Application of Surface Treatments to Improve Fuel Efficiency of Internal Combustion Engines

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APPLICATION OF SURFACE TREATMENTS TO IMPROVE FUEL EFFICIENCY OF INTERNAL COMBUSTION ENGINES

A Thesis

Submitted to the Graduate Faculty of the Louisiana State University and Agricultural and Mechanical College in partial fulfillment of the requirements for the degree of Master of Science

in

The Mechanical Engineering Department

by

Amirabbas Akbarzadeh

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Abstract

To improve the tribological performance of contacting surfaces, different surface modification methods can be employed. Surface texturing and surface coating are examples of viable techniques developed for this purpose. Surface texturing involves creating micro patterns on the contacting surfaces while surface coating requires depositing a thin layer of a suitable material on the surface(s) to improve the component’s friction and wear characteristics. The performance of textured surfaces is affected by the geometric characteristic of textures. When dealing with surface coating parameters, the parameters of interests are the type of coating materials and their thicknesses.

The current study aims to experimentally study the friction variation as a result of surface texturing and surface coating on the piston rings by using a custom made piston cylinder machine. Of particular interest are their running-in behavior and the associated transient friction characteristics. A new type of texturing is also proposed for the application in piston oil control rings to reduce engine friction. Experimental studies are conducted to compare their performance of grooves with three structural angles (0, 30 and 60 degrees) and two different depths (5 and 10 microns). The optimum texture with grooves parallel to the sliding direction and depth of 5 microns was found to yield a 10% reduction in the friction force. Combination of the texturing and coating showed 12.5% improvement in frictional behavior when compared to cases when only one of them was applied.
Chapter 1 Introduction

Merriam Webster dictionary defines “efficiency” as “effective operation as measured by a comparison of production with cost (as in energy, time, and money)” and when it comes to internal combustion engines —which are the main type of powertrain in vehicles in today’s transportation industry,— the fuel efficiency plays a key role in popularity of a specific type of vehicle.

According to the National Academies Press entitled “Assessment of Fuel Economy Technologies for Light-duty Vehicles”, as early as 1918, Charles Kettering —the automotive pioneer in General Motors Company— predicted the demise of the internal combustion engine within 5 years because of its wasteful use of fuel energy [1]. In the 1920s through the 1950s peak efficiencies increased from 10 percent to as much as 40 percent, with improvements in fuels, combustion system design, friction reduction, and more precise manufacturing processes. As a result, engines became more powerful, and vehicles became heavier, bigger, and faster [1].

In the wake of the 1973 oil crisis, the issue of energy security became center of attentions, and the U.S. Department of Transportation (DOT) set appropriate standards for light trucks, and forced automobile manufacturers to increase the average fuel economy of passenger cars. Figure 1-1 illustrates the extent to which the developed countries around the world are concerned with energy efficiency and its associated financial and environmental impacts [2]. Most of these countries have specified goals and deadlines for their transportation manufacturers. Figure 1-2 present the United States’ defined MPG goal for 2025 [3].
Figure 1-1: MPG targets for different developed countries (International Council on Clean Transportation) [2]

Figure 1-2: Targets of US fuel consumption rate (National Highway Traffic and Safety Administration) [3]
In the United States, the Department of Energy sponsors annual workshops on industrial research needs for reducing friction and wear, focusing on industrial research as primary steps to reduce friction and respectfully wear in transportation to reach a desired fuel efficiency target [3]. Estimations show that by reducing the friction in engines and other moving components, such as drive trains, the US economy will save as much as 120 million dollars per year [4].

In the following sections, each automotive component is discussed in an appropriate detail. The frictional loss in an internal combustion engine is the most important factor in determining the fuel economy and performance of the vehicle utilizing the power of the engine. Approximately 50% of the friction losses in an internal combustion engine are due to the piston/cylinder system, of which 70–80% comes from the piston rings [5].

Improvements to the tribological performance of engine components can reduce fuel and oil consumption and exhaust emission, increase engine power outputs, improve durability, and reduce vehicle maintenance. Improvement in the tribological performance can be achieved in different ways such as enhancing the tribological properties of the materials used for the mechanical parts, coating the surfaces, and by developing lubricants capable of improving tribological behavior [5].

![Diagram](image)

Figure 1-3: On the distribution of fuel energy in engine from [6]
1.1 Engine tribology

The reciprocating internal combustion engine, shown in Figure 1-4, is a major source of friction loss in the motor vehicles as well as in many other transportation devices [7]. Due to its relatively high performance and reliability, internal combustion engines are the most popular one. Nevertheless, there are also some drawbacks that cannot be neglected. In these type of motors, mechanical and thermal efficiencies are not very high since a major portion of energy is dissipated due to friction and heat loss.

![Engine components diagram]

Figure 1-4: Main engine components in an internal combustion engine [7].

1.2 Importance of Engine Tribology

With a huge number of internal combustion engines in use, even a small amount of improvement in fuel efficiency or emission level can make an appreciable difference in oil consumption and reduction of detrimental environmental impact. The combustion of gasoline is the driving force for distributing energy in the powertrain system. It is thus worthwhile to examine how the energy derived from combustion of the fuel is consumed in an engine. With this in mind,
Anderson [2] analyzed the distribution of fuel energy in a medium size passenger car during an urban cycle (Figure 1-5). According to his paper, only 12% of the available energy in the fuel is available to drive the wheels, with some 15% being dissipated as mechanical and frictional losses. Based on the fuel consumption data presented by Anderson [2], a 10% reduction in mechanical losses would lead to 1.5% reduction in fuel consumption. The worldwide economic implications of this seemingly small reduction are startling in both resource and financial terms and the prospect for significant improvement in efficiency by modest reductions in friction [5, 8].

Examination of the energy consumption within an engine, Figure 1-5, reveals that frictional loss is the main portion (48%), and followed by the acceleration resistance (35%) and the cruising resistance (17%). Of the entire friction loss, the engine friction loss accounts for 41% and the transmission and gears are approximately 7%. Concerning the frictional losses alone, the sliding of the piston rings and piston skirt against the cylinder wall are undoubtedly responsible for the largest contribution in an engine. Frictional losses arising from the rotating engine bearings (notably the crankshaft and camshaft journal bearings) are the next most significant factor, followed by the valve train (principally at the cam and follower interface), and the auxiliaries such as the oil pump, water pump, and alternator.

The relative proportions of these losses, and their total, vary with engine type, component design, operating conditions, choice of engine lubricant, and the service history of the vehicle (i.e., worn condition of the components) [5, 9].

The relative proportions of these losses, and their total, vary with the engine type, component design, operating conditions, choice of engine lubricant and the service history of the vehicle, i.e. worn condition of the components. It should be mentioned that auxiliaries should not be overlooked, since they can account for 20% or more of the mechanical friction losses.
Figure 1-5: a) Distribution of energy consumption in a light-duty vehicle, b) Distribution of friction lost in engine and transmission [5].

1.4 Lubrication Regimes in the Engine

According to Tung and McMillan [10], the key operating tribological parameter in a powertrain system is the lubricant film thickness separating the interacting component surfaces. Figure 1-6 shows a plot of the relationship between the coefficient of friction and the oil film thickness ratio or Sommerfeld number. The diagrams at the top of the figure provide a visual...
example of the lubrication of two surfaces that are in relative motion to each other and that are separated by an oil film [10]. In the region on the top left, labeled boundary lubrication, there is considerable contact between surfaces, whereas the region on the far right, the fluid-film (hydrodynamic) region, designates the region where the surfaces are completely separated with a layer of lubricant. Between these two extremes there is partial lubrication with intermittent contact at the asperity levels. The curved line below the lubrication regimes indicates the relationship between the friction coefficient and the Sommerfeld number. Different automotive components rely upon the modes of lubrication to achieve acceptable performance, and each may experience more than one regime of lubrication during a single cycle. Figure 1-6 shows the so-called Strubeck curve which demonstrates the relationship between the coefficient of friction and the bearing parameter $\eta N/P_{avrm}$ (where $\eta$ is dynamic viscosity of the lubricant, $N$ is rotation speed, and $P_{avrm}$ is average bearing pressure [10]. The ratio of the oil film thickness $h$ to the surface roughness $R_a$ determines the type of the lubrication regime.

![Lubrication regimes diagram](www.subtech.com)

Figure 1-6: The three lubrication regimes are clearly distinguished in the Strubeck curve [11]
1.4.1 Boundary lubrication ($h < R_a$)

A considerable contact between micro asperities of the surfaces is the main feature of this lubrication regime. The coefficient of friction is expected to be the highest in this regime, leading to the greatest energy loss. With increasing load, non-uniform distribution of the bearing load (localized pressure peaks) can occur with severe wear and the possibility of bearing seizure. Boundary lubrication typically occurs at low speeds and also at engine start up and shutdown or under high loads [11]. The presence of Nano-particle additives in the lubricant can improve lubrication and prevent seizure conditions when direct metal-to-metal contact occurs between the components parts operating in the boundary lubrication regime [12].

1.4.2 Mixed lubrication ($h \sim R_a$)

The intermediate regime between the boundary lubrication and hydrodynamic friction is mixed lubrication where an intermittent contact between the friction surfaces at few high surface points (micro asperities) occurs [11]. This regime deals with the condition when the speed is low, the load is high or the temperature is large to reduce lubricant viscosity. The tallest asperities of the bounding surfaces will protrude through the film and occasionally come in contact.

