchetting (accumulation of plastic flow with repeated loading) acts in two ways: the soft surface becomes cyclically loaded by stress concentrations associated with the edges of the harder slider and pummelling action of the softer surface by asperities of harder slider.
Wear can also occur by ratchetting failure. In this mechanism, the near surface material loses its integrity after accumulating large plastic deformation and detaches as wear debris. Kapoor and co-authors have modelled this wear [3,4,5,6].

A softer, rough surface at the start became quickly “run-in” while the remaining contact occurred between the rough hard slider and flattened tops of soft asperities, Kapoor and Johnson [7]. The rough surfaces at the harder slider came in contact with asperity peaks where the contact pressure became much higher than the nominal contact pressure. From Johnson [8], the contact stress below these asperity contacts was immediately severed, but decayed rapidly as the square of distance.

1.2 RCF damages

When a cyclic load is applied at a wheel/rail contact the material will respond in 4 ways. No plastic deformation occurs if the stress is below the materials yield point. If the stress exceed the yield point but stay less than the elastic shakedown limit the material will undergo some plastic deformation. This deformation might introduce work hardening and also protective residual stresses. Stress beneath the shakedown limit leads to certain loads cycled with plastic deformation, though the elasticity of the following load cycles returns. This material response is called elastic shakedown. If the load introduces stress over the plastic shakedown limit it results in plastic deformation for each load cycle until the stress-strain curve becomes a stabilised loop. This material response is called plastic shakedown. Stress levels over the plastic shakedown limit add plastic deformation for each load cycle and are known as ratchetting. High cycle failure might occur if loadings are beneath the plastic shakedown limit, while low cycle failure would then occur when the load exceeds the shakedown limit. When materials are unable to sustain further plastic deformation, ratchetting failure occurs.

A lubricated contact imply low friction and can not be expected to give any plastic deformation on the surface but it is more likely that the subsurface will achieve some plastic strain according to shakedown maps as seen in Fig. 1, Johnson [9].

The yield stress at rail material 900A is initially 469MPa (Telliskivi and Olofsson [10]) and so the shaking limit is 4 x 469 MPa for low friction coefficient if the effect of roughness is ignored. The material will respond elastically in the steady state if the contact pressure is below this limit, and will last a long time, failing ultimately by high cycle fatigue. The effect of roughness cause plastic flow in the subsurface layer, where crack-like flaws may initiate, see Kapoor and co-authors [11,12,13,14].

Such cracks have been studied in a twin-disc machine by Fletcher and Beynon [15]. Crack propagation under lubricated and intermittent lubricated conditions was measured and a lubricated contact showed crack propagation of about 0.3 nm/load cycle at about 20-30° angle, which corresponds to a crack growth rate of ~0.12 nm/cycle perpendicular to the surface in the tested steel. Intermittent lubrication showed a
crack growth rate which correspond up to ~4.3 nm/cycle perpendicular to the surface, while a dry contact gave up to ~22 nm/cycle. When a crack has been initiated the crack growth rate is a function of the crack length; at a critical crack length the crack propagates rapidly, Kapoor [16].

There is an interaction between fatigue and wear. Wear is the dominating surface damage for instance in sharp curves at the rail gauge face while fatigue is dominating at the rail head. One of those surface damages will dominate depending on the wear or crack speed. At the rail head the geometric situation lead to very little sliding wear and therefore the most common damage type is rolling contact fatigue. Grinding is considered as artificial wear and widely is used to remove cracks from the rail track. The idea in this paper is to find a lubricant that in a mild way gives a sufficient wear rate to avoid crack initiations.

1.3 Work hardening

Repeated loading will lead to rolling contact fatigue (RCF). Squats, shelling, and head check’s are all forms of RCF, but head check’s are prevalent in curves where the wheel root comes in contact with the rail gauge corner and results in high pressure. The work hardening process depends on the material microstructure and is driven by the plastic deformation on the steel. The yield point indicates were the material start to deform plastically. Hardness measurements shows not the connection between stress and strain but indicate that some plastic strain have happened in the material. As an example a field test done by Olofsson and Telliskivi [17] there hardness measurements indicated were the material was worked hardest. A slightly higher hardness was found in unlubricated rail curves compared to lubricated. From Olofsson and Telliskivi [17], it was shown that the gauge face and the gauge corner had higher hardness compared the rail head.

