THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

A Surface Engineering Approach to Reduction of Frictional Losses of Heavy Duty Diesel Engines

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Göteborg, Sweden 2012

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ISBN 978-91-7385-713-0

Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr: 3394
ISSN 0346-718X

Published and distributed by:
Department of Materials and Manufacturing Technology
Chalmers University of Technology
S-412 96 Göteborg, Sweden

Printed in Sweden by Reproservice at Chalmers University of Technology
Göteborg 2012

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Abstract

Reducing emissions is a top priority within heavy duty diesel engine development and research. The aim of the work is to decrease the fuel consumption by decreasing frictional losses of the Power Cylinder Unit.

Full scale testing of engine components improvements is time-consuming and costly. It is however possible to simplify the testing of engine components by pilot tribometer testing, enabling cost-effective screening of candidate material concepts. This thesis work answers the following research questions:

1. How should a pilot tribometer test be constructed in order to replicate the frictional and wear behaviour of the engine in the boundary, mixed and hydrodynamic lubrication regimes?
2. What part of the surface morphology of the cylinder liner surface affects the frictional behaviour of the different lubrication regimes?

In this thesis work the tribometer test approach was further developed to study a wider range of the Stribeck curve. Several different surfaces were analysed using the developed tribometer test approach, the results showed that the plateau part of the cylinder liner surface was responsible for controlling the frictional response in the boundary and mixed lubrication regimes. The results of these experiments were compared with single cylinder engine tests which were also conducted in this thesis work. The result of the engine tests and the tribometer test were in contradiction, the surface exhibiting low frictional losses in the tribometer exhibited high fuel consumption in the engine test. In evaluating this difference it was determined that the majority of the frictional losses were governed by the contribution of hydrodynamic friction and that a smoother plateau surface increased the hydrodynamic friction. The results of the engine testing were reproduced using a tribological simulation tool.

It is possible to decrease the hydrodynamic friction losses by decreasing the viscosity of the engine oil; however, this measure could increase the boundary frictional losses. To decrease the hydrodynamic friction losses in this thesis work a novel type of texturing was investigated in tribometer experiments. A DoE setup was developed with focus on analysis of the hydrodynamic lubrication regime. The results from the tribometer test show that a significant reduction in the hydrodynamic friction can be accomplished by applying textures on the cylinder liner surface. Based on the results from the experiments with textures a design proposal is put forward, in this a specification texture design in full scale is given.

Suggestions for future work include development of manufacturing techniques for machining textures on the cylinder liner, optimization of texture geometry by e.g. using a mesh-free calculation method, and design of a tribometer test with the aim of only distinguishing the hydrodynamic friction response of different surface morphologies.

Keywords: Friction, Wear, Lubrication, Oil consumption, Piston Ring, Cylinder Liner, Diesel Engine, Design of Experiments, Rig testing, Tribometer

Acknowledgements

First of all I would like to thank my supervisors, Bengt-Göran Rosén, Robert Ohlsson and Per H Nilsson, without you this work would not have been possible.

I would like to express my gratitude to the CAPE PhD Research Academy as well as the Knowledge Foundation (KK-stiftelsen) for giving me their support.

Tom Thomas is the gentleman that took the time to make the language correct and the illustrations understandable.

Thanks to Torbjörn Fridh at HB Mekaniska, thank you for an outstanding machining of components.

Thanks to Johan de Bruin, Ries de Leeuw and Stefan Rosén.

I would like to thank Josef Schmid and Thomas Witt, ELGAN-Diamantwerkzeuge GmbH & Co. KG, for many rewarding discussions on the topic of machining of cylinder liners.

I would like to thank Andreas Killinger, Andrei-Sorin Manzat, Rainer Gadow and Johannes Rauch from the University of Stuttgart for great collaboration in the Nanoker project.

I would like to show my appreciation to Roland Larsson and Braham Prakash and their colleagues at Luleå Technical University, you have been of great assistance.

Thanks Sören Andersson at KTH for helping me understanding a thing or two about wear mechanisms.

I would like to give my appreciation to Tim Benham and Peter Andersson (VTT) for giving me an insight of how tribological testing works when it is at its best.

I would like to thank the four patent engineers that assisted me in writing the patent application. Mr B, you have been most helpful, I would be grateful if I get the opportunity to visit you some day in Alexandria.

I would like to thank my department colleagues, I'm especially thinking of Erland Max, Bengt Olsson, Krister Fredriksson, Kristina Tullgren, Lars Mattsson, Mats Elmroth, Claes Frennfelt, Bertil Stenbom, Joachim Lövgren and Markus Söderfjäll, thank you for your never-ending assistance. I would also like to show my appreciation to my colleagues at other departments; Magnus Christensen, Bengt Otterholm, Jan Eismark, Ulla Särnbratt, Anders Persson, Roald Karlsson, Lars Lundin, Anders Danielsson, John Danielsson, Daniel Grunditz and Olof Domstad.

I would like to show my appreciation to my fellow PhD students and to my friends, I'm sorry for not having as much time for you as I'd like.

The three persons that are most responsible for the existence of this work and for which I will probably never be able to show my true appreciation are my wife Monica, my daughter Irma and my son Theo.

Appended papers

The results presented in this thesis are based on the work in the following appended papers:

Paper I S. Johansson, P H. Nilsson, R. Ohlsson, C. Anderberg, B-G. Rosén, New cylinder liner surfaces for low oil consumption. Tribology International, Vol. 41, Issues 9-10, pp. 854-859, (2008)

Paper II S. Johansson, P. H. Nilsson, R. Ohlsson, B.-G. Rosén., Experimental friction evaluation of cylinder liner/piston ring contact, Wear, Vol. 271, pp. 625–633, (2011)

Paper III S. Johansson, C. Frennfelt, A. Killinger, P.H. Nilsson, R. Ohlsson, B.G. Rosén., Frictional evaluation of thermally sprayed coatings applied on the cylinder liner of a heavy duty diesel engine: Pilot tribometer analysis and full scale engine test. Wear, Vol. 273, pp. 82– 92, (2011)

Paper IV S. Johansson, P. H. Nilsson, R. Ohlsson, B.-G. Rosén. Simulation and Experimental Analysis of the Contact between Oil Control Ring and Cylinder Liner in a Heavy Duty Diesel Engine. Proceedings of 18th International Colloquium Tribology 10–12 January 2012 Stuttgart / Ostfildern, Germany. Accepted for publication in Tribologie & Schmierungstechnik, August 2012.

Paper V S. Johansson, P. H. Nilsson, R. Ohlsson, B.-G. Rosén.A Novel Approach to Reduction of Frictional Losses in a Heavy Duty Diesel Engine by Reducing the Hydrodynamic Frictional Losses. Submitted for Wear, May 2012.

Additional papers (not included in thesis)

Paper VI S. Johansson, P. H. Nilsson, R. Ohlsson, C. Anderberg, B.-G. Rosén Optimization of the cylinder liner surface for reduction of oil of oil consumption. Proceedings of WTC 2005 World Tribology Congress III, September 12-16, 2005, Washington D.C. USA. pp. 559-560

Paper VII S. Johansson, P H. Nilsson, R. Ohlsson, C. Anderberg, B-G. Rosén. New cylinder liner surfaces for low oil consumption. Proceedings of Nordic Symposium on Tribology, NORDTRIB, 2006, pp. 854-859

Paper VIII S. Johansson, R. Ohlsson, Z. Dimkovski, B.G. Rosén, B.-G. Manufacturing mechanisms of cylinder liner surface roughness. Proceedings of the 11th International Conference on Metrology and Properties of Engineering Surfaces, 2007, pp. 338.

Paper XI S. Johansson, P. H. Nilsson, R. Ohlsson, B.- G. Rosén. Experimental friction evaluation for cylinder liner/piston ring contact: Results. Proceedings of 12th International Conference on Metrology and Properties of Engineering Surfaces Rzeszów, Poland, 2009, pp. 193-199.

Paper X S. Johansson, C. Frennfelt, A. Killinger, P. H. Nilsson, R. Ohlsson, B.G. Rosén, Frictional evaluation of thermally sprayed coatings applied on the cylinder liner of a heavy duty diesel engine; pilot tribometer analysis and full scale engine test. Proceedings of 14th Nordic Symposium on Tribology Nordtrib 2010, Storforsen, Sweden

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Abbreviations

AFM	Atomic Force Microscope
APS	Air Plasma Spraying
BDC	Bottom Dead Centre
BL	Boundary Lubrication
CAD	Crank Angle Degree
CFD	Computational Fluid Dynamics
CH	Chemical Machining
CI	Compression Ignition
CR	Capacitor Resistor (filter type)
CTDC	Combustion Top Dead Centre
DLC	Diamond Like Carbon
DoE	Design of Experiments
EMB	Electro Beam Machining
ECM	Electro Chemical Machining
LVDT	Linear Variable Differential Transformer
HL	Hydrodynamic Lubrication
HVOF	High Velocity Oxygen Fuel
HVSFS	High Velocity Suspension Flame Spraying
ML	Mixed Lubrication
MVA	Multi Variate Analysis
PCU	Power Cylinder Unit
PM	Particulate Matter
RSM	Response Surface Modelling
SI	Spark Ignition
STM	Scanning Tunnelling Microscope
TDC	Top Dead Centre
TWA	Twin Wire Arc system

List of symbols

Greek indicators

Indicator	Description	Dimension
γ	Local shear rate (Cross)	s^{-1}
γ_c	Critical shear rate (Cross)	s^{-1}
μ	Friction coefficient	-
μ or η	Dynamic viscosity[1]	Pa*s
μ_0	Dynamic viscosity for low shear	Pa*s
μ_∞	Dynamic viscosity for high shear	Pa*s
μ_k	Kinetic friction coefficient	-
μ_s	Static friction coefficient	-
ν	Kinematic viscosity	mm^2/s
ρ	Density	kg/m^3

Other indicators

Indicator	Description	Dimension
A	Area	m^2
a_0	Correlation Parameters (Vogel)	-
h	Oil film thickness	µm
P	Contact pressure	Pa
S	Shear ratio	-
T	Temperature	^0C
T1	Correlation Parameter (Vogel)	^0C
T2	Correlation Parameter (Vogel)	^0C
v, v_0	Sliding speed	m/s

[1] μ is commonly used in describing the dynamic viscosity in lubrication equations, η is commonly used in describing shear independent dynamic viscosity, used e.g. in characterisation of the Hersey parameter

1. General

1.1. Background

Reducing emissions is a top priority within heavy duty diesel engine development and research. The power cylinder unit (PCU) is a main contributor to emissions. The PCU unit consists of piston rings, piston, piston pin, connecting rod and cylinder liner and these components account for about 50% of the total frictional losses [I, II, III, IV] found in the engine. Reducing frictional losses means reduced fuel consumption and this means reduced CO_2 emission. Reduction of frictional losses is addressed in research question 2 in this thesis work (see section 1.3)

Full scale testing of engine components improvements is time-consuming and costly. It is however possible to simplify the testing of engine components by pilot tribometer testing, enabling cost-effective screening of candidate material concepts. In component specific pilot tribometer tests, components are studied individually; this results in a decreased number of noise factors compared to full scale testing. However, it is uncertain how to transfer the results from tribometer testing into full scale, one of the aims of this thesis work is to narrow the gap between lab tribometer testing and full scale engine testing.

One aspect of the emission legislation (previous [V], current [VI] and future [VII]) for the heavy duty diesel engine regulates the allowed amount of particulate matter (PM), the amount of particulate matter is to an extent affected by the consumption of engine oil [VIII]. The emission level according each emission legislation, with implementation year, can be seen in Figure 1. There are several components that contribute to the consumption of oil, e.g. the piston rings design is of importance [IX]. However, one of the single most influential factors in controlling oil consumption is the surface roughness of the cylinder liner [X, XI]. Uncontrolled wear of the cylinder liner surface causes seizure of the engine. Thus, minimization of wear is of vital importance for retained durability [XII].

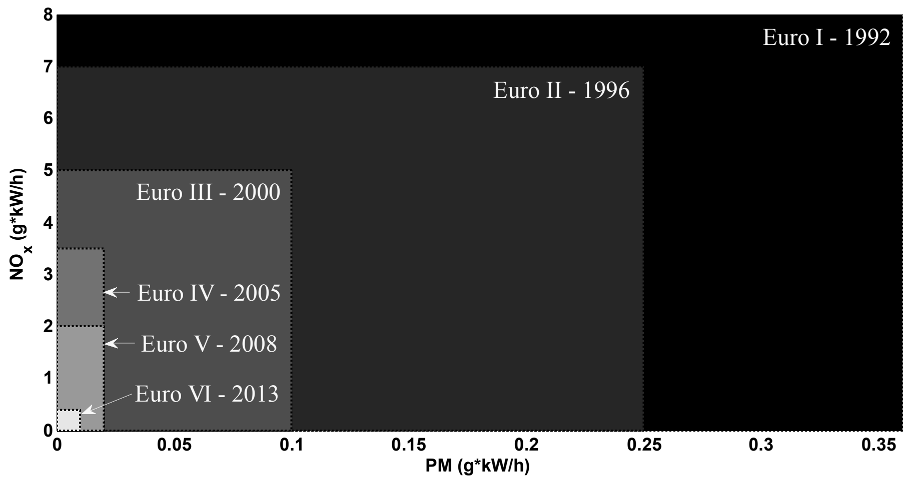

Figure 1. Emission legislations for heavy duty diesel vehicles with respective implementation year.

1.2. Aims

The primary aim of this thesis is to reduce the frictional losses and emissions of the power cylinder unit of heavy duty diesel engines by optimising the interior cylinder liner surface roughness morphology. The secondary aim is to narrow the methodological gap between pilot tribometer testing and full scale engine testing.

1.3. Research questions

The vision for general tribological testing is to create a test environment capable of reproducing full scale engine conditions. The research questions in this thesis are as follows:

1. How should a pilot tribometer test be constructed in order to replicate the frictional and wear behaviour of the engine in boundary, mixed and hydrodynamic lubrication regimes?
2. What part of the surface morphology of the cylinder liner surface affects the frictional behaviour of the different lubrication regimes?

This work focuses on the development of test rig methodology and the investigation of present and future cylinder liner candidate materials.

1.4. Research approach

The current state-of-the-art test methodology was examined in an early study (Paper I), this methodology was considered to be insufficient for frictional studies of the power cylinder unit since only a small part of the Stribeck curve could be analysed. A new methodology was created using a design of experiments to study a wider range of experimental input parameter configurations (Paper II). The basis for the developed methodology uses the connection between simulation results and input experimental data to accurately mimic the different lubrication regimes for the top piston ring during a normal operating cycle. In comparing the developed tribometer test approach with results from full scale engine testing the overall research approach shifted from focusing on boundary and mixed lubrication regime (Paper I, II, III) to focusing on the hydrodynamic lubrication regime (Paper IV, V). The progress of the research approach in this thesis work is schematically shown in Figure 2.

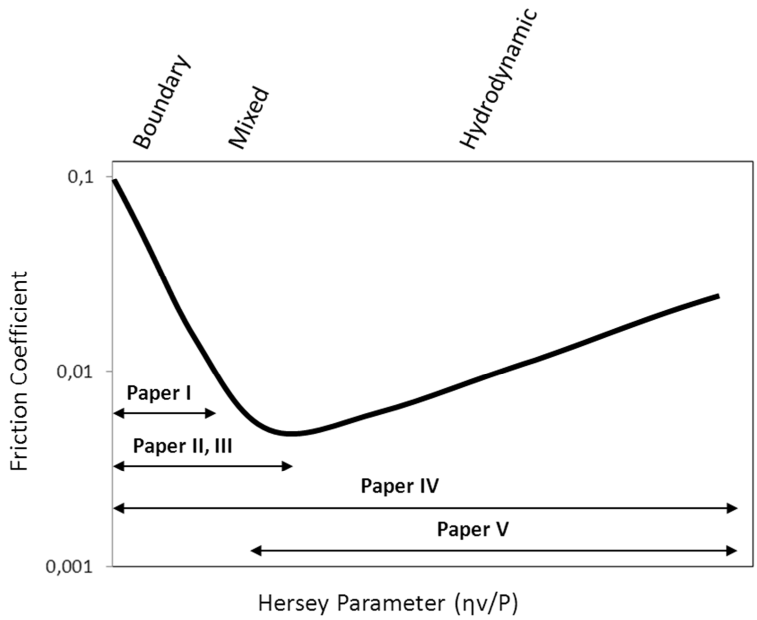

Figure 2. Schematic overview of the lubrication regimes analysed in each individual paper.

The surface engineering loop, as proposed by Stout [XIII], symbolizes the close connection between function, characterization and the manufacturing process where characterization can be used to understand and control function (see Figure 3). In this thesis special attention has been paid to the link between function (frictional response) and surface characterization (roughness parameters). Statistical analysis was applied in this thesis work to study the correlation between a multitude of surface roughness parameters and frictional response of different surfaces.

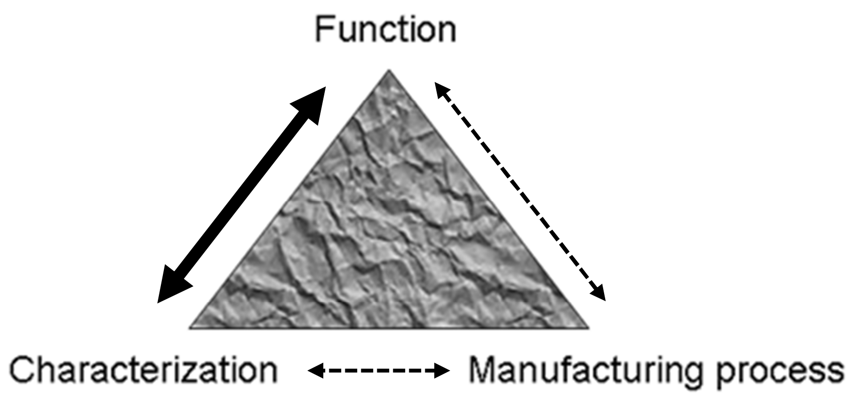

Figure 3. Surface engineering loop as proposed by Stout. Adapted from [XIII].

1.5. Delimitations

The thesis will be limited to mechanical contact losses and hydrodynamic losses for the top piston ring and the oil control ring. For the top ring this thesis focuses on mechanical contact losses at and near the upper reversal zone of the top piston ring and the cylinder liner, where the viscous losses are small. The methodology is exemplified on the selected material in paper I-III. In this thesis the frictional effects resulting from form deviations, such as cylinder bore out of roundness, were not analysed.

This thesis aims to analyse the correlation between surface morphology and friction. Material properties of different cylinder liner materials were left out of the scope of this thesis.

Wear is of course an important feature in examining the PCU, however, wear has been monitored and evaluated but was not been focused on in this study.

1.6. Outline of Thesis

Chapter 1 gives a general description of the background of this thesis and gives a description of the aim, the research questions, research approach, and the delimitations relating to this thesis.

Chapter 2 gives an introduction to the research field and to the experimental and numerical methods that were used and developed within the scope of this work.

Chapter 3 gives an introduction to surface roughness instrumentation and the analysis of surface roughness, this section also describes surface roughness filtering techniques and describes the functional surface roughness parameters used to characterise surface roughness morphology.

Chapter 4 describes the current process of finishing machining of the interior surface of cylinder liners and exemplifies additional manufacturing techniques with which special surface features could be obtained on the interior surface of the cylinder liner.

Chapter 5 gives an introduction to the functionality of the diesel engine and highlights frictional losses, blow by and engine oil consumption.

Chapter 6 describes the existing test methods for frictional characterization of the PCU; The tribometer test approach, the simulation approach and the full scale engine test approach.

Chapter 7 describes the two statistical analysis methods that have been used in this work. Design of Experiments was here used to generate a relevant experimental input matrix, the analysis of output data was performed by using Multi Variate Analysis with which one can simultaneously analyse the connection between several input and output parameters.

Chapter 8 describes the main results of the appended papers.

Chapter 9 provides a discussion of the results.

Chapter 10 provides suggestions for future work.

Chapter 11 shows the conclusions of this thesis.

Chapter 12 lists the references used in this thesis.

1.7. Author’s Contribution to Appended Papers

Paper I: Johansson created the simulation model, performed the calculations, the experimental work and wrote the paper.

Paper II: Johansson updated the experimental equipment and created the test methodology described in this paper, apart from the calculation of the elastic matrix. Johansson took part in creating the simulation model, Johansson also wrote the paper.

Paper III: Johansson performed all the experimental work and measurements as well as wrote the paper.

Paper IV: Johansson created the simulation model, performed the calculations and the statistical analysis, the experimental work and wrote the paper.

Paper V: Johansson performed the experimental work, carried out detailed analysis of the experimental results leading to the formulation of viscous friction in textures, Johansson also wrote the paper.