1.4.3 Hydrodynamic lubrication ($h > R_a$)

In hydrodynamic lubrication there is no contact between the surfaces. Typically, high rotational speeds at relatively low bearing loads results in hydrodynamic friction. Hydrodynamic lubrication is characterized by stable lubricant film (oil film) between the rubbing surfaces. Due to the generation of hydrodynamic pressure the bearing, and the shaft surface stay apart [11]. Bearings working under the conditions of hydrodynamic lubrication are called hydrodynamic bearings [5]. Generally, journal and thrust bearings are designed to operate in the hydrodynamic lubrication regime in which bearing surfaces are separated by a lubricant film. Actual metal-to-
metal contact is expected to take place only at low speeds and high loads and with low-viscosity lubricants. In contrast, valve train, piston ring assembly and transmission clutch sliding generally take place under mixed or boundary lubrication conditions where surface contact occurs, and chemical films or reaction products may be an important means of surface protection. In addition, the importance of different lubrication regimes for each engine component may change with the surface roughness at the interface, wear of critical interacting surfaces, and lubricant degradation. In Chapter 2 an experimental is reported to determine the lubrication regime between piston rings and cylinder wall.

1.5 Piston Assembly

Tung and McMillan describe piston ring as the heart of the piston assembly in a reciprocating internal combustion engine [10], since it plays a critical role in transforming the energy generated by combustion of the fuel and air mixture into useful kinetic energy. Piston assembly consists of ring packs, which is a series of metallic rings, with primary role of effective gas sealing [10].

The piston rings essentially form a labyrinth seal by closely conforming to their grooves in the piston and to the cylinder wall. A secondary role of the piston ring is to transfer heat from the piston into the cylinder wall, and thence into the coolant in order to limit the amount of oil that is transported from the crankcase to the combustion chamber. This flow path is probably the largest contributor to engine oil consumption and leads to an increase in harmful exhaust emissions as the oil mixes and reacts with the other contents of the combustion chamber.
The objectives to extend engine service intervals and minimize harmful exhaust emissions for meeting more stringent legislative requirements are to reduce the oil consumption to a low level compared to their predecessors of 20 years ago. The left side of Figure 1-7 is a schematic representation of a piston assembly from a modern automotive engine. From a tribological perspective, the main piston’s features are the grooves that hold the piston rings, and the region of the piston below the ring pack (the piston skirt), which transmits the transverse loads on the piston to the cylinder wall. The top two piston rings are referred to as the compression rings. As shown in the Figure 1-7:

![Piston assembly and piston ring function from an internal combustion engine.](image)

Tung and McMillian state that entire ring face engages the cylinder wall as a result of firing pressure. The top compression ring usually has a barrel-faced profile recent common coating for first pressure rings is flame-sprayed molybdenum. The second compression ring, which is sometimes referred to as the scraper ring, is designed to assist in oil flow in addition to providing a gas seal [10].
In Figure 1-7 the two bottom rings in the ring assembly are the oil control rings, which have two running faces, and a spring is attached to one of them in order to enhance radial load. The major role of these rings is to limit the amount of oil transported from the crankcase to the combustion chamber, with no gas sealing ability.

Compression rings are internally stepped in order to improve oil control. Smaller heights would reduce contact surface and friction achieved by double edge oil control rings. Large variations of the load, speed, temperature and the lubricant render the analysis of piston ring complicated. In terms of the lubrication regime described above, the piston ring interface with the cylinder wall may experience boundary, mixed, and full fluid film lubrication during each cycle.

A mathematical model for the cyclic variation of minimum film thickness between a piston ring and cylinder wall, where zero degrees of crank angle is at the top dead center, has been presented by Ruddy et al. [13]. The predictions made using their model assumes a plentiful supply of lubricant for the top compression ring of a four-stroke gasoline engine. In their experiment, the combined surface roughness of the piston ring and cylinder wall was 0.1 µm. The solution exhibits a characteristic curve with a small film thickness around the dead center positions where the sliding velocity and lubricant entrainment velocity are low. In their analysis, a large film thickness was observed at the mid-stroke positions where the sliding and entrainment velocities are large.

During the engine cycle the piston itself exhibits a complex secondary motion with transverse movement toward the cylinder wall and tilting about the main piston pin axis [14]. This movement results in the formation of fluid film or mixed lubrication between the piston skirt and the cylinder wall. Modern, light-weight pistons have skirts that may deform elastically during interaction with the cylinder wall, leading to the application of elastohydrodynamic lubrication theory to these components [15, 16]. Frictional losses between the skirt and the cylinder wall are
significant, amounting to roughly 30% of total piston assembly friction. The piston skirt interaction with the cylinder wall can lead to noise generation, i.e. the so-called piston slap.

Gray cast iron, malleable/nodular iron and carbides/malleable iron are the most common base materials for all types of compression piston rings and single piece oil-control rings. However, steel is becoming popular as a piston ring material due to its high strength and fatigue properties. Steel is used for top compression rings and the rails of multi-piece oil control rings. There are a variety of surface treatments and coatings for piston ring running-in processes. Chromium plating and flame sprayed molybdenum are the most common wear resistant coatings, although plasma sprayed molybdenum, metal matrix composites, cermet and ceramics are becoming popular as their technology progresses. Aluminum–silicon alloy is the major material used for automotive pistons, with additions of other elements to enhance particular properties, e.g. copper to increase fatigue strength. Piston skirt coatings include polytetrafluorethylene (PTFE) or graphite.

In contrast to the piston ring and the cylinder liner coatings, piston skirt coatings are generally intended to reduce friction rather than wear. Controlling friction and wear of the piston assembly is crucial to successful engine performance. Several research opportunities in piston ring/cylinder bore contact can be explored by investigating low friction coatings, wear resistant coatings for aluminum bores, surface texture control, and better surface finish or treatments. Manufacturers have come to rely upon early life wear of the piston rings and cylinder wall to modify the profile and roughness of the interacting surfaces to achieve acceptable performance as part of the running-in process.

A clear understanding of the complex interactions between lubrication and wear of these components is indeed important for lubrication engineers. Manufactured surface finish of piston rings and cylinders can have a major influence on wear behavior and thereby the success or failure
of an engine. Despite the evolving wide range of surface finishing techniques for both piston rings and cylinder liners, the objectives of these processes appear to be the same: to improve oil retention at or within the surface, to minimize scuffing, and to promote ring profile formation during the running-in period. Plain cast iron piston rings are often used in a fine turned condition and electroplated; flame sprayed and plasma sprayed rings are generally ground to the desired finish.

1.6 Surface Texturing on a Material’s Surface

Laser surface texturing (LST) is a surface engineering process used to improve tribological characteristics of materials. Using a laser to create patterned microstructures on the surface of the materials can improve load capacity, wear rates, lubrication lifetime, and reduce friction coefficient. The use of surface irregularities to improve tribological properties was first discussed in the 1960’s and is implemented via several manufacturing techniques. Given that friction creates unavoidable losses and wear in countless processes and devices, the opportunities for improving efficiency and lifespan with LST technologies are extensive. This reduction of friction has been tested for lubricated friction, since it is the case for piston rings. Solid lubricants also can be used with LST surfaces and benefit from friction reduction. The reduction of friction results in several benefits. First, the energy saved from being lost to frictional heat can lower energy demands of the application. Secondly, a lower friction produces less heat, and as a result thermal stresses and strains at the surface are reduced. Finally, a lower friction coefficient reduces adhesion. Another benefit is an improvement of fatigue life. The micro cavities act as debris traps, saving small loose particles from initiating micro fissures and damage. It has been observed that wear life of a component treated with LST has fatigue lifetime improvements of upwards of three times longer than a standard component. Wear caused by repeated small surface motions, known as fretting wear, can be substantially reduced when LST is applied. Experiments have found that fretting
fatigue life doubled with LST application [17]. In what follows, the debris tracking mechanism, which works as lubrication reservoir, is explained.

1.6.1 Debris Trapping

The micro cavities provide a sink for debris particles to fit into and reduce the associated additional friction of debris in the contact zone. The function of the pores as debris traps is found in both lubricated and non-lubricated applications, and is the main positive effect of surface texturing for non-lubricated applications.

1.6.2 Lubricant Reservoirs

If an area of surface contact loses lubrication, the micro cavities can provide additional lubricant that is drawn to the starved area through capillary action. The patterned geometry allows for countless miniature lubricant reservoirs, providing direct and immediate lubricant relief for starved areas. In order for these reservoirs to exist, the geometries of the pattern microstructure must be closed to prevent lubricant from being forced out through channels.

1.6.3 Benefits

Laser surface texturing has many benefits that could potentially save enormous amounts of energy and improve efficiencies of many mechanical systems. The most obvious benefit is a reduction of friction. Since 1965 many researches have been reported in the field of laser texturing (Figure 1-6), particularly with emphasis on theoretical aspects. Analytical/computational treatments involve considerable time and effort since the governing equation are quite complex and a large number of variables are involved [18]. Although the initial models were not very
sophisticated, newer models capable of handling the inter asperity cavitation, deformation, and asperity contact models are now available to provide more accurate predictions.

Figure 1-6: Worldwide research effort on surface texturing over the last 50 years: (a) Number of publications per year and (b) Research method [18].

In this thesis, results have illustrated a substantial reduction in friction in the range of 20%. The exact reduction in friction depends on many variables including texture’s geometry, piston’s velocity, and the materials used.

1.7 Piston Pressure Rings

Neal [19] describes piston rings as: “mechanical sealing devices used for sealing pistons, piston plungers, reciprocating rods, etc., inside cylinders.” In gasoline and diesel engines and lubricated reciprocating type compressor pumps, the rings are generally split-type compression metal rings. When they are placed in the grooves of the piston and provided with a lubricant, a moving seal is formed between the piston and the cylinder bore [19].
Piston rings are divided into two categories: compression rings and oil-control rings. Compression rings, generally two or more, are located near the top of the piston to block the downward flow of gases from the combustion chamber. Oil rings, generally one or more, are placed below the compression rings to prevent the passage of excessive lubricating oil into the combustion chamber yet provide adequate lubrication for the piston rings. In typical lubricated situations, the piston skirt is in direct contact with the cylinder and acts as a bearing member that supports its own weight and takes thrust loads. In unlubricated arrangements, it is necessary to keep these two surfaces separated because they are not frictionally compatible. This is usually accomplished with a rider ring that supports the piston for a more complete review on Neale (1973), and Taylor and Eyre (1979) are recommended [19, 20].