1.4 Surface chemistry

The load influences formation of the boundary layer and the choice of additives. Even if the load is mild it may require friction modifiers in the form of polar HC-molecules as fatty acids, alcohols, or esters into the lubricant to physically adsorb to the surface. The function of these layers is limited by their thermal instability. The occurrence of desorption depends on the HC-molecule type at temperatures from 50°C up to 170°C. Moderate loading renders antiwear additives useful, thereby ensuring mild or low wear. Here, the chemical composition is polar HC-molecules that include a phosphor or a sulphur element. This molecule will physically adsorb to the surface, but form a metal soap after chemical reaction. These types of layers are stable over 400°C. Extreme loading requires anti-scuff or extreme pressure additives to permit excessive wear protection. The chemical composition is basically complex compounds based on phosphorus, sulphur, chlorine, and boron that react with the surface and build multi layers of metal salts.

The additives give thermal stability under load but another important factor is the possibility to influence shear strength. Adsorbed boundary layers of stearic acid forming Fe stearates gave lower shear strength compared to chemical reacted Fe-oxides or –sulphides at a similar pressure, see Briscoe et al [18] and Bridgman [19]. For that reason stearic acid is usually used as friction modifier in lubricants.

Recently, a matrix of synthetic ester formulations was evaluated in a reciprocating rail steel contact (HV ~300), see Waara et al [20]. TMP-Oleate added with 2 wt% stearic, azaleic and decanoic acid gave tribochemical wear with very smooth surfaces. The volume of acid may be reduced because the acid did not completely dissolve in the ester solution.

2 Experimental method

Dry conditions in high pressure rolling sliding contacts generate high wear rates. Well lubricated contacts diminish wear to very low levels, but crack growth does not decease at the same magnitude, in fact it can increase due to fluid entering the crack and pressuring the tip, making it easier for the crack faces to slide.
The idea in this paper was to use a lubricant that provides a sufficiently high wear rate. This wear rate was thought to continuously shear off thin surface layers. By continuously removing surface layers the cracks should not be able to initiate at the surface. Such a lubricant is thought to be physically adsorbed to the surface and then easily sheared off. This is useful at the rail head therefore example head checking usually appear. Liquids similar to the lubricants tested in Waara et al [20] was chosen because synthetic esters with the addition of fatty acids seem to be an interesting choice. This lubricant formulation was assumed to function well at the interface between gauge corner and wheel root. To simulate such a rolling/sliding high loaded contact, a twin-disc machine was used.

2.1 The lubricants

Five different lubricants were tested with Table 1 showing their formulations. Three lubricant formulations were mixtures of fatty acids and synthetic ester while two were pure base fluids of the type TMP-Oleate and Poly alfa olefin (PAO). TMP is trimethylol propane (the polyol alcohol part of the ester), oleate is a monounsaturated C18:1 fatty acid, and C₈-C₁₀ is a mixed fraction Octanoic-Decanoic (Caprylic-Capric) saturated fatty acids (making up the acid part of the ester). The synthetic esters as base fluids were basically chosen because they have excellent lubricating properties, good biodegradability, and are non-toxic. The base fluids were blended one by one with Oleic, Stearic and Azaleic acid (added by 0.3 wt%). From [20], the addition of 0.3 wt% was assumed to produce an effect. PAO was chosen since it contains no sulphur or phosphor and the results should reflect the lubricants viscosity.

<table>
<thead>
<tr>
<th>Table 1. Viscosity properties and contact parameters.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lable</td>
</tr>
<tr>
<td>-------</td>
</tr>
<tr>
<td>S1</td>
</tr>
<tr>
<td>S2</td>
</tr>
<tr>
<td>S3</td>
</tr>
<tr>
<td>S4</td>
</tr>
<tr>
<td>PAO</td>
</tr>
</tbody>
</table>

2.2 The SUROS machine and test parameters

The test rig used in this investigation was a twin-disc machine, known as the SUROS machine (Sheffield University Rolling Sliding machine). A complete description of the machine is found in Fletcher and Beynon [21]. In this paper, the description is limited to the machines basic functions. A schematic picture of the machine is given in Fig. 2 and the operating parameters in the test machine and the disc specimen’s surface roughness are shown in Table 2.

