1 INTRODUCTION

1.1 HISTORICAL BACKGROUND TO PISTON RINGS AND HÄGGLUNDS HYDRAULIC MOTORS

Let us take a step back in time, and examine the historical development of the piston ring. In the early days of humankind, making the drawing of water more efficient was an important goal. In 1556, the Renaissance author Agricola published a detailed account of the piston pumps that miners were using to raise water. He referred to the use of metal discs and leather washers, and to the use of leather to prevent friction on pump rods. The rapidly expanding urban population of the time led to a great need for pumps to raise water. The first waterwheel-driven pump was installed in London in 1582. By the end of the seventeenth century, the serious problems the mining industry in England was beginning to experience with water accumulation in deeper shafts provided the incentive to start using steam power to drive reciprocating engines. Since then, reciprocating engines have continued to develop, and there is no foreseeable end to their use and continued improvement.

Over the years, one particular problem has prompted extensive development efforts, namely, fluid leakage between the piston and the cylinder bore. This problem occurs in many types of machinery, water pumps, combustion engines, air compressors, hydraulic motors, hydraulic pumps, and cylinders. At first, attempts were made to have an extremely narrow gap, but such a sealing method had very low efficiency. It proved impossible to manufacture a piston and bore with small enough tolerances to ensure low leakage. The solution was to isolate the sealing function and make a separate machine element, the piston ring, that could better conform to the contact surface of the cylinder bore or cylinder liner. The first attempt to make such a ring took place in 1774, when rope packing was used in a steam engine to make a seal that increased the thermal efficiency of the engine to 1.4%. The next step was to make the piston ring out of metal Dowson [1]. The internal combustion (IC) engine, which operates at much higher pressures and temperatures than a steam engine does, was patented in 1860 by Etienne Lenoir. Development of this type of engine prompted the use of ferrous material for the rings, the mechanical properties of which are less affected by high temperatures. Since then, piston rings have commonly been made of cast iron or steel. The ongoing effort to improve piston rings has led to the production of numerous scientific papers, since at least 1889 (Hersey [2]). There is still considerable research interest in increasing our understanding of tribological phenomena occurring between piston rings and cylinder liners and piston in such places as internal combustion engines (Andersson et al. [3]). Performance improved early on because of improved form and reduced waviness of the cylinder bore. Later, the degree of surface roughness was improved by the development of honing technique. The cylinder bore in internal combustion engines has been honed for almost a century, and improved plateau-honing technique has been in use for the last half century (Suzuki [4]). Plateau honing produces a flat land surface between a cross-hatched pattern of grooves. The grooves or valleys serve primarily to retain oil. To reduce the amount of retained oil and thereby reduce oil consumption, there is a trend toward the use of a land surface between the valleys having a higher bearing capacity—which also implies an improved scuffing resistance. Thus the valleys became shallower and narrower and the land surface becomes smoother (Lenthall [5]). What forces are driving the improvement of the function and performance of piston rings in IC engines, currently used in the production of roughly 50
million automobiles every year? The main driving forces are certainly stricter environmental regulations and concurrent demands for improvements such as less wear and friction loss. What about piston seals or piston rings in Hägglunds hydraulic motors? A huge number of processes around require the input of mechanical work, torque, and rotation to function. Mechanical work produced by machines depends on the principles of how they convert energy. In an IC engine, chemical energy, conserved in the fuel, is released as heat in a thermodynamic process so as to accomplish mechanical work, torque, and rotation. Another usual way to convert energy to mechanical work is by means of an electric motor. Another common way is to convert hydrostatic to mechanical energy by means of a hydrostatic transmission of the radial piston type, such as the Hägglunds hydraulic motor, in which torque and rotation are produced from hydrostatic pressure and flow. The complete drive system consists of a cabinet, containing an electric motor driving a hydraulic pump, an oil tank, and filter and control equipment, connected by pipes and hoses to the hydraulic motor (see Figure 1).

![Figure 1: The complete Hägglunds drive system.](image)

The design of the radial piston hydraulic motor begins in 1957 with a patent for an IC radial piston engine using diesel fuel (Bergström and Omnell [6]). This patent was bought by Hägglunds, and the operating principle was adapted from chemical energy conversion to that of a hydrostatic transmission. The first hydraulic motor prototype was laboratory tested in 1959; it incorporated new piston assembly concepts, for transmitting the tangential force producing the torque (see Figure 2).
This newly developed hydraulic motor, the Viking model, was adapted to meet the basic requirements for the smooth and effective operation of winches in deck cranes (see Figure 3).
To seal off the high hydrostatic pressure between piston and cylinder bore a long and narrow gap is commonly used (Ivantysyn and Ivantysynova [7]). The state of the art for hydrostatic pump and motor design involves having either a long or short piston guide. The advantages of a long over a short piston guide are lower leakage and providing side load support with less frictional loss. A long piston guide will also produce higher volumetric and mechanical efficiency than a short piston guide will. Pistons could have three possible types of gap geometry for seal off pressure, each of which contributes different to leakage; these are presented in cross section in Figure 4.

![Figure 4: Cross sections of gap geometries for different piston sealing designs: A depicts a fully eccentric gap design, B depicts a rectangular gap design, and C depicts a concentric gap design. Dimension e represents eccentricity.](image)

Design A, which is of a solid piston that tilts in the cylinder bore to support side loads, has the highest leakage. Design B is of a slit piston ring, commonly used in internal combustion engines, in which the slit has almost a rectangular cross section, the width being the circumferential gap dimension and the gap being dictated by the clearance between piston and cylinder bore. Design B shows 3 times less leakage than design A does. In reality, however, the gap in design B is not open, but takes the form of a trap to reduce leakage, so the difference should be greater. In this simple comparison, the least leakage is achieved by design C, which is of a closed, i.e., un-slit, piston ring that can approach closely so as to be concentric within the cylinder bore, according to Skytte af Sättra [8]. Design C (see Figure 4) offers several advantages, such as:

- leakage, in terms of Poiseuille flow, that does not depend on the position in the stroke
- less leakage
- reduction of the size of the piston and of the whole hydraulic motor
- reduction in cost of the piston
To reduce leakage and obtain other benefits, one piston ring could be mounted on side of the piston, near the top (see Figure 5).

![Diagram of piston ring and hydraulic motor components](image)

**Figure 5: Piston ring on piston side near top.**

A seal on the piston has shown its usefulness since it was introduced to the market 1972 in the wheel-hub hydraulic motor from Hägglunds. Another nice feature of the piston ring is that it makes size reduction possible. The length of a piston can be shorter when a piston ring is used, and such a compact design can feasibly be fit into a wheel for direct torque transmission. This hydraulic motor was primarily designed to meet the requirements of mobile applications with intermittent duty. Hägglunds’ self-developed hydraulic motor, Marathon, was introduced to the market in 1984 to meet the technical demands of industrial use: i.e., continuous operation for 40,000 hours at the rated speed and rated torque. Continuous development led to the introduction of the piston ring in hydraulic motors. It started with the Marathon model in November 1991 and some years later in the Viking model. The latest developed industrial hydraulic motor with piston rings, Compact, was introduced to the market in 1994 (see Figure 6).
The Compact model hydraulic motor displays all the benefits of size reduction—as suggested by the name. Designs without piston rings, as constantly used for 32 years in Hägglunds hydraulic motors (excluding the wheel-hub type), have now became part of history.

With more than 100,000 Hägglunds hydraulic motors having been manufactured over the past 45 years, and put into service in a great variety of applications, the hydraulic drive systems have proven themselves to be good energy converters (see Figure 7).

**Figure 6:** The Hägglunds Compact model hydraulic motor in operation.

**Figure 7:** A variety of applications using Hägglunds complete hydraulic drive systems. From top left to bottom right, the applications are in the following industries: cruise ships, pulp & paper, sugar and bulk materials handling.
Ending this retrospective survey with a very general comparison of the characteristics of piston rings in hydraulic motors versus IC engines, as shown in Table 1.