An ideal piston-ring material must meet the requirements such as low friction and wear losses, superior scuffing resistance, tolerances for marginal lubrication and rapidly varying environments, good running-in wear behavior, long-term reliability and consistency of performance, long maintenance-free life, and low production cost. This study aims to experimentally compare the reduction in friction as a result of surface texturing by using a piston cylinder custom built machine. A new series of texturing with different depth and angle have also been proposed, tested and compared for the application in piston oil rings to reduce engine friction.

1.8 Previous Studies

Primarily theoretical works have been done to investigate the frictional behavior of piston rings. One of the first analytical work predicted the oil film thickness (the film thickness is predicted by solving Reynolds equation considering the “squeeze film effect” at TDC and BDC where it is more significant). Fully flooded lubrication was assumed for the entire stroke length for a four stroke engine at full load and no-load conditions and the results were compared with the
findings of other researchers [21]. Later researches have focused on the effect of roughness. For this purpose models have been proposed to employ stochastic theory and simulate the effect of one dimensional roughness of the piston ring surface on lubrication and friction. The applications of the model to an actual diesel engine indicate that about 8.3% -9.4% increase in friction power loss can be expected with rough surface (d = 0.6 μm) rather than the smooth surface [22]. Quan-bao et al. showed that the peak friction forces in compression and power stroke occur very near the fired top dead center (FTDC) [22]. Since in their research the speeds of their tests were relatively slow (under 400 rpm), the mentioned frictional behavior is expected. In those ranges of speeds, boundary lubrication would carry the majority of contact force, and friction is the highest near the dead center. An increase in roughness will result in an increase in surface-to-surface contact and would result in an increase in power loss.

Models were developed to study the IC engine piston ring friction. The instantaneous friction force of a piston assembly under the firing engine conditions was measured by an improved floating liner method [23]. Ideas to reduce friction in IC engines have been investigated. In one of the successful researches, a two-ring package were applied instead of the standard three ring packages. Further improvements were made by developing low viscosity engine oil, reducing the piston mass by using an alternative material with similar strength but higher strength to weight ratios, changing the piston ring width and piston ring tension, and reducing the piston ring friction losses. Experiment showed that reduction of piston ring tension and using two ring packages are effective in reducing piston ring friction, and that the reduction of piston ring friction can contribute to reducing the fuel consumption by 7% [9].

Some studies have focused on the modeling of the major frictional components of the automobile engine, e.g. bearings, the valve train and the piston assembly, and prediction of the
overall engine friction. Each of these components have been addressed and the specific issues of modeling of lubricant behavior and the role of surface topography touched upon for greater efficiency and reduced environmental impact have been studied [24]. Attempts have been made to discuss some of the challenges facing primary researches in tribology [25, 26, 27, and 28]. One of the attempts was based on extrapolation of existing trends, though more speculatively considering the possible driving forces over subsequent decades [25].

Later studies focused on the current status and the future trends in automotive lubricants [11]. An overview of various lubrication aspects of a typical power train system including the engine, transmission, driveline, and other components to improve the efficiency and productivity is presented [10].

In an experimental approach, a commercial six cylinder, medium speed diesel engine as a test engine was hired to investigate the oil film thicknesses between the compression rings and the cylinder liner. Tests were repeated at different loading conditions but with constant speed (900 rev/min) [26]. In another attempt, results from semi-empirical models were compared with the results of prediction models. In early studies, the squeeze film effect was neglected and a simplified hydrodynamic lubrication theory was applied to predict the oil film thickness. The model proposed by the authors [27] shows that the complete ring pack can be reduced to a set of several compression rings and one twin-rail oil control ring. Each rail of the oil control ring is manipulated as a separate single ring. For the simulation of the oil film action between the piston ring and the cylinder liner, the one-dimensional Reynolds equation was used considering the sliding and squeeze ring motion [27].

Later, tribological behavior of piston rings at different locations was investigated. By studying the compression ring tribology at the vicinity of top dead center in compression and
power stroke transition to attain full fluid film lubrication was studied with the aim of reduction in friction [28]. In this study a set of experiments were carried out on developed experimental setup at laboratory scale to measure PRA friction of multi cylinder 800 cc engine system indirectly by measurement of power consumption by Strip Method with variable frequency drive (VFD) is used to vary the engine speed. Frictional power loss contribution by individual piston ring varies under different speed [29].

In later studies, it was observed that the interactions between the control factors do not have significant influence on the weight loss and friction of the cylinder liner (CL) and PR pair. Some researcher applied the Taguchi design method with respect to three process parameters (sliding velocity, applied load and oil type) in order to optimize the reciprocating wear test for different commercial oil conditions of cylinder liner (CL)/piston [30]. Studies have compared an analytic solution to the average flow model [31]. In that paper, a new analytical thermal model was represented for the piston/Cylinder contact and an analysis was carried out that corresponds to a typical cylinder of a V12 engine with an output power of 510 BHP [31].

Researchers have also attempted at examining the performance of additives in oil, for example, by including nano Titanium Dioxide (TiO$_2$) and P25 additives to base oil to improve friction and wear characteristics [31]. Experimental set up was operated at constant load and constant reciprocating motion using the pin-on-disk apparatus. From this investigation it was found that the TiO$_2$ particles addition to the base oil slightly reduced the coefficient of friction [32]. To avoid high friction and subsequent wear, the liner surface was textured and the ring was coated with thermal and wear resistant coatings. In some of the other recent studies, a four stroke four cylinder engine was modeled for lubrication performance. The detailed parameters related to engine friction and lubrication is computed numerically for the 1-3-4-2 engine firing order [33].
In another study, a sewing machine oil was used to blend oil in the Castrol GTX oil in order to perform oil analysis a four ball tribotester. The coefficient of friction, wear scar diameter and frictional torque these parameters are measured with five different blending ratios two different loads and of oils was tested [34]. Studying the effect of compression ring profile on friction force of internal combustion engine was the purpose of these studies. In one study, three different ring profiles were selected and the ring film thickness, the ring twist angles, the friction force and the friction coefficient for the compression ring were analyzed. Hydrodynamic lubrication was found to prevail for most parts of the stroke [35] except at the dead center where mixed lubrication regime occurred and friction force increased due to reduced film thickness.

By measuring the “power consumption” in the piston ring assembly, the friction under the different operating parameter like speed and lubricant on a motorized multi cylinder engine test rig were measured. Initially the power consumption was reduces until 900 rpm but then it increased with increasing engine speed [36].

Broad literature survey is carried out on the simulations and experimental methods developed to study the performance of piston compression rings. Experimental and theoretical works have been devoted to analyze the tribology of piston compression rings. In several cases results illustrated that 80% of total power loss in an engine cycle is due to friction of piston rings [37].

In another study, the effects of piston scuffing fault on engine performance and vibrations was investigated in an internal combustion (IC) engine. Piston scuffing fault was produced by the three body abrasive wear mechanism, and this caused the engine performance to deteriorate significantly. Piston scuffing fault appeared in the scales of 7–14 (frequency band of 2.4–4.7 kHz)
and more at the scale of 9 (frequency of 3.7 kHz), according to Continuous wavelet transform (CWT) analysis wavelength [38].

Other than the mentioned experiments, some researcher have focused on reviewing different processes. For instance, Shah et al. [3] have done a complete literature review over studies on importance of tribology in internal combustion engines. In most of the previous experimental work on textured piston rings, a reciprocating test rig was utilized to measure the friction between the piston ring segments and cylinder liner segment where the ring specimens were fixed in a holder while the cylinder liner had a reciprocating motion. This test configuration oversimplified the motion of the piston rings and did not account for the secondary motion of piston and piston rings (radial motion, elastic deformation and twist) in a cylinder. Motorized engine tests using a piston and a piston ring-pack is desirable for evaluation of textured piston rings since it can better simulate the piston ring motion and lubrication in real engines. Moreover, the sealing performance of textured piston ring should were also assessed, since the primary function of piston rings is to form a moving seal between the piston and cylinder wall. If the design of surface textures reduces the sealing capability of piston rings, it would increase the blow-by and lubricant oil consumption and hence decrease the engine efficiency, even though the texturing could help reduce friction loss.

1.9 Objective of this thesis

In order to improve the fuel efficiency of internal combustion engines, one of the best methods is to reduce engine friction. Since friction of piston rings is responsible for the 70% of engine friction lost [5]. In this research, an experimental study was performed to investigate the effect of layered grooves and surface coating on the frictional performance of oil control and pressure piston rings. A piston ring set, piston, cylinder liner and connecting rod from a diesel engine were utilized in a newly developed test apparatus for the measurement of friction force.
Results from the literature were reproduced and by improving experimental set up an improvement in the results were observed.

The present research has concentrated on three aspects related to the tribological behavior of piston rings:

A. Tribological Behavior of Pocketed Piston Rings

   Micro-pockets with optimal geometries were selected based on a previous study and fabricated on the running surface of compression rings using a laser.

B. Frictional Behavior of Grooved Oil Control Rings

   Six different grooves were proposed to be applied on 6 oil control rings. Length, depth, and orientation of the proposed grooves were different in each case. Series of the frictional tests were performed with this different rings and different results were observed from different oil control rings.

C. Frictional Behavior of Coated Piston Rings

   Plasma coating and micro-pockets with optimal geometries were selected based on a previous study and fabricated on the running surface of compression rings using a laser.
Chapter 2 Test Apparatus and Experimental Procedure

In this chapter, the experimental procedure is described. The majority of the current research is experimental, and different geometries and different methods of surface treatment have been tested in order to determine the optimum surface texturing and treatment. Also presented in this chapter is the description of different instruments required for the experiments.