2. Tribology

2.1. History of Tribology

The word Tribology originates from the Greek language where “tribo” translates into “I rub”. Knowledge of the practical aspects of tribology has a long history, the principle of reducing sliding friction by using lubricants dates back to 2400 B.C. where lubricant was used for wooden logs carrying a sledge used for transporting building material [XIV]. Amontons formulated the first law of friction in 1699, which states that force of friction is directly proportional to the applied load (Eqn. (1)). Coulomb (1781) verified Amontons conclusions and made the distinction between static and kinetic friction. The static friction was here described with the static friction coefficient, μ_s which describes the boundary value of friction coefficient required to initiate sliding and the kinetic friction coefficient μ_k which describes the friction coefficient during sliding.

$$\mu = \frac{F_\mu}{F_N} \tag{1}$$

Amontons first law of friction describes the frictional interaction of a dry sliding contact, for lubricated sliding contacts the frictional behaviour is more complicated. For a lubricated sliding contact the friction is dependent on both the mechanical contact friction and the hydrodynamic friction from the fluid separating the surfaces. Richard Stribeck and Mayo D Hersey described the frictional behaviour in what has been termed the “Stribeck curve” (see Figure 4) [XV], this curve describes the combined friction of both mechanical contact and hydrodynamic friction. Commonly the Hersey parameter is used in the Stribeck curve for describing alterations of sliding velocity (v), dynamic viscosity of oil (η) and contact pressure (P) for lubricated sliding contact. The Stribeck curve is divided into lubrication regimes; Boundary lubrication (BL) describes the relatively high friction coefficient when mating surfaces are only partially separated by a small amount of boundary lubricant. In the mixed lubrication regime (ML) a larger amount of oil is carrying the contact load, thus friction decreases. The minimum value of friction is obtained when the majority of contact pressure is carried by the lubricant with still a small influence from the shear resistance of the lubricant. In the hydrodynamic lubrication regime the friction increases as a function of sliding velocity or more precisely as a function of increased viscous shear, the friction from mechanical contact in the hydrodynamic lubrication regime is very small or zero. The appearance of the Stribeck curve is dependent on application; since the frictional increase in the hydrodynamic lubrication regime is dependent lubricant film thickness the Stribeck curve will be different for conformal and non-conformal contacts. The amplitude of surface roughness will also alter the Stribeck curve. Since a rough surface could increase the film break through thus increasing the amount of contact between surfaces, a higher value of friction coefficient is expected in the mixed lubrication regime in comparison to a smooth surface [XVI].

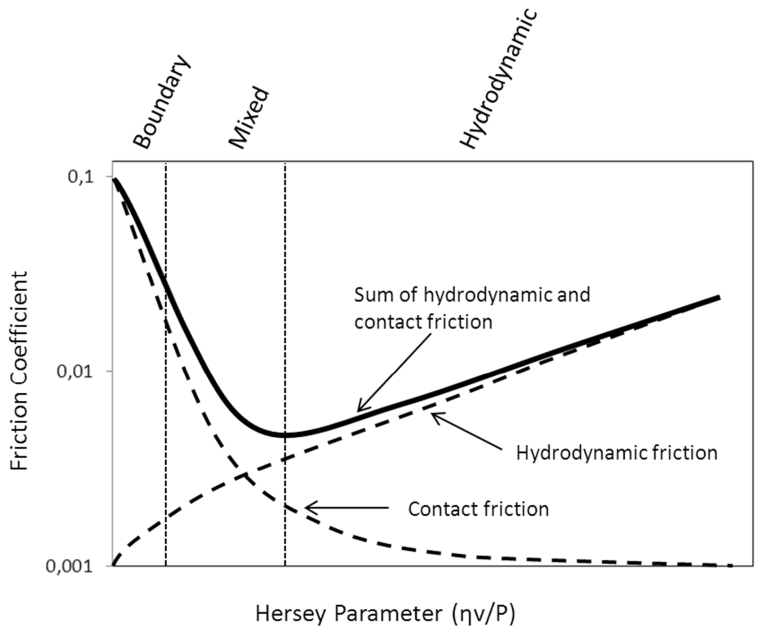

Figure 4. Illustration of the Stribeck curve showing the lubrication regimes and the interaction between contact and hydrodynamic friction.

It was not until March 9, 1966 that the term Tribology sprang into existence. Tribology was first used in the "Jost Report" [XVII], in this report it was determined that losses due to friction, wear and break-down had a great economic impact (several per cent of Britain's GDP at the time). In the Jost report Tribology was originally defined as "The science and technology of interacting surfaces in relative motion - and of associated subjects and practices". One alternative and perhaps more current definition of Tribology is "the science of friction, lubrication and wear". Tribology provides a natural link between different research disciplines that could be considered fundamental when the engagement between two surfaces is to be analysed. Under the Tribology "umbrella" research disciplines such as solid and fluid dynamics, chemistry, material science find collaborative and synergetic effects.

In more recent historical terms one of the most important developments in tribology include the understanding of the fundamental mechanisms of oil film generation between the heavily loaded tribological contacts under elastohydrodynamic lubrication conditions, existing in roller bearings and gears [XVIII]. Although the steady state situation of rolling element bearings has been initially more thoroughly quantified than transient conditions of a gear contact [XIX]. In the elastohydrodynamic lubrication regime the combination of surface flattening and immense increase in viscosity permits a heavily loaded tribological element to sustain its functionality.

In present terms one great achievement within tribology is the development of thin Diamond Like Carbon (DLC) coatings used in combination with special oil additives. The combination of thin coatings and oil additives has shown great potential for tribomechanical systems operating in boundary and mixed lubrication regimes, such as the contact between roller and tappet in the valve mechanism of a passenger car [XX]. Due to the availability of tailor made coatings with properties that are unique to different tribological contact situations [XXI] it is highly likely that the amount and usage of thin coating DLC coatings will increase in future design of machine elements.

2.2. Lubricants and lubrication

Lubricants are used to reduce the friction forces between mating surfaces. The amount of lubricant between surfaces is often very small; surfaces are typically only separated by a lubricant layer with a thickness of a few µm or parts of a µm. A lubricant can be of gaseous, liquid or solid form. Besides reducing friction between components a lubricant also; performs cooling of components by transport of thermal energy, removes wear particles and prohibits corrosion.

The main component of an automotive lubricant is base oil. However, base oil only is not sufficient to account for the high demands that are put on the automotive lubricants and therefore additives are added to the base oil to increase the lubricants performance. Additives are commonly divided into three groups [XXII] with the following functions:

- Maintenance of cleanliness
- Wear reduction
- Alteration of the physical properties of the oil (e.g. reduction of the viscosity decrease at high temperatures)

The following types of additives are used in automotive lubricants:

- Detergents, dispersants, added to decrease the formation of deposits and to maintain cleanliness
- Anti oxidants; added to reduce oxidative oil degradation
- Anti wear additives (such as ZDDP) and extreme pressure (EP) additives; added to reduce mechanical wear
- Friction modifiers (FM); added to decrease friction in metal-to-metal contact (boundary and mixed lubrication regime)
- Viscosity Index (VI) improvers (viscosity modifiers); counteract a decrease in viscosity at high temperatures
- Pour depressants, anti foam additives, rust inhibitors; added to increase or modify inherent properties of the base oil

The most important parameter to characterise a lubricant is the viscosity [XXIII]. Viscosity is defined as the shear force needed to overcome the internal cohesiveness of the material. For liquid lubricants viscosity is commonly referred to as the internal friction of a fluid or fluid thickness. Two different measures are used to quantify viscous properties; dynamic viscosity and kinematic viscosity. Dynamic viscosity is measured using a capillary viscometer in which the flow rate is quantified for a controlled amount of flow from a container through a small hole under the force of gravity. In a capillary viscometer all fluids experience approximately the same shear stress, however, the gravitational influence will cause a liquid with higher density to flow faster, since liquids with higher density are affected by a larger gravitational force. The kinematic viscosity is calculated by dividing the dynamic viscosity by the density (Eqn. (2)), thus a value of viscosity independent of gravitational influence is obtained.

$$v = {}^{\mu}/_{\rho} \tag{2}$$

A common definition of the automotive lubricant viscosity uses the SAE viscosity grades (10W30, 15W40, 20W50 etc.). Two classes are used to define the viscosity grade; the first figure in the term in this classification represents the oil's ability to flow to the oil pump and the cranking resistance at low temperatures. The second term in the classification represents the viscosity at high temperature, typically 100 °C [XXIV].

Thermal variations have a large effect on the value of viscosity. By measuring the kinematic viscosity at two (or more) temperatures (commonly 40 °C and 100 °C) [XXV] an accurate quantification of the temperature dependency of kinematic viscosity can be obtained using the Vogel equation [XXVI] (Eqn. (3)) as illustrated in Figure 5.

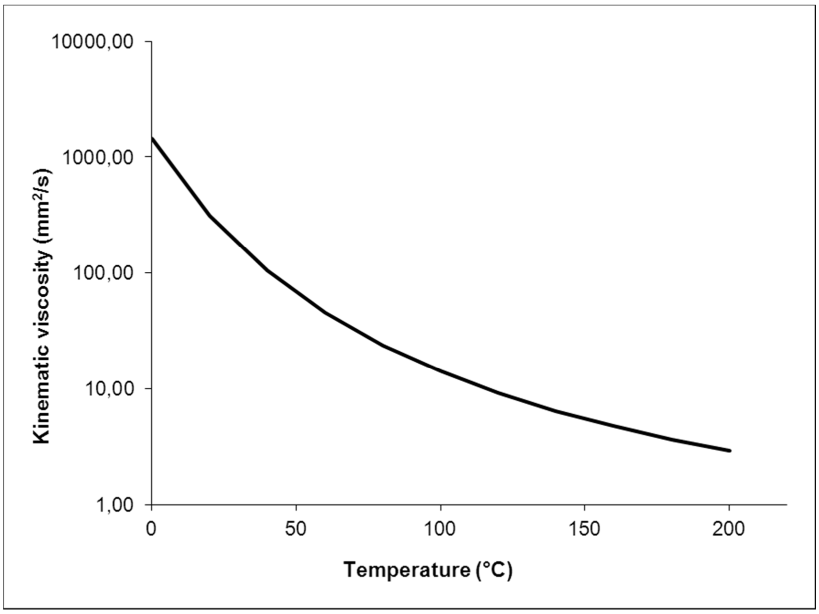

Figure 5. Thermal dependency of kinematic viscosity for a typical 15W40 automotive lubricant.

$$\mu_0 = a_0 \exp\left(\frac{T1}{T2+T}\right) \tag{3}$$

2.2.1. Shear rate dependency of dynamic viscosity

The viscosity of Newtonian liquids is independent of the shear rate. For laminar flow shear rate in its simplest form is defined as the ratio between the sliding velocity of two parallel surfaces and the height or thickness of the medium separating the two parallel surfaces. A pure mineral oil is a Newtonian fluid, however, due to the additive content of a typical automotive lubricant it has a shear thinning non-Newtonian behaviour. This means that the value of dynamic viscosity will decrease with increasing shear ratio. The value of the shear stress increases with the sliding speed and decreases with the oil film thickness. For low values of shear rate the value of dynamic viscosity is equal to the μ_o, the zero shear plateau.

As shear rate increases, either by an increase in sliding speed or a decrease in film thickness (Eqn. (4)) the dynamic viscosity decreases. For high levels of shear the dynamic viscosity assumes the value of μ_∞, the infinite shear plateau. Figure 6 shows the shear rate dependency of typical 15W40 automotive lubricant.

$$S = \frac{v_0}{h} \tag{4}$$

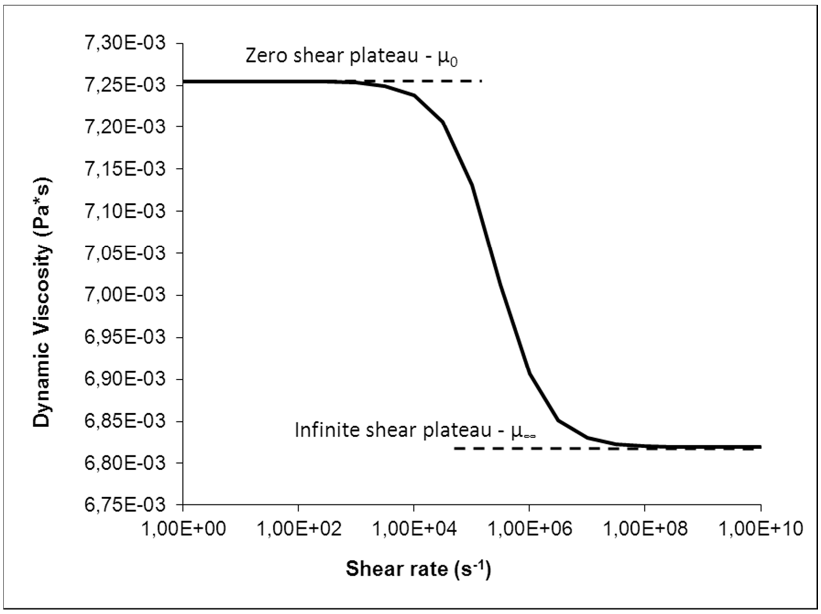

Figure 6. Shear rate dependency of dynamic viscosity at 100 °C of a typical 15W40 automotive lubricant.

It is possible to characterise the dynamic viscosity for an automotive lubricant, accounting for both temperature and shear rate. This type of characterisation is useful in e.g. tribological simulations of the PCU in which different sliding speeds and temperatures are present at different parts of the stroke. The characterisation is done by using the Cross equation [XXVII] (Eqn. (5)). A comparative quantification of the shear rate behaviour of different automotive lubricants is the HTHS (High Temperature High Shear) value, the HTHS value is the dynamic viscosity of the oil at a temperature of 150 ^{o}C and at a shear rate of 10^6 s^{-1} [XXVIII]. The HTHS is an important viscosity parameter for determining the frictional loss characteristics of an operating engine since the majority of the engine components, including the piston rings, operate in the hydrodynamic lubrication regime at high shear rates [XXIX].

$$\mu = \mu_\infty + \frac{\mu_0 - \mu_\infty}{1 + {}^{\gamma}/_{\gamma_c}} \tag{5}$$

2.3. *Wear*

Lubrication of a tribosystem decreases friction, however, for surfaces in partial contact as occurring in the boundary and mixed lubrication regime wear of surfaces is inevitable. Wear is defined as “the progressive loss of substance from the operating surface of a body occurring as a result of relative motion of the surfaces” [XXXII]. The process of wear is commonly described with a classification that relates to the wear mechanism, however, wear is an intricate process involving many processes and physical parameters thus it is not always easy to establish the originating mechanism and the cause of wear. This section aims to describe the most common wear classifications relevant to the piston ring/cylinder liner interface, namely abrasive, adhesive and corrosive wear [XXX]. Other types of wear classifications include erosive wear, fretting and surface fatigue and are not subjected to further characterisation in this thesis. Gates [XXXI] has argued that the common wear classification of two and three body abrasive wear is inconsistent, he proposes that the classification of abrasive mechanisms should depend on the severity of the specific wear process. However, the wear classification in this thesis builds on the classical definition of wear classification [XXXII].

2.3.1. Abrasive Wear

Abrasive wear is the process in which one surface of commonly a hard material ploughs the top asperities into a softer mating surface, leaving a scratch and/or a wear particle. The abrasive wear mechanism is not always considered a negative characteristic, actually this type of wear process has positive effects for functionality of engines; the running in of engine components is an abrasive wear process which produces a smooth surface thus reducing frictional losses in boundary and mixed lubrication regimes. The abrasive running in process has been investigated up to a point such that it is today possible to, with prior knowledge of the surface morphology, predict the surface morphology alteration during the running in process [XXXIII].

There are two types of abrasive wear processes; In a tribosystem where the asperities or hard protuberances on one of the surfaces plough through the mating surface this is commonly defined as two-body abrasive wear, if wear particles are moving freely between the contacting surfaces this process is termed three body abrasion.

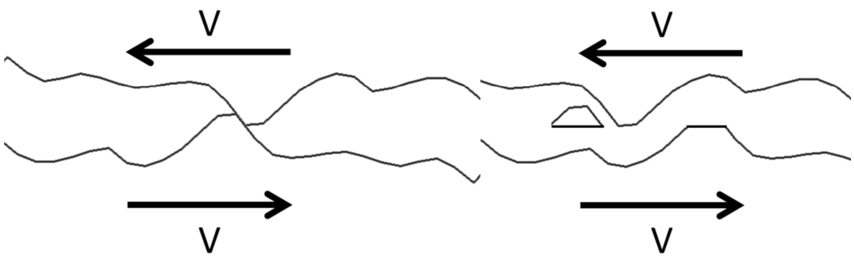

Figure 7. Illustration of two body abrasive wear in sliding interface.

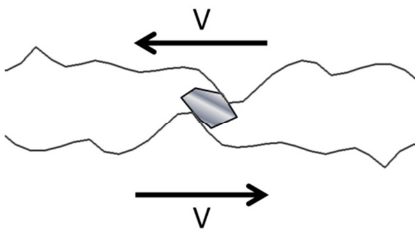

Figure 8. Illustration of three body abrasive wear in sliding interface.

2.3.2. Adhesive Wear

For adhesive wear the material properties of the mating surfaces are of importance, the crystal structure, the crystal orientation and the amount of alloying elements of metals influence the adhesive wear behaviour. The adhesive wear process is a combination of adhesion (sticking) and fracture of the subsurface of the material. In the adhesive wear process material is transferred from one surface to the other (as shown in Figure 9). Compared to abrasion which generally takes some time to develop adhesive wear can reach critical levels in a short period of time, resulting in scuffing or seizure. The abrasive wear process is a normal part of engine component wear. For adhesive wear to occur in engine components an extreme contact situation is required with one or several of the following prerequisites; high contact pressure, high overall temperature/frictional heating and starved (lack of) lubrication.

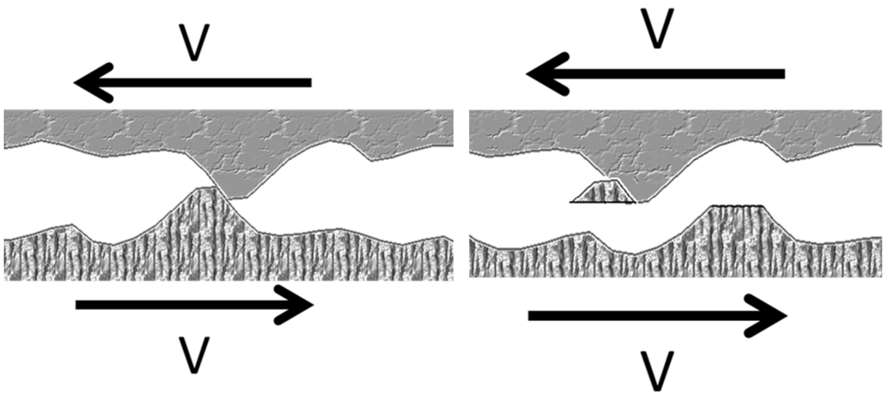

Figure 9. Illustration of adhesive wear in sliding interface.

2.3.3. Oxidative wear

Oxidative wear is also commonly known as corrosive wear or tribochemical wear, it is a process in which one or both of the surfaces reacts with the environment forming reaction products, oxides, at the surface (see Figure 10). The oxidation of the surface asperities could increase the possibility of brittle fracture which causes an accelerated wear process. It is not certain that just because an oxide layer forms on the surface the severity of wear increases, an oxide layer is generally quite hard and can have a wear protective effect. If an oxide layer is worn down new oxides are free form on the worn surface. Oxidative wear can have a large effect on the wear of the cylinder liner; exhaust gas recycling (EGR) which creates an acidic environment could partially dissolve the material of the cylinder liner causing large wear levels of the piston ring/cylinder liner interface [XXXIV].

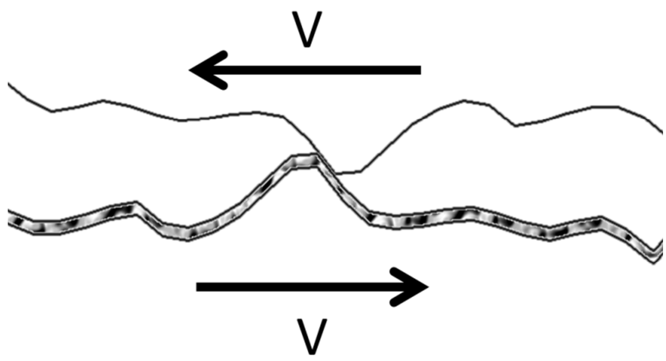

Figure 10. Illustration of oxidative wear in sliding interface.

3. Characterisation of surface roughness

In this thesis the correlation between surface morphology and engine component functionality is analysed, however, the characterisation of surfaces has found its way into many different applications and fields of science. Surface characterization is today used in: zoology (animal digestive systems), food processing (determination of the correlation between surface morphology and visual appearance and perception), medicine (durability and functionality of implants such as the knee prosthesis), infrastructure (surface roughness effects of asphalt pavement and train rails), sports (roughness of skis, flow characteristics of competition swimsuits), finance (security marking of bank notes), criminology (analysis of which handgun has fired a specific bullet) etc. [XXXVI].