<table>
<thead>
<tr>
<th>Table 2. Operating parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Po [MPa]</td>
</tr>
<tr>
<td>1500</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>

Blue light is steel material aimed for wheels made by Lucchini. 900A is steel material aimed for rail.

The twin-disc machine is based on a Colchester Mascot 1600 lathe, providing the drive system for the rail disc. An AC motor mounted on the machine bed drives the wheel disc. A hydraulic cylinder fitted with a load cell applies and controls the load, and presses the discs together. The AC motor operates together with a frequency controller to attain the right slip speed. A torque transducer is fitted on the axle between the rail disc and the lathe. From calibration, it was found that the real torque lies within ± 5% of a reading value of ~6 Nm, considered valid for oil lubricated contacts. The discs width was 10 mm with a diameter of 41.97 ± 0.01mm. The tested lubricant floods the inlet region and separates the discs in the contact
region. A real rail/wheel contact is generally more starved lubricated but this test evaluate the lubricants performance and from that view starved lubrication will disturb the results to much.

![Schematics of the Twin disc machine](image)

**Fig. 2. Schematics of the Twin disc machine, 1) lathe, 2) AC motor, 3) torque transducer, 4) hydraulic cylinder, 5) rail disc, 6) wheel disc, 7) gear, 8) bearing assemblies.**

The film thickness $h_{\text{min}}$ is calculated according to Pan and Hamrock’s [22] formula for rectangular conjunction while the relation between $h_{\text{min}}$ and the Root mean square (rms) roughness is described by the dimensionless film parameter ($\Lambda$). The Herzian pressure $p_o$ is calculated by Timoshenko and Goodier’s formula [23] for dry contacts.

$$p_o = 0.418 \left( \frac{WE}{BR} \right)$$  \hspace{1cm} (1)

where $W$ is the load, $B$ is the line contact length, $E$ is the elasticity modulus, and $R$ the disc radius. The calculated film thickness and the dimensionless film parameter are shown in Table 3. As seen $\Lambda$ increase 4-5 times when the surfaces becomes smoother by “running in” the surfaces.

<table>
<thead>
<tr>
<th></th>
<th>$h_{\text{min}}$ [(\mu\text{m})]</th>
<th>$\Lambda$</th>
<th>Initially</th>
<th>100000 cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1</td>
<td>0.156</td>
<td>0.11</td>
<td>0.49</td>
<td></td>
</tr>
<tr>
<td>S2</td>
<td>*</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>S3</td>
<td>*</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>S4</td>
<td>0.0830</td>
<td>0.056</td>
<td>0.26</td>
<td></td>
</tr>
<tr>
<td>PA0</td>
<td>0.147</td>
<td>0.099</td>
<td>0.46</td>
<td></td>
</tr>
</tbody>
</table>

Table 3. Lubricating film properties.

The slip in the contact was controlled cumulatively as well as instantaneously. The cumulative slip $S_c$ is defined by the following formula, see Ref. [21].

$$S_c(\%) = 200 \times \left( \frac{R_{\text{rail}} \cdot N_{\text{rail}} - R_{\text{wheel}} \cdot N_{\text{wheel}}}{R_{\text{rail}} \cdot N_{\text{rail}} + R_{\text{wheel}} \cdot N_{\text{wheel}}} \right)$$  \hspace{1cm} (2)

where $R$ is radius of the disc specimen and $N$ is the revolution count of the discs. To get a fast control of the slip, the instant slip $S_i$ is used to control the ac motor, defined by the formula

$$S_i(\%) = 200 \times \left( \frac{R_{\text{rail}} \cdot V_{\text{rail}} - R_{\text{wheel}} \cdot V_{\text{wheel}}}{R_{\text{rail}} \cdot V_{\text{rail}} + R_{\text{wheel}} \cdot V_{\text{wheel}}} \right)$$  \hspace{1cm} (3)