2.1 Laser Texturing Machine

Laser texturing offers unimagined possibilities for improving the performance of mechanical components in a cost-effective manner. Surface texturing began as a manual etching procedure, with a lot of labor required to emboss the structure or graining in the forming tool. On the other hand, in the laser technology the entire process is digitally handled, and the result is recognizably better in quality in terms of precision, accuracy, repeatability, and reliability of the process. The AgieCharmilles laser from GF Machining Solutions (the machine used in current study) uses a completely digital process for texturing and engraving with five axes. A fiber laser with wavelength of 1064 nm was used to fabricate micro textures. The shape and distribution of the pockets are first designed using a CAD software and then imported into the laser device. The texture’s depth can be controlled by setting the power of the laser and number of the repeats that surface beam scans the surface. The laser machine used for this research is the so-called MaxBox unit and is designed for mid-range, manually-loaded production and is equipped with a rugged sheet metal enclosure, interlocked access door with laser safe viewport. It possesses a fully programmable Z-axis, interior work light, a t-slot table and an exhaust port fitting.

Using intelligent software (Scriba), the operator generates textured areas in such a way that a homogenous design of the end product is ensured. This software is developed and maintained within the Electrox organization and allows importation of most common graphic files. A picture
of the laser machine used for this experiment is presented in Figure 2-1. This extensive software package is already integrated in the machine workplace in a user-friendly manner. Complete digitalization of the process makes it possible to calculate and visualize the entire work piece with the desired surface structure on the computer, so that the results can be seen and changed in advance. Since laser manufacturing can be laser-textured directly without incurring costs for tools, the costs for the manufacture of prototypes or design samples are reduced. The machine’s 5-axis makes it possible to produce the textures directly in almost any prototype part. The laser head is very flexible due to the large swivel range of the rotation axes. It can also make complex textures with a geometric or organic appearance. Three-dimensional structures can be engraved on all kinds of materials like aluminum, copper, steel as well as graphite, hard metal, brass or ceramics. Creating variate graining depths of layer thicknesses (even less than 0.001 mm) is also possible.

![Electrox 600 laser machine](image)

**Figure 2-1 Electrox 600 laser machine**

In order to import a design two methods can be used. They are:

a) By using a software, for example AutoCAD, primary draw of the texture shape can be drawn in AutoCAD and imported into the computer connected to the laser Machine. It is
possible to make small modifications with the laser machine drawing software and Scriba3 software.

b) The laser machine’s software (Scriba3) has some drawing tools that can be used for designing simpler textures without taking advantage of engineering drawing software such as AutoCAD.

While the laser stands still, specimen would rotate by an electrical motor. This feature would help with the texturing of circular specimens such as piston rings Figure 2-2.

![A piston ring is being Surface textured using Electrox 600 machine](image)

**Figure 2-2: A piston ring is being Surface textured using Electrox 600 machine**

### 2.2 Piston Tests

Experiments were conducted using a custom-built reciprocating piston test apparatus that enables both friction and compression pressure measurements. This test apparatus directly measures the piston friction and utilizes the cylinder liner, piston, piston rings and connecting rod from a diesel engine. The design set up is similar to the engine model Perkins 4.236, which was first built in the sixties and has been modified several times during last 60 years. This model is very well known and has been frequently used in agriculture applications. The reciprocating
motion of the piston assembly is created with an electric motor driving the crankshaft through a set of pulleys. The rotating speed and angular position of the crankshaft is recorded by a rotary encoder, attached to the end of the crankshaft. The pulley drive ratio used to be 3.4:1 in former studies done with this machine, and as parts of the improvement the ratio was reduced to 2:1. The large pulley connected to the crankshaft can work as a flywheel to ensure the smooth motion of the piston. The stroke length used in the study is similar to that in a typical engine (114.3 mm).

The test apparatus has two configurations for the cylinder liner: a suspended configuration for measuring the friction force and a fixed configuration for measuring the compression pressure. A load cell is installed on the top of the cylinder liner to measure the axial friction force (Figure 2-3). In this research a Futek load cell (model: LCM300) with features such as: minimal mounting clearance, 17-4 PH stainless-steel construction, capable for being used in both tension and compression, utilizes metal foil strain gauge technology, adheres to RoHS directive 2011/65/EU was used.

![Figure 2-3: Load cell model LCM300](image)

As shown in Figure 2-4, the radial motion of the cylinder liner is prevented by three lateral supports positioned at 120° spacing. Two ball bearings are installed on each lateral support to maintain the center position of the liner. Smooth motion in the axial direction can be achieved by the above-mentioned supports. The weight of the cylinder liner and its fixture are counterbalanced.
by a weight assembly to eliminate its influence on the friction measurement. The unwanted effects of the weight of the cylinder be modified, since the zero point in friction force graph can be defined by the operator of the machine in a way that the unwanted weights gets balanced. The cylinder liner is open at its end without a cylinder head in the friction tests so that the compression gas is not sealed and the friction is the only force acting on the suspended liner.

Figure 2-4: Schematic of the test rig [39]
Figure 2-5: Final assembly of the test rig for the suspended-liner configuration for friction test

Figure 2-6 shows the lubrication arrangement of the test rig. The lubricating oil is pumped through a pipe and sprayed to the bottom of the piston and cylinder liner by using a nozzle. Some of the lubricant could flow through the holes in the piston and in the oil control ring to the cylinder wall and then be regulated by the oil control rings. In a real engine, oil is splashed onto the cylinder due to the circular movement of the crankshaft, similar to the nozzle set up. The lubricant oil is stored in an oil tank before being pumped through a filter and its temperature is controlled by a heater attached to the tank. A thermocouple inserted in the chamber so that the temperature of the oil can be monitored online. An oil reservoir is used to collect the lubricant that falls out of the cylinder liner and drains it back to the tank. The oil flow rate is controlled by a valve available next to the gear pump and monitored by a flowmeter.

Figure 2-6: Lubrication arrangement of the test rig [39].

All measurement data (including the data from the rotary encoder, load cell, and the thermocouple in the oil tank) are captured using a data acquisition system and a LabVIEW program.
Real-time data processing is also achievable with the help of this program for reducing noise in the signal.

### 2.3 Untampered Plasma Coating

Another method for improving frictional behavior of surfaces is coating. Several coating processes are available based on their unique features. Coating methods are classified in Figure 2-7. In the current research, four options are available for coating:

![Figure 2-7 General coating methods in producing coating materials for joint prostheses [40]](image-url)
Sputter (metal) - Pt or Au/Pd was not used since by using this method only the upper or lower surface would be covered. In Figure 2-8 a schematic of coating process for TiO2Ps is presented.

![Schematic diagram of coating process](image)

Figure 2-8 Preparation method sputtering of Au and Pd onto RTIL of [BMIm][BF4], in which TiO2 NPs are pre-dispersed [41].

Evaporative Carbon (from a Carbon thread) probe, has only a low tendency for adhesive wear or tribochemical reactions. In most tribological systems these properties result in a lower coefficient of friction under the boundary lubricated conditions when compared to other coating systems (such as metallic layers, hard coatings and others). As the damage mechanism is most frequently seen under cold start conditions, this carbon-based coating is typically chosen and used as a running-in coating applied onto a wear resistant coating (nitride layer, PVD layer, etc. This method is effective but for the current study the same problem of not being able to coat outer radius remained.

Hard coating is vastly used due to its beneficial features such as corrosion resistance and for the ability to function in various thicknesses, offering high hardness (Rockwell scale rating of
C - 60 /70), good wear resistance property, high resistance to chemicals, solvents and petroleum based products, and low friction coefficient.

Untampered plasma coating has a number of benefits such as improving the adhesion and wetting of surfaces. It is convenient and lightweight and can be applied via a handheld device. It is also available in robotic mountable option to improve productivity and repeatability. Since it uses “cold plasma”, it is suitable to treat temperature-sensitive substrates. For the experiments in this thesis a Piezo brush PZ2 was used. Using Piezoelectric Direct Discharge (PDD®) Technology, developed by Relyon Plasma, the device transforms low input voltage into high electric field strengths, dissociating and ionizing the ambient process gas (typically air). For this method a Piezo Brush PZ2 as shown in Figure 2-9 was used.

![Figure 2-9 Piezo Brush PZ2](image)

In Figure 2-10 schematic of a pressure ring with square textures on it is presented.

![Figure2-10 schematic of textured pressure ring](image)
In this chapter, all the experimental equipment such as laser machine, custom-built engine tester and untampered plasma coater are described. Technical characteristics and reason for using this specific equipment were described. Also presented are the schematic of a textured pressure ring for further analysis to be described in the following chapter.
Chapter 3 Effect of Micro-grooves on the Friction Reduction of Oil Control Rings

The main outcome of the test procedure is to obtain quantitative frictional characteristics of oil control rings, which is an important indicator of the performance of piston rings. Former members of Center for Rotating Machinery (CeRoM) at Louisiana state university have designed and optimized the shape of textures for piston ring application [39, 42, 43]. In one of the papers the performance of different shapes textures (sphere, triangle, square and trapezoidal shape) was compared. It was shown that trapezoidal shape textures had the most improvement in frictional performance of piston rings [42]. Later, the effect of depth of trapezoidal on the performance was investigated, and it was shown than shallow textures (5 µm depth) showed improved performance compared to deeper ones (10 and 15µm) [39].

In the current study, several validation tests have been performed, and the results of Shen et al. [39] were reproduced. The range of the speed in previous studies with this rig was 60 rpm to 600 rpm, which is relatively small. In this study, textured (trapezoidal with the shallow depth) piston rings were tested in speeds as high as 950 rpm (Table 3-1). In addition, the lubricant temperature was increased to a temperature near a typical engine operating temperature (85 ºC) to further validate that textured surfaces (Figure 3-1). In both cases results were promising.
Figure 3-1 Comparison between current research (right) and Shen et al. results [39] (left) at 25°C at various speed (a and b) and at 60 rpm (c and d).