A surface consists of numerous wavelengths, to analyse different aspects of surface roughness a division of the surface wavelengths into groups that depend on the wavelength is necessary. The traditional division of surface features is based on the lateral scale of the surface features; roughness is the product of the machining operation, waviness is the product of imperfect operation of a machine tool and form deviations are generated by larger scale distortions [XXXVIII] (see Figure 11). In a measurement of a piston ring or a cylinder liner the cylindrical form of the component represents the largest wavelength, since surface roughness typically focuses on the smaller wavelengths form is commonly removed by subtracting a polynomial shape from the original measurement. The sampling distance between measurement points is dependent on the wavelength to be quantified, with smaller spacing shorter wavelengths can be quantified but using close spacing which increases the number of data points for a specific area. The wavelength to be quantified is thus dependent on sampling distance but also on the choice of surface roughness measurement device and filtering technique as illustrated in this section.

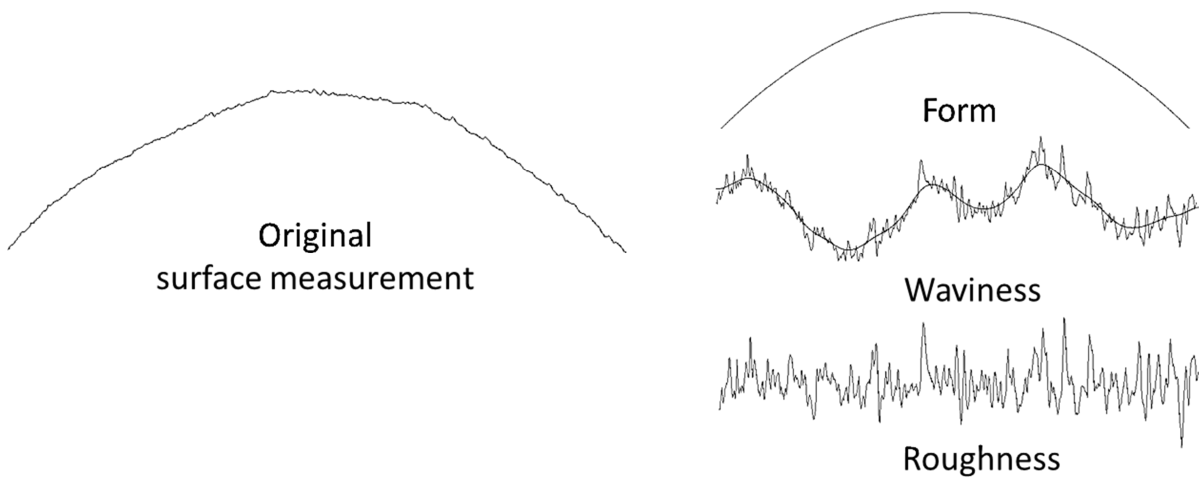

Figure 11. Wavelength components of surface measurement.

Each surface has a unique topography, by using surface metrology it is possible to characterise this unique topography by monitoring the change e.g. during operation or manufacturing. This offers understanding of how the surface topography controls the functional performance of components in a system [XXXV].The amount of data generated in a surface roughness measurement is often quite large, filtration enables an extraction of the wavelengths of interest from the large amount of data. A calculation of roughness parameters gives quantification of the wavelengths of interest with only a few significant scalar values. Using roughness parameters thus simplifies the analysis of the link between characterization and function. Numerous surface roughness measurements were conducted in this thesis, including before and after tribometer experiments and engine tests. This enables a statistical correlation analysis between surface roughness morphology, friction and wear (the specifics of the statistical correlation analysis are described in section 7).

3.1.1. Measurement equipment

There exist numerous devices for surface roughness measurement. The most commonly used measurement instrument in mechanical industry, both in past and present terms [XXXVI], the tactile stylus was beginning to be used around 1933 [XXXV]. The stylus operates similarly to a gramophone where a sharp tip (with a specific radius) traverses the surface picking up the surface irregularities as it moves along the surface. Using a linear variable differential transformer (LVDT) gauge the amplitude for each specific measurement point is quantified. This amplitude information is sent to a recording device (computer) for storage and further processing. A tactile stylus profilometer is capable of high resolution, a measurement uncertainty of as low as ± 1 nm in both the vertical and lateral axis has been reported [XXXVII]. Since the stylus profilometer does not have a theoretical limitation in the area size of the measurement, however, measurements using a stylus profilometer are time consuming. The first areal surface topography system capable of measuring micrometre surface topography was built by Williamson in 1968, this system was based on the conventional 2D stylus system but with the addition of making parallel traces for 3D representation. Since areal measurements often contain large amounts of data it was not until the introduction of the personal computers in the 1980s that areal measurements were beginning to become more common [XXXVIII].

Measurement techniques based on optical phenomena, such as white light interferometers, scanning confocal microscopes and chromatic confocal profilometers have had a rapid development and are increasing in popularity. The chromatic confocal probe is similar to the stylus profilometer in the respect that this device also traverses the surface but not with a stylus tip but a focused colour spectrum of light. Depending on the lateral height a distribution of wavelengths (of light) are reflected from the surface, the reflected wavelengths are collected on a CCD chip. A computer analyses the distribution of wavelengths and returns a single height value for each specific lateral position of the surface [XXXIX] (see Figure 12). As the confocal probe traverses the surface each amplitude value for each specific lateral position is recorded, by summarizing this data a surface amplitude matrix is obtained.

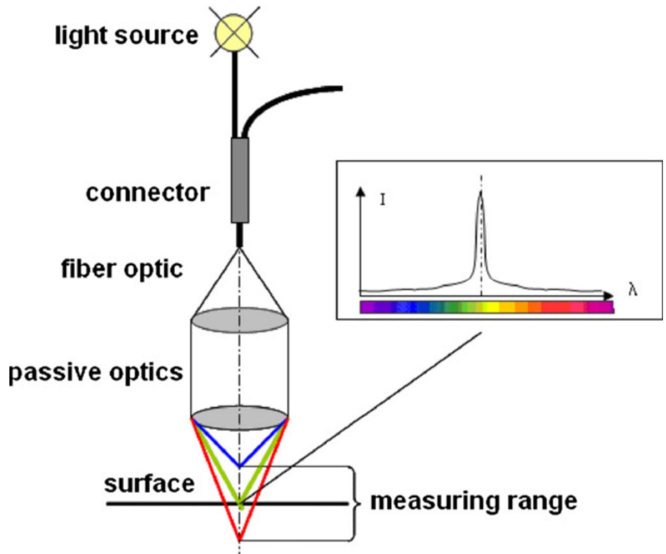

Figure 12. Overview of the operating principle of a chromatic confocal profilometer [XXXVI].

There are two types of surface measurement equipment capable of analysing the surface with very high vertical (nanometre and below) and lateral resolution; atomic force microscope (AFM) and scanning tunnelling microscope (SPM). An AFM consists of an elastic cantilever with a tip with a very small tip radius, several orders of magnitude smaller than the radius of the stylus tip of a stylus profilometer, typically 2 to 60 nm [XXXVI]. As the AFM tip traverses the surface it picks up the irregularities of the surface. The altitude values in an AFM are obtained by measuring the vertical position of the cantilever beam with a laser beam which is focused on to a photodiode detector. An STM uses the concept of quantum tunnelling, as the tip of a STM moves close to the surface (without contact between the two bodies) electrons are capable of tunnelling between the tip and the sample thus generating a tunnelling current. The tunnelling current is dependent on the distance between the surface and the tip, the tunnelling current is kept constant by displacement of the tip, and from the values of this displacement the altitude values of a surface can be obtained.

In this thesis two types of 3D profilometers have been used, the stylus profilometer which was mainly used for characterization of surface roughness and the chromatic confocal profilometer which was mainly used for larger areal surfaces measurements used for characterization of wear depth. Since the characterisation of wear depth requires measurement of the complete wear scar, which was quite large considering the stroke length of the eccentric tribometer, it was considered too time consuming to use the stylus profilometer for this task.

3.1.2. Preprocessing

Preprocessing of surface roughness measurement, enabling surface parameterisation, is commonly done in two steps; levelling/form removal and filtration. Levelling and form removal are the initial steps carried out so that the waviness and roughness can be studied individually. Preprocessing of surfaces can also include steps such as; truncation, rotation, inversion etc. Levelling of a surface that does not contain curvature is often performed by using a least square plane [XL]. Form removal is required when the surface includes a curvature (which is not part of the roughness morphology), for a wide range of uni-curved surfaces, such as the cylinder liner, a second order polynomial form removal has proved to be suitable [XL].

Filtration is the process in which features of interest are extracted from measured data for further analysis [XXXV]. In 1986 an agreement was made to use Gaussian filter for wavelength separation, this resulted in ISO 11562:1996 [XLI]. Before this time much effort was put on developing analogue CR (capacitor resistor) filters, the first versions of these filters were not phase correct, although CR filters with smaller phase error has been developed these types of filters are seldom used today.

Gaussian filtering is a viable method for working with large range of different surface types. The surface of a plateau honed cylinder liner has a negatively skewed distribution of amplitude values, meaning that a larger number of measurement points is located above the mean line of the surface (for a standardly distributed surface the amount of data points above and below the mean surface line would be the same, see Figure 13).

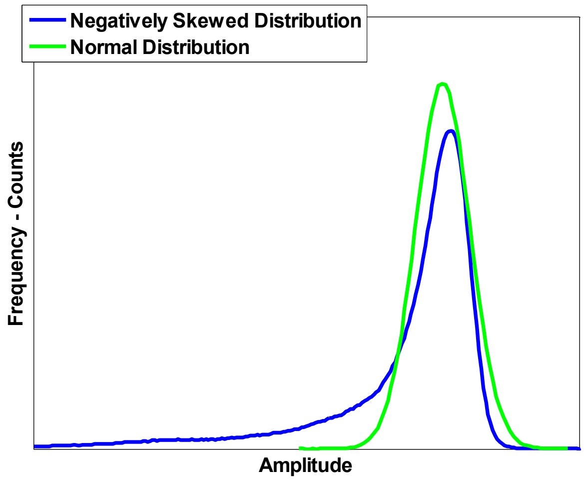

Figure 13. Histogram example of the negatively skewed distribution of a plateau honed cylinder liner surface and normal distribution.

The Gaussian filter is not ideal for filtering of skewed surfaces, such as the surface of the plateau honed cylinder liner since artificial waves are introduced in the filtered signal. This results in an uneven topography after filtration. The Robust Gaussian filter corrects this by performing a modified filtering sequence [XLII]; in this the plateau part of the surface is partially segmented from the surface by using a weight function. This function assigns the extreme values of the surface (in the case of a plateau honed surface extreme values are the deep valleys) a lower weighting, Gaussian filtering is thus more or less only applied on the plateau part of the surface. Using Robust Gaussian it is possible to obtain a filtered surface with much smaller form deviations (as shown in Figure 15) in comparison to Gaussian filtering (as shown in Figure 14). For illustrative purposes to better visualise the effects of filtering profiles (not surfaces) are portrayed as examples in this section.

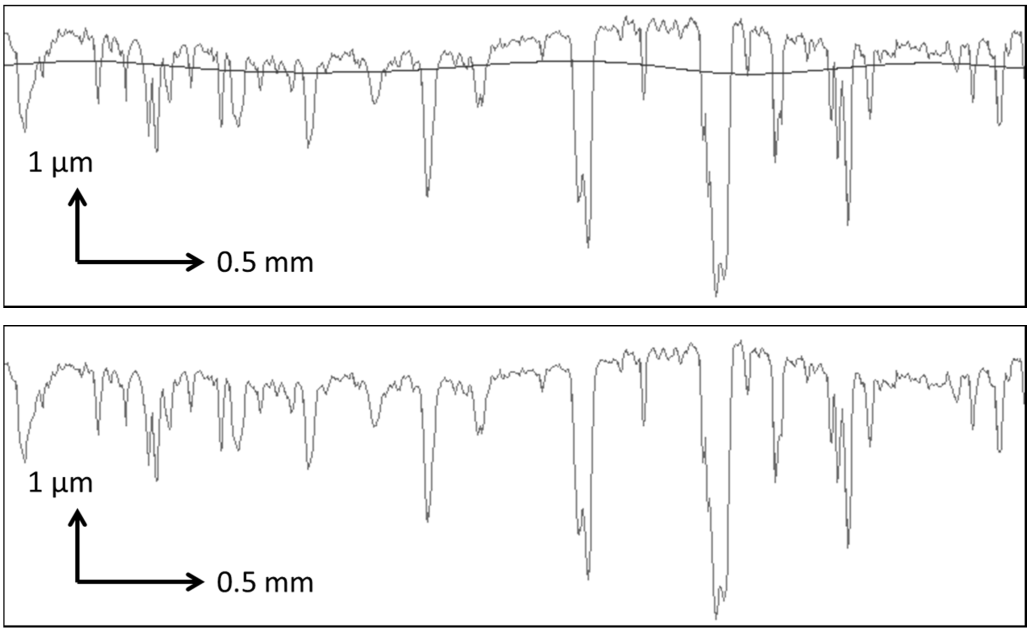

Figure 14. Cylinder liner profile measurement filtered with standard Gaussian filter. Top: Original measurement and waviness profile. Bottom: Filtered surface.

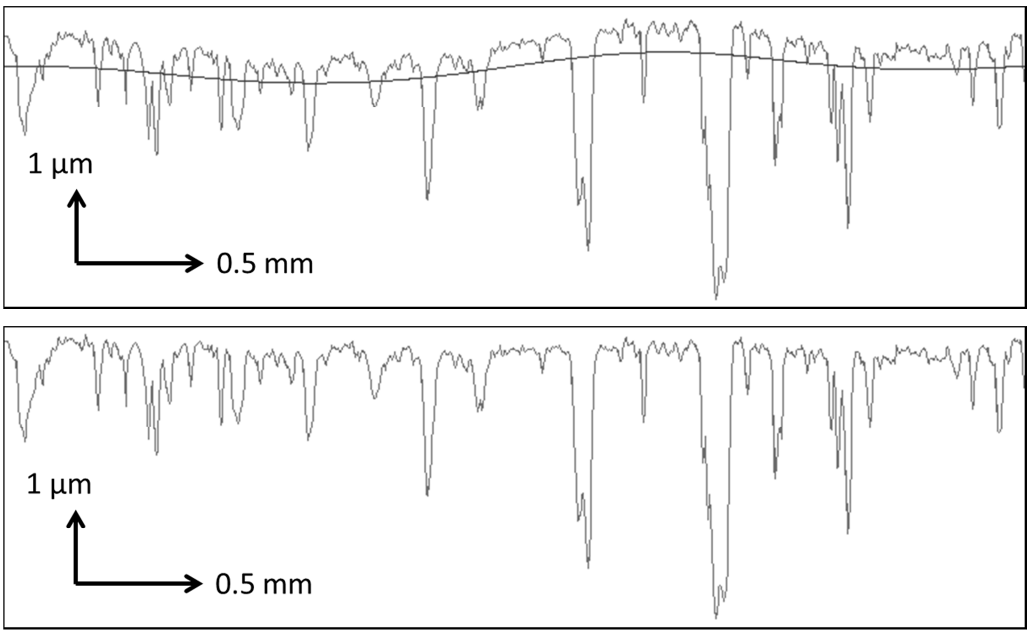

Figure 15. Cylinder liner profile measurement filtered with Robust Gaussian filter. Top: Original measurement and waviness profile. Bottom: Filtered surface.

It is likely that future filtration techniques, including wavelets filtration [XLIII] will enable further improvement of the segmentation of different surface features.

3.1.3. Roughness parameters

Roughness parameters for surface roughness measurements in two dimensions, surface profiles, are characterized by the letter R. This indicates that the profiles are filtered according to the specific standard of the measurement. The letter W signifies the roughness parameters of the waviness profile (filter residue) and the letter P signifies roughness parameters of the primary profile (unfiltered) (see Figure 11).

The most common surface roughness parameter both in historic and present terms is Ra (see Figure 16), the arithmetic mean deviation of the profile heights [XLIV]. A surface is a complicated component, and using only this parameter to describe the surface is in most cases not sufficient. The Ra parameter does not take skewness or spatial properties into account. In Figure 17 this issue is exemplified, showing two versions of the same profile. The differences between the profiles are that the one on the left is inverted around the y-axis. If a rigid body, like a piston ring, would slide over these two surfaces the frictional outcome would be quite different with smooth plateaus in contact on the left figure and sharp peaks on the right figure.

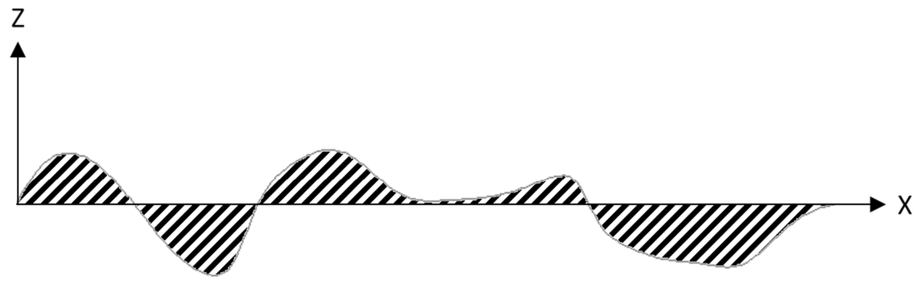

Figure 16. Illustration of the calculation of surface roughness parameter Ra, arithmetic mean deviation of surface heights.

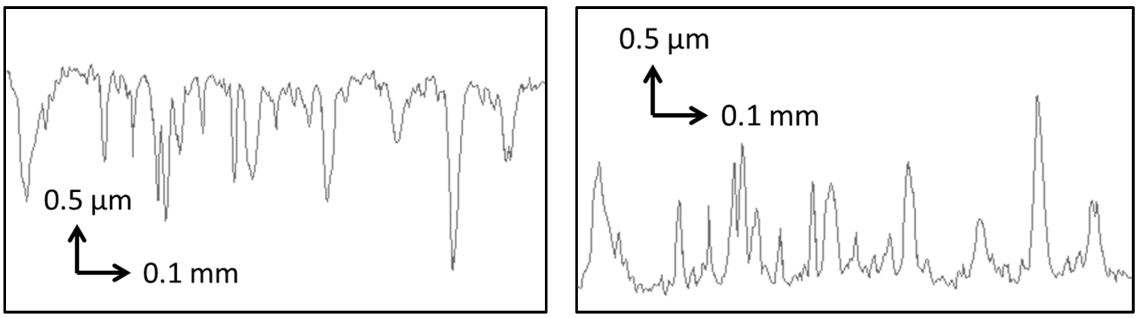

Figure 17. Two profiles showing the same two profiles with the profile on the left is inverted. These two profiles have the same Ra value (Ra=0,3 µm) but will have different frictional behaviour.

There is however not only one or a few 2D surface roughness parameters, in fact there are currently several hundreds of 2D surface roughness parameters [XXXVI].

One practical characterisation method for analysing surface morphology is done by quantifying the specific amplitude at different lateral levels of the surface. The Abbott-Firestone curve (1933), is a plot of the material/void ratio as a function of height value. The Abbott-Firestone curve offers a way to make a more specific quantification of the surface by describing the material distribution as a function of surface height. From the Abbott-Firestone curve it is possible to calculate the Rk/Sk family of parameters [XLV] which describes roughness amplitude for different height regions of the surface; peaks (Rpk, Spk), core/plateau part of the surface (Rk/Sk) and the valley part of the surface (Rvk/Svk) (see Figure 18). The Abbott-Firestone curve offers the possibility to analyse the roughness amplitude at different levels of the surface, however, this quantification does not take spatial properties into account. The Rk/Sk family of parameters is currently widely used in industry, but according to Malburg et al., a correlation between engine performance and the Rk/Sk family of parameters has not been shown [XLVI]. Whether or not this is an overstatement is not be elaborated further in this thesis, however, it could not be considered optimal to characterise the product of a material distribution created by a two-step finishing honing process (plateau honed surface) with three amplitude parameters. Previous research has shown that the Rk/Sk parameter set suffers from internal correlation [XLVII].

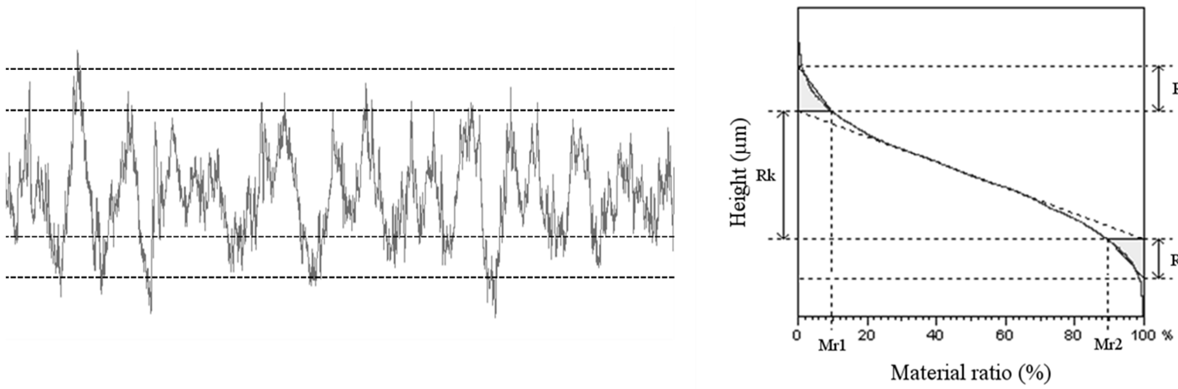

Figure 18. Abbott-Firestone curve with calculation of Rk surface roughness parameters.