where $V$ is the disc speed in rpm.
2.3 Experimental work

Before and after testing, the discs were cleaned in an ultrasonic bath using a solvent (Ronseal Tri-flow Orange 2). Finally the discs were cleaned with acetone in an ultrasonic bath for another five minutes. After the cleaning process the discs were separately weighed to evaluate the wear with a high resolution scale (0,0001g). The disc surfaces were examined using an optical microscope and a scanning electron microscope (SEM) to indicate the wear type and cracking. To evaluate the influence from friction and plastic deformation of the surfaces, cross sections from the rail discs were cut out. The surface roughness was measured perpendicular to the rolling/sliding direction that coincides with the direction of the grinding marks, before and after running a test. The initial surface roughness for the individual discs is shown in Table 4. Friction was monitored continuously during the test. The rail disc hardness was measured from the surface down to 1 mm at the subsurface by using a micro Vickers hardness indenter.

Table 4. The initial surface roughness for the individual discs measured perpendicular to the disc grinding marks. The grinding marks direction coincides with the discs sliding direction.

<table>
<thead>
<tr>
<th>Lubricant (Disc nr)</th>
<th>S1 (13)</th>
<th>S1 (16)</th>
<th>S2 (50)</th>
<th>S3 (55)</th>
<th>PAO (96)</th>
<th>S4 (97)</th>
<th>S1 (18)</th>
<th>S2 (19)</th>
<th>S2 (15)</th>
<th>S3 (11)</th>
<th>S4 (12)</th>
<th>PAO (14)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Rq (wheel)</td>
<td>0,96</td>
<td>0,98</td>
<td>0,97</td>
<td>0,99</td>
<td>1,16</td>
<td>0,70</td>
<td>0,86</td>
<td>1,06</td>
<td>0,85</td>
<td>0,80</td>
<td>1,06</td>
<td>1,02</td>
</tr>
<tr>
<td>Initial Rq (rail)</td>
<td>0,78</td>
<td>1,06</td>
<td>1,03</td>
<td>0,99</td>
<td>1,07</td>
<td>1,01</td>
<td>1,09</td>
<td>0,79</td>
<td>1,02</td>
<td>1,20</td>
<td>1,05</td>
<td>1,22</td>
</tr>
</tbody>
</table>

3 Results

3.1 The wear rates

The wear rates from the twin-disc tests summarised in Fig. 3 separate the rail and wheel disc wear for different lubricating fluids. Material transfer between the discs is not considered. This test showed that by using synthetic formulations S3 and S4 the wear rate was of the same magnitude as PAO, 0,003-0,009g (or 0,012-0,035 nm/cycle) in the 25,000 cycle tests. With these lubricants, wear rates gradually decreased due to “run-in” with the 100,000 cycle tests giving wear rates in the range ~0,01g or 0,01 nm/cycle. The wear rate 0,035 nm/cycle corresponds to removing 2,134 nm/sliding meter in the 25,000 cycle test. The results from the disc test lubricated with S1 and S2 showed partly high wear rates, but with large differences.

![Fig. 3. The diagram shows the quantity of worn off material from individual discs. The test was divided into two series there the first was running for 25 000 cycles and the second for 100 000 cycles. Each row in the diagram represents the pair of discs which was tested together there the left hand side shows the wheel disc while the right hand side shows the rail disc. Some of the tests were repeated as can be seen in the diagram.](image-url)
3.2 The coefficient of friction

The coefficient of friction depended on two mechanisms. The surface roughness initially influenced the friction until the asperities were flattened down. After the “run in” phase the influence of the asperities became weaker with the lubricants’ properties influencing the coefficient of friction, see Fig. 4.

Fig. 4. In the left diagram the coefficient of friction for the 25 000 cycle test is presented. In the right diagram the coefficient of friction was monitored for the 100 000 cycle test.