Figure 3-1 compares the friction traces for untextured and pocketed piston rings. The results show that the friction-reduction effect of the pockets vary depending on the operating
conditions. At lower speeds, their major effect is to reduce the friction spikes near the top and bottom dead centers. However, at higher speeds they can help lower the magnitude of the hump in the friction trace that occurs around the mid-stroke. These two effects can be explained by the friction reduction mechanisms of the laser pockets discussed above. Each piston ring and the piston skirt contribute to the measured friction force. The friction force acting on each part includes the friction force from the contacting asperities and the viscous shear force in the lubricant film. By increasing the speed the hydrodynamic action is improved, hence the fluid film separation is increased and as a result a reduction in asperity contact and friction is observed. On the other hand, an increase in the speed results in an increase in the viscous shear force, since it is proportional to the shear rate and viscosity. These two competing factors influence the total measured force. The trend of increased friction with speed indicates that the increase of viscous force is more dominant in the total friction [43].

3.1 Improvements

The highest speed reached in the previously done experiments was 600 rpm [39]. Some modifications, such as changing the ratio in the motor gears, had to be implemented to reach to the speed of 1100 rpm (Figure 3-2). It is worthy to note that at higher speeds, the problem of vibration is a major concern. To overcome such a concern, additional deadweight (almost 100 kg) had to be applied to decrease the vibration. Also, the variation in results was reduced by a filtration codes presented in appendix A. The major function of the program is to filter the noises such as vibration on the final result.
Figure 3-2: Friction force versus crank angle at 1100 RPM at room temperature (25°C)

In the next step, the oil temperature was increased to engine temperature (85°C) with using an electric heating element and a sensor to monitor the temperature change. Smooth and trapezoidal pocket textured ring set were tested at 85°C and a similar trend as in 25°C was observed (Figure 3-3). Friction force was recorded as the average of friction at each speed. A small difference in friction force in Figures 3-3a, b is due to using different piston ring brands. Both brands of piston ring yield the same trend.
Figure 3-3 illustrates the friction test results under lubrication of SAE 30 oil at temperature of 85 °C (left) and 60 °C (right). As the oil temperature is increased from 25 °C to 60 °C and 85 °C, its viscosity is decreased from 0.17 to 0.03 and 0.02 Pa·s, respectively.

3.2 Lubrication Regimes

As can be seen in Figure 3-4, when the speed is increased from 60 rpm to 480 rpm, the friction force exhibits different trends. Under low speeds (60 rpm), the friction force reaches its highest value at the beginning and the end of the reciprocating strokes (top and bottom dead centers), where the sliding velocity is the lowest and the friction is dominated by the asperity contacts. Increasing the crank speed to 120 rpm and 240 rpm results in lowering the friction spikes at the dead centers and friction build-up around mid-stroke. This trend is attributed to the increased hydrodynamic action at the piston ring/cylinder liner interface and the rise in viscous shear force due to higher sliding speeds. Further increase of the crank speed to 480 rpm leads to a friction trace with a sinusoidal shape.

The asperity interaction is greatly reduced near the dead centers, and the friction force has its highest value in the middle of the stroke, where the sliding velocity is the highest throughout the reciprocating cycle. This indicates that the viscous shear force is dominant in the total friction under high speeds. Since the shear force is proportional to the shear rate (sliding speed divided by the film thickness), a higher speed usually results in a higher shear force. Although the film
thickness could also increase with speed, this effect is less significant than the impact of increased speed.

Figure 3-4: Change in the trend of friction force in accordance with the change in the speed

The friction coefficient of the piston rings at each speed is calculated. The purpose of such calculations was to clarify changes in lubrication regime as mentioned in the introduction. Figure 3-5 shows the variation of the friction coefficient as a function of the speed, a plot that is generally referred to as the Striebeck curve. As can be seen in Figure 3-5, hydrodynamic lubrication starts when the operating speed reaches 200 rpm. Note that due to the transition in the lubrication regime, the shape of frictional forces changes as a function of the crank angle (Figure 4-3).
3.3 Improving Tribological Behavior of Oil Control Rings

The oil control rings have the largest contribution to the frictional losses in the piston ring pack, making it very interesting when focusing on reduction of fuel consumption. Figure 3-6 shows the distribution of ring friction in a diesel engine. Our engine tester uses the piston from a Perkins diesel engine, which has two oil control ring. The friction reduction with oil control ring could have more significant influence on the total friction. In Figure 3-7 a picture of installed oil control ring is presented. Figure 3-8 shows a single oil control ring.
Engine tester used for this study, is equipped with two twin-land oil control rings. The outer lands of the rings are flat and parallel to the sliding direction.

Figure 3-6: Distribution of oil control ring’s friction [6]

Figure 3-7: Single cylinder engine tester (Perkins diesel engine)

Figure 3-8: Untextured oil control ring
The method used to reduce oil ring friction is by means of laser texturing its surface. Since the oil ring surface is very thin, the textures had to be carefully designed in order to achieve the friction reduction effect. The following micro-grooves are proposed for the twin land of the oil ring. As shown in Figure 3-9, the distribution of the groove on the two lands can be staggered to prevent excessive oil left on the liner surface. The width, depth, pitch and orientation of the grooves can be investigated to find the optimum design for friction reduction, as shown in Table 3-2.

Figure 3-9: Micro-grooves on the twin land surfaces of oil control ring

Table 3-2: Geometric parameters for micro grooves

<table>
<thead>
<tr>
<th></th>
<th>Groove width (micron)</th>
<th>Groove Depth (micron)</th>
<th>Pitch (micron)</th>
<th>Angle (degree)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>100</td>
<td>5</td>
<td>1000</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>100</td>
<td>5</td>
<td>1000</td>
<td>30</td>
</tr>
<tr>
<td>3</td>
<td>100</td>
<td>5</td>
<td>1000</td>
<td>60</td>
</tr>
<tr>
<td>4</td>
<td>100</td>
<td>10</td>
<td>1000</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>100</td>
<td>10</td>
<td>1000</td>
<td>30</td>
</tr>
<tr>
<td>6</td>
<td>100</td>
<td>10</td>
<td>1000</td>
<td>60</td>
</tr>
</tbody>
</table>
The groove configuration numbers 1, 2 and 3 in Table 3-2 have the same depth but different angles. In order to get a better understanding of the grooves of Table 3-2, a drawing of a part of the ring which has been modeled in ANSYS is presented (Figure 3-10). Schematic of the geometry of the grooves is presented in Figure 3-11.

Figure 3-10: Model of the 6 grooves in the Table 3-2

Figure 3-11: Drawing of the first suggested groove geometry in Table 3-2
Chapter 4 Results and discussion

In this chapter conducted results as well as a brief discussion on the reasoning behind the results will be presented.

4.1 Effect of groove geometry

One of the researches with most similarity to this geometry and test set-up is presented by Zapl et al. [44]. In their research, they have defined a new parameter GIR (groove influence ratio) with the purpose of quantitatively studying the effect of grooves on the film thickness. This parameter was defined as a ratio of minimum film thickness influenced by the groove in centerline ($h_{Gl_{min}}$) divided by the central film thickness of smooth surface under steady state conditions ($h_{smooth}$). The value of $h_{Gl_{min}}$ was evaluated in the central zone of the contact (out of the outlet constriction). The aim was to capture the groove effect in highly loaded zone where thin film thickness can have significant effect on wear and other failures. Groove influence ratio is given by [44]:

$$R_{GIR} = \frac{h_{Gl_{min}}}{h_{smooth}}$$  \hspace{1cm} (4-1)

4.2 Effect of Material

In the current experiments, the textures were designed and applied on the oil ring surfaces using the laser machine. Newly textured rings showed different behavior compared to the used ring due to build up edges caused by using the laser machine. One method to overcome the effect of built up edge was to run a break-in procedure in which rings would be installed and work at relatively low speeds (120 rpm) for a short period of time since at this speed a great portion of the
total load is carried by the asperity contact. The friction force was recorded online and as soon as the running-in period was stabilized.

Figure 4-1 compares the friction force before and after running-in. Red lines show the friction verses time when the breaking in period has just started and black line shows the friction force after breaking in. Running-in period last for three hours and the oil flow rate was maintained at 1.2 L/min for all the experimental tests. Speed was set to 120 rpm at room temperature (25° C).

Figure 4-1 Comparison of friction force before and after running-in

Another set of test was performed to make sure that the piston rings have entered the steady state regime. For this purpose, a new set of pressure rings were textured with the subject to trapezoidal patterns (Figure 4-2) and were installed on the machine and the speed of 120 rpm.
Figure 4-2 Image of pocketed compression ring [39]

Table 4-1 Geometric parameters of lasered pockets [39]

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pocket shape</td>
<td>Trapezoidal</td>
</tr>
<tr>
<td>Characteristic length</td>
<td>$a = 0.6$ mm; $b = 0.36$ mm; $h = 0.75$ mm</td>
</tr>
<tr>
<td>Pocket depth</td>
<td>$3.9 - 4.2$ µm</td>
</tr>
<tr>
<td>Pocket spacing</td>
<td>$2.8^\circ$ (circumferentially)</td>
</tr>
<tr>
<td>Total no. of pockets</td>
<td>516</td>
</tr>
<tr>
<td>Area ratio</td>
<td>25.3%</td>
</tr>
</tbody>
</table>

The purpose of this test was to capture friction behavior of the piston rings. A total of 22 friction force values were recorded. For the first 200 minutes, the friction force was measured every 10 minutes and from 200 minutes to 360 minutes it was measured every 20 minutes. For each one of these data points, a graph similar to Figure 4-3 was created by the software of the machine and then was filtered by a MATLAB code. In the Figure (4-4 and 4-5), max and min points that have been used to draw the graphs shown in Figure 4-6 are shown. Since the graphs are not symmetrical with respect to zero, min + max/2 has been used to get an accurate estimation of the friction force.
Figure 4-3: A sample of output of LabVIEW connected to machine after filtering using a MATLAB program (After 200 mins).