Table 1: Comparison of piston ring characteristics in hydraulic motors versus IC engines.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Limit</th>
<th>Hägglunds hydraulic motor</th>
<th>Otto or Diesel engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid to seal</td>
<td></td>
<td>oil</td>
<td></td>
<td>gas &amp; oil</td>
</tr>
<tr>
<td>Pressure</td>
<td>MPa</td>
<td>max 35 continuous 42 peak</td>
<td>15–20</td>
<td></td>
</tr>
<tr>
<td>Type of service</td>
<td></td>
<td>Constant fast decaying</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Service life</td>
<td>[h]</td>
<td>40,000</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td>[m]</td>
<td>5·10⁸</td>
<td>1·10⁹</td>
<td></td>
</tr>
<tr>
<td>Velocity</td>
<td>m/s</td>
<td>max 6.3</td>
<td>20–40</td>
<td></td>
</tr>
<tr>
<td></td>
<td>min</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Temperature</td>
<td>°C</td>
<td>max 50</td>
<td>155</td>
<td></td>
</tr>
<tr>
<td>Oil</td>
<td></td>
<td>Shell Tellus S 68</td>
<td>Mobil 1</td>
<td></td>
</tr>
<tr>
<td>Kin. Viscosity</td>
<td>cSt</td>
<td>43</td>
<td>4.3</td>
<td></td>
</tr>
</tbody>
</table>

The above values for an IC engine are estimated. The two piston ring applications have little in common, aside from the main function of sealing properly and the characteristic reciprocating motion that gives zero sliding velocity at the end positions.

1.2 OPERATION OF A RADIAL PISTON HYDRAULIC MOTOR

The Hägglunds hydraulic motor has radial pistons with constant displacement, as seen in Figure 8.
This type of hydraulic motor is used in low-speed, high-torque applications and is called a low-speed, high-torque (LSHT) hydraulic motor. It has an even number of pistons that operate alternately under working pressure and charge pressure in a reciprocating motion. The hydrostatic pressure and oil flow correspond to the actual torque and speed, respectively. When torque and rotational speed are produced, as in the driving mode, the pistons move outwards under working pressure and inwards under charge pressure. The hydrostatic pressure is constant, but differs in magnitude between the inward and outward movements. Either the casing is stationary, which means that the cylinder block and piston assembly rotate, or the reverse is the case. At both end positions of a stroke, the piston tilts $2\beta$ due to the changed direction of the tangential force and clearance (see Figure 9).
**Figure 9:** Piston movement during a stroke in driving mode. The piston moves outward under high pressure and inward under charge pressure.

As seen in Figure 10, the piston ring surfaces in contact with the cylinder bore and the piston groove are called the face and the flank, respectively. The sliding distance along the flank is 0.26 to 1.3% of the sliding distance along the face; this range applies to all different motor types.

**Figure 10:** Contacts on the piston ring face and flank.
1.3 PURPOSE

This research examines the type of piston ring wear occurring in hydraulic motors and how it progresses, depending on different operational parameters. This will help us better understand the wear and friction behavior of piston rings. A reliable simulation model for use in calculating wear under actual operating conditions will be formulated, based on the findings of the study. Laboratory testing conducted using special test-rig equipment was also validated using an actual hydraulic motor. Wear was gauged by measuring loss of mass, change in form, and change in surface roughness. The ultimate goal of this research is to enable the lifetime of piston rings to exceed, with a certain margin of safety, the general service life of a hydraulic motor, as determined by the rolling contact fatigue life of the bearings.

1.4 SUMMARY OF THE RESEARCH

The research was mostly experimental, but ended with the formulation of a wear model. As a starting point, we characterized the piston ring wear occurring in a model test in sliding test-rig equipment, and the results were compared with those of the characterization of piston ring wear occurring during actual motor operation. This was followed by a model test in the sliding test-rig equipment to achieve the initial wear and monitor the process by which it occurred. Then, we examined wear in an actual hydraulic motor running for an extended time.

To gain an understanding of wear on the flank contact, the wear interaction between the piston ring flank and the piston groove was reproduced in a model test using tilting test-rig equipment. Next, the cylinder bore surfaces were evaluated in a texture test performed in a pin-on-disc machine. The pin-on-disc machine was also used to test cylinder bore surfaces made of actual cylinder bore material, to determine the influence of coatings, oil additives, and surface roughness. Also, we successfully measured oil film thickness using an ultrasonic technique. Finally, an experimental study was conducted of piston ring face wear due to different surface properties of the cylinder bore in the sliding movement test-rig.

1.5 CONTRIBUTING WORKS

While the present research was being conducted, the following, related M.Sc. research was also being performed:

- Studying wear caused by the tilting movement of the piston: the experiment was performed in the tilting movement test-rig, and the contact examined was between the piston ring flank and the piston groove (Byström and Vukovic [9]).
- Evaluating the scuffing resistance of actual cylinder bore material: the experiment was carried out as a model test using the scuffing lathe test-rig (Törnqvist [10]).
- Oil film thickness measurement using capacitive gauge technique: the experiment was performed in the sliding movement test-rig.
2 ABOUT LUBRICATION, FRICTION AND WEAR

It may be incorrect to distinguish three objects—lubrication, friction, and wear—which are so intimately related; however, I will maintain the distinction to make the discussion easier to express and grasp.

2.1 LUBRICATION

What is lubrication? A normal and simple definition is as follows: it is the introduction of a lubricant into the solid contact between two surfaces in relative motion, in order to reduce friction and prevent damage to the surfaces. According to Blau [11], the practice of lubrication science involves the proper design of interfacial geometry, the selection of running conditions, and the selection of lubricant compositions that work best under those conditions. Or expressed more popularly, lubrication is helping bodies in contact to operate in the most favorable way in the desired state. According to the Organisation for Economic Co-operation and Development (OECD), International Research Group (IRG) on Wear of Engineering Materials, a lubricant is defined as follows:

Any substance interposed between two surfaces in relative motion for the purpose of reducing the friction and wear between them.

So, the lubricant has to be applied intentionally, otherwise the substance is merely considered to be lubricious, like, for example, oxides formed on a surface or water condensed from humid air.

A well-known description of lubricated contacts was formulated by Stribeck for a journal bearing. Its normally depicted as a Stribeck curve, with the coefficient of friction set against velocity, viscosity, and load (also called the Hersey number). Figure 11 presents such a Stribeck curve, in which the coefficient of friction is set against the dimensionless film parameter, but with the behavior of a hydraulic motor indicated by the boldface portion of the curve. The first boldface portion for Λ up to 0.6 is the sliding motion in the stroke and the second portion for Λ up to 2 is the squeeze motion at the end positions.

![Stribeck curve](image)

*Figure 11: Stribeck curve, i.e., coefficient of friction versus film thickness, with hydraulic motor operation indicated by the boldface portion of the line. Λ up to 0.6 represent the sliding motion in the stroke and Λ up to 2 represent the squeeze motion at the end positions.*
This behavior of cyclic movement from $\Lambda = 0$ to $\Lambda = \Lambda_{\text{max}}$ represents poor operating conditions. Other types of machine elements, such as roller and journal bearings in IC engines, operate almost all the time under elastohydrodynamic or hydrodynamic conditions, and briefly travel through mixed and boundary lubrication conditions at the start and stop of the cycle.