The Rq/Sq family of parameters [XLVIII] also uses the Abbott-Firestone curve for parameter calculation, however, with modifications. A linearization of the probability plot of the Abbott-Firestone curve is used to calculate the Rq/Sq family of parameters (see Figure 19). In the Rq/Sq family of parameters two amplitude levels are used to characterise a surface. The Rq/Sq family of parameters consists of three parameters; Rpq/Spq (plateau Root-Mean-Square (RMS) roughness), Rvq/Svq (valley Root-Mean-Square (RMS) roughness) and Rmq/Smq (material ratio at plateau-to-valley transition).

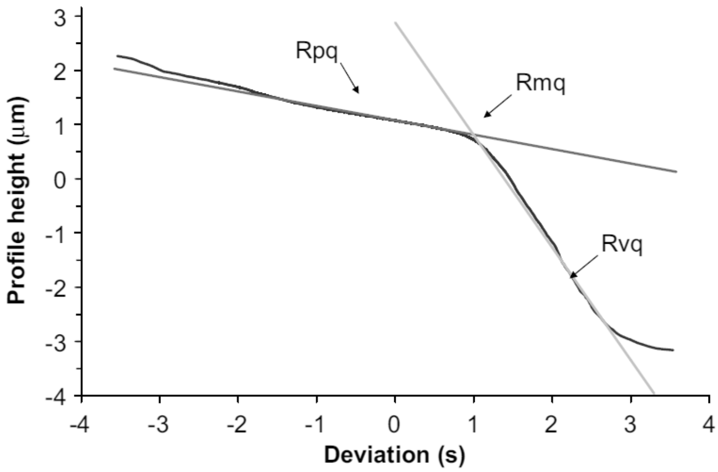

Figure 19. Material probability curve used for calculation of the Rq/Sq family of parameters. The unit of the abscissa is the material ratio on the standard probability scale [XLIX].

A surface is truly a 3D-feature and naturally a 2D representation of a surface, such as a profile, has clearly limited use. Researchers started to experiment with characterization techniques for surfaces in three dimensions in the 1980s [L] (it should however be noted that 2D surface roughness parameters could correlate with the corresponding 3D surface roughness parameter). The work later resulted in two parameters sets able to describe numerous properties of a measured surface such as spatial features, texture direction, roughness at different height sections of the surface etc. commonly known as the "Birmingham 14 parameters", this set of parameters was updated to contain the 25 roughness parameters which forms the core part of the ISO technical standard for areal surface texture (ISO/TS CD 25178-2: 2006) [XXXVIII]. The parameter sets in the ISO standard are divided into the S-Parameters (surface) and the V – Parameters (volume) (see Figure 20). The S-parameter set describes amplitude and spatial information, the V-parameter set describes volumetric information based on the Abbott-Firestone curve. The proposed 3D roughness standard consists of a limited number of parameter of which all were used in evaluating the surfaces analysed in the tribometer tests (in the appendix of paper III a list is provided covering all the surface roughness parameters used in this thesis).

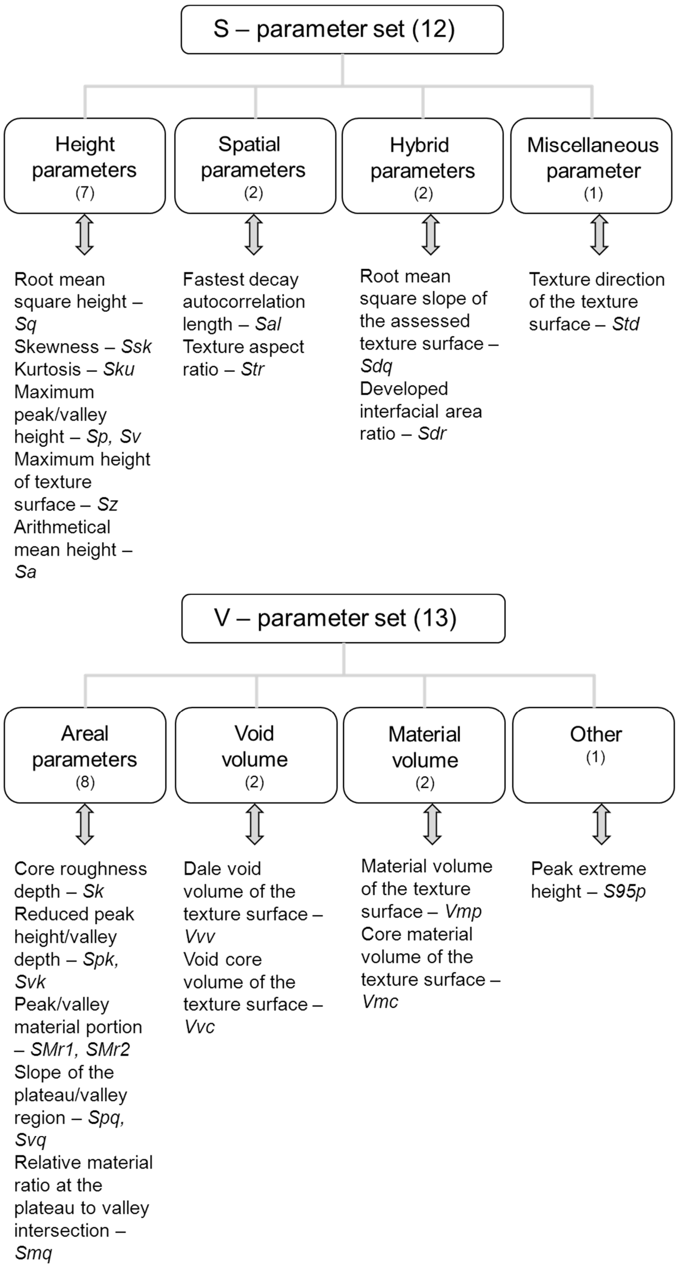

Figure 20. V- and S- surface roughness parameter sets. Adapted from [XXXVIII]

4. Finishing machining of cylinder liners

The production of cylinder liners is a process of several stages. Commonly, cylinder liners are centrifugally casted. In this initial production step mould is injected to a water cooled drum which is rotated at high speed. The rotational movement causes the mould to be pressed against the wall of the rotating drum, the water cooling allows the mould to solidify quite quickly. The outer diameter of the cylindrical cast geometry is determined by the diameter of the rotating drum, the inner diameter is determined by the amount of mould injected into the drum. Centrifugal casting creates a material with more homogeneous microstructure and more isotropic mechanical properties compared to other production processes such as rolled, welded or forged components [LI].

As the cylindrical pipe has cooled several machining steps are employed to machine the outer diameter and the axial ends to achieve the correct shape and roughness for fitting in the engine block. Since the engine type analysed in this thesis uses sleeve fitted liners the description of machining processes will be confined to this specific part, if machining is performed directly in the engine block the steps in the machining processes can differ.

The inner diameter of the cylinder liner is machined in different steps, each consecutive machining step removes material thus increasing the inner diameter of the cylinder liner. The machining steps earlier in the process chain remove more material than the machining steps later in the process chain. Each consecutive machining step also alters the form deviation; improving on cylindricity and straightness (although plateau honing has a smaller effect on these parameters due to the small amount of stock removal). Boring or drilling is the first step in producing the inner surface of the cylinder liner, this section will however focus on the two last steps in machining of the inner diameter of the cylinder liner; base honing and plateau honing.

4.1. Honing

Honing is a finishing machining process for the interior surface of cylinder liners, honing improves the form and the surface quality of the product. In honing the tool is constantly in contact with the surface and is simultaneously moved in the tangential (rotation) and axial direction. The main parameters that affect the production outcome, the material removal rate and the wear of the honing tool are: the contact pressure between the tool and the surface, the cutting speed, the axial stroke length of the tool and the machining time [LII].

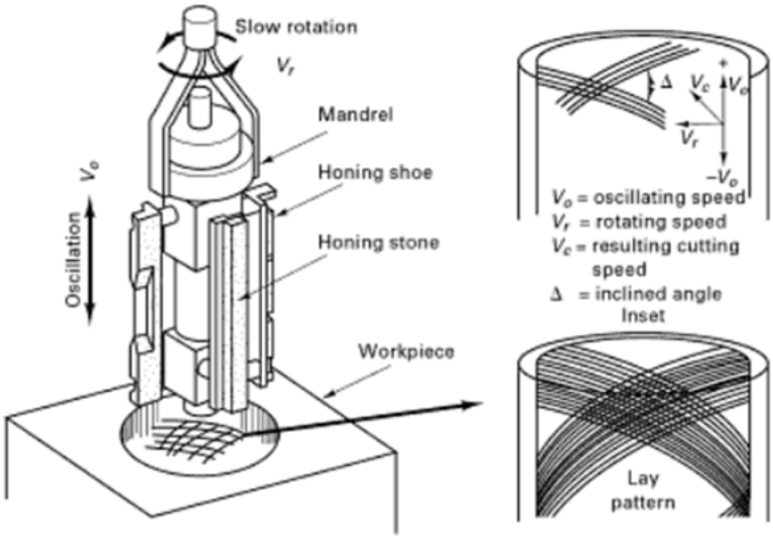

Figure 21. Overview of honing machine and the surface lay pattern resulting from honing of cylinder liners [LIII]

A honing tool consists of abrasives that are fixed in a tool using a bonding agent, it is the abrasive grains that are responsible for the chip formation and the cutting process. The abrasives in the tool can consist of different materials; corundum, silicon carbide, boron nitride and diamond [LII], honing ledges with diamonds as abrasives are expensive but have several benefits; more liners can be honed with one set of diamond honing ledges which means less frequent changes are needed, diamond tools also have a more consistent production outcome with smaller variations in the surface roughness in between liners [LIV]. In the machining process the bonding material is worn more rapidly than the abrasive grains causing the grains to be exposed on the cutting surface of the tool. The abrasive grains produce a cross hatched surface pattern on the interior surface of the cylinder liner. The angle of this pattern or the surface lay will be a product of the axial speed (oscillation) and the tangential speed (rotation). An illustration of honing angle can be seen in Figure 22.

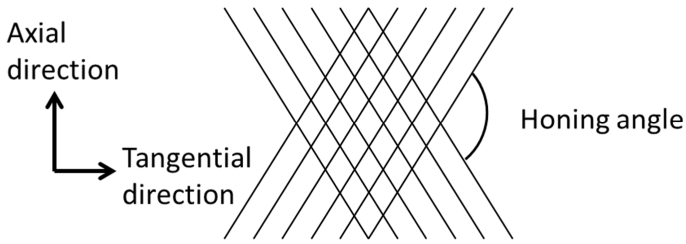

Figure 22. Schematic illustration of the honing angle on the cross hatched pattern of a cylinder liner surface.

Base honing is the second to last production step of the interior surface of the cylinder liner. In base honing a set of honing tools with coarse abrasive grains (coarse in comparison to the last production step, plateau honing) is used producing a rough surface. In base honing the final form dimensions are achieved.

Plateau honing is the last production step of the interior surface of the cylinder liner. The large difference between plateau honing and the previous production steps is that in plateau honing the material removal depth is smaller than the initial surface amplitude prior to machining. In plateau honing only a part of the surface amplitude from base honing is removed. In plateau honing a honing tool with much smaller grains is used which gives smooth plateaus with intersecting deep grooves remaining from base honing, these deep grooves or voids act as a reservoir for lubricant [LV]. Plateau honing decreases the oil consumption, the risk of scuffing, the cylinder liner wear and piston ring wear [LVI ,LVII]. An illustration† of the roughness of base honing and plateau honing can be seen in Figure 23.

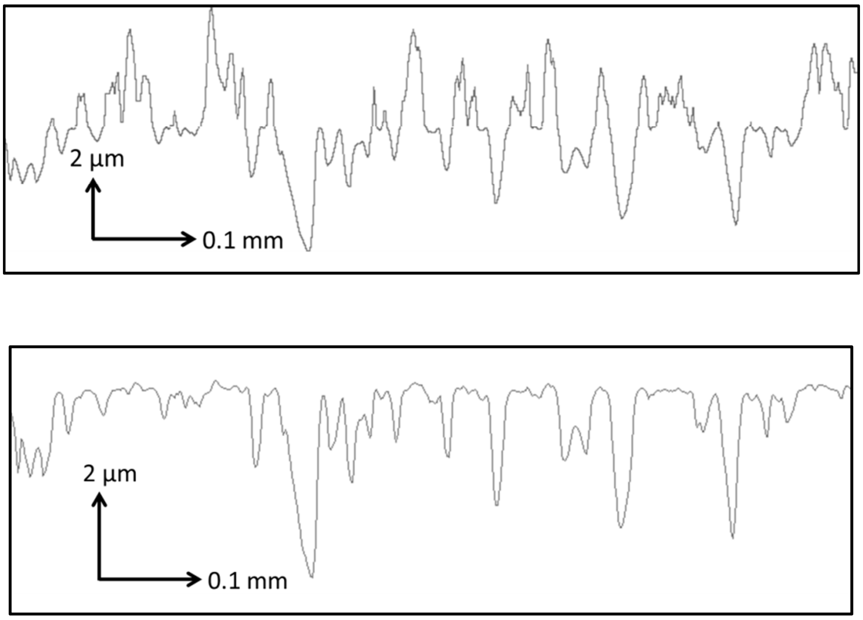

Figure 23. Illustration of different honing steps. Upper figure: After base honing. Lower figure: After plateau honing of the base honed cylinder liner.

In recent development and optimization of the honing process it has been possible to create what is considered to be an "ideal plateau" by using slide honing [LVIII]. In slide honing the amplitude of the plateau surface is greatly reduced compared to a conventional plateau honed cylinder liner, although the valley part of the surface is maintained. The advances of slide honing mainly lie in the development of the honing abrasives (mainly of diamond composition) [LVIII]. Besides reducing the plateau amplitude it is also today possible to produce a cylinder liner with increased honing angle using helical slide honing. Engine test results have shown that a helical slide honed cylinder liner has significantly less wear and oil consumption compared to a conventional plateau honed liner [LVIII].

† The plateau profile is here used as a basis in illustrating base honing, in the profile showing base honing the amplitude values of the plateau part of the plateau honed profile are increased with a zoom factor of 13, by this the same amplitude is obtained for the upper and lower part of the profile showing base honing.

4.2. *Additional manufacturing techniques*

4.2.1. Laser honing

There are alternative production methods for the interior surface of the cylinder liner. By using a high powered laser material can be evaporated from a surface, this creates a set of closed voids or valleys. In contrast to normal honing, laser honing is not confined to produce a characteristic surface pattern (e.g. cross hatched) , theoretically all types of surface voids are possible to manufacture, the only restriction is the movement capability of the laser beam and the adjacent manipulator. A laser honed surface consists of closed or open voids and a smooth plateau surface, the smooth plateaus allow a high percentage of contact area and the voids account for the adhesion of the lubricant to the contact zone [LII]. One disadvantage of laser honing is the introduction of at least one additional production step. Before laser honing the liner is pre-honed in the conventional manner, after the laser honing one further honing step is required; this last honing step is for generating the smooth plateaus and to remove the fusion- and oxide bulging generated in the laser process [LII].

4.2.2. Thermal spray

Thermal spraying of cylinder liners is a coating process in which both the material properties and the surface morphology can be improved. In this process melted or heated particles, are sprayed on to a surface. In the thermal spray process it is possible to use a wide variety of materials [LIX]. It is important to achieve high bond strength between the thermally sprayed material and the bulk cylinder liner material, to increase this bond strength the surface of the cylinder liner can be grit-blasted prior to the coating process. There are currently plenty of different thermal spray techniques; HVOF (High Velocity Oxygen Fuel), TWA (Twin Wire Arc system), APS (Air Plasma Spraying), HVSFS (High Velocity Suspension Flame Spraying) etc., some of which are currently used in serial production [LX].

Thermally sprayed coatings can consist of a certain level of porosity, when the surfaces are honed after thermal spraying these porosities open and form closed voids. Instead of honing grooves in a plateau honed surface acting as the oil reservoir the voids originating from the porosities can act as the lubricant reservoirs [LXI]. It is possible to reach a very smooth plateau roughness on the cylinder liner surface using thermal spray but still retaining the valley component of the surface. Figure 24 shows a measurement of a plateau honed cylinder liner surface and a thermally sprayed cylinder liner surface. Although these surfaces have roughly the same value of peak-to-valley amplitude the morphological properties differ. The plateau honed surface has a Ssk of -2,1, the thermally sprayed surface is much more negatively skewed with an Ssk value of -4,8, indicating a much lower amplitude of the plateau part of the surface compared to the amplitude of the valley part of the surface. The honing scratches on the plateau part of the plateau honed liner can clearly be observed in Figure 24, while the honing scratches on the plateau part of the thermally sprayed surface is hardly visible. This is because almost all of the plateau roughness are on the same value of vertical amplitude, thus the plateau surface of the thermally sprayed liner is much smoother compared to the plateau honed liner.

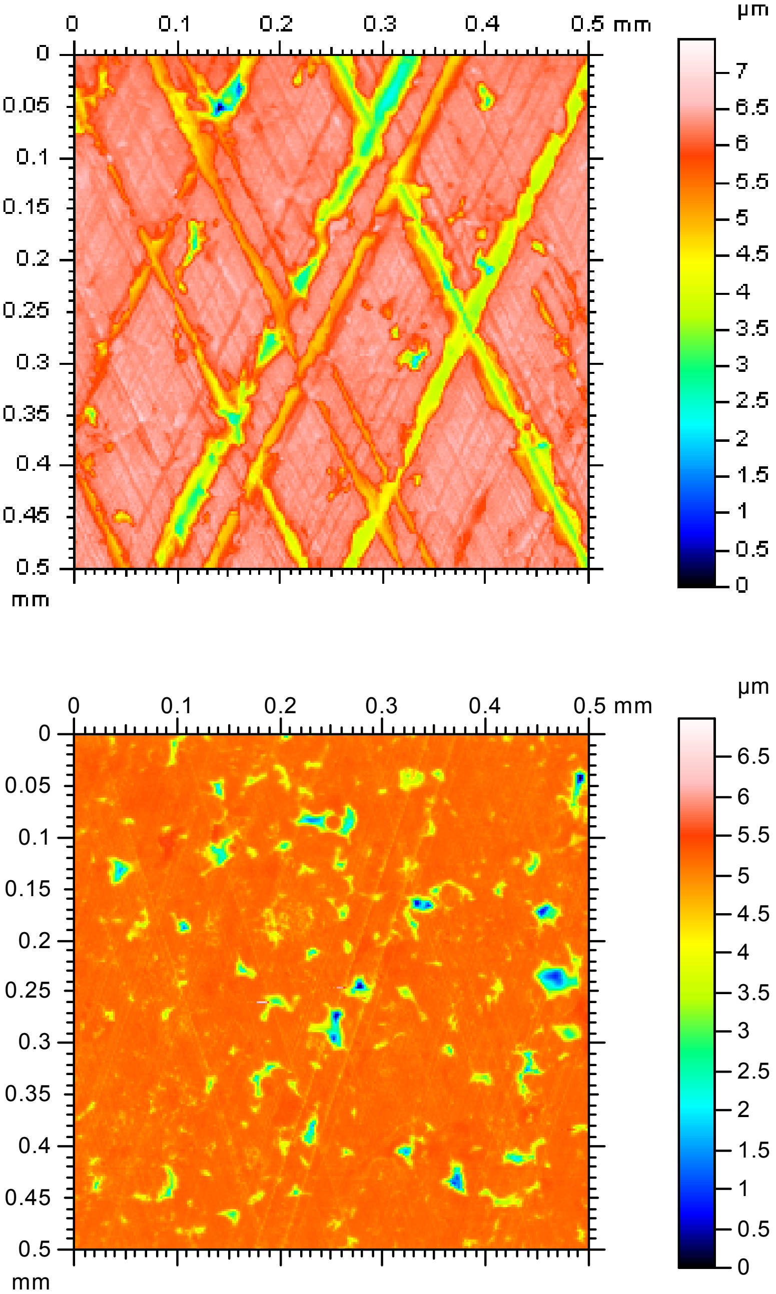

Figure 24. Examples of surface roughness measurements of plateau honed cylinder liner (top) and thermally sprayed cylinder liner (bottom).

5. The Diesel Engine - Functionality

5.1. Historical development of the Diesel Engine

The four stroke spark ignition (SI) engine was developed by Nikolaus Otto in 1876, a few years later, in 1892, Rudolf Diesel invented the compression ignition (CI) Diesel engine. By igniting fuel with air heated from compression the Diesel engine permitted a doubling in efficiency compared to other internal combustion engines at the time [I]. The main parameter accountable for the increased efficiency is the allowable increase in compression ratio of CI engines; an SI engine has typically a compression ratio of between 8 and 12 whereas a CI engine has a compression ratio of between 12 and 24 [I]. Also the Diesel engine has improved gas exchange since no throttle, which is responsible to control the load for SI-engines, is needed.