Figure 4-4: Change in the minimum friction force
Figure 4-5: Change in the maximum friction force

Figure 4-6: Trend of the change in the Max and Min of friction force verses Time
As it can be seen in the Figures 4-6 after 50 minutes of testing, there is no further reduction in the amount of friction force and the process entered into the steady state phase. All 6 pattern designs as well as an untextured ring are used for comparison. For each design, 10 data points at 10 different speeds were captured which are shown in Figure 4-6. A possible reason for the friction reduction of pocketed rings at low speed is that these groove can act as oil reservoirs during sliding, and help to supply oil to the contact surface and reduce asperity contacts. At high speeds where the hydrodynamic action is more dominant, the pockets are expected to work as tiny step bearings and generate additional hydrodynamic pressure and load-carrying capacity.

4.3 Effect of Groove Orientation

As can be seen in Figure 4-7, the pattern with 0 degree grooves has the best performance. Previous studies have investigated the effect of texture orientation on the frictional performance. Studies have shown that textures oriented perpendicular to the sliding direction would have the best performance [45]. Likewise, in the current experiments grooves with different orientation showed relatively similar behavior. All of the test characteristics such as speed and temperature were identical and the only difference was oil control rings with different grooves orientation. Parallel grooves are the optimum with friction force near 65 N, oil control ring with smooth surface is the second one with the friction near 70 N, 60 degree is the close third with 72 N, and finally for the 30 degree 78N was captured.
4.4 Effect of groove depths and width

Previous studies concentrated on the effect of patterns depth on the frictional behavior of the ring [42]. The effect of pocket depth on the average friction coefficient operating under different loads and rotational speeds is presented in Figures 4-9 and 4-10. The specimens (R2, R4, and R5) have an identical area ratio of 25%, their pocket depths are varied from 5.3 to 25.6 µm. The results show that, under the conditions tested, the specimen with smallest pocket depth has the lowest overall friction coefficient. While the specimen with the highest pocket depth exhibits the highest friction coefficient. In addition, the friction coefficient of deep pockets is consistently higher than that of the plain surface specimen, even under relatively high-speed conditions. This indicates that the pocket depth plays an important role on the tribological performance of the pocketed surfaces, and that deep pockets do not improve the frictional characteristics. In fact, their
performance in terms of friction coefficient is inferior to untextured surfaces. The reason for these results is thought to be related to the hydrodynamic lubrication of the pockets, which work similar to the step bearings. According to the Rayleigh step-bearing theory, if the ratio between step height and lubricant film thickness is too high, the step bearing can hardly generate any load-carrying capacity [39]. Hence, deep pockets in this study cannot produce additional hydrodynamic pressure and yield higher COF.

![Figure 4-9: The effect of pocket depth on the frictional performance [39].](image)

![Figure 4-10: Friction force comparison between specimens with different pocket depths [39].](image)
Figure 4-11 shows the friction force comparison between the specimens with different pocket depths. It can be seen that shallow pockets (with depth of 5.3 µm) show superior performance compared to others throughout the full cycle of the reciprocating motion while deep pockets (with depth of 25.6 µm) yield higher friction [39].

It can be concluded that the first groove with the following characteristics has the best performance.

Table 4-3 characteristic of the optimum groove

<table>
<thead>
<tr>
<th>Groove 1 width micron</th>
<th>Groove depth micron</th>
<th>Groove pitch micron</th>
<th>angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>5</td>
<td>1000</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 4-12: Picture of the optimum groove

4.5 The effect of untampered plasma coating

The goal of this part of the research is to study the effect of coating on the tribological behavior of piston rings. Of particular interest to determine if a combination of coating and laser texturing would improve the piston ring’s performance.
Coating procedure was done by the help of a piezo gun, which creates a plasma coating on the surface. In the current research, a three-minute treatment under the plasma gun was used for each point all over the surface of each piston ring. Five sets of piston rings were tested. The first set was coated with the plasma gun, the next set was just textured, the third set was coated identical to the first set but then coated, and the fourth set was textured and then coated just like the set number 1 and 3. Finally, the fifth set was neither coated nor textured for the reference purposes.

One concern was that if the sequence of coating/texturing versus texturing and then coating would affect the running in process of rings. Prior to recoding data, the rings were run at a relatively low speed (100 rpm) for some time (between one to two hours) or until the friction coefficient stabilized. The goal for such a procedure was to make sure the friction between the piston rings and piston skirt had completed its running-in state and stabilized by entering into the steady state. Figure 4-13 presents the behavior of different sets during running in.
Figure 4-13: running in behavior of: textured, coated, coated- textured and textured-coated piston rings

As can be seen in Figure 4-13, the running-in behavior of different sets are different. It took close to 75 minutes for uncoated textured ring set to reach to the steady state. For the textured ring, it took 50 minutes while the time for similar ring after being coated was reduced to only 30 minutes. One application of this would be in reducing the running-in period of mechanical element in an engine. According to Table 4-4, the running-in period of new cars can be reduce up to 60% with application of mentioned methods.

Table 4-4: a comparison between running-in behavior of different ring sets

<table>
<thead>
<tr>
<th>Surface Treatment</th>
<th>Running in Time (Min)</th>
<th>Reduction in Running in time Compare to Conventional Rings</th>
<th>Initial Friction Force (N)</th>
<th>Steady State Friction Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>75</td>
<td>0</td>
<td>185</td>
<td>145</td>
</tr>
<tr>
<td>Textured</td>
<td>50</td>
<td>33%</td>
<td>180</td>
<td>141</td>
</tr>
<tr>
<td>Coated</td>
<td>30</td>
<td>60%</td>
<td>160</td>
<td>123</td>
</tr>
<tr>
<td>Textured then Coated</td>
<td>27</td>
<td>61%</td>
<td>169</td>
<td>140</td>
</tr>
<tr>
<td>Coated then Textured</td>
<td>23</td>
<td>62%</td>
<td>173</td>
<td>137</td>
</tr>
</tbody>
</table>

Combination of coating and texturing not only reduced the running in time but also yielded the lowest frictional forces in an engine (Figure 4-14). A 15% reduction in friction force was
observed where traditional rings were replaced by rings that have been textured and coated. The close second best behavior was observed when the rings were coated and then textured. In higher speed difference between the rings sets with coating and texturing was reduced. It can be assumed that in higher speeds both ring set would share a friction force amount. Texturing and coating alone would also improve frictional behavior as well.

Figure 4-14: Frictional behavior of piston ring sets under after various surface treatment
Chapter 5 FEM Model

5.1 Finite element method

ANSYS is a numerical method for solving problems of engineering and mathematical physics using the finite element method (FEM). Typical problems include structural analysis, heat transfer, fluid flow, mass transport, and electromagnetic potential. The basic concept of the finite element method which is applied on a physical problem is dividing the domain into smaller subdomains which are defined as the finite elements and solve the governing equations on these elements.

In this study, two different models were developed. The first one is a 2D model of a simplified shaped ring connected to a piston sliding against cylinder and the second model is a modeling reaction of a piston under heat fluxes similar to one that are applied to it in engine tester machine. In the 3D model of piston and cylinder, components such as oil rings, have the similar size as the real components.

For each crank angle, the corresponding cylinder liner diameter and pressures are used for prediction of the piston ring behavior. All simulations are performed on a workstation PC with a dual core 2.8 GHz CPU, 8 Gb RAM and a 64-bit Windows 7 operating system.

5.1.1 Main processes

This section describes the procedure in each step of the simulation work flow. The geometry of the piston ring and the boundary conditions can easily be modified in the simulation environment and it is also possible to adapt specific simulation work flows. For the advanced user it is recommended to analyze the simulation tool thoroughly to obtain a deeper understanding with help of the reference manual for ADPL (ANSYS Parametric Design Language) commands, which
is the programming language in ANSYS. During a simulation there are mainly four different processes which are executed which are

- Design of the piston ring geometry
- Solid element mesh generation
- Generation of contact surfaces
- Application of the specific boundary conditions

5.1 2D model of piston ring and cylinder

A 2D model was developed to study the deformation of rings under pressure caused by sliding piston on cylinder skirt. A 2D model of a portion of piston and a portion of cylinder connected to each other through a ring which got modeled as circle have been presented in Figure 5-1.

![Figure 5-1 Schematic of the model.](image)

All the characteristics of material involved in the model are as presented in Table 5-1:

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>7850</td>
<td>kg.m$^{-3}$</td>
</tr>
</tbody>
</table>
Meshing the 2D model is done in the mesh tool as shown in Figure 6-2. Piston ring was meshed finer due to its potential to deform more and also since its deformation is the main purpose of this modeling.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tensile yield strength</td>
<td>2.5 ×10^8 Pa</td>
</tr>
<tr>
<td>Tensile ultimate strength</td>
<td>4.6 ×10^8 Pa</td>
</tr>
<tr>
<td>Initial shear moduli</td>
<td>0.1 MPa</td>
</tr>
<tr>
<td>Incompressibility parameter</td>
<td>0.15 MPa^-1</td>
</tr>
<tr>
<td>Young moduli</td>
<td>180 GPa</td>
</tr>
</tbody>
</table>

### 5.1.2 Results

The change in the pressure of the pressure ring as the result of sliding is presented in Figure 5-3. A nonlinear increase in the pressure is observed. Nonlinearity is resulted from rapid deformation of the rings under specific pressures.
Figure 5-3 change in the pressure as the result of contact

The total sliding distance (SLIDE) is the amplitude of total accumulated slip increments (a geometrical measurement) when the contact status is sticking or sliding (STAT = 2, 3). It contains contributions from the elastic slip and the frictional slip. Elastic slip due to sticking represents the reversible tangential motion from the point of zero tangential stresses. Ideally, the equivalent elastic slip does not exceed the user-defined absolute limit.
Figure 5-4 Chang in the sliding distance versus time

5.2 3D model

In the next step, a combination of static and thermal loading has been applied to the piston setup. This load has been defined similar to the real loads in an actual engine.