Many different parameters determine the state of lubrication of the contact between the piston ring face and the cylinder bore. Some of them are relative sliding velocity, operational mode, surface roughness, load, dynamic viscosity, temperature, and the hardness of the material in contact. The lubricating regions have different limits depending on whether the contact is rolling or sliding. For rolling element bearings, the $\Lambda$ value has been set along the abscissa. The dimensionless film thickness parameter, $\Lambda$, is the minimum oil film thickness divided by the composite surface roughness, as in Eq. 1.

$$\Lambda = \frac{h_{\text{min}}}{\sqrt{R_{q1}^2 + R_{q2}^2}}$$  \hspace{1cm} (1)

where $R_{q1}$ and $R_{q2}$ is the Root-Mean-Square value of the surfaces, see [12]. This has been generalized for rolling bearings as follows by Khonsari and Booser [13]:

- boundary lubrication for $\Lambda < 1$
- partial (mixed-film) lubrication for $1 < \Lambda < 3$
- elastohydrodynamic lubrication for $\Lambda > 3$

For hydrodynamic journal and thrust bearings, the following generalization can be used (according to Khonsari [13]):

- partial- or mixed-film and boundary lubrication for $\Lambda < 3–10$
- full-film lubrication for $\Lambda > 3–10$

Another rough estimate of the four important lubrication regimes has been formulated by Hamrock [14], as follows:

- boundary lubrication for $\Lambda < 1$
- partial lubrication for $1 \leq \Lambda < 5$
- elastohydrodynamic lubrication for $3 \leq \Lambda < 10$
- hydrodynamic lubrication for $5 \leq \Lambda < 100$

In modeling of wear of the sliding piston ring the following division of lubrication regimes will be used:

- boundary lubrication for $\Lambda \leq 1$
- mixed lubrication for $1 < \Lambda \leq 3$
- elastohydrodynamic lubrication for $\Lambda > 3$
Boundary lubrication occurs where the load is supported by surface-to-surface contact, elastic and plastic, and molecules from the lubricant are adsorbed on the surfaces. The opposite is the case when the load is supported by the lubricant, shearing a thick film, which separates the surfaces as in hydrodynamic lubrication. The minimum coefficient of friction is achieved when the supporting lubricant is reduced to the minimum film thickness without asperity contact, as in elastohydrodynamic lubrication. The partial or mixed region extends from conditions of no separative film, as in boundary lubrication, to a complete separative film as in elastohydrodynamic and hydrodynamic lubrication.

Lubrication regimes can also be described according to Johnson [15] and Dowson et al. [16]. According to this description there are four regimes in nonconformal contact influenced by two physical effects: elastic deformation of the solids in contact under a load, and the increase of fluid viscosity with pressure. Those regimes are defined as:

1. Rigid isoviscous (RIV): very little elastic deformation compared to the film thickness and pressure is too low to increase the viscosity significantly.
2. Rigid variable viscosity (RVV): the contact pressure is high enough to increase the fluid viscosity in the conjunction.
3. Elastic isoviscous (EIV): the elastic deformation is significant compared to the thickness of the separating fluid film, but the contact pressure in the conjunction is insufficient to increase the fluid viscosity.
4. Elastic variable viscosity (EVV): the elastic deformation of the solids is significant compared to the film thickness and the contact pressure is sufficient to increase the fluid viscosity.

The parameters are as follows:

\[ g_v = \frac{W}{U^{1.2}} \] is a dimensionless viscosity parameter that depends on the dimensionless parameters of load, \( W \), and speed, \( U \).

\[ g_e = G \left( \frac{W^3}{U} \right)^{1.2} \] is a dimensionless elasticity parameter that depends on the dimensionless parameters of load, \( W \), speed, \( U \), and materials, \( G \).

In a diagram, with the elasticity parameter along the abscissa and viscosity along the ordinate, we may discern four regions corresponding to these four regimes. Superimposed on this diagram is the performance of a typical hydraulic motor, with a hydrostatic high pressure of 35 MPa and different maximum velocities and radial gap (tolerances and elastic deformation of cylinder bore) calculated and presented, as in Figure 12.
Figure 12: Map of fluid film lubrication regimes: viscosity parameter, $g_v$, versus elasticity parameter, $g_e$. The different regions represent the rigid isoviscous (RIV), rigid variable viscosity (RVV), elastic isoviscous (EIV), and elastic variable viscosity (EVV) regimes. Typical hydraulic motor operation in driving mode for three gap heights and seven maximum sliding velocities.

The diagram depicts how the major part of typical hydraulic motor operation, using the mean radial gap and maximum velocities of 0.1, 1, and 3 m/s, lies in the RIV and EVV regimes, or close to the borders of these regimes. In particular, the end of the stroke is very much in the EVV region, while the middle of the stroke, when the velocity is highest, lies in the RIV regime region.

It would be preferable to operate in the RIV regime, in which the elastic deformation is small compared to the film thickness and the pressure is too low to increase the viscosity significantly.

2.2 FRICITION

Friction processes have long been part of our daily lives, even 100,000 years BC when mankind had recently discovered how to light and harness fires (Burakowski [17]). Early people would have noticed that high friction conditions, unlubricated by water, were good for generating heat or sparks. Mankind has at all times put great effort into restraining friction, minimizing it when it is wasteful and maximizing it when it is useful, and optimizing it where appropriate. It counteracts every relative movement between the surfaces of bodies. Friction represents a loss of energy; it generates traction force and heat, and is a source of wear and damage to contacting surfaces. The friction force under dry conditions follows Amontons’ basic laws (originally discovered by da Vinci):
• friction force is directly proportional to normal applied load by coefficient of friction as constant of proportionality.
• friction force is constant and independent of contact area
• static friction force is generally greater than dynamic friction force

In a lubricated contact, the model of friction force comprises the adhesive, ploughing, hysteresis, and viscous components.

• Adhesion: This is the resistance to shear at the actual contact points, whether elastic or plastic deformed. The adhesion of sliding materials, according to the classical theory, cannot, on its own, model the coefficient of friction or explain experimental results. Material adhesion between metals, measured as the coefficient of friction, is up to 0.4 for identical metals without surface contaminants or oxide layers, and decreases according to the lubrication condition. Asperity deformation gives a coefficient of friction of up to 0.7 and is a major component of the static coefficient of friction. In the steady-state condition, asperity deformation only contributes partially if new asperities are generated by a wear process.
• Ploughing: This is the resistance to surface asperities or entrapped particles that are ploughing, by means of plastic deformation, into the counter-surface, and thus producing grooves, chips, and loose wear debris. The friction coefficient of ploughing by wear particles can be as high as 0.4 in a typical situation, depending on the depth of penetration.
• Hysteresis: This is the loss when a stressed moving contact makes a cycle on not perfectly elastic materials.
• Viscous: This component is shearing a viscous fluid film.

The value of the friction force varies greatly depending on material combinations, lubrication conditions, and other particular circumstances; the coefficient of friction is thus a parameter representing the complete system, not only the materials.

2.3 WEAR

Wear is a process that causes surface damage or the removal of material from one or both of the interacting contact surfaces in relative motion. The ultimate goal of this specific study is to minimize wear. To keep the piston rings functioning well over the lifetime of the hydraulic motor one must minimize wear, thus ensuring that only small and acceptable changes occur, first in the surface roughness of and later in the form of the piston rings. Another reason for minimizing piston ring wear is that is reduces the risk that wear particles from the piston ring contacts may damage other machine elements in the motor.

Wear is often expressed as the change of a material’s shape or mass (normally expressed as loss of material) at a given operational time. Wear could be classified in two ranges, as described in Williams [18]:
• Mild wear – produces extremely smooth surfaces, possibly smoother than the original surfaces
• Severe wear – produces surfaces with ploughing marks, much rougher than the original surfaces

Wear is not the only process that causes changes in surface roughness and form. Plastic deformation of the surfaces in contact is also possible and sometimes probable. Initial local deformations of asperities are often treated as plastic.

Different types of wear for a sliding contact are:
• Adhesive: when the junction of two materials forming something like a "cold weld" is broken, and the wear debris loosens or becomes adhesively attached to one surface.
• Abrasive: when a harder surface removes material from a softer one; this can happen via two mechanisms,
  - 2-body: when one surface or the particles embedded in it are harder than the other surface
  - 3-body: when a particle harder than one surface is freely moving (sliding and rolling) within the contact
• Oxidational wear: when the wear of the surfaces is within the thickness of the oxide for every pass of a contact spot; this can happen via two mechanisms,
  - mild adhesive wear: when the oxide regrowth is greater than the wear rate or called adhesive wear in low state
  - severe adhesive wear: when the layer of material removed is thicker than the oxide layer and when the wear rate is higher than the regrowth rate; this type of wear comes after mild adhesive wear.
• Atomistic wear is extremely mild wear in which material is removed atom by atom from the material lattice.