5.2. Functionality and Components of the Power Cylinder Unit

The power cylinder unit consists (PCU) of piston, piston rings, gudgeon pin, connecting rod and cylinder liner. The power cylinder unit has the following responsibilities:

- To transfer the combustion pressure into torque
- To seal the combustion chamber from the crankcase
- To remove heat from the piston (piston rings and cylinder liner)
- To control the consumption of engine oil

The diesel engine analysed in this thesis work operates according to the four stroke principle. Each stroke performs different tasks; the operating- and tribological contact conditions (pressures and temperatures) are different for the four strokes:

- *Intake stroke:* This stroke starts at – 360 crank angle degrees with the piston at the upper reversal zone and ends at -180 crank angle degrees when the piston is at the lower reversal zone. The inlet valve(s) are opened just before the stroke starts and closed just after the stroke ends to maximize the amount of air introduced in the cylinder.
- *Compression stroke:* This stroke starts at – 180 crank angle degrees with the piston at the lower reversal zone and ends at 0 crank angle degrees when the piston is at the upper reversal zone. In this stroke both the inlet and outlet valve(s) are closed, due to the piston motion the air injected in the intake stroke is compressed to a small volume and heated.
- *Power stroke:* This stroke starts at 0 crank angle degrees (combustion top dead centre, CTDC) with the piston at the upper reversal zone and ends at +180 crank angle degrees when the piston is at the upper lower zone. The amount of torque generated in a Diesel engine is controlled by the amount of fuel injected in the power stroke. The pressure from combustion pushes the piston down, at high engine load around five times as much work is done on the piston in the power stroke compared to the work in the compression stroke. The pressure behind the compression ring(s) increases with the engine load [LXII], in a piston system with two compression rings the gas pressure acting behind the upper compression ring is significantly higher compared to the gas pressure behind the lower compression ring in the vicinity of the CTDC position. The

gas pressure behind the compression ring(s) pushes the ring in the radial direction generating radial force acting between the outer diameter of the piston ring and the cylinder liner.

- *Exhaust stroke:* This stroke starts at + 180 crank angle degrees with the piston at the lower reversal zone and ends at +360 crank angle degrees when the piston is at the upper reversal zone. In this stroke the exhaust valve(s) is opened to let out the exhaust gas. As the piston approaches the upper position the cycle starts again with the intake stroke.

An overview of the four strokes can be seen in Figure 25.

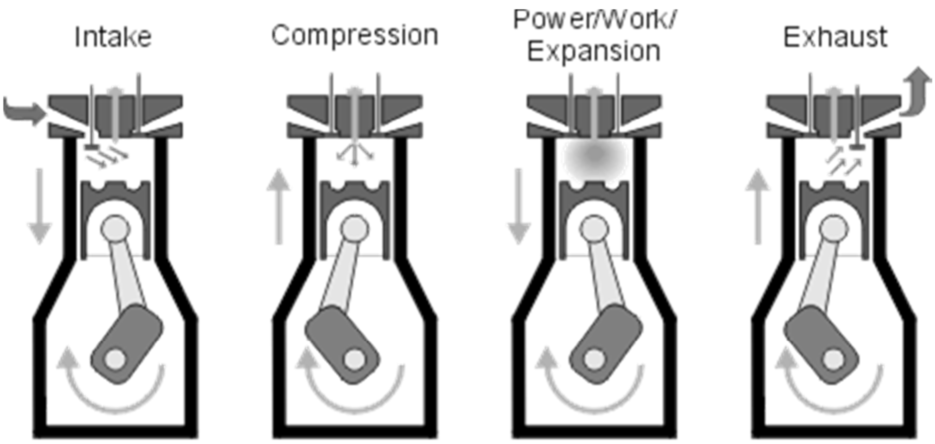

Figure 25. Operating principle of the four stroke diesel engine. [LXIII]

The engine investigated in this thesis work, Volvo MD13, has a piston with three piston rings; two compression rings and one oil control ring (see Figure 26). The two compression rings (top and second) are mainly responsible for sealing the crankcase from the combustion chamber, the oil control ring (third) is mainly responsible for scraping oil towards the crankcase and distributing a layer of oil film on the cylinder liner. To ensure a tight fit between piston ring and cylinder liner the piston rings are manufactured with a preload (piston ring tangential force) which acts between the rings and the cylinder liner. The tangential force of the oil control ring for the engine in this study is more than twice the tangential force of either of the two compression rings.

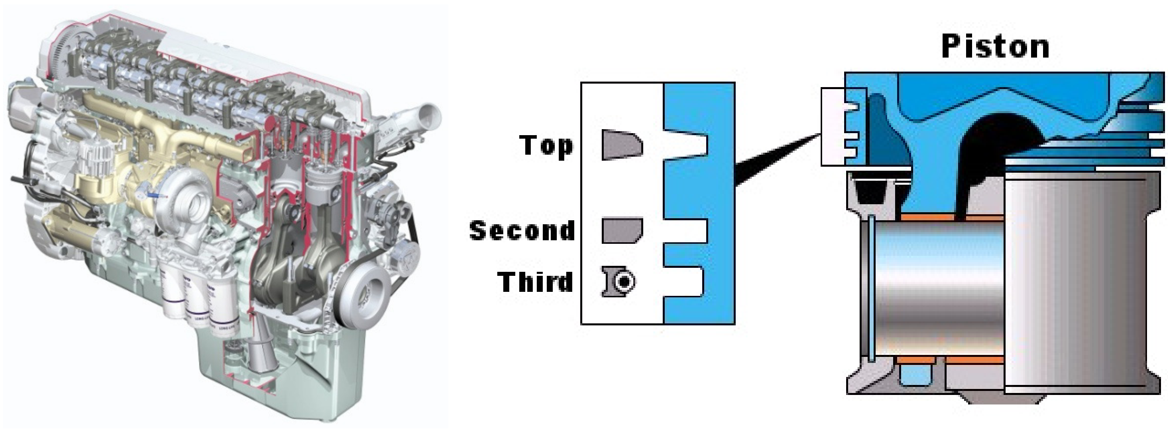

Figure 26. Left: Volvo D13 diesel engine. Right: Overview of the piston and piston ring design.

5.3. Piston Rings and Frictional Losses

The PCU is responsible for 50 % of the total frictional losses in the engine [I, II, III, IV]. The frictional power losses in the PCU are not only to be considered as a direct loss of torque, but also a thermal load caused by frictional heating of rubbing components has to be removed by the engine cooling system, thus the frictional thermal load is also to be considered as a loss mechanism.

As stated previously in this section the force behind the top piston ring is dependent on the cylinder pressure, thus the top ring in an engine with high cylinder pressure will have a high radial force in the vicinity of the CTDC position. The contact pressure between the top piston ring and the cylinder liner will increase with the axial width of the top piston ring since a larger axial width creates a larger area on the backside (inner diameter) of the ring. In a previous study [III] it was shown that the oil control ring is accountable for 75 % of the total frictional losses from the piston rings, however, this value is dependent on the running condition and the geometry of PCU components. For a high loaded heavy-duty diesel engine it is likely that the frictional power loss from the top piston ring is higher than any of the other rings [LXIV], at least for operating points with higher load.

A study [III] has shown that within the industry frictional losses in engines are currently being reduced mainly employing two measures; reducing the tangential force of the piston rings (mainly referring to oil control ring) and reducing the axial height of the top piston ring. Although these measures present a significant reduction in frictional losses it is also possible to significantly reduce frictional losses by reducing the hydrodynamic losses between piston ring and cylinder liner. The majority of the frictional losses between piston rings and cylinder liners are hydrodynamic losses [LXV], boundary friction losses are present mainly at low engine speeds in the reversal zones [LXVI, LXVII]. However, since the sliding speed is small at the reversal zones the boundary friction has little effect on the frictional power loss [LXXX].

The governing parameter for controlling the friction in the hydrodynamic lubrication regime is the dynamic viscosity of the engine oil [LXVIII]. Decreasing the viscosity has a large effect on reducing the hydrodynamic friction losses of the power cylinder unit [LXIX], however, a large reduction of the dynamic viscosity could significantly increase the frictional losses in the boundary and mixed lubrication regimes [LXX] and possibly increase wear.

Large wear levels of the outer diameter profile of the piston rings could shift the main lubrication condition from hydrodynamic to boundary [LXXI].

5.4. Oil consumption and blow by

The emission regulations of heavy duty diesel engines include, among other substances, particulate matter (PM). The consumption of engine oil has an important impact on the amount of PM, the primary factor in generation of PM is by incomplete combustion of fuel hydrocarbons (soot), however, since the combustion process is being intensively improved it is likely that the contribution of soot emissions will decrease [LXXII]. Decreasing the amount of engine oil consumption will have positive effects not only for emission reduction but also for the function of the complete drivetrain; less oil in the exhaust gases will prevent both clogging of the particulate filter and poisoning of the catalytic converter. Oil may be consumed from various sources in the engine, including the turbocharger, the valve stem seals and the crankcase ventilation, the oil mass transport mechanisms from the power cylinder unit to the exhaust can be described with the following [LXXIII]:

- Evaporation from the surface of the cylinder liner
- Throw-off of oil that has accumulated on the piston above the top ring
- Oil containing reverse gas flow, from the piston ring gap into the combustion chamber
- Scraping of oil from the top edge of the piston (upper position on the top piston land)

Blow by gases are combustion gases which flow through the ring pack into the crankcase. A minimization of blow by gases is desired since an increase in blow by gases causes a decrease in torque, an unnecessary heating of the piston rings and a deterioration of the engine oil (and possibly also an increase in fuel dilution of the engine oil). There is thus a need to decrease both oil consumption and blow by, however, there is a characteristic trade-off between the amount of blow by gases and the amount of oil consumption. Since the blow by gases push down the oil towards the crankcase an increase in blow by causes a decrease in oil consumption. To control this equilibrium it is possible to alter the gap clearance for the top piston ring in order to gain the lowest possible levels of both blow by and oil consumption [LXXIII].

The surface roughness of the cylinder liner is an important factor in controlling the oil consumption. Experiments have shown that by altering the surface morphology of the cylinder liner a significant decrease in oil consumption can be obtained without an increase in the amount of blow by gases [LXXIV]. Although this experimental study only points to the cause of decreased oil consumption it does not provide a physical explanation for the decrease in oil consumption. In said study the oil consumption is likely decreased by means of reduction of the oil film between piston rings and cylinder liner. The oil film thickness between the piston rings and the cylinder liner is an important property in controlling oil consumption [LXXV]. Besides the oil film thickness of the piston rings other factors influencing oil consumption include rheological properties of the engine oil [LXXVI], conformability of the piston rings [LXXVII], cylinder liner out of roundness [LXXVIII] and the geometry of the piston skirt [LXXIX].

6. The Diesel Engine - Experimental Techniques

6.1. Experimental Tribometer Testing

A tribometer is an experimental device used to analyse the friction and wear of surfaces in relative motion, most commonly a load is applied to one of the components where the mating component supports this load, thus causing a contact pressure between the components. Tribometer testing presents the opportunity to conduct isolated analysis of engine components and is a common procedure for analysis of frictional effects and the durability of PCU components [LXXX, LXXXI, LXXXII]. There are several types of tribometer configurations, the selection of configuration is dependent on the full-scale application and the lubrication regime of interest, in analysis of the PCU components a reciprocating tribometer is most commonly used. An overview of the most common tribometer configurations is shown in Figure 27.

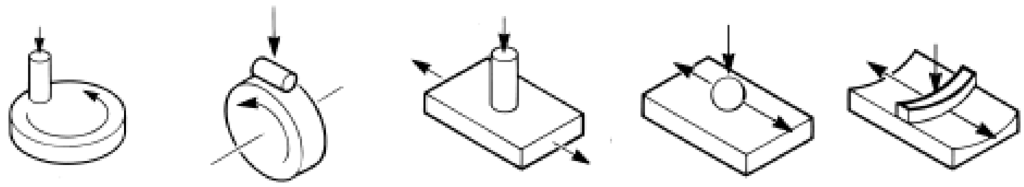

Figure 27. Illustration of the most common tribological test configurations. Left-to-right: Pin-on-disc, block-on-ring, reciprocating pin-on-plate, reciprocating ball-on-plate and reciprocating piston ring segment on cylinder liner specimen [LXXXIII, LXXXIV].

Two different reciprocating tribometers were used in this thesis work, both tribometers were reciprocating and both used a complete piston ring and a segment of a cylinder liner as experiment samples. Both tribometers also used piezo electric transducers to measure the friction (shear) force and a strain gage transducer to measure the normal load. Both the vibrator tribometer and the eccentric tribometer were built and developed in previous projects, however, the eccentric tribometer was modified in this thesis work to include different frequency settings in the DoE setup, also the lubrication system was modified in this rig to ensure a constant amount of oil supply to the piston ring/cylinder liner interface throughout the experiment.

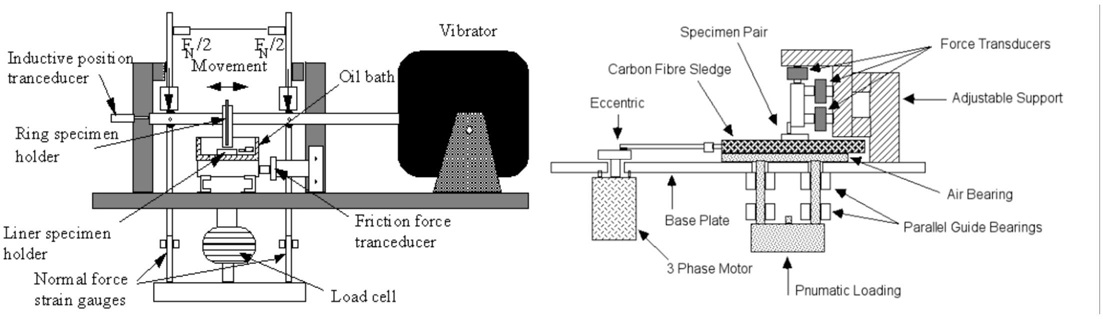

Figure 28. Schematic illustrations of the tribometers used in the thesis work. Left: Vibrator reciprocating tribometer. Right: Eccentric reciprocating tribometer

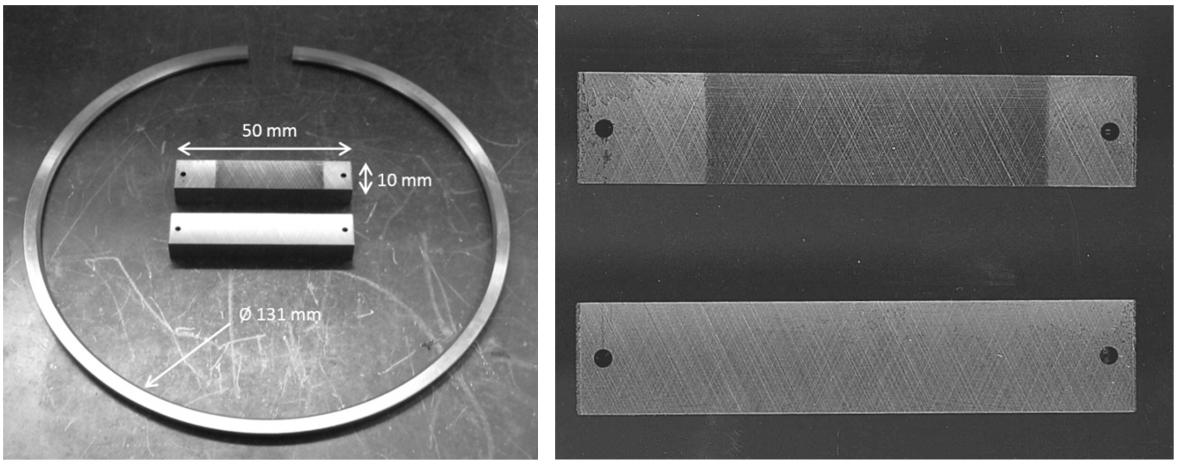

Figure 29. Overview of used test specimens in tribometer testing. Left: Top piston ring and two cylinder liner samples. Right: Close up of two cylinder liner samples, the upper sample has been used in tribometer test, the lower sample is unused.

6.2. Simulation of the Piston Ring/Cylinder Liner Interface

It is possible to numerically analyse the frictional behaviour in tribological simulations thus enabling less costly full-scale verification of component alterations. Tribological calculations/simulations often require high detail and a high level of complexity. However, since computing power has steadily increased so has the possibility to investigate tribological phenomena using tribological simulations. The rise in computing power has not only decreased the duration of calculations and allowed for an increased sophistication of possible algorithms to be used but has also allowed for more detailed types of visualisation aids in presenting the calculation result. Under current prognosis it is estimated that in the year 2030 computing power will have increased by a factor of one million compared to the reference year of 2001 [LXXXV], this presents a next to unlimited opportunity for the advancement of tribological calculations.

Within this thesis work the contacts and the functionality of the piston system have been analysed using two different simulation tools. The main purpose of the calculations was to analyse the frictional response of different cylinder liner surfaces (Paper I and Paper IV), also the correlation between cylinder liner roughness and oil consumption was analysed (Paper I).

6.2.1. Piston Simulation

In the initial part of this thesis (Paper I) the simulation tool “Piston Simulation” [LXXXVI, LXXXVII] was used, with this simulation tool it is possible to analyse the gas flows, piston ring dynamics and the surface interactions within the power cylinder unit. The simulation model in Piston Simulation uses a range of input parameters to quantify the geometry and functionality of the power cylinder unit: (1) detailed component geometries and material properties of the three piston rings, the cylinder liner and the piston including surface characterisation, (2) general component geometries of the PCU (such as bore, stroke, connecting rod length etc.), (3) the rheological properties of the engine oil (4), thermal deformations and (5) the temperature of the surface interfaces. Using the value of ring load, sliding speed and oil viscosity Piston Simulation solves the average Reynolds equation [LXXXVIII]. Using this equation it is possible to obtain values of the hydrodynamic pressure, the shear stress of the oil film, the roughness dependent flow factors and the oil film thicknesses. The illustrations in Figure 30 show the top piston ring at two locations of the stroke with areas indicating the contact pressure between piston ring/piston ring groove and piston ring/cylinder liner, the difference in contact load on different parts of the stroke can clearly be observed.

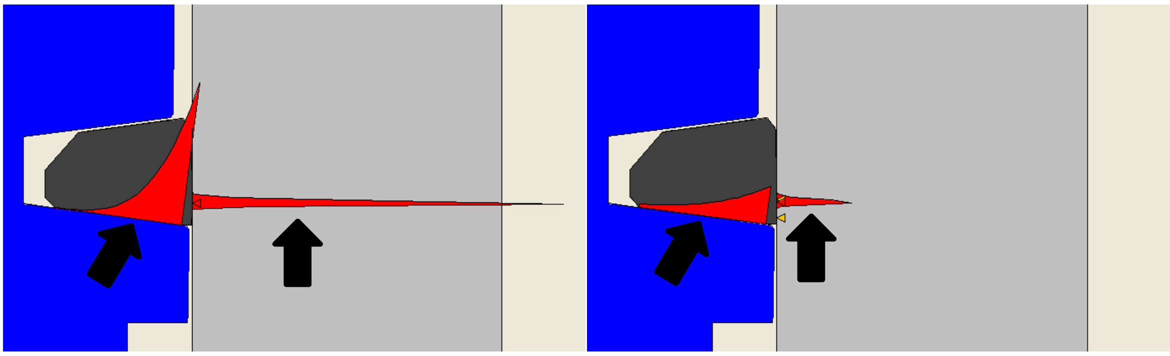

Figure 30. Illustration of simulation result showing how contact load decreases with stroke position. Images are showing the top piston ring at two different locations in the expansion stroke. The red triangular areas (indicated with arrows) in the piston ring groove and the pointy red areas (indicated with arrows) in the contact between cylinder liner and piston ring represent the contact load acting between the components. Left: 0 crank angle degrees (combustion top dead centre). Right: 58 crank angle degrees (approximately 45 mm below top dead centre position).

The characterization of surface roughness within Piston Simulation is done by calculating the root mean square of the combined roughness amplitude standard deviation (surface parameter Rq) for the piston rings and the cylinder liner. Since the surface morphology of cylinder liner surface is complex this characterisation suffers from some limitations. Firstly, the cylinder liner surface is ordinarily negatively skewed (non-standardly distributed), resulting from the final machining process of the cylinder liner (plateau honing, see section 4). Secondly, it is not possible to characterise the surface lay (honing angle) of the cylinder liner using the Rq parameter.