5.2.1 Geometry

The goal was to study the behavior of piston rings under variant thermal and statically loadings. For this purpose, the geometry of the piston and piston rings was precisely defined in the software. This model consists of 6 parts: piston, piston skirt, oil control ring and 3 pressure rings. Each one was separately designed and then assembled in the model. Frictional contact between each two elements was also defined. In Table 5-2 a list of the geometry of components in the model is presented.
Table 5-2 Geometry of Modeled Components

<table>
<thead>
<tr>
<th>Object Name</th>
<th>cylinder</th>
<th>piston</th>
<th>pressure ring 3</th>
<th>oil control ring</th>
<th>pressure ring 1</th>
<th>pressure ring 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length X</td>
<td>268.33 mm</td>
<td>258.8 mm</td>
<td>259. mm</td>
<td></td>
<td></td>
<td>259.09 mm</td>
</tr>
<tr>
<td>Length Y</td>
<td>193.93 mm</td>
<td>108.94 mm</td>
<td>8.4096 mm</td>
<td>6.39 mm</td>
<td>3.19 mm</td>
<td>2.39 mm</td>
</tr>
<tr>
<td>Length Z</td>
<td>268.33 mm</td>
<td>258.8 mm</td>
<td>259. mm</td>
<td></td>
<td></td>
<td>259.09 mm</td>
</tr>
<tr>
<td>Volume</td>
<td>0.14 m³</td>
<td>0.139 m³</td>
<td>86886 mm³</td>
<td>48682 mm³</td>
<td>32889 mm³</td>
<td>24731 mm³</td>
</tr>
<tr>
<td>Centroid Y</td>
<td>25.527 mm</td>
<td>7.1854 mm</td>
<td>3.0504 mm</td>
<td>-7.8884 mm</td>
<td>26.17 mm</td>
<td>13 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Statistics of meshes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nodes</td>
</tr>
<tr>
<td>Elements</td>
</tr>
</tbody>
</table>

Every geometric detail was addressed in the model as well. This would improve the accuracy of the model, with the help of Figure 5-5, a comparison between the actual ring and the model is possible.

a) Modeled piston with rings assembled on it  
b) Piston
Figure 5-5 a) modeled piston, and b) a picture of the piston installed on this research’s test instrument.

Geometry of the assembled particles in total, number of particles, total number of nodes and elements are presented in Table 5-3.

<table>
<thead>
<tr>
<th>Table 5-3. Geometry of the model</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Bounding Box</strong></td>
</tr>
<tr>
<td>Length X</td>
</tr>
<tr>
<td>Length Y</td>
</tr>
<tr>
<td>Length Z</td>
</tr>
<tr>
<td><strong>Properties</strong></td>
</tr>
<tr>
<td>Volume</td>
</tr>
<tr>
<td>Mass</td>
</tr>
<tr>
<td><strong>Statistics</strong></td>
</tr>
<tr>
<td>Bodies</td>
</tr>
<tr>
<td>Active Bodies</td>
</tr>
<tr>
<td>Nodes</td>
</tr>
<tr>
<td>Elements</td>
</tr>
<tr>
<td>Mesh Metric</td>
</tr>
</tbody>
</table>

### 5.2.2 Material

In each model, an important step is to define the characteristics of the material. For this model, structural steel was defined for piston skirt and cast iron was define as the material for the piston. In ANSYS software, a demo characteristic for most of manufacturing material exists and it is capable of being updated by the user. In Table 5-5, mechanical characteristic of structural steel at 22 °C is presented.
Table 5-5 Structural Steel, Compressive Ultimate Strength

<table>
<thead>
<tr>
<th>Reference Temperature</th>
<th>22 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressive Yield Strength</td>
<td>400-500 MPa</td>
</tr>
<tr>
<td>Tensile Yield Strength</td>
<td>250 MPa</td>
</tr>
<tr>
<td>Tensile Ultimate Strength</td>
<td>400-500 MPa</td>
</tr>
</tbody>
</table>

5.2.3 Friction

In order to study the piston rings one of the key point is to simulate the frictional behavior of the piston rings. For doing so, ANSYS is equipped with different frictional tools. Piston rings are designed in a way that are constantly under pressure and in contact with the piston skirt. While there may be some flaws in the geometry and meshing, a menu named “adjust to touch” was activated for all the rings this menu would fix any separation between piston rings and piston skirt by adjusting the geometry (gap is zero for all the contacts in Table 6-4). In Table 6-4 characteristics of each of the contacts is presented.

Table 5-6 contact information for all components

<table>
<thead>
<tr>
<th>Name</th>
<th>Type</th>
<th>Number Contacting</th>
<th>Penetration (mm)</th>
<th>Gap (mm)</th>
<th>Geometric Penetration (mm)</th>
<th>Geometric Gap (mm)</th>
<th>Contact Depth (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frictional - cylinder To pressure ring 3</td>
<td>Frictional</td>
<td>2.</td>
<td>0.</td>
<td>0.</td>
<td>0.</td>
<td>0.053334</td>
<td>4.5907</td>
</tr>
<tr>
<td>Frictional - cylinder To pressure ring 3</td>
<td>Frictional</td>
<td>17.</td>
<td>2.842*10^{-14}</td>
<td>0.</td>
<td>0.</td>
<td>0.0689</td>
<td>6.3371</td>
</tr>
<tr>
<td>Frictional - cylinder To oil control ring</td>
<td>Frictional</td>
<td>0.</td>
<td>0.</td>
<td>0.</td>
<td>0.</td>
<td>0.005</td>
<td>4.5907</td>
</tr>
<tr>
<td>Frictional - cylinder To oil control ring</td>
<td>Frictional</td>
<td>522.</td>
<td>5.901*10^{-14}</td>
<td>0.</td>
<td>0.</td>
<td>0.005</td>
<td>3.5756</td>
</tr>
<tr>
<td>Frictional - cylinder To pressure ring 1</td>
<td>Frictional</td>
<td>1.</td>
<td>0.</td>
<td>0.</td>
<td>0.</td>
<td>0.0070284</td>
<td>4.5907</td>
</tr>
<tr>
<td>Frictional - cylinder To pressure ring 1</td>
<td>Frictional</td>
<td>8.</td>
<td>2.842*10^{-14}</td>
<td>0.</td>
<td>0.</td>
<td>0.0070284</td>
<td>6.3222</td>
</tr>
</tbody>
</table>
5.2.4 Meshing

Due to complexity of the shapes, conventional methods for meshing would not result in accurate outcomes. For such cases, ANSYS is equipped with alternative methods for meshing. Each of these methods is ideal for a specific geometry. For circular ring, manual hexagonal meshing was used. In Figure 6-6 the complete geometry after being meshed is presented. It has been cut for better presentation.

As can be seen in Figure 5-7, different rings have been meshed separately for more accurate results.
Figure 5-7 Meshed piston with meshed rings installed on it.

In Figure 5-8, 3 pressure rings and 2 control rings can be seen. Since each ring has specific geometry which is unique, shape of the elements are different for each rings.
Figure 5-8 Closer look at the oil control ring’s geometry and meshed elements.

Piston components’ manufacturers, make sure that the surface of products are as smooth as possible before products leave the company. This is not only a safety concern but also it would increase the efficiency of the engines. As can be seen in Table 5-7, in the modeled piston set up, all surfaces are set to be very smooth and surface elements are set to be very fine, which is the feature of smooth surfaces.

Table 5-7 characteristic of meshing

<table>
<thead>
<tr>
<th>Sizing</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Use Advanced Size Function</td>
<td>On: Curvature</td>
</tr>
<tr>
<td>Relevance Center</td>
<td>Fine</td>
</tr>
<tr>
<td>Initial Size Seed</td>
<td>Active Assembly</td>
</tr>
<tr>
<td>Smoothing</td>
<td>High</td>
</tr>
<tr>
<td>Transition</td>
<td>Slow</td>
</tr>
<tr>
<td>Span Angle Center</td>
<td>Fine</td>
</tr>
<tr>
<td>Curvature Normal Angle</td>
<td>Default (18.0 °)</td>
</tr>
<tr>
<td>Min Size</td>
<td>Default (6.2213e-002 mm)</td>
</tr>
<tr>
<td>Max Face Size</td>
<td>Default (6.22130 mm)</td>
</tr>
<tr>
<td>Max Size</td>
<td>Default (12.4430 mm)</td>
</tr>
<tr>
<td>Growth Rate</td>
<td>Default (1.20 )</td>
</tr>
<tr>
<td>Minimum Edge Length</td>
<td>678.890 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Statistics</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Nodes</td>
<td>43927</td>
</tr>
<tr>
<td>Elements</td>
<td>10296</td>
</tr>
<tr>
<td>Mesh Metric</td>
<td>None</td>
</tr>
</tbody>
</table>

5.2.5 Thermal Studies

In the experimental set up instead of combustion chamber, the source of the heat is the by physically heating the oil. Therefore, the piston ring may behave differently from what has been
observed in an actual engine. In other words, thermal models that have so far explained the thermal behavior of conventional piston rings cannot be used for this experiment. A thermal boundary condition was applied to the model. In this study for simplicity, thermal Radiation and conduction were neglected. Thermal convention constants were defined for each of the elements separately, due to difference in the material and therefore thermal coefficients.

In order to model the heat source, the inside wall of the piston was set to get an increase in the amount of temperature since hot oil is pumped thru the nozzle in inner surface of piston and oil control ring pass to outer wall of the piston and splash it over the piston skirt.

Change in the temperature was modeled in 3 steps. Steps and assigned temperature to each step can be seen in Table 6-8.

<table>
<thead>
<tr>
<th>Step</th>
<th>Time [s]</th>
<th>Temperature [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.</td>
<td>0.</td>
<td>22.</td>
</tr>
<tr>
<td>0.</td>
<td>2.</td>
<td>85.</td>
</tr>
</tbody>
</table>

Change in the temperature in the piston skirt due the change in temperature of the internal surface of piston is presented in Figure 5-10.
Figure 5-10 Change in the temperature, after the change in oil temperature.