Piston rings should display mild wear at most. The mild wear process is sometimes explained as a process in which the rubbing asperity surfaces are oxidized to create an oxide film, followed by new interaction with another asperity or simply by an elastic recovery of the deformed asperity. Although many wear models and theories are described in the literature, the wear process is complex and difficult to capture. Normally, more than one mechanism is active in any given contact. In boundary lubricated contacts, the wear process is strongly influenced by such things as lubricant additives, surface treatments, and coatings.
3 METHODS USED TO CHARACTERIZE FRICTION AND WEAR

The coefficient of friction was calculated as the tangential force divided by the force of the normal load. The tangential force was continuously measured during testing in the pin-on-disc machine. The wear on the piston rings was observed at different inspection points, using the quantitative and qualitative methods described here. Mass loss was measured by weighing the rings before test and at different inspection points. The weight loss value obtained gives a mean value for the wear on a piston ring, but no information about its distribution. In order to determine the wear distribution, the surface topography and form of the rings were measured. An atomic force microscope (AFM) was also useful in that it enabled quantification of what was seen with the scanning electron microscope (SEM) and also gave some insight into the wear mechanism at work. Each ring was carefully cleaned by ultrasonic washing before it was measured and mounted in the test equipment.

Direct observations were made as follows:

- Unassisted visual observation: Visual examination of the surface gave a valuable initial idea of the wear distribution. As well, significant signs, such as the oil trace on a disc after contact with a sliding ball under very lean lubricant conditions, could be easily and quickly observed.
- Touch: Both skin and nails of the fingers were used to assess, for example, the texture of the surface finish, whether new or worn.
- Listening: For example, listening to the sound of a sliding contact operating close to seizure.

Qualitative observations were made as follows:

- Light optical microscopy (LOM): Examination of the surface with the LOM gave a valuable idea of the wear distribution. It also allowed us to document, photographically, the effects of wear.
- Scanning electron microscopy: The surfaces were examined more closely using an SEM, which gave some idea of the actual wear mechanism. It also allowed us to document, photographically, in greater detail the effects of wear and its processes.

Quantitative measurements were made as follows:

- Gravimetric weighing of mass: A Sartorius 2604 (Sartorius AG, Goettingen, Germany) balance with an accuracy of 0.00006% was used to measure the mass (see Figure 13). The equipment is traceable to national standards. Every piston ring was weighed 5 times; a master piston ring serving as the weight reference was weighed before and after each time the piston ring was weighed. By calculating the standard deviation between the weights of these two rings at each weighing, a standard deviation of 0.000110 g based on the pooled sample variance was achieved. This value defines the precision of the mass measuring process.
Measurement of surface roughness and profile form: A Rank Taylor Hobson Form Talysurf Mk 1 (Taylor Hobson Ltd., Leicester, England) contacting device was used for the 2-D measurement of form and surface roughness. This equipment is traceable to national standards. The stylus tip radius was 2 µm. The surface roughness measurement had a cut-off length of 0.08 mm due to restrictions on the length that could be measured. Measurements were made at each inspection point on the contacting surfaces of the piston ring. The piston ring was mounted on a special apparatus that allowed it to be turned to 16 distinct positions, equally spaced 22.5° apart (see Figure 14). Sixteen distinct measurements could thus be made on the outer surface; however, only 8 measurements could be made on the side flank, the latter being spaced 45° apart, because of the design of the special apparatus. The repeatability of the form measurements was evaluated by making 6 independent form measurements at one position on a master piston ring used as the form reference. The MATLAB program was used to plot graphs from the raw measurement data, and then evaluate the difference in section area between two graphs; this revealed that the mean value of the difference was 0.00018 mm², with a standard deviation of 0.000132 mm². This area was swept through one complete turn to calculate the wear volume, which, in combination with the density of the steel, gave a mean value of 0.00033 g for the difference in mass, with a standard deviation of 0.000243 g. The form measurement process has higher scatter than the mass measurement process does; however, form measurement of the piston ring permits study of the geometric location of mass loss due to wear.
Figure 14: Surface roughness and form measurements were made using a Rank Taylor Hobson Form Talysurf, a special apparatus for specifically positioning the piston ring for measurement.

- Measurement of surface roughness with an atomic force microscope (AFM): A DTE Dual Scope DS5 (Danish Micro Engineering, Herlev, Denmark) mounted in an optical microscope was used to examine nano-scale surface roughness on the outer surface of new and worn piston rings (see Figure 15). The equipment had a vertical resolution of 0.1 nm and a surface wavelength space of 1 nm; this allowed us to undertake detailed study of surface wear mechanisms and to precisely locate the surface spot investigated with a precision of 1 µm.

Figure 15: AFM mounted in an optical microscope
Measurement of surface roughness with a non-contacting device: An optical profiler WYKO NT2000 (Veeco Instruments Inc., Woodbury New York, USA) using white light interferometry gives surface roughness and form values evaluated over an area of the surface. As well, comparison of before- and after-wear test results allows the worn volume to be calculated.

The coefficient of friction was determined in a model test with a pin-on-disc machine. The coefficient of friction was calculated as the tangential force divided by the force of the normal load. A load cell was used to measure the tangential force, and the normal load was applied by dead weights (see Figure 16).

*Figure 16: Pin-on-disc machine. A load cell, which contacts the dead weight loaded lever, records the tangential friction force.*
4 EXPERIMENTAL EQUIPMENT

The aim of this research has been to isolate the wear occurring either on the face or the flank of the piston ring. In testing actual hydraulic motors (HM), wear is found to be simultaneously present on both contacts; the wear debris from one contact surface could thus impinge upon wear on the other surface, making it difficult to isolate wear components. To conduct simpler model testing, however, the tribometer, or pin-on-disc machine (POD), is very useful. Ease of handling makes it most appropriate for screening tests, for example, of different surface roughness properties.

To reproduce, in simplified tests of component, the wear that happens during actual hydraulic motor operation, two different test rigs were developed and built: the sliding movement test rig (SMTR) and the tilting movement test rig (TMTR). The sliding movement test rig reproduces wear occurring in the contact between the piston ring and cylinder bore. The tilting movement test rig reproduces wear occurring in the contact between the piston ring and piston groove.

4.1 THE SLIDING MOVEMENT TEST RIG

The first piece of equipment, the sliding movement test rig, was especially designed to simulate wear in the contact between the piston ring face and the cylinder bore. This equipment, see Olsson [19] and Sjödin [20], has two separate cylinders, the reciprocating motion being produced by an external electrical motor, equipped with a gearbox, via a crankshaft (see Figure 17).

Figure 17: Sliding movement test rig (SMTR). In the magnified view, the piston rings and the direction of reciprocating movement are indicated.
The cylinder bore sections were machined from an actual cylinder block. The piston was machined from an actual piston blank, but is unique in having two piston rings, one at each end. To balance the force on the piston, the hydrostatic high pressure inlet is in the middle of the cylinder bore. The cylinder bores are oriented horizontally, and gravity applies unidirectional mass force on the piston rings and pistons. Both piston rings are intended to perform in both driving and braking modes, depending on the direction of movement. The piston groove supporting the piston ring flank is flat and in the normal direction to the cylinder bore.

The motion in the SMTR is nearly sinusoidal, and differs slightly from that of the hydraulic motor, especially in the first half of the stroke (see Figure 18).

![Comparison between the sliding velocities in a HM and in the SMTR, and a pure sinusoidal motion.](image)

The maximum velocity is used as reference when the motor velocity is simulated in the SMTR. Typical test data are: piston ring outer diameter, 75 or 68 mm; stroke, 43 mm; hydrostatic high pressure, constant up to 35 MPa; oil temperature, 50°C. The parameters to be controlled during the test were working pressure, piston velocity, friction force, temperature, oil flow from leakage, and time.