6.2.2. Deterministic Simulation

The novel "Deterministic Simulation" software [LXXXIX, XC, XCI], used in Paper IV, has the possibility to more accurately describe the frictional interaction between oil control ring and cylinder liner. Deterministic Simulation is a simulation tool which calculates the operational frictional properties in the contact of the twin land oil control ring and the cylinder liner. The oil control ring in Deterministic Simulation is considered to be fully flooded (as opposed to starved) at all parts of the stroke. This means that oil always is considered to be available for the inlet of the contact between piston ring and cylinder liner. In this thesis work the Deterministic Simulation tool has been used for analysing the contact between oil control ring and cylinder liner, however, a complete model including the piston, top and second piston ring is currently in development [C]. Since Deterministic Simulation only models the oil control ring the complete model of the piston and the two compression rings was not needed, however all the other input data that also exists in Piston Simulation (such as detailed component geometry of the oil control ring and the piston oil control ring groove, material properties of the engine oil etc.) is also used in the Deterministic Simulation model. The main input in the Deterministic Simulation model was a cylinder liner 3D surface measurement of arbitrary point spacing and area size. In this work a surface measurement of 3 mm (sliding direction) * 2 mm (tangential direction) with a point spacing of 2 μm was used, the surfaces were measured with a stylus with 2 μm tip radius. To analyse the frictional effects of different cylinder liner surfaces (both with different surface morphology and general production method) numerous simulations at different operating conditions were performed in this thesis. The result from the calculations using Deterministic Simulation was of great importance in this thesis since the result of the engine test and the following discussions and conclusions (Paper III) was numerically confirmed (Paper IV) by using Deterministic Simulation.

6.3. Engine Testing

It is desired to characterise the frictional behaviour of component alterations in engine tests. Frictional losses can be quantified both in motored engine tests and in engine tests that include combustion, however, the true representation of frictional losses can only be obtained by using a fired engine since [I]:

- The temperatures are higher in combustion compared to motoring which alters the viscosity of the oil. Temperatures can also influence geometrical properties of the components (mainly referring to cylinder liner roundness)
- The pressures are higher in test with combustion, in motoring only the compression pressure is acting on the piston, the piston rings and the bearings.

Measurement methods of characterising frictional losses in engines with combustion include:

- *Measurement of Friction Mean Effective Pressure (FMEP) from Indicated Mean Effective pressure (IMEP) [I]*:. By measuring the Brake Mean Effective Pressure (BMEP) and subtracting BMEP from IMEP the frictional losses can be quantified.
- *Floating Liner Technique [XCII, C]*. In this setup the cylinder liner is fixed to piezo electric transducers, no additional modification of the piston system is necessary. All axial forces acting between the piston rings/piston and the cylinder liner are measured

with this technique. A comparative method, not that dissimilar from the Floating Liner technique, is the method in which the elongation of the connection rod is measured with crank angle resolution during fired operation of the engine [XCIII].
By measuring forces directly on the cylinder liner (or directly on the connecting rod) the Floating Liner technique offers the best possibility of quantification of frictional losses with crank angle resolution. A frictional analysis with high detail enables the analysis of different lubrication regime transitions on different parts of the stroke with different operating conditions and also the analysis of the transitions of lubrication regime during the running in process (see Figure 31).

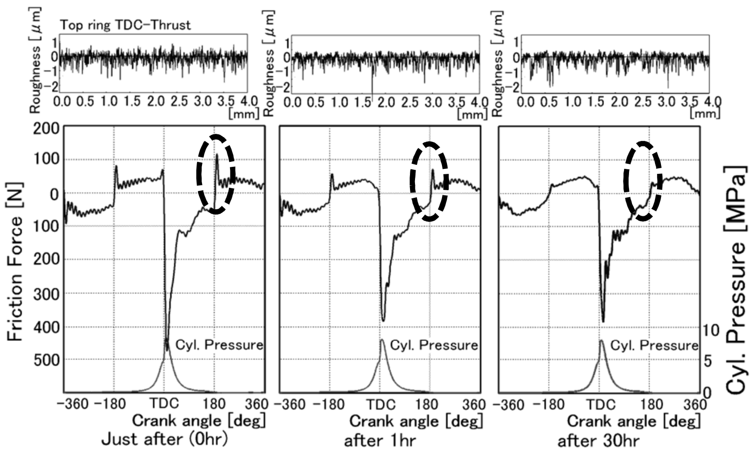

Figure 31. Overview of the Floating Liner measurement showing the decrease in frictional losses at the reversal zones (exemplified with encircled areas) and the alteration of cylinder liner surface roughness during running in [XCII]. Reprinted with permission from SAE Paper No. 2004-01-0604 © 2004 SAE International.

Measuring frictional losses directly in a fired engine is the only method with which true frictional losses can be quantified; however measuring friction in fired engine tests could prove difficult. There are different alternative motored methods to measure frictional losses in motored engine tests; in such tests an electrical dynamometer runs the engine. A summary of motored tests are as follows:

- *Direct motoring test method [I].* In this type of test the engine is motored at a constant speed and torque is measured, to mimic an engine with combustion temperatures of oil and coolant are kept as close to real conditions as possible.
- *Willans line [I, XCIV].* In this type of test the engine is motored for a specific set of speed intervals, torque is measured for each engine speed. A linear regression is performed for the measurement data and is extrapolated back to “zero” engine speed which gives a representation of the torque losses. Willans line is possible to use for tests with combustion in which the engine speed is constant and a range of different engine loads measured and quantified with the extrapolation technique.
- *Strip down method [XCV].* In this type of test the engine is operated at constant speed. The baseline torque of the engine is acquired with all components, after the baseline value has been acquired components are successively removed and torque is measured for each component removal. This technique offers the possibility to analyse the friction contribution of each component in the power cylinder unit, however, this method might not be ideal when a measurement with high sensitivity is required. If e.g. one piston ring is removed this would alter the amount of oil

available for the other piston rings which could be of importance for the frictional response of the engine.

- *Morse/misfire test [I]*. In this type of test a multi cylinder engine is required. Firstly, the torque is measured generated from the engine at constant speed with combustion. After this is completed the injector from one of the cylinders is disconnected, the engine then run by the remaining cylinders at constant speed. This is repeated for all cylinders. From the loss of torque from the disconnected cylinders it is possible to calculate the frictional loss. By using this method a close value of real friction can be obtained since temperatures close to the combustion temperatures can be achieved.

In this thesis work the engine testing was performed on a single cylinder D13 engine. The engine has primarily been used for combustion development [XCVI] but due to the high controllability of the engine it is also ideal for investigations involving measurement of fuel consumption and torque. The single cylinder engine corresponds to (in terms of power and torque) the commercial Volvo D13 six cylinder truck engine. By measuring Indicated Mean Effective pressure (IMEP) and the Brake Mean Effective pressure (BMEP) the Friction Mean effective Pressure (FMEP) could be calculated. The torque was set to a pre-determined constant level for each experimental point and the resulting Brake Specific Fuel Consumption (BSFC) that was required to sustain the level of torque was measured. Direct motoring test method was used in this thesis work.

7. Statistical Design and Analysis

7.1. Design of Experiments

A measurement is the ultimate test of a theory or a design and also the platform from which we build our understanding of the physical world, if physical variables are not measured they can only be estimated [XCVII]. In a system with several input parameters the procedure of altering one separate parameter at a time (often referred to as trial-and-error) is truly inefficient, both because this type of method might not be capable of interpreting (finding) an optimum and also that the ratio of the number of experiments divided by the precision of the result is higher compared to a statistical design model [XCVIII]. The biggest difference between Design of Experiments (DoE) and the trial-and-error approach is that all factors are varied simultaneously in the total number of experiments in the experimental design. The order of experiments could be randomized to make sure that time trends are left out from the experimental outcome. There are three types of designs within DoE; screening, optimization and response surface modelling (RSM). Both optimization and RSM requires knowledge of the potential important factors, thus only screening was used in this thesis work. A screening approach with a factorial two level design (high and low level, + and -) was used in order to study the influence of input parameters. Included in the screening model were also a set of centre points both to evaluate the experimental reproducibility and to account for changes over time (see Paper II). Within statistical analysis of surfaces it is usually not possible to generate input parameters that are in line with a factorial DoE approach. In this case it is important to maximize the amount of experiments (Paper IV) to create a statistical foundation which is used to analyse the correlation between input and output parameters.

7.2. Multi Variate Data Analysis

Multivariate analysis (MVA) is a statistical tool with which one can investigate the relations between an arbitrary number of input and output variables. There are several types of different MVA methods, the type of MVA method that was used in this thesis work was partial least squares regression (PLS). PLS is used to find the fundamental relation between input data, X and output data, Y. Using PLS it is possible both to analyse linear and non-linear correlations between parameters, a non-linear analysis naturally requires a more complex DoE setup. It is important to judge whether or not the generated statistical models are useful in correlating input parameters with output parameters. Two variables, R^2 and Q^2, are used to indicate the usefulness of the statistical models; R^2 could be expressed as the model fit of experimental parameters and Q^2 is the usability of the model showing how well an experimental parameter could be predicted with a model. What is used to a large extent in this thesis work is the scaled coefficients and the correlation to a single output parameter. By using scaled coefficient it is possible to judge whether or not an experimental input parameter is significant, the scaled coefficients are also useable to rank the level of significance in a set of input parameters (this is exemplified in Figure 37).

8. Main Results of Appended Papers

Paper I

In this study the frictional effect of different cylinder liner surfaces was investigated using both a simulation tool and experimental tribometer tests.

A complete model of the PCU was constructed in the simulation tool “Piston Simulation”, several simulation cases were analysed in which the surface roughness of the cylinder liner was varied. The simulation result showed that the oil film thickness decreases with decreasing cylinder liner surface roughness. It was also shown that the friction could be reduced with decreased cylinder liner roughness. Engine tests (internal results) confirmed a decrease in oil consumption with a decrease in cylinder liner surface roughness.

In the tribometer analysis friction between top piston ring and cylinder liner was studied. Cylinder liner samples with different honing angles were analysed in combination with different coatings, applied on the top piston ring. To measure the effects of running in each experiment was run for 3 hours, the wear volume of the cylinder liner sample was measured and the experiment was again run for an additional 13 hours with an additional measurement of wear. All experiments showed a higher wear rate for the initial experimental duration which is expected in the running in phase. The result from the experiments showed that the PVD (physical vapour deposition) coated top piston ring exhibited superior wear resistance, other coatings CCC (chromium ceramic coating) and HVOF (high velocity oxygen fuel) exhibited large scratches on the surface after the experiment (see Figure 32).

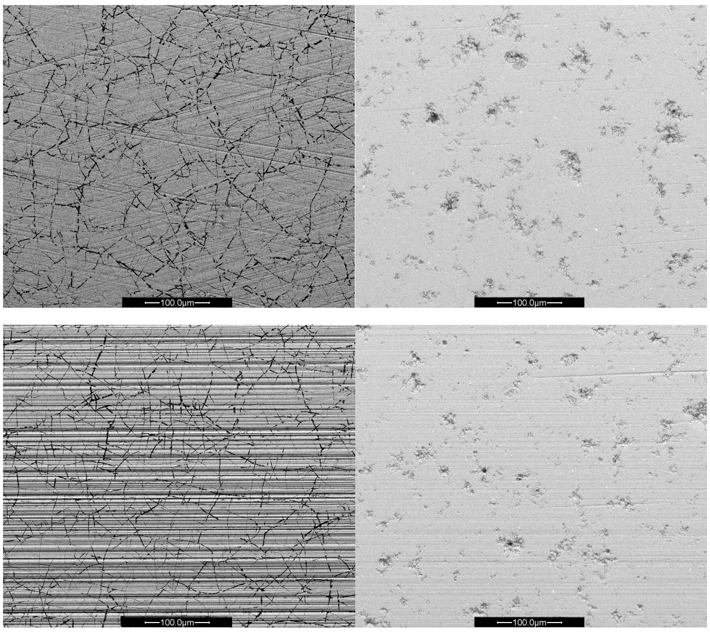

Figure 32. SEM images of piston rings. Top left: CCC coated ring outside of wear scar. Bottom left: CCC coated ring inside wear scar. Top right: PVD coated ring outside of wear scar. Bottom right: PVD coated ring inside wear scar.

The cylinder liner sample used with the PVD coated top piston rings showed little signs of wear, in comparison to the other material combinations investigated, which exhibited roughly the same wear amount in the experiment. The cylinder liner with the PVD coated top piston ring exhibited 1 % of the wear (see Figure 33). Due to the high complexity of measuring wear volume for extremely small wear amounts a new parameter Rktot was proposed. With this parameter the wear is quantified by using a simplified integration of the Abbott Firestone curve. The parameter might prove useful to quantify wear, however, the parameter is dependent on the wear depth being smaller than the initial surface roughness amplitude.

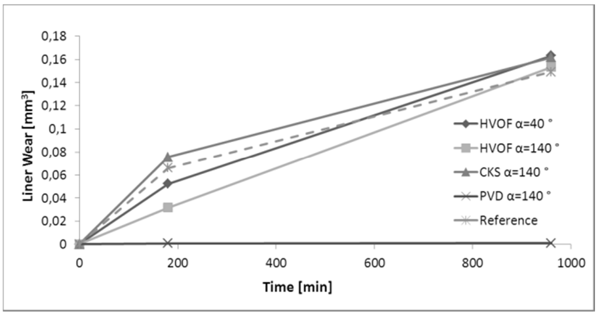

Figure 33. Wear volume of cylinder liner at 180 minutes and 960 minutes experiment duration.

Paper II

The aim with tribometer testing is to produce results which are comparable with the full scale engine test, related to research question 2. The restricted 8 mm stroke length of the vibrator reciprocating tribometer in Paper I results in studies being limited to the boundary and mixed lubrication regimes. The experimental input parameters (load, reciprocating frequency and temperature) were held constant in the vibrator tribometer tests. The tribometer used in Paper II was the eccentric tribometer. The experimental input parameters were; reciprocating frequency, normal force acting between cylinder liner and piston ring and temperature. These input parameters were continuously measured during the experiment and were controlled using a feedback loop which was active throughout the experimental duration. This procedure ensured a more consistent set of experimental input parameters. There were also two main operational differences between the vibrator and eccentric tribometers, firstly, the vibrator tribometer operated with constant experimental parameters whereas the eccentric tribometer used a predetermined Design of Experimental (DoE) experimental setup in which different levels of load, reciprocating frequency and temperature were varied in experiments. Secondly the reciprocating eccentric tribometer was capable of a stroke length of 30 mm, using the vibrator tribometer the stroke length was confined to 8 mm.

Based on values obtained from piston simulation it was determined that the highest frictional power loss for the top piston ring occurs approximately between 10 – 30 CAD (see Figure 34).

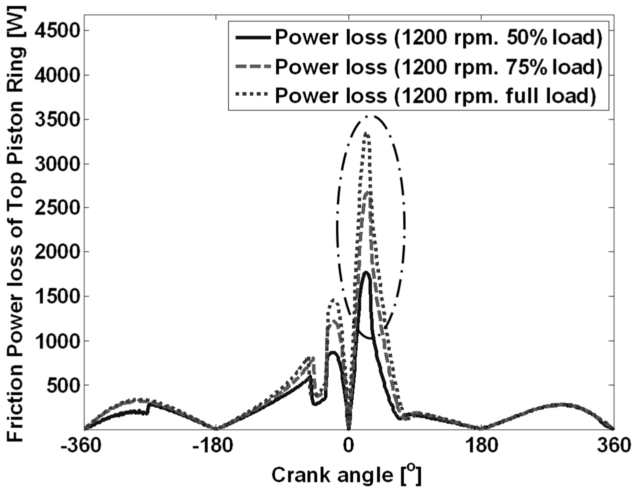

Figure 34. Friction power loss of top piston ring at 1200 RPM and 50%, 75% and full load.

The dynamic viscosity, the contact pressure and the sliding speed were calculated for the top piston ring in the engine. Using the values of these parameters the Hersey parameter was used as a parameter to compare the operating conditions for the top piston ring in the engine with the operating conditions of the top piston ring in the tribometer. The values of the input parameters in the DoE setup of the tribometer were selected in such a way that the average values of the Hersey parameter in the tribometer represented the engine at the 3 – 28 CAD for full load and 3 – 22 CAD for 50 % load CAD (see Figure 35).

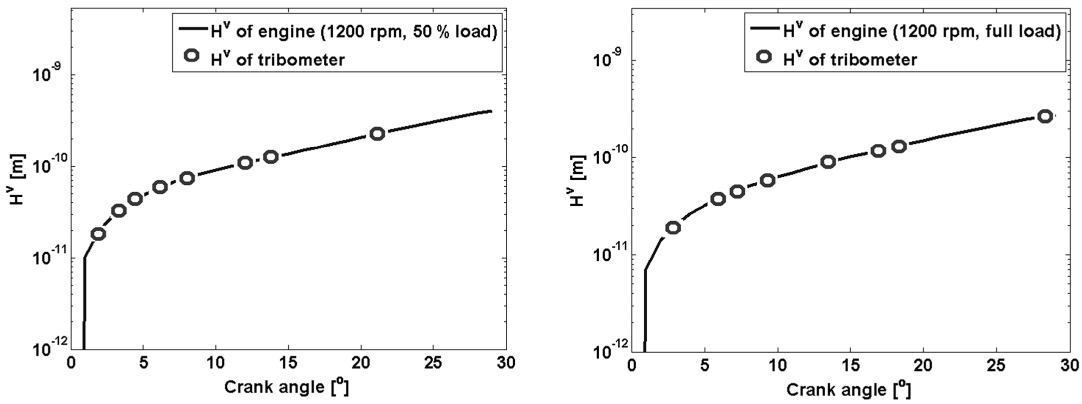

Figure 35. Calculation of the Hersey parameter for engine and tribometer.

Three different surface types were analysed using the developed approach, five experimental repetitions of each material type were conducted. There were relatively large frictional differences between the material types as shown in Figure 36. The frictional result correlated well with surface roughness parameters describing the surface, however, due to the multitude of surface roughness parameters that show a significant correlation to friction in combination with the relatively few number of material combinations it was difficult to draw conclusions of what surface characteristic (e.g. plateau or valley amplitude) is of most importance to decrease friction.

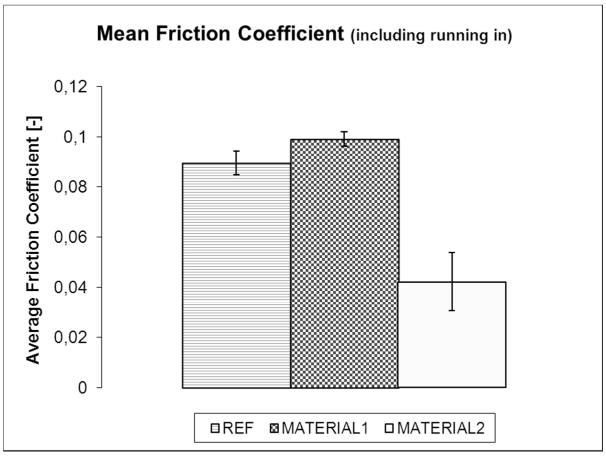

Figure 36. Average friction coefficient including running in stage. The error bars represent the standard deviation of the five experiments.

Using statistical analysis it was possible to determine that the interaction of dynamic viscosity, velocity and contact pressure could be studied within one experiment. For a material/surface with lower friction (MATERIAL2) the importance of dynamic viscosity and velocity increases. For a material/surface with higher friction only contact pressure is of importance. This means that a surface has to be able to generate conditions for oil film build-up. If this is not accomplished the properties of oil and the velocity have little (REF) or no (MATERIAL1) significance.

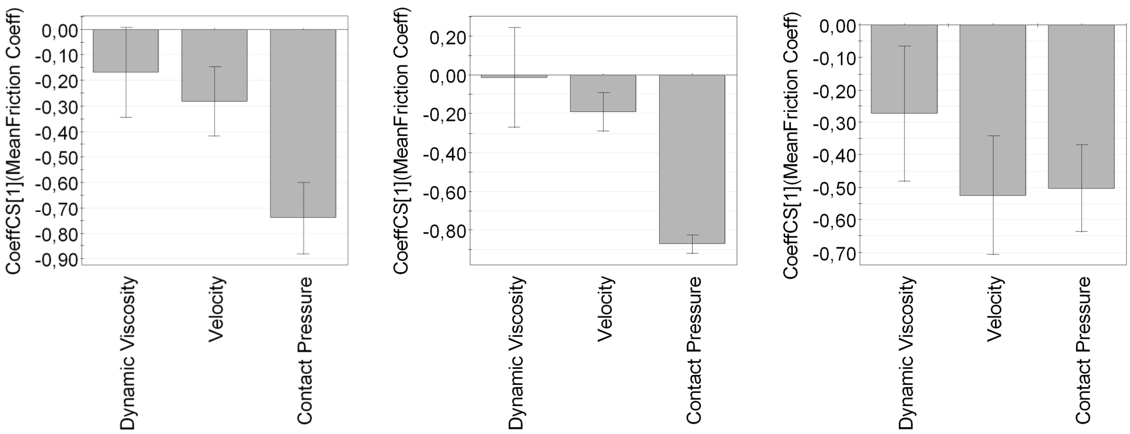

Figure 37. Coefficient plots for describing the correlation between dynamic viscosity, velocity and contact pressure on mean friction coefficient for the three investigated materials. Left: REF. Centre: MATERIAL1. Right: MATERIAL2. The error bars in the coefficient plots represents 95 % confidence interval.