Results of the model in terms of temperature change and heat flux is presented in Table 5-9
Table 5-9 overview of the thermal behavior of the model

<table>
<thead>
<tr>
<th>Type</th>
<th>Temperature</th>
<th>Total Heat Flux</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum</td>
<td>22. °C</td>
<td>7.4199*10^{-18} W/mm²</td>
</tr>
<tr>
<td>Maximum</td>
<td>22. °C</td>
<td>1.2177*10^{-18} W/mm²</td>
</tr>
<tr>
<td>Minimum Occurs On</td>
<td>cylinder</td>
<td>pressure ring 2</td>
</tr>
<tr>
<td>Maximum Occurs On</td>
<td>cylinder</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Minimum Value Over Time</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum</td>
<td>22. °C</td>
<td>7.4199 * 10^{-18} W/mm²</td>
</tr>
<tr>
<td>Maximum</td>
<td>22. °C</td>
<td>9.6362 * 10^{-18} W/mm²</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Maximum Value Over Time</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum</td>
<td>22. °C</td>
<td>1.2177 * 10^{-12} W/mm²</td>
</tr>
<tr>
<td>Maximum</td>
<td>22. °C</td>
<td>1.2177 * 10^{-12} W/mm²</td>
</tr>
</tbody>
</table>

Maximum amount of heat flux due to increase in the temperature is presented in Table 5-10.

Table 5-10 Total Heat Flux

<table>
<thead>
<tr>
<th>Time [s]</th>
<th>Minimum [W/mm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>9.6362 * 10^{18}</td>
</tr>
<tr>
<td>2.</td>
<td>7.4199 * 10^{18}</td>
</tr>
</tbody>
</table>

5.2.6 Displacement

Movement of the piston from dead end stroke to mid stroke is defined by a vertical displacement of piston. As can be seen in Table 5-11, piston starts at Y=0 point (dead end stroke) and moves toward mid stroke (Y=25). These numbers are easily changeable and can be modified with different piston’s geometries.
TABLE 5-11 Displacement

<table>
<thead>
<tr>
<th>Steps</th>
<th>Time [s]</th>
<th>Y [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.</td>
<td>0.</td>
</tr>
<tr>
<td>1.</td>
<td>25.</td>
<td></td>
</tr>
</tbody>
</table>

5.2.7 Lubrication film

For simplicity, a portion of the piston journey inside a cylinder skirt was modeled. Due to the linear behavior of the lubrication film in the portion of the route, the movement of piston from dead end stroke to mid stroke which would be the 25% of the whole journey was chosen. In reality, due to linear increase in the speed from zero (dead stroke) to maximum speed (mid stroke) a linear increase in the hydraulic pressure due to existence of lubrication film will be observed. In order to model the lubricant film, pressure was applied to the piston and to the cylinder. Applied pressure to the piston skirt (Figure 5-11) is normal to the inner surface of the piston skirt. Pressure applied to the oil ring and piston was normal to outer surface of mentioned elements (Figure 5-12)

Figure 5-11 Pressure is defined normal to the inner surface of piston skirt
Figure 5- 12 Pressure is defined normal to the pressure rings and piston

Pressure is set to be increased from zero to the specifies amount in one second. One second is approximately the time needed for the piston to complete this displacement at a speed close to 60 rpm.

5.2.8 Results

A sample of the results is presented in Figure 6-14, by importing accurate characteristic of the model into the software, such as friction coefficient, speed, displacement and material characteristics one would expect accurate results from the model. In the current model additional important value is the amount of pressure that is supposed to replace hydrodynamic pressure. Method to measure the hydrodynamic pressure for different surface textures is solving the Reynolds equation for one unit area (usually unit area consist of one or two textures) and integrate the calculated value over the surface. Amount of pressure in the model easily can be changed. So
in future works if one needs to use the model can calculate the hydrodynamic pressure and adjust the numbers in the software.

Figure 5-14 results from the model
Conclusions

An experimental study was performed to investigate the effect of layered grooves and surface coating on the frictional performance of oil control and pressure piston rings. A piston ring set, piston, cylinder liner and connecting rod from a diesel engine were utilized in a newly developed test apparatus for the measurement of friction force. Results from literature were reproduced and by improving experimental set up an improvement in the results were observed. The present research has concentrated on three aspects related to the tribological behavior of piston rings:

A. Tribological Behavior of Pocketed Piston Rings

Comparing with the smooth piston rings, the pocketed rings have a consistently lower value of averaged friction over the entire speed range. The friction reduction resulted from the layered pockets is about 11‒15% of the total friction. The mechanism for the reduced friction is thought to be different depending on the operating conditions. The lubrication arrangement in this study is similar to that in real engines. Excess oil on the cylinder wall falls down to the oil catcher due to gravity and is also wiped off by the oil control ring, leaving only a thin layer of oil retained on the surface to provide lubrication. As a result, there would be a certain degree of oil starvation at the piston ring/cylinder liner interface, especially when the rotational speed is low and oil cannot be splashed onto the cylinder wall. The possible reason for the friction reduction of pocketed rings at low speed is that these pockets can act as oil-reservoirs during sliding, which helps to supply oil to the contact surface and reduce asperity contacts. However, at high speeds where the hydrodynamic action is more dominant, the pockets are expected to work as tiny step bearings and generate additional hydrodynamic pressure and load-carrying capacity

B. Frictional Behavior of Grooved Oil Control Rings
Six different grooves were proposed to be applied on 6 oil control rings. Length, depth, and orientation of the proposed grooves were different in each case. Series of the frictional tests were performed with this different rings and different results were observed from different oil control rings. In four cases a reduction in friction was observed. A possible reason for the friction reduction of pocketed rings at low speed is that these grooves can act as oil reservoirs during sliding, thus help to supply oil to the contact surface and reduce asperity contacts. At high speeds where the hydrodynamic action is more dominant, the pockets are expected to work as tiny step bearings and generate additional hydrodynamic pressure and load-carrying capacity. This effect can be attributed to mechanisms of the grooves which not only would reduce the contact surface but also can act as micro oil reservoirs under starved lubrication and work as tiny step bearings under full film lubrication.

C. Frictional Behavior of Coated Piston Rings

Plasma coating and micro-pockets with optimal geometries were selected based on a previous study and fabricated on the running surface of compression rings using a laser. Combination of coating and texturing not only reduced the running in time but also yielded the lowest frictional forces in an engine. This represent a 15% reduction in friction force when traditional rings were replaced by rings that have been textured and coated. The close second best behavior was observed when rings were coated and then textured, in higher speed difference between the rings sets with coating and texturing was reduced. It can be assumed that in higher speeds both ring set would share a friction force amount. Texturing and coating alone would also improve frictional behavior as well the friction test results showed that combination of coating and laser texturing yielded a reduction of 11–15% in the total friction between cylinder liner and piston assembly over a wide
speed range. Also noteworthy is the favorable results of 40 percent reduction in the break-in
duration.
Reference


Appendix A

Mat lab code for filtering the friction forces

```matlab
%% Import data from text file.
close all
clear all
clc

%% Initialize variables.
filename = 'C:\Users\aakbar1\Desktop\Trial\Trial_240rpm 4';
delimiter = '\t';
startRow = 3;

%% Format string for each line of text:
%    column1: double (%f)
%    column2: double (%f)
formatSpec = '%f%f'[^n\r]';
%% Open the text file.
fileID = fopen(filename,'r');
%% Read columns of data according to format string.
textscan(fileID, '%[^n\r]', startRow-1, 'ReturnOnError', 'false');
dataArray = textscan(fileID, formatSpec, 'Delimiter', delimiter, 'EmptyValue', NaN,'ReturnOnError', false);

%% Close the text file.
fclose(fileID);

%% Allocate imported array to column variable names
t1 = dataArray[7];
t = t1-t1(1);
x = dataArray[7];

%% Clear temporary variables
clearvars filename delimiter startRow formatSpec fileID dataArray ans;

%% Signal processing

% Step 1 Median filtering
y = medfilt1(x,8);

% Step 2 Low-pass filtering

% Fs = 8000;          % Sampling Frequency
% Fpass = 50;         % Passband Frequency
```
% Fstop = 100; % Stopband Frequency
% Apass = 1; % Passband Ripple (dB)
% Astop = 80; % Stopband Attenuation (dB)
% match = 'stopband'; % Band to match exactly
% % Construct an FDESIGN object and call its BUTTER method.
% h = fdesign.lowpass(Fpass, Fstop, Apass, Astop, Fs);
% hd = design(h, 'butter', 'MatchExactly', match);
% y1 = filtfilt(hd.sosMatrix, hd.ScaleValues, y);

h1 = fdesign.lowpass('Fp,Fst,Ap,Ast', 0.01, 0.2, 1, 80);
hd1 = design(h1, 'butter', 'MatchExactly', 'stopband');
y1 = filtfilt(hd1.sosMatrix, hd1.ScaleValues, y);

figure
plot(t(1:4000), y(1:4000), t(1:4000), y1(1:4000), legend('Filtered Signal'),
legend('Original Signal'))

% Test Results and Export
a0 = mean(y1);
y2 = y1 - a0;
y = y - a0;
friction = mean(abs(y2));

D = zeros(10000, 4);
D(:,1) = t(90001:100000);
D(:,2) = y(90001:100000);
D(:,3) = y2(90001:100000);
D(1,4) = friction;

filename2 = 'Trial_360rpm.xls';
xlswrite(filename2, D);
Vita

Amirabbas Akbarzadeh grew up in Islamic republic of Iran and received his bachelor’s degree in December 2013 at the Isfahan University of Technology in Mechanical engineering with a concentration in Manufacturing and tribology. He is a candidate to receive his Master’s Degree in December 2017 and plan to continue his education in the field of transportation engineering.