As well, oil film thickness was measured ultrasonically in this test rig. The transducer was mounted on the horizontal top side of the cylinder, in the middle of the stroke (see Figure 19).
4.2 THE TILTING MOVEMENT TEST RIG

The second piece of equipment was specially designed to simulate wear in the contact between the piston ring flank and the piston groove, see Byström and Vukovic [9] and Sjödin and Olofsson [22]. An axial movement, an external electrical motor produces rotating movement via a gearbox to an eccentric shaft and a push-pull rod in the centre of the cylinder block, produce the tilting of the pistons (see Figure 20). Twelve separate cylinders and pistons are used and twenty-four piston rings are tested at the same time. The tilting movement of the piston in the test rig differs in some respects from that in an actual hydraulic motor. The cylinder bores are oriented horizontally, and gravity thus exerts a unidirectional mass force on the piston rings and pistons. The hydrostatic pressure on the rig is constant, and the tilting movement is the result of an external force; in contrast, the movement in a hydraulic motor results from changes in hydrostatic pressure from high to low and vice versa. The piston is force balanced by the placement of the pressure inlet in the middle of the cylinder bore. The cylinder bores were manufactured from actual cylinder blocks and the pistons from actual piston blanks. Each piston has two piston rings, one at each end. The tilting movement is only in the vertical direction and the angular amplitude is $\alpha$. The tilting of the pistons was determined by the play between the cylinder bore and the supporting piston land outside the piston rings.

Figure 19: Schematic diagram of the transducer location, showing the focusing of the sound wave through the water bath and cylinder wall onto the piston ring.
Figure 20: Tilting movement test rig (TMTR). In the magnified view the piston rings and the direction of tilting movement (angle $2\alpha$) are indicated. The tilting movement test rig contains twelve pistons.

The TMTR allowed different piston groove designs to be evaluated. One type of groove had a short support plane, length A, with the rest of the surface being inclined, by angle B, from the piston ring flank (see Figure 21).

Figure 21: Design parameters of a piston groove.
4.3 THE PIN-ON-DISC MACHINE

The pin-on-disc machine is a conventional tribometer (VTT Industrial Systems, Espoo, Finland), with a horizontal rotating disc and a dead weight-loaded pin (see Figure 22).

*Figure 22: Pin-on-disc machine.*

The pin-on-disc machine was installed in a climate room where temperature and humidity could be controlled. The test specimens used were standard bearing components (see Figure 23).

*Figure 23: Pin holder with a mounted ball displaying a circular wear scar. On top of the pin holder lies the disc, a bearing washer with a dark trace of oil in the sliding trace.*

The discs and pins used in testing were also manufactured from the actual materials used in a hydraulic motor.
4.4 THE HYDRAULIC MOTOR

Testing was performed on an actual hydraulic motor at the Hägglunds Drives hydraulic laboratory in Mellansel. The type of motor tested contains 16 piston rings and pistons which reciprocate 20 times every revolution (see Figure 24).

Tests using actual hydraulic motors were conducted in order to verify the results obtained from the test rigs. The results obtained from the hydraulic motor in field operation were similar to those obtained from the test rig, except that the magnitude of the hydrostatic pressure in the test rig did not change with different sliding directions. Hydrostatic pressure thus remained constant throughout the stroke cycle during the test rig testing; consequently, the change in the tangential contact force will differ from that occurring in actual motors in operation.

4.5 TEST PROGRAM

The planned test program included two series of tests using the SMTR. The first series examined the running-in behavior of the piston rings. The second series involved a long-duration test designed to replicate the first series, and to be long enough so the results could be compared with those of a long-duration test of an actual hydraulic motor. The hydraulic motor test was of fairly long duration, in order to provide reference data for the SMTR testing.

SMTR testing was conducted at the Department of Machine Design, KTH, in Stockholm. Testing using the TMTR and the hydraulic motor were conducted at the Hägglunds Drives laboratory in Mellansel. The mass, surface topography, and form of the piston rings were measured at the Department of Machine Design, KTH. The piston grooves and cylinder bores were also subject to observation and measurement. Some of the observations were made using the scanning electron microscope at the Materials Science Division, Uppsala University, Uppsala.
The following tests were conducted to create wear, except for test number 6 which measured oil film thickness:

1. Sliding movement test  
   Run time, 16 h
2. Sliding movement test  
   Run time, 516 h
3. Hydraulic motor test  
   Run time, 2046 h
4. Tilting movement test  
   Run time, 513 h
5. Pin-on-disc test, no. 1  
   Run time, 3 h
6. Sliding movement test  
6. Sliding movement test  
7. Pin-on-disc test, no. 2  
   Run time, 1 h
8. Pin-on-disc test, no. 3  
   Run time, 128 min
9. Sliding movement test  
   Run time, 128 h

For all tests, the intervals between each inspection and measurement of the studied surfaces are listed in Table 2.

### Table 2: Intervals, in sliding distance [m], at inspections during all tests.

<table>
<thead>
<tr>
<th>Paper</th>
<th>A</th>
<th>A</th>
<th>B</th>
<th>G</th>
<th>C</th>
<th>D</th>
<th>F</th>
<th>H</th>
<th>G</th>
</tr>
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<td>SMTR</td>
<td>HM</td>
<td>SMTR</td>
<td>SMTR</td>
<td>HM</td>
<td>TMTR</td>
<td>POD</td>
<td>POD</td>
<td>POD</td>
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<tr>
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<td>4</td>
<td>5</td>
<td>7</td>
<td>8</td>
<td>9</td>
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<td>closed</td>
<td>closed</td>
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<td>37.5</td>
<td>37.5</td>
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<td>Pin radius [mm]</td>
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<td>3 h</td>
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<td>26,980</td>
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<td>220 h</td>
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<td>512 h</td>
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<td>102,790</td>
<td>102,790</td>
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<td>513 h</td>
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<td>2,046 h</td>
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<td>10,700 h</td>
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<td>5.48⋅10^6</td>
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</table>
5 RESULTS

One way to represent wear is to plot the wear coefficient versus sliding distance. The wear coefficient is the measured loss of mass divided by the sliding distance, density, and force. Results of all the tests are presented in Figure 25. The diagram, which uses a logarithmic ordinate and abscissa, shows that the highest rate of wear occurs early on in the sliding distance, at a distance of approximately 10 to 100 m. Note that the required calculated service life of a hydraulic motor is $4 \cdot 10^4$ h, which corresponds to a sliding distance of $5 \cdot 10^8$ m.

The wear coefficient measured using the SMTR, in the last point in test 2, is in good agreement with the wear coefficient measured in a hydraulic motor, as seen in the middle point in test 4. The wear coefficient was low in the hydraulic motor, even though it experiences wear on two contacts, instead of one as in the other tests. Testing by means of a simplified test of components or a model test seems conservative in terms of showing higher wear coefficients compared to wear coefficients in the real hydraulic motor. It is desirable to achieve lower wear coefficients in real hydraulic motor operation. In actual tests, the wear coefficients are over a four- to five-decade span, with a general trend toward decreasing wear with increasing sliding distance. Changing from a coated, grey cast iron pin specimen, to a coated steel ball decreases the wear coefficient by the equivalent of at least one decade, as seen in the bottom position in tests 7 and 8. A polished steel disc surface in contact with a steel ball has a wear coefficient close to that of a roller-burnished disc surface in contact with a coated steel ball, as seen in the bottom point in test 5 and the top point in test 8. A roller-
burnished cylinder bore surface produces a piston ring wear coefficient in the range of $1 \cdot 10^{-18}$ to $4 \cdot 10^{-19}$ $m^2/N$ as seen in test 2 and 4, at a sliding distance of $25 \cdot 10^3$ to $4 \cdot 10^6$ m.

Another way to observe wear is to measure changes in form. Accordingly, two different designs (of piston grooves, with and without a support plane) used in the factorial analysis of the piston groove were compared in the tilting movement test rig. In the case of a piston groove without a support plane, the main form of change was wear of asperities on the outer circumference. This form change is clearly noticeable and is uniform around the entire circumference of the piston groove. The flank surface of the piston rings showed no measurable change in form. In the case of a piston groove with a support plane, the form change is significant and the surface is worn smooth (see Figure 26). In this case, too, the flank surface of the piston ring showed no measurable change in form (see Figure 27).