Paper III

Paper III included tribometer testing as well as single cylinder engine testing. The objective of the analysis was to evaluate the most promising materials from the tribometer test in single cylinder engine tests. The experimental approach developed in Paper II was further used in Paper II, five different cylinder liner materials were experimentally evaluated. Of these four materials were thermally sprayed materials, the reference material was grey cast iron. The oil used in the tribometer experiments in paper II was fresh oil. To gain more representative wear levels and to accelerate the wear process in this study, oil extracted from an engine test containing wear particles was used in the tribometer tests. Comparing the wear of results of the reference material from Paper II-III indicated the importance of three body abrasion on the wear of the cylinder liner surface. The introduction of wear particles, increasing the three body abrasion, causes a wear depth of the reference gray cast iron surface which was more than five times higher for the test including wear particles (see Figure 38).

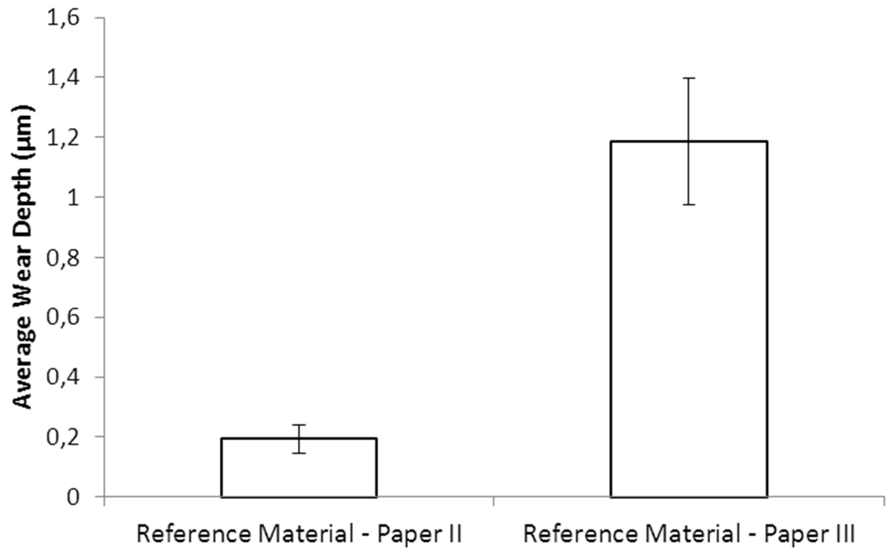

Figure 38. Comparison of wear in tribometer test. In the tribometer test of paper II fresh oil was used, in paper III oil from an engine test, containing wear particles, was used.

The only difference in the experimental setup between paper II and paper III, apart from the oil, was the experimental duration; the experiment duration in paper II was 8 hours (DoE cycle step duration: 30 minutes), the experimental duration of paper III was 13 hours (DoE cycle step duration: 60 minutes). Using longer duration of each DoE cycle more data representing each cycle step can be acquired, this allows for an improved statistical analysis.

The results from the analysis show that two of the thermally sprayed cylinder material exhibited much lower friction compared to all other investigated materials (see Figure 39, left), these thermally sprayed materials and the reference material were further evaluated in single cylinder engine tests. The frictional behavior measured in the tribometer correlated with the surface roughness parameters describing the plateau amplitude of the surface (see Figure 39, right)

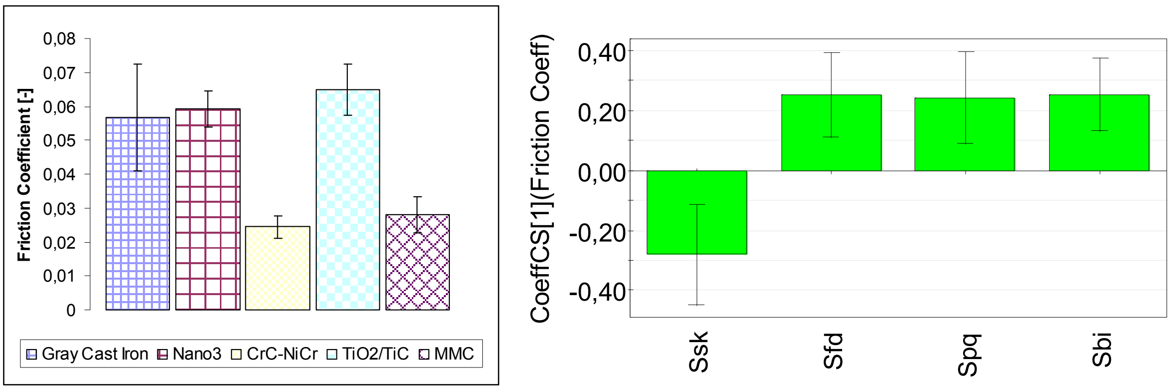

Figure 39. Left: Measurement of frictional coefficient in tribometer test. Right: Coefficient plot of surface roughness parameters and the correlation to friction coefficient. The error bars reflect 95% confidence interval of the calculated surface roughness parameters.

The results from the engine test show the opposite trends in comparison to the results of the tribometer tests. The MMC cylinder liner material which exhibited roughly 50 % lower frictional values in the tribometer tests showed high fuel consumption in the engine test for all but one experimental point (see Figure 40). The increase in friction with increased fuel consumption could not be reconciled with an increase in wear on the MMC thermally sprayed cylinder liner.

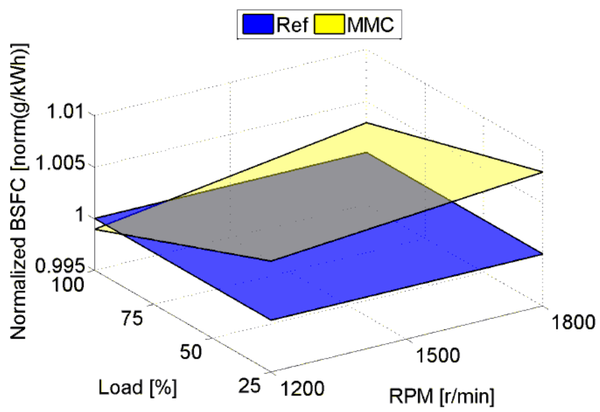

Figure 40. Measurement of Brake Specific Fuel Consumption (BSFC) in single cylinder engine for reference cylinder liner and MMC thermally sprayed cylinder liner. Plot is showing difference between reference surface and MMC thermally sprayed cylinder liner surface.

The cause of the increased frictional losses in the engine test compared to the tribometer tests was thought to be increased viscous losses for the MMC cylinder liner material. This assumption correlates with other research in which the floating liner approach has been used to quantify frictional losses [XCII]. In said study the frictional losses of a smoother cylinder liner surface were compared with the frictional losses of a rougher surface. It was shown that the smooth surface showed lower frictional force at reversal zones whereas the rougher surface showed lower friction force at mid stroke positions (see Figure 41).

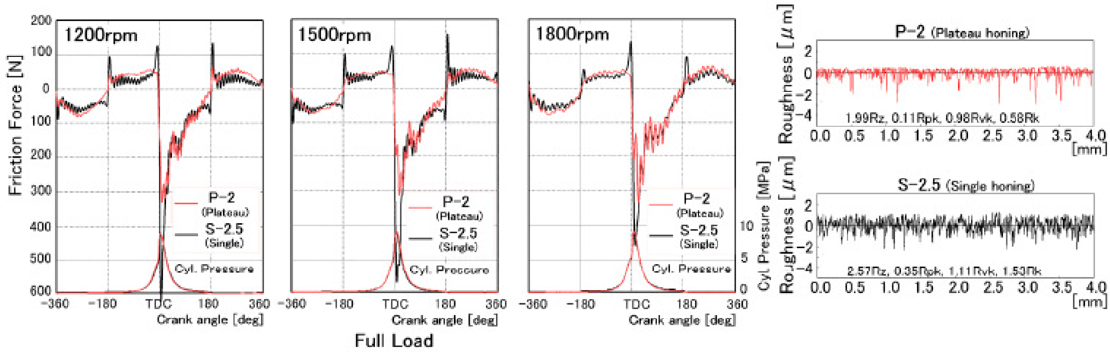

Figure 41. Friction force measurement in engine test (with combustion) by "floating liner" technique [XCII]. Two different surfaces are investigated in this test, P-2; smoother plateau honed surface and S-2.5; rougher single honed surface. The smoother surface exhibits a lower frictional force at the reversal zones whereas the rougher surface shows a lower frictional force at mid stroke. Reprinted with permission from SAE Paper No. 2004-01-0604 © 2004 SAE International.

However, since the sliding speed is low at and around the reversal zones a large value of friction force at these positions has little impact on the total frictional power and the fuel consumption of the engine.

The main result of this paper was not only that higher fuel consumption and frictional losses could be obtained with a smooth cylinder liner surface but also indicated the large contribution of viscous friction losses to the total friction losses and the fuel consumption.

Paper IV

In this paper the frictional loss of the contact between oil control ring and cylinder liner was investigated using the novel Deterministic simulation software. Tribometer experiments were also conducted to compare/validate the frictional output from the tribometer and the frictional output from the simulation software. With the simulation tool it was possible to study alterations of boundary friction and hydrodynamic friction separately for different cylinder liner surfaces. A complete simulation model was built of the oil control ring and the cylinder liner including thermal deformation, temperatures etc. (see section 5.6). The main input in the simulation model was cylinder liner surface measurements. In statistical analysis it is desired to have the experimental input data arranged according to a DoE approach, however, this is not possible when the measurement input data is measurements of real surfaces. Thus the focus was to generate a large amount of input data for improving outcome of the statistical analysis. A total of 18 surface measurements were analysed at five different engine speeds and with two different oil control ring beam widths which gave a total of 180 simulation cases.

The result from the simulation showed that the surface parameters Spq and Smq, describing the amplitude on the plateaus of the cylinder liner surface correlate well with friction. Plateau roughness governs both boundary and hydrodynamic friction. Boundary friction increases with increasing plateau roughness amplitude (Spq), however the hydrodynamic friction decreases with increasing plateau roughness amplitude (Spq). Decreasing surface lay (|Std|) decreases both boundary and hydrodynamic friction (see Figure 42).

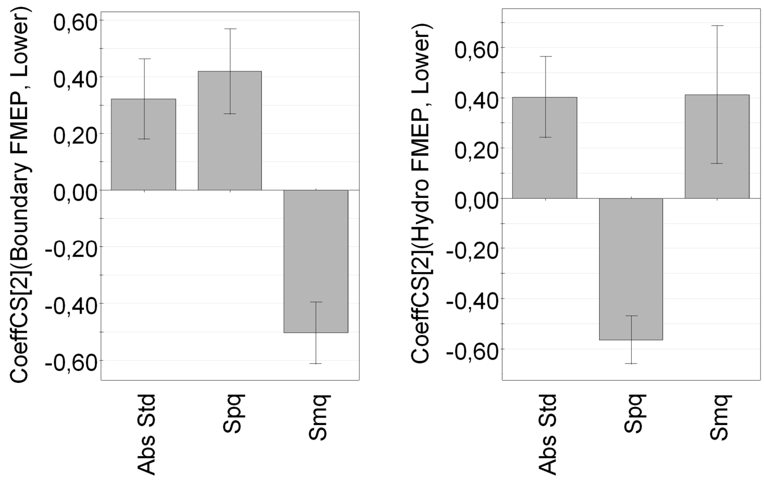

Figure 42. Scaled coefficient showing the correlation between boundary friction (left) and hydrodynamic friction (right) for three surface roughness parameters.

In Paper III higher frictional losses were measured in tribometer experiments and single cylinder engine tests for a thermally sprayed cylinder liner (S6/MMC), this cylinder liner and the plateau honed reference cylinder liner (S1/REF) were also evaluated using the Deterministic Simulation tool. The radial load on the oil control ring is not influenced by cylinder pressure, thus stroke dependent load variations were not included in the simulation. The result from the simulation shows the same trend as the engine test result; at lower engine speeds the frictional losses are lower for the S6, as speed increases the frictional losses increase and surpass S1 at 1200 RPM (see Figure 43).

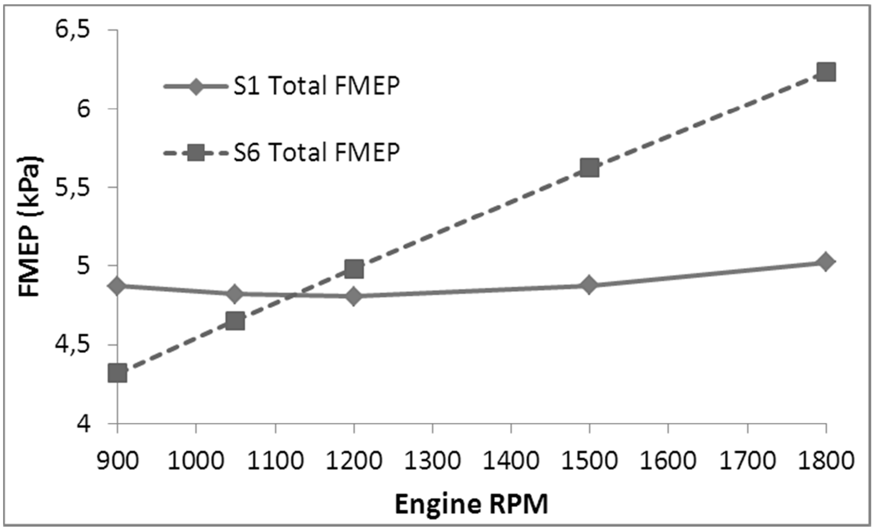

Figure 43. Average FMEP values for the lower of the two oil ring beams with standard beam width for plateau honed surface (S1) and thermally sprayed surface (S6).

The lambda ratio (oil film thickness divided by plateau surface roughness amplitude) is generally higher for S6, this gives a partial explanation as to why no wear was measured for S6 in Paper III. This together with the fact that S6 showed a higher wear resistance in tribometer tests gives an explanation for why no wear was measured for S6 in Paper III. Since the plateau surface roughness amplitude of S6 is much smaller than S1 the hydrodynamic friction losses are higher for S6 compared to S1. The increase in viscous friction with a smooth surface provides an explanation for the increased fuel consumption for the thermally sprayed surface measured in the engine test in Paper III (see Figure 40).

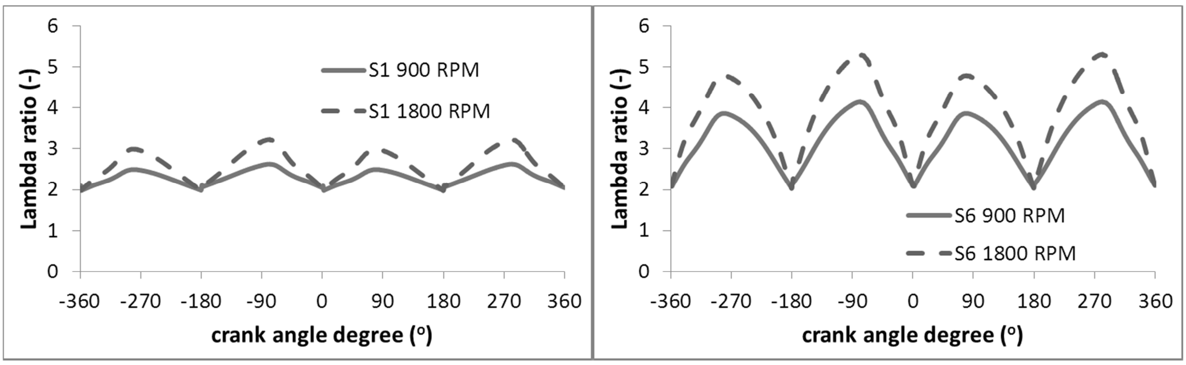

Figure 44. Lambda ratio of surface S1 and surface S6 plotted vs. crank angle degree.

A DoE tribometer setup was constructed in order to replicate the tribological contact between oil control ring and cylinder liner. Two surfaces were evaluated in tribometer experiments, a plateau honed reference surface and a smoother surface. Additionally to the tribometer experiments a simulation model was constructed, replicating the running conditions of the tribometer DoE setup. Although there are known physical differences between the tribometer and the simulation model an agreement between the average value of friction coefficient in the tribometer and the average value of friction coefficient for the plateau honed reference surface was found. The agreement between the experimental and simulation output for the smooth surface was not as good as for the plateau honed surface. It was suggested to repeat this experiment using different surfaces, e.g. comparing plateau honed cylinder liners with varying plateau roughness amplitude.

Paper V

This paper aimed at decreasing hydrodynamic frictional losses by analysing the frictional effects of a novel type of surface texturing thus giving answers to the second research question. Based on the result from all previous papers it was evident that frictional losses were governed by the plateau surface roughness, however, result from other researches showed that it was possible to decrease hydrodynamic friction losses by using surface texturing. The frictional effects of textures were analysed in tribometer tests, the texture design was chosen with the following prerequisites:

- Large sized features
- Closed circular- or elliptical textures
- A significant texture fraction of the total area
- A surface lay of the textures which was adjacent to the direction of motion

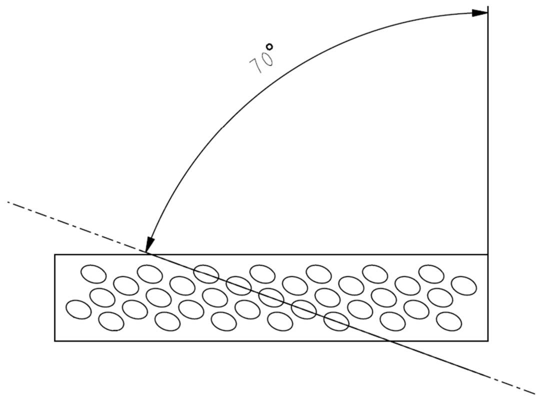

Figure 45. Overview of texturing on cylinder liner samples.

The textures were manufactured by milling, an overview of the texturing on the cylinder liner sample can be seen in Figure 45. Two different depths (20 µm and 100 µm) of texturing were tested. Measurement of the textured samples showed that only the depth varied, besides texture depth other texture parameters such as the perimeter, the lay of textures, the area of textures etc. were constant for the two different manufactured samples.

The frictional measurement showed that the textured samples exhibited much lower frictional values compared to the reference surface (see Figure 46). It was also observed that the amount of metal-to-metal contact, as quantified by the resistive coefficient, increased for the textured surfaces. By comparing the measured frictional values from the experiments with high and low values of input parameters it was seen that the majority of both the reference surface and the textured surfaces mainly operated in the hydrodynamic lubrication regime.

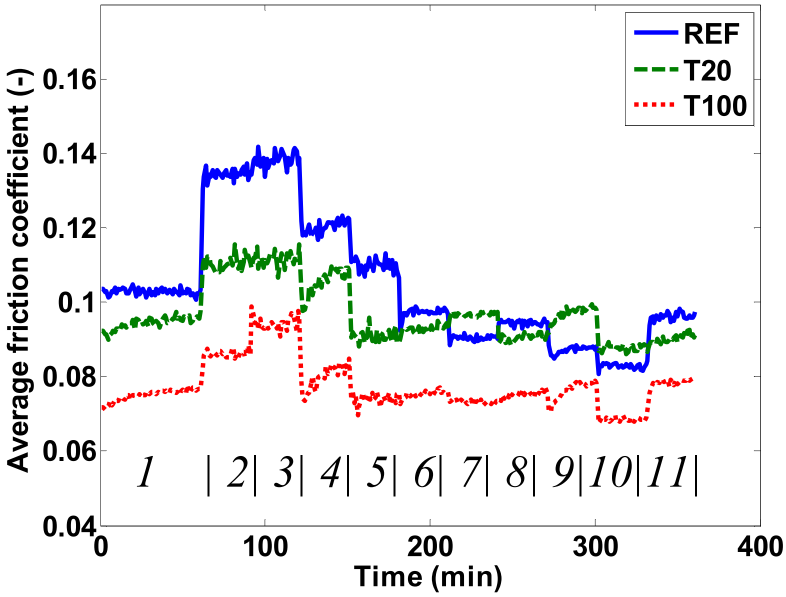

Figure 46. Measured friction coefficient, average of experiments for each surface type. The numbers above the x-axis (italic) indicates the duration of each experimental cycle step.

Based on the experimental results a new hypothesis of how to decrease hydrodynamic friction with surface texturing was formed. The assumption was based on the equation describing shear force for two parallel planes fully separated by a Newtonian fluid (Eqn. (6)).

$$F_T = \frac{\eta v A}{h} = \eta S A \tag{6}$$

In consideration of the terms of this equation it is here proposed that the introduction of textures does not alter the dynamic viscosity or the area of the contacting surface, however, the average oil film thickness is significantly increased considering that it could be assumed that the oil film thickness is approximately the same as the texturing depth (see Figure 47). Due to the increased metal-to-metal contact of surfaces it can be assumed that the oil film thickness in the contact between plateau of cylinder liner (untextured part) and the piston ring is lower for a textured cylinder liner surface compared to a typical plateau honed surface (see Figure 47). This means that the total boundary friction will be higher for the textured surface. However, as can be concluded from the results of the tribometer experiments, the decrease in hydrodynamic friction for the textures with 100 µm depth is greater than the increase in boundary friction, thus the total friction in the hydrodynamic lubrication regime will be significantly lower for the texturing with 100 µm depth compared to the reference surface.