Compared to pure esters the addition of fatty acids did not influence friction. A significant difference was found between PAO and the synthetic formulations. When the same lubricant solution was tested for 25 000 and 100 000 cycles, it was expected that the coefficient of friction would not diverge much. However, the friction results indicated a significant divergence that was suspected to be caused by the discs surface roughness. A linear regression analysis was made to find out if there was any correlation between the initial surface roughness and the friction coefficient. In Fig. 5 a-d, the monitored coefficients of friction after 800, 3500, 25 000 and 100 000 cycles were plotted against the initial surface roughness (wheel disc) of each individual disc couple. The fitted black line is drawn by least square analysis. Any slope on the fitted line indicates a relation between surface roughness and friction. The analysis showed a moderately strong relation between the initial surface roughness (rms) and the coefficient of friction at 800, 3 500 and 25 000 cycles, but not after 100 000.
Fig. 5. The correlation between the surface roughness and the monitored coefficient of friction was analysed using linear regression. The test showed a moderately strong relation in Figures a-c while Figure d showed no correlation.

The coefficient of friction is initially at its highest level and decreases during the “run in” phase. If the harder disc had a rough surface the friction decreased slowly, whereas the friction coefficient diminished rapidly if the harder surface was initially smooth. The synthetic formulations S1 that ran for 25 000 cycles and S2 for 100 000 cycles did “run out” instead of being “run in”. “Run out” means that the surfaces gets a lot of cracks combined with high wear rates. The discs that “run out” had higher initial friction compared to ester based lubricants. PAO showed the highest long term friction, though these rail discs did not “run out”. The proposition is that “run out” of surfaces appears when the harder surface is rougher compared to the softer surface. High local friction gave local plastic deformation at the surface and caused cracks.

3.3 The surfaces

When examining the surfaces through the optical microscope and the scanning electron microscope (SEM), it was found that the surfaces could be divided into two groups, since PAO differed from synthetic ester formulations. SEM micrographs show the general differences between the surfaces. In Fig. 6a, the surface, lubricated by S1 for 100 000 cycles, was covered by cracks and was representative of all surfaces lubricated by synthetic ester formulations tested here. Fig. 6b shows a surface lubricated by PAO after 100 000 cycles with far less cracks than any surface lubricated by synthetic ester formulations.
3.4 The hardness

The subsurface hardness was measured for rail disc tested using PAO and synthetic ester formulation S2 as lubricant. Micro Vickers technique was used at a polished cross section with loads between 25-50 g. From Fig. 4, the coefficient of friction seems to emphasise the differences with PAO showing an overall higher friction coefficient ($\mu_{\text{average}}=0.050$) and S2 an overall lower friction ($\mu_{\text{average}}=0.034$).

In Fig. 7 the subsurface’s hardness was plotted vs. its depth. The hardness of the disc surfaces was also measured and added to this diagram. A polynomial was fitted to the subsurface hardness to evaluate the hardness distribution, indicating a local hardness maximum a bit lower in the material. It is known that increased friction increases the stress intensity, as the local stress maximum moves closer to the surface, see Johnson [8]. The maximum work hardening of the subsurface material may be affected by the stress resulting from friction.

![Diagram](image1)

**Fig. 7.** The diagrams show subsurface hardness of two rail discs. The rail disc in diagram a) had a friction coefficient of $\mu_{\text{average}}=0.050$ (lubricated by PAO) while diagram b) had a friction coefficient of $\mu_{\text{average}}=0.034$ (lubricated by S2).

A rail disc cross section shows how the pearlitic grain structure has been influenced by the lubricated rolling/sliding contact, see Fig. 8. In the interval 100-400 µm, the pearlitic grains have been deformed by plastic flow. Within this interval the maximum hardness is located (see Fig. 7) at the depth 200-250 µm. Below ~600 µm the pearlitic grains are unaffected and the bulk material properties would be expected. This is indeed the same as observed in Fig. 7.
The hardness measurements in Fig. 7 indicate that material very close to the surface was almost unaffected by work hardening. In Fig. 9 a rail disc cross section was examined in an SEM. The laminar pearlitic structure below the white dotted line at depth ~8 μm seemed unaffected which agrees with decreasing subsurface hardness close to the surface in Fig. 7 (subsurface hardness was not measured closer than 23 μm from the surface). Above the white dotted line in Fig. 9 the laminar pearlitic structure has deformed plastically in the sliding direction. The hardness measurements from the rail disc surfaces showed this surface layer to be work harden as can be seen in Fig 7.