![Figure 26: Change of form of groove, piston 6B, position 1](image)
Piston ring wear, in a real hydraulic motor, does not only affect the surface roughness of the face and flank of the piston ring, but also produces a form change. Calculating the area of the worn surface of a section (see Figures 28–29) at eight locations, and multiplying this by the length of the sector, approximates the volume of wear. The volume of wear on the flank is ten times greater than that on the face. The total calculated, mass loss from form measurement, on the face and flank accounts for 90% of the weighed loss of mass.

Figure 27: Change of form of flank, piston ring 41, position 1
Figure 28: Change of form of flank, piston ring IC 5, position 5.

Figure 29: Change of form of face, piston ring IC 5, position 5.
This difference in volume of wear is mainly due to the tilting movement of the piston. The piston itself could be more affected by wear than the piston ring is—because the latter is made of harder material.

A worn face surface is shown in Figure 30. The sliding direction is horizontal, as shown by the abrasive scratches. The wear has not penetrated beneath the surface roughness, as traces of grinding marks are still visible in the vertical direction.

![Figure 30: SEM micrograph. Worn surface of the face. Grinding marks are in the horizontal direction, normal to sliding direction. Sliding rig test series no. 1, piston ring IC 1, position 1. Magnification 1000×.](image)

The results of the AFM measurements are presented in Figures 31 to 33. A new, unworn surface with grinding grooves has a top to bottom value of 1.24 µm, compared to that of a worn surface which is 0.46 µm. The representation of the unworn surface derived from the AFM measurements clearly shows the deep groove produced by the grinding operation, and also short- and long-wavelength waviness (see Figure 31). Comparing this image to that of the mildly worn surface, it can be seen that the bottoms of the valleys remain and that the waviness is shorter in wavelength between the grinding grooves (see Figures 32 and 33). There is also an abrasive groove that is not parallel to the sliding direction, which could be the trace of a third body ploughing between the two contacting surfaces (see Fig. 33).

The results of calculations of the surface roughness parameters of the face are presented in Table 3. The plasticity index (PSI) is far higher than 0.7 for both the unworn and worn piston rings, indicating that plastic deformation will take place in the contact. This can be clearly seen in Figure 34, where the material shows tongue-shaped plastic deformation in the direction of sliding. The change in the functional parameters indicates that the surface bearing
index ($S_{bi}$) experiences the least alteration; this is advantageous, as a larger $S_{bi}$ is a property of a good load-bearing capacity.

Table 3: Face surface roughness 3-D parameters of unworn and worn piston rings, no. IC 1, after 16 h.

<table>
<thead>
<tr>
<th></th>
<th>$S_a$</th>
<th>$S_q$</th>
<th>$S_{bi}$</th>
<th>$S_{ci}$</th>
<th>$S_{vi}$</th>
<th>$S_{dq}$</th>
<th>PSI=8.62-$S_{dq}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unworn</td>
<td>0.195</td>
<td>0.257</td>
<td>0.71</td>
<td>0.067</td>
<td>1.35</td>
<td>0.60</td>
<td>5.2</td>
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<tr>
<td>IC 1</td>
<td>0.060</td>
<td>0.075</td>
<td>0.38</td>
<td>0.022</td>
<td>0.259</td>
<td>0.38</td>
<td>3.3</td>
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</table>

Figure 31: Unworn piston ring face surface. Profile from left to right, in sliding direction.

Figure 32: Worn piston ring face surface. Profile from left to right, in sliding direction.
Figure 33: Worn piston ring face surface. Profile from top to bottom, perpendicular to sliding direction.

Figure 34: SEM micrograph. Worn surface of the face made of plastically deformed material. Grinding marks are in the horizontal direction, which is normal to the sliding direction. Hydraulic motor test, piston ring IC5, position 1. Magnification 5000×.

The appearance of the worn flank surface is shown in Figure 35. The mild wear has not penetrated the surface roughness, as traces of grinding marks are still visible. The wear mechanism on the flank, was observed to be similar in both the rig tests and the actual hydraulic motor. It can be seen that wear has produced plateaus in the deepest grooves in the profile of the worn surface (see Figure 36, in the middle of the image profile).
The wear on the flank is greatly influenced by the design of the piston groove. A piston groove without a support plane results in a surface on the piston ring flank that is similar to an unworn surface (see Figure 37). In such a case, the wear takes place on the tops of asperities in contact at the outermost diameter.

The results of calculations of the surface roughness parameters of the piston ring flank are presented in Table 4. The PSI is far higher than 0.7 for both the unworn and worn piston rings, indicating that plastic deformation will take place in the contact.

The changes in the functional parameters indicate that the SBI has experienced the smallest decrease; this is advantageous, as a higher Sbi is a property of good load-bearing capacity.

Table 4: Flank surface roughness 3-D parameters for unworn and worn piston rings, nos. IC43 and IC42, respectively.

<table>
<thead>
<tr>
<th></th>
<th>Sa</th>
<th>Sq</th>
<th>Sbi</th>
<th>Scai</th>
<th>Svi</th>
<th>Sdq</th>
<th>PSI=8.62⋅Sdq</th>
</tr>
</thead>
<tbody>
<tr>
<td>IC43</td>
<td>0.207</td>
<td>0.257</td>
<td>0.46</td>
<td>0.024</td>
<td>2.17</td>
<td>0.39</td>
<td>3.4</td>
</tr>
<tr>
<td>IC42</td>
<td>0.082</td>
<td>0.107</td>
<td>0.33</td>
<td>0.11</td>
<td>2.90</td>
<td>0.18</td>
<td>1.6</td>
</tr>
</tbody>
</table>

Figure 35: SEM micrograph. Worn surface of the flank. Hydraulic motor test, end of test, piston ring IC5, position 1. Sliding in horizontal direction. Magnification 1000×.
The friction and wear behavior of a surface, either with or without a specific texture, and grinding treatment or polish treatment, were investigated in tests of factorial designs. The two textures were made by laser ablation in commercial bearing component, such as washers. The worn surfaces on the disc surfaces after test where abrasive scratches are in the sliding direction (see Figure 38).
Figure 38: SEM micrograph of the pockets in washer surfaces of polished texture A (left) and B (right). Pocket width, in sliding direction from left to right, is 0.07 mm for both textures.

In Figure 39 the wear rate is presented as mass loss per distance slid for each type of surface texture and surface treatment.

![Wear rate as mass loss per sliding distance for different surface textures and surface treatments. Surface textures are; No texture (None); texture A (A); or texture B (B). Surface treatment are; Ground (Gr.) or polished (Pol.). Error bars represent ± one standard deviation.](image)

The lowest wear rate was displayed by the polished non-textured surface. All the polished surfaces have similar levels of variation in wear rate. However, surface texture B displays the lowest variation of all the non-polished surfaces. In each pair of surfaces, the polished surface displays the lower wear rate and variation in wear rate.
The worst case of coefficient of friction shown in Figure 40 is that with the highest mean value over the whole sliding distance; in contrast, the best case in Figure 41 is the one with the lowest value. In the worst case, the non-textured ground surface peaks to very high values about 0.7.

A noticed favorable result regarding the polished texture B is that the coefficient of friction tends to decrease a little bit more over time compared to the polished non-textured surface (see Figure 41).

A surface with pockets should be advantageous in contacts where the oil is prone to leaking and drainage, for example, at start-up after a long stationary period or during intermittent operation. Such operation occurs between the flank of the piston ring and the piston groove, where only small relative movement occurs at the ends of the stroke.

*Figure 40: Coefficients of friction for a none texture ground surface and for a ground surface with texture A. Worst case is highest mean of coefficient of friction.*
Figure 41: Coefficients of friction for a none textured polished surface and for a polished surface with texture B. Best case is lowest mean of coefficient of friction.