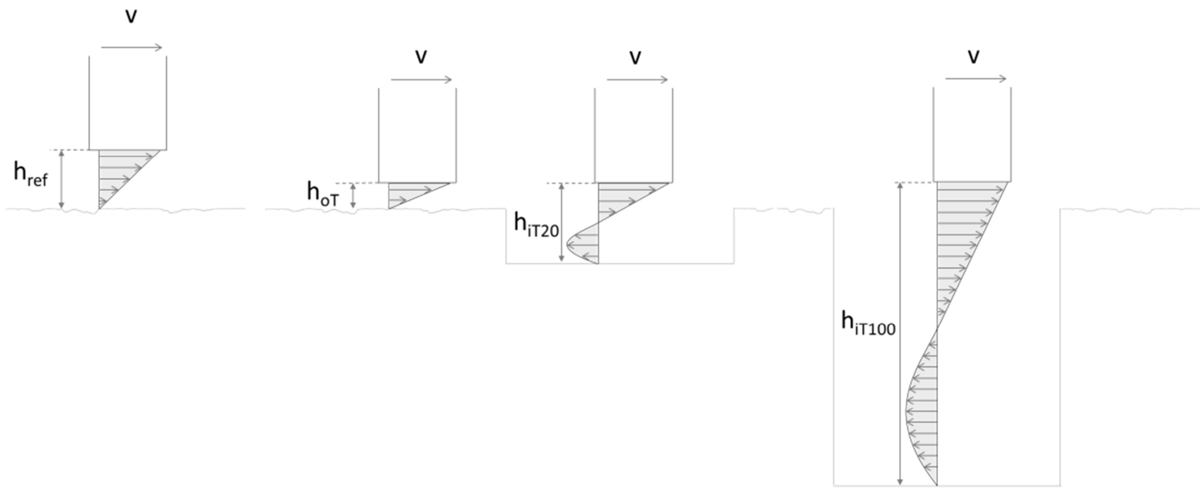

Figure 47. Schematic overview of the oil film build-up for the reference surface (left in image) and a piston ring passage over the plateau of a textured surface, the piston ring passage over a 20 µm deep texture and the piston ring passage over a 100 µm deep texture (right in image).

Based on the hypothesis that hydrodynamic frictional losses can be reduced with increased oil film thickness within textures a recommendation of texturing design on the cylinder liner surface was proposed. This design included that no texturing should be placed in the vicinity of the reversal zones because of the hydrodynamic friction losses is small at these positions. The design also included an increase of the area density of textures with increasing piston speed. By using this design, experimentally verified in Paper V, it is believed that the hydrodynamic friction losses in the full scale PCU can be significantly reduced. Based on this texture design and the result of Paper V a patent application [XCIX] was filed.

9. Discussion of Results

9.1. Measurement of the cylinder liner surface morphology

The surface roughness analysis in this thesis was constrained to the roughness wavelength output of Gaussian and Robust Gaussian filtering (additional to this is geometrical analysis of the textures in Paper 5). By using alternative filtering methods it is possible that the values of the surface rough ness parameters would have been somewhat different, however, it is not considered that this would have any influence on the conclusions of this thesis. Also, in this thesis work it could have been possible to use different types of surface roughness measurement equipment; however, when measuring the outcome of machining such as honing the equipment used has proved to be appropriate. The effectiveness of the surface roughness measurement methodology used is illustrated with the high correlation between surface roughness parameters and friction in the tribometer experiments of Paper III.

9.2. Optimal surface of the cylinder liner, in reference to tribological simulation and manufacturing

In the simulation model of Deterministic Simulation the surface of the oil control ring was modelled as nominally flat (both in terms of roughness and ring face profile). In the average Reynolds equation, used in Piston Simulation, a nominally flat surface is not capable of generating hydrodynamic pressure when the oil supply to the ring is smaller than the oil film thickness. In Deterministic Simulation it is possible to obtain a hydrodynamic pressure for a nominally flat surface because Deterministic Simulation is capable of calculating the hydrodynamic pressure generated at inter-asperity level [C]. In this respect Deterministic Simulation offers a much more detailed calculation approach in analysing frictional behaviour compared to Piston Simulation.

In considering the different tribological contact conditions between the piston ring and the cylinder liner that are present at different parts of the stroke it would be surprising if only one type of surface morphology is optimal independently of the tribological contact conditions. In the simulations carried out in this thesis it was not possible to use different surfaces for different parts of the stroke. It would be practically difficult both to machine different surfaces on different parts of the stroke and also to include several surfaces in one simulation model. However, until these steps have been accomplished it is challenging to see how frictional losses in the contact between piston ring and the cylinder liner can be systematically optimized.

The simulation results of Paper IV indicated that friction (both boundary and hydrodynamic) could be decreased with a decrease in surface lay (lay oriented perpendicular to the direction of motion). The result of decreasing hydrodynamic friction with decreased surface lay has not to date been experimentally verified. Experiments, which could be considered to be partially related to the frictional issue discussed in this thesis indicate the opposite; an increase in surface/texture lay (grooves which are parallel to the sliding direction) decreases hydrodynamic friction at a high shear ratio [CVIII]. Since the generation of hydrodynamic pressure decreases with increasing angle of the surface lay [CI] it would not be practical to use a too high angle of the surface lay because this would most likely increase the mechanical contact thus increasing the amount of wear. In combining the statements above; It is here considered that an optimal surface lay would be depending on the surface feature; textures should be manufactured having a high angle of the surface lay and the plateau surface should be manufactured having a low angle of surface lay.

There are difficulties in simulating the frictional behaviour of the novel textures described in Paper V. Using the traditional simulation with average Reynolds equation on the textures analysed in Paper V is not possible since Reynolds equation is not valid when the depth of the texture is significantly larger than the oil film thickness [CII] (as is the case for the analysed texture with 100 μm depth). However, modifications of the Reynolds equation for analysing the frictional effects of textures are currently being developed [CIII]. In developing a calculation method for textures it is important to understand the physical aspects of the oil in relation to texture design. It is thus important to experimentally analyse the behaviour of the oil film in and around the point of contact, similar to studies conducted by Dellis et al. [CIV]. In summary, it is here considered important to increase the knowledge of the frictional behaviour of textures for sliding contacts operating in the hydrodynamic lubrication regime. A step towards this would be to analyse the frictional behaviour of oil for textured surfaces in an analysis which is similar to the one conducted by Dellis et al. Results from such a study could be used to develop an alternative calculation approach for accurate representation of the frictional behaviour of textures.

9.3. Significant Contribution of Different Lubrication Regimes on Fuel Consumption

The typical tribometer test uses a fixed set of experimental input parameters during an experiment. This approach is sufficient for analysing e.g. only the frictional characteristics of the top ring at CTDC. To optimize the friction between piston ring and cylinder liner it is important to analyse the frictional contribution of all lubrication regimes. In a tribological system in which the normal load is constant, such as for the oil control ring/cylinder liner contact, it is very likely that hydrodynamic friction will be the most significant contributor to the total friction power loss. This is illustrated in Figure 48 which shows a schematic overview of the contribution of different lubrication regimes to the frictional power. It is important to recognize that friction power is not the same as frictional coefficient, it is the frictional force multiplied by the instantaneous sliding speed, and thus the hydrodynamic frictional power losses have a significant impact on the total frictional power losses.

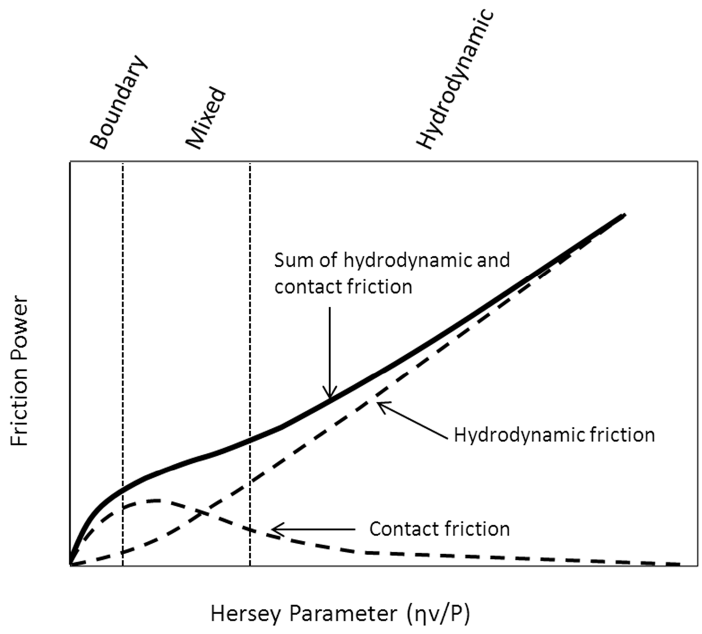

Figure 48. Schematic illustration of frictional power loss with the contribution of contact friction and hydrodynamic friction.

9.4. Additional research questions

A number of relevant additional research questions can be raised and discussed based on the findings of this thesis:

Does running in decrease the frictional losses? The answer to this question is yes only if the contribution of mechanical frictional loss decreases more than the contribution of hydrodynamic friction losses increases.

Could the hydrodynamic friction losses increase during running in? The answer to this question is yes if the plateau surface roughness of the cylinder liner is reduced during running in. By very close inspection of Figure 31 and by comparing T=0 h and T=30 h it can be seen that friction force increases at the mid stroke positions. This is true except for the mid stroke position after CTDC (90 crank angle degrees). The reason why the viscous friction does not increase at this position is most likely due to the increase[‡] in temperature and pressure resulting from combustion resulting in mixed lubrication. The increase/difference in hydrodynamic friction is small and it could be possible that the measured increase in hydrodynamic friction is smaller than the measurement error. However, if the plateau surface at the majority of the stroke length is smoothened during running in it is here considered possible that the hydrodynamic friction losses could increase during running in. This assumption is in line with the results of Paper IV.

[‡] It is worth keeping in mind that all frictional measurements in this paper were conducted at full load. If a lower load level would have been used more hydrodynamic/less boundary lubrication conditions would have been expected.

Why do the frictional losses normally decrease during the running in process? The contribution of contact friction is highest at the reversal zones. Since the plateau amplitude of the surface at the reversal zones will experience more wear (more smoothening) than the plateau amplitude at mid stroke, it is highly likely that the frictional contribution of contact friction will decrease more than the assumed contribution of hydrodynamic friction.

In reference to statements above; If textures are used in such a way that the hydrodynamic friction decrease is governed by textures located around the centre position of the stroke it could be possible to a gain larger frictional decrease during running without the increase of the hydrodynamic friction losses during running in.

9.5. Additional Consequences of Applying Texturing Elements on the Cylinder Liner Surface

In the design of a texturing on the cylinder liner surface it is important to keep in mind that the texturing should survive the wear procedure, the initial running in wear rate can be rather high. Due to the large texturing depth in the proposed texturing design it is possible to sustain the decrease in hydrodynamic friction throughout the lifespan of the engine. Also, as was shown in Paper V the wear particles generated in the contact between piston ring and cylinder liner were trapped in the textures. This reduced the three body abrasion which meant the plateaus between the textures showed little signs of wear. However, adding textures in a tribological contact could increase the wear [CV] thus it could be important to increase the wear resistance of the cylinder liner (and perhaps also the wear resistance of the piston rings) by using different materials. Thermally sprayed cylinder liners have exhibited great wear resistance (Paper II and III), thus it might be fruitful to use thermally sprayed cylinder liners in combination with textured surface, as the results of Paper II and Paper III indicate.

The textures analysed in this thesis work are of a special geometry, consequently much larger than the majority of other textures (see Paper V, introduction section). The textures introduce an additional volume in the cylinder liner surface. Also the axial length of the textures is larger than the axial Hertzian contact length of any of the piston rings. Practically this could result in increased blow by and increased oil consumption. However, previous research has shown that the surface morphology of the cylinder liner has little effect on the amount of blow by [LXXIV], but in this work different honed surfaces were analysed and not larger scale textured features. An potential increase in blow by and oil consumption could be reduced by redesign of the piston ring package. An initial solution procedure might include gas tight piston rings [CVI] and letting the volume of the textures act as the ring gap ensuring a positive gas flow towards the crankcase.

10. Suggestions for future work

The following suggestions for future work based on the results from this thesis work are as follows:

10.1. Optimize manufacturing of textures

As an example, by using a novel type of a grinding texturing method [CVII] it should be possible to produce a texturing on the mid stroke of the cylinder liner. To date this method has been applied on flat surfaces and the outer diameter of cylinders, to apply this method on the inner diameter of a cylinder requires development of the machining procedure However, it should not be impossible to accomplish this. There are two main benefits of applying the novel type of texturing method. Firstly the novel texturing method does not create any burrs on the surface, this means that no additional machining step is required (as often required in e.g. laser honing). Secondly the machining method is extremely fast, thus providing a solution that could be cost effective in large scale production.

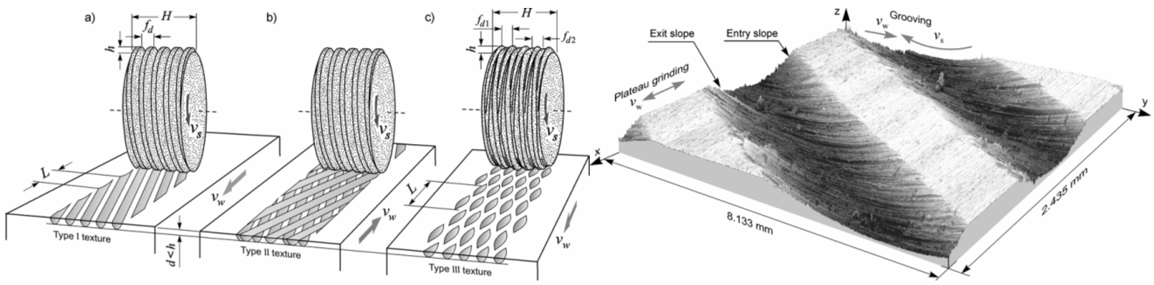

Figure 49. Surface patterning produced by a special grinding process. Left: Overview of different cutting geometries and the resulting surface pattern. Right: Detailed view of the surface after machining of surface pattern[CVII].

Naturally alternative machining methods like laser and EBM machining or etching by ECM or CM exist, and future tests needs to be carried out to judge the performance of these methods.

10.2. Optimize/develop/invent tribometer test setup to analyse hydrodynamic friction in the power cylinder unit

To analyse the frictional aspects of hydrodynamic lubrication it is here proposed to analyse the hydrodynamic lubrication separately (without also studying the contribution of boundary friction as measured in a reciprocating tribometer). Hydrodynamic effects can be analysed separately by keeping a constant separation between mating components, similar to the analysis conducted by Chen et al. [CVIII]. In this approach load is not included in the experiment thus frictional coefficient is not quantified. Instead the hydrodynamic drag between the two surfaces is measured. By keeping a small separation between surfaces it should be possible to quantify the hydrodynamic drag of various texture geometries.

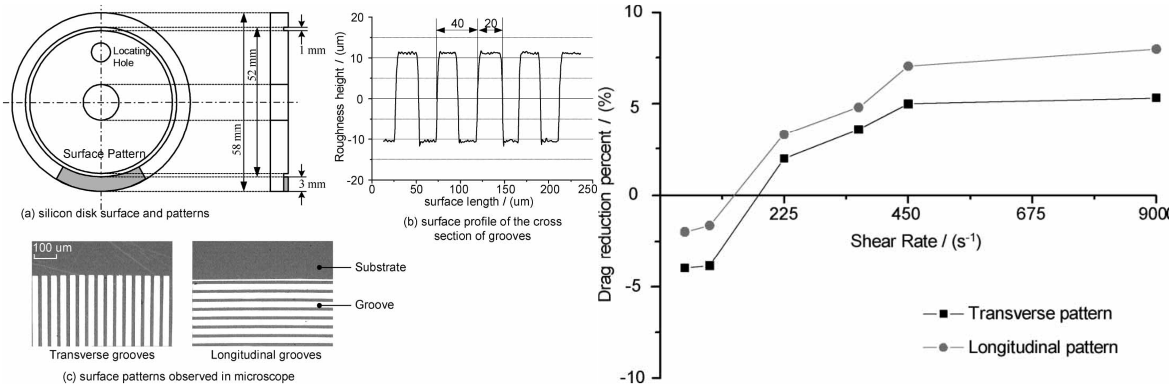

Figure 50. Overview of a novel type of tribometer test setup, this setup or similar could be used to analyse hydrodynamic friction of PCU components [CVIII].

10.3. Analysis of the effects of blow by and oil consumption for novel textured surfaces

As mentioned in section 9.3 the introduction of textures on the cylinder liner surface creates an additional volume in the surface which could generate an increase in blow by and oil consumption. Firstly it should be investigated if the proposed texturing design has a negative effect on blow by and oil consumption. If this proves to be the case it could be possible to solve this issue with a redesign of the piston ring package, with minimised gas leakage in the ring gap (gas tight piston ring). This solution could be used to compensate the possible effects of increased blow by with the novel texture design.

10.4. Optimization of texture geometry; with experimental analysis and by calculation

In Paper V a tentative example is given, this example shows how textures density is varied along the stroke length of the cylinder liner. For an accurate optimization of the texture design it could be possible to use/design a calculation approach for the issue. Since the average Reynolds equation is not valid when the oil film thickness is much smaller than the texturing depth perhaps it could be possible to utilize a meshfree method [CIX] for calculating and determining the optimal texture geometry for different positions of the stroke length.

11. Conclusions

Within this thesis work the following research questions were posed:

1. How should a pilot tribometer test be constructed in order to replicate the frictional and wear behaviour of the engine at boundary, mixed and hydrodynamic lubrication regime?
2. What part of the surface morphology of the cylinder liner surface affects the frictional behaviour of the different lubrication regimes?

In relation to the research questions the following apply:

1. A tribometer test should be constructed in such a way that frictional effects in all lubrication regimes can be quantified. Currently, there is a strong focus, both in academia and industry, on analysing frictional effects in the boundary and mixed lubrication regimes. The results from this work have shown that the friction losses in the hydrodynamic lubrication regime have a significant contribution to the total frictional losses, thus future development of tribometers should be aimed at replicating hydrodynamic friction losses.
2. For the boundary and mixed lubrication regime decreasing the plateau surface roughness decreases the friction in said lubrication regimes. For the hydrodynamic lubrication regime a decrease in plateau surface roughness increases friction. Using a novel texture design it is possible to decrease friction in the hydrodynamic lubrication regime.

The overall conclusions in this thesis can be summarized as follows[§]:

- [1]The Piston Simulation software shows how oil film thickness decreases with decreasing cylinder liner surface roughness. This is also implied from engine tests where oil consumption decreases with cylinder liner surface roughness.
- [1]PVD-coated top ring has proved to confer exceptional wear resistance, in an accelerated tribometer test almost no signs of wear were detected on the piston ring or the counterpart, the cylinder liner sample.
- [2]With the developed DoE tribometer test approach it was possible to measure and evaluate the frictional effects of sliding speed, dynamic viscosity and contact pressure in one tribometer experiment. Using the developed approach determined that:
  - [2]A surface has to be able to generate the necessary conditions for build-up of the oil film, if the surface does not accomplish this an increase in sliding velocity or a decrease in the oil viscosity has no effect on decreasing friction.
  - [3]The difference in friction coefficient correlates with surface roughness parameters describing the plateau part of the cylinder liner surface roughness, as plateau roughness decreases, friction decreases.
- [3]The frictional measurements of the single cylinder engine test presented a contradiction in comparison to the results of the tribometer, a higher frictional loss

[§] Numbers in superscript indicate the conclusions related to each specific paper.

was obtained in the single cylinder test using a cylinder liner with smooth plateaus. The cause of increased friction was explained as an increase in viscous losses.

- [4]Tribological simulation confirmed the cause of increased hydrodynamic friction using a cylinder liner with a smooth surface. The simulation result also showed that both boundary and hydrodynamic friction could be reduced by reducing the angle of surface lay.
- [5]A novel texturing design was developed and experimentally analysed using a tribometer test.
    - It was shown that both textured surfaces and the reference plateau surface operated in the hydrodynamic lubrication regime.
    - An increase of wear was not detected for the textured surface, on the contrary textured surfaces exhibited smaller amounts of abrasive scratches, this was due to entrapment of wear particles within the textures.
    - Significant reduction in hydrodynamic friction was obtained with the novel texturing design.
- [5]Based on the result of the textured surfaces a design suggestion was made, detailing how a texture pattern could be designed to decrease hydrodynamic friction losses. In essence this suggestion includes an untextured area in the vicinity of the reversal zones (where the contribution of hydrodynamic friction on total friction is small), and an increasing texture area density with increasing piston speed. A patent application was filed based on the experimental findings.

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