Additional tests on surface hardness demonstrated rail discs lubricated by synthetic formulations of the type S1 and S2 to have very hard and very soft spots at the surface, while rail discs lubricated by PAO generally resulted in a slightly harder surface with less variation in hardness, see Fig. 10. This result may be coupled to work hardening caused by the higher friction given by PAO during the whole test.
Fig. 10. Rail discs lubricated by PAO resulted in higher hardness at the surface compared to discs lubricated by synthetic formulations. The central boxes cover 50% of the measurements and all the measurements are included inside the whiskers (outliers are marked by squares). The line in the box is the median and the plus signs represent the average values.

What is the difference between the twin-disc subsurface hardness distributions built up under laboratory conditions compared to actual conditions? Samples of rail from a partly lubricated heavy haul track showed the highest hardness at the gauge corner surface, while the hardness decreased almost linearly on the subsurface below the gauge corner, see Fig. 11. The highest hardness in Fig. 11 is comparable with the highest subsurface hardness at the rail disc seen in Fig. 7. The difference is that grinding activities and wear has continuously removed surface layers until the material with maximum hardness become the new surface.

Fig. 11. Hardness distribution at a rail section taken out from an heavy loaded ore track. The hardness distribution has a maximum close to the surface and decreases somewhat linearly.

4 Discussion

The examination made by [15] showed crack growth from 0.3 nm/load cycle for the lubricated conditions and up to 52 nm/load cycle for intermittent lubrication. The wear rates in this rolling/sliding test normally remained under ~0.035 nm/load cycle (except for the surfaces that “run-out”), which is too low to work in a controlled wear situation. The wear rates in this twin-disc test stayed about 10-150 times lower than the actual needed to continuously wear off surface cracks.

Similar synthetic base fluids but with other additives was tested in a reciprocating sliding contact (see Waara et al [25]). The contact pressure was much lower and the wear rates correspond to ~0.08-0.32 nm/load cycle. Work hardening of the rail steel and the fact that a rolling/sliding contact is less prone to be worn may cause the difference.

The increase in hardness indicate how much strain the material have been exposed to. The surface layer and band of layers (at depth 100-400 μm) seem to flow plastically when the coefficient of friction is as low as in this test. However, the cracks found in this test all started at the surface or a few microns under
the surface. For that reason the work hardening at subsurface (at depth 100-400 μm) could not influence the crack initiation. It is more likely that the surface roughness of the harder disc plastically deformed the pearlitic structure down to 8-10μm from the surface, which lead to initiation of cracks. This deformation at the surface agrees with the high contact pressures under harder asperities in a contact described by Johnson [8]. Another finding was that the surface roughness at the harder disc did influence the friction, and this agrees with Ref [7].

If the rails gauge corner is temporarily exposed to wheels with rough surfaces the rail surface will risk to become “run-out”. After “running out” the surface becomes rough or becomes damaged by crack initiation.

The fact that synthetic ester formulations caused more RCF damage in the present test does not mean that ester formulations should be avoided. This test used experimental blends of synthetic lubricants suited to fulfil just one purpose, get a wear rate sufficiently high to remove crack initiations from the surface. However, this approach to create a lubricant aimed to control the wear rate is in an early stage and more research is needed.

5 Conclusions

Surface roughness seems to be an important factor in these lubricated tests. A rough surface on the harder disc gave a slow “run-in” while a smoother surface on the harder disc gave rapid “run-in”. A rough, hard surface combined with a smoother, soft surface could cause “run-out” of the softer surface. Evaluating lubricated contacts requires very careful control of the surfaces roughness.

After “run-in”, the synthetic ester formulations generally gave low friction and surfaces slightly covered by cracks. However PAO gave higher long term friction and surfaces with relatively fewer surface cracks.

The wear stayed at a very low level for both PAO and synthetic esters and regardless if fatty acids were added to the fluid or not. In the rolling/sliding test, sufficiently high wear rates able to remove cracks never appeared.

6 Acknowledgement

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7 References