Other ways to obtain higher wear resistance and reduced friction loss are to put a hard coating on one of the contact surfaces, make the surfaces smoother, or use a suitable additive in the lubricant. This was done in test performed as factorial designs, in which the pins were made of actual piston blank material and the discs were made of actual cylinder bore material. The three lubricants were a base oil without additives, a base oil with an anti-wear additive, and a base oil with an extreme-pressure additive. The wear spots on the ball are shown in Figure 42, for the three pins and the two surfaces of the disc.

Pin wear improved the most when the surface changed from uncoated to coated with a base oil and an roller burnished surface (see Figure 43). The wear displayed a weak dependency on the type of coating, Me-C or Me-C:H; however, the Me-C:H coating produced a better wear improvement.
Figure 42: Worn contact spot on the pins. The least worn pin (upper row) was in contact with a lapped disc surface. The most worn pin (lower row) was in contact with a roller-burnished disc surface. The sliding direction is from the bottom to the top in the pictures. The size of the contact spot for the Me-C coated pin and the L disc is 0.7 mm in the sliding direction. L: lapped surface; RB: roller-burnished surface.

Figure 43: Ratio of wear coefficient for an uncoated pin compared to a coated pin, versus lubricant and disc surface. Coating on the pin are Me-C:H or Me-C. BO: Base oil without added additive; AW: base oil with added anti-wear additive; EP: base oil with added extreme-pressure additive; RB: roller burnished disc surface; L: lapped disc surface; and LM: laser melted disc surface.
The greatest improvement in performance, i.e., reduction in wear, was obtained by changing both the pin and surface at the same time (see Figure 44). An uncoated pin in contact with an roller burnished surface was replaced with a coated pin in contact with an lapped surface. This give the greatest improvement in wear, the wear coefficient being reduced by 500 times for a lubricant with an AW additive. Interestingly, the base oil lubricant produced a greater wear reduction than did the lubricant with an EP additive.

![Figure 44: Ratio of wear coefficient for an uncoated pin in contact with a roller-burnished disc surface, compared to a coated pin in contact with a lapped disc surface, versus lubricant. Coating on the pin are Me-C:H or Me-C. BO: Base oil without additive; AW: base oil with anti-wear additive; EP: base oil with extreme-pressure additive;](image)

The friction behavior displayed in Figure 45 indicates the detrimental sliding of the coating when the wear comes through the coating thickness. This also is seen irrespective of lubricant, so the benefits of a smoother surface, such as the lapped surface, are obvious.
Figure 45: Coefficient of friction versus sliding distance. The upper graph shows a worst case with a roller-burnished disc surface and a Me-C coated pin surface with AW lubricant. The coating is worn away. The lower graph shows a best case with a lapped disc surface and a Me-C coated pin surface with a BO lubricant. The coating is intact.

The wear of the piston ring face could be strongly dependant on the type of cylinder bore surfaces. The wear coefficients for five different cylinder bores are presented in Figure 46. The result for laser honed (LH) surface consist of two surfaces with different texture density; a fine and a coarse texture. The roller-burnished surface gave the lowest wear coefficient, of $4 \times 10^{-19}$ m$^2$/N.

This study found the wear reduction, in terms of the lowest wear coefficient, to be 99% when a roller burnished surface rather than a plateau honed surface was used. As well, a piston ring sliding against a polished honed surface, polish honed and laser melted surface, results in a significantly lower wear coefficient compared to when the ring slides against a plateau honed surface. However, a piston ring sliding against a laser honed surface results in no significant wear reduction compared to when it slides against a plateau honed surface.

One characteristic feature of the roller burnished surface was the low $R_{sk}$ value, i.e., the surface with fewest peaks and the most valleys. Another feature of the roller burnished surface was the low $R_{pk}/R_{vk}$ ratio, which indicates that this cylinder bore surface has the highest number of valleys relative to the number of peaks. This is favorable in terms of low wear, since the valleys can act as oil reservoirs. The roller burnished surface is most similar to
the piston ring surface with respect to the $R_{sk}$ parameter, which may be a beneficial parameter under boundary sliding contacts.

![Figure 46: Wear coefficient for the different cylinder bore surfaces. H: plateau honed; LH: laser honed; PH: polish honed; LM: laser melted; and RB: roller burnished cylinder bore surfaces. Error bars represent ± one standard deviation.](image)

In sliding contact like that which occurs between the piston ring and cylinder bore, the build-up of a separating oil film is of great interest. To enable future improvements in performance, a measurement of the actual parameter is essential. A successful attempt was made to measure the oil film thickness, between piston ring and cylinder bore, of the reciprocating piston ring using a stationary ultrasonic transducer. The film thickness remained at similar levels throughout most of the testing. Both increasing speed and reducing pressure caused a thicker oil film, though a great deal of scatter was observed. Figure 47 shows a series of film thickness measurements made under changing speed and pressure conditions.

![Figure 47: Oil film thickness between the piston ring and cylinder versus pressure at three different speeds.](image)
The oil film thickness could also be calculated by numerically solving the Reynold’s equation in one dimension for an infinitely long bearing approximation. The oil film thickness is determined when the forces, in radial direction, are in equilibrium. The active forces are the force on the inner diameter exerted by the hydrostatic pressure and the force exerted on the outer diameter by the Reynold’s pressure field. The oil film thickness is calculated for different conditions, such as, in the piston groove contact is the flank unaffected by friction or is the friction coefficient $\mu = 0.1$. As well, are different hydrostatic pressure loads on the flank side, the piston groove contact for the two piston groove designs, applied as a no-pressure load for design I (a flat groove), or as a load from a rectangular pressure field for design II (minor a flat groove and major a surface sloping away from the piston ring). The hydrostatic pressure is assumed to be 35 MPa and the kinematic viscosity 43 cSt. The friction always opposes the motion, so the conditions include the piston ring approaching the cylinder bore (the friction force cooperates with the Reynold’s force) or recess away from the cylinder bore (the friction force obstructs the Reynold’s force). This is depicted in Figure 48, where the zero friction curve is in the middle, with the curves for approach the cylinder bore or recess the cylinder bore on each side.

Figure 48: Calculated oil film thickness between the piston ring and cylinder bore versus sliding velocity under different loadings with the friction coefficient $\mu = 0.1$. The loads depend on the design of the piston groove as; design I has zero hydrostatic pressure on the flank while design II has hydrostatic pressure on the flank. Frictionless conditions, i.e., $\mu = 0$ are also depicted.

Sensitivity to the different conditions of the oil film thickness build-up is not particularly great. The calculated oil film thickness is small compared to a typical composite surface roughness value $R_q$ of 0.6 $\mu$m. For a typical hydraulic motor operating at approximately 0.5 m/s sliding velocity, the calculated oil film thickness is only a tenth of the composite surface roughness. The dimensionless film parameter, $\Lambda$, will be small and variable during the stroke.
In Figure 49 the dimensionless film parameter, $\Lambda$, is presented for a whole stroke at two different maximum sliding velocities.

![Diagram showing film parameter versus working stroke for two velocities (maximum velocity in mid stroke) in hydraulic motor operation. For a composite surface roughness $R_q$ of 0.6 µm.](image)

**Figure 49:** Film parameter versus working stroke for two velocities (maximum velocity in mid stroke) in hydraulic motor operation. For a composite surface roughness $R_q$ of 0.6 µm.

When the film parameter is equal to or greater than three times the composite roughness, the lubrication mode is defined as the state occurring when an oil film completely separates the solid surfaces. This state is present in elastohydrodynamic lubrication as well as in full film lubrication. When $\Lambda$ greater than 1 and less than 3, the lubrication mode is mixed lubrication and when $\Lambda$ is less than 1 a boundary lubrication condition is operating.

This case displays the highest $\Lambda$ value of 0.63 at a sliding velocity of 6 m/s, and that results in boundary lubrication for the stroke as a whole. At a sliding velocity of 0.6 m/s, the value of $\Lambda$ will be 0.13. Irrespective of velocity the lubricated piston ring face contact is boundary lubricated for the sliding in the stroke. The squeeze effect in the end positions will give a reasonable value of $\Lambda$ of approximately 2.
6 MODELING OF PISTON RING WEAR

The use of wear modeling allows less time to be spent on time-consuming experiments, especially in the early stages of developing new and existing components. Here I will adapt a very simple model of mild wear to simulate wear on a piston ring throughout the full service life of a hydraulic motor. This will allow comparisons of the wear that occurs under different surface roughness and coating conditions which could ultimately lead to positive improvements. The restrictions valid in this model are as follows:

- wear mechanism is independent of time
- surface roughness is independent of time
- hardness is constant throughout the worn depth.

Holm’s [23] commonly used model for wear in sliding contacts states that the wear volume is proportional to the load and the sliding distance, and inversely proportional to the hardness, according to

\[ V = Z \frac{F \cdot s}{H} \]  

(2)

where \( V \) is the total wear volume, \( s \) is the sliding distance, \( Z \) is a dimensionless wear coefficient, \( F \) is the normal load, and \( H \) is the hardness of the softer material.

Holm’s wear model is often referred to as Archard’s wear law (Archard [24]). If \( Z/H \) is replaced by the wear coefficient \( k \) [m²/N], we get

\[ V = k \cdot F \cdot s \]  

(3)

Essentially, in every stroke the velocity, which is variable, acts on the sliding distance, which is also variable; both velocity and sliding distance are variable because their behavior is determined by the geometry of the cam ring. Analysis is easier if we discretize the stroke into a manageable number of parts; hence, we will divide the stroke into nine parts, equal with respect to the turning angle of the hydraulic motor (see Figure 50).
Likewise, the sliding distance, $s$, is also discretized into nine parts (see Figure 51).

The wear coefficient, $k_i$, depends on the velocity of each discretized part of a stroke, $i=1, 2, \ldots, 9$. A simple model of the relationship between the sliding velocity and wear coefficient is assumed. At a velocity at which the film thickness is less than or equal to one times depth of the composite surface roughness, the contact is boundary lubricated and a wear coefficient factor is unity. If the film thickness is greater than one up to three times the depth of the composite roughness, the contact zone is mixed lubricated, and we can assume a linear gradient of the wear coefficient factor, from unity to zero. If the film thickness is greater than three times the depth of the composite roughness, the contact zone is full-film lubricated, and we can assume no actual contact and thus no wear, i.e., the wear coefficient factor is zero.
This can be graphically represented in a diagram, in which a wear coefficient factor takes the value of unity or zero or somewhere in-between, depending on the velocity for three different composite surface roughness (see Figure 52).

![Figure 52: Wear coefficient factor versus sliding velocity.](image)

The wear volume expressed by Eq. 3 represents a worn area on the top, or cap, of the piston ring face; this is a volume of wear that extends around the outer circumference of the ring (see Figure 53).

![Figure 53: Worn area on a cross section of the piston ring face. The worn area on the cap extends around the full outer circumference of the ring.](image)
The volume of a worn cap is
\[ V_{\text{cap}} = \frac{\pi l^3}{6} \]  (4)
which together with the equation to relate worn depth to length, \( l \), gives
\[ d = r - \sqrt{r^2 - \left(\frac{6V_{\text{cap}}}{\pi} \right)^{2/3}} \]  (5)
Then the worn cap volume is made equal to the worn volume in Eq. 3, and together with the sum over the discretized sliding distances gives the worn depth per stroke as
\[ d_{\text{stroke}} = r - \sqrt{r^2 - \left[ \frac{3kF_{\text{stroke}} \sum_{i=1}^{9} \varepsilon_i \kappa_i}{4\pi} \right]^{2/3}} \]  (6)
The simple model states that each stroke is similar to the first one, which indicates a linearly increasing volume. Thus, after the \( N \)th strokes or after a total sliding distance of \( Ns_{\text{stroke}} \) the depth will be
\[ d_{N} = r - \sqrt{r^2 - \left[ \frac{3NkF_{\text{stroke}} \sum_{i=1}^{9} \varepsilon_i \kappa_i}{4\pi} \right]^{2/3}} \]  (7)
where
\[ \varepsilon_i = 1 \quad \text{if} \quad U_i \leq U_{A=1} \]
\[ \varepsilon_i = 1 - \frac{U_i - U_{A=1}}{U_{A=3} - U_{A=1}} \quad \text{if} \quad U_{A=1} < U_i \leq U_{A=3} \]
\[ \varepsilon_i = 0 \quad \text{if} \quad U_i > U_{A=3} \]
Equation 7 is evaluated as depth of wear versus total sliding distance. The initial load condition is set assuming a gap of zero. When the depth of wear reaches 1 µm, a new lower appropriate load is applied. When the depth of wear reaches 2 µm, yet another new lower appropriate load is applied, and so on.

From an experiment, in the sliding movement test rig, the wear coefficient was determined for a piston ring sliding in a cylinder bore with a roller burnished surface, (see table 5). The piston rings were uncoated but the question is what could the wear coefficient be if the piston rings were coated?
Table 5. Wear coefficients for piston ring sliding against a roller burnished cylinder bore surface. Obtained from the sliding movement test rig.

<table>
<thead>
<tr>
<th></th>
<th>Sliding distance from 0 to 3.3 km [m^2/N]</th>
<th>Sliding distance from 3.3 to 25 km [m^2/N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>uncoated</td>
<td>2.1 \times 10^{-18}</td>
<td>4.0 \times 10^{-19}</td>
</tr>
<tr>
<td>coated</td>
<td>(k_1)</td>
<td>(k_2)</td>
</tr>
</tbody>
</table>

To make an estimation of the wear coefficient for a coated piston ring a pin-on-disc test was performed using both coated (with Me-C:H coating) and uncoated bearing balls as pins. A 12.5 mm radius ball was sliding at a constant average velocity of 0.05 m/s with a 100 N load on a disc of the same material as the cylinder bore, roller burnished to obtain a surface finish similar to that of the cylinder bore. The wear coefficients are presented in Table 6.

Table 6: Wear coefficients for balls sliding against a roller burnished disc surface. Obtained in test number 8.

<table>
<thead>
<tr>
<th></th>
<th>Sliding distance, from 48 to 384 m [m^2/N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>uncoated</td>
<td>1.16 \times 10^{-17}</td>
</tr>
<tr>
<td>coated</td>
<td>1.08 \times 10^{-18}</td>
</tr>
</tbody>
</table>

The wear coefficient for a coated piston ring is calculated by a proportional scaling from the wear coefficient of a coated and an uncoated ball in the pin-on-disc test. The quotient, \(q\), between a coated and an uncoated ball is

\[
q = \frac{1.08 \times 10^{-18}}{1.16 \times 10^{-17}} = 0.093
\]

The scaling from an uncoated piston ring to a coated piston ring is accomplished using quotient \(q\). A piston ring with a similar coating as the ball is assumed to have wear coefficients as follows:

\[
k_1 = q \times 2.1 \times 10^{-18} = 0.093 \times 2.1 \times 10^{-18} = 1.96 \times 10^{-19} \text{ [m}^2\text{/N]}.
\]

During the first 3300 m of sliding distance, the wear coefficient is \(k = k_1\). After sliding distance 3300 and on is the wear coefficient \(k = k_2\),

\[
k_2 = q \times 4.0 \times 10^{-19} = 0.093 \times 4.0 \times 10^{-19} = 3.72 \times 10^{-20} \text{ [m}^2\text{/N]}.
\]

A wear simulation using different values on composite surface roughness and kinematic viscosity and without and with a coating on the piston ring was performed. Calculated wear depth for a composite surface roughness \(R_q = 0.1 \mu m\) with kinematic viscosities 43 and 86 cSt and a coating on the piston ring is shown in Fig. 54. When the piston ring does not come in contact with the cylinder bore, that is when the curves level out, a wear coefficient of \(1 \times 10^{-20}\) is assumed. When the wear depth reaches 4 \(\mu m\), the coated piston is treated as if it were uncoated, as seen in the rapid increase of wear depth. The
gain in sliding distance by doubling the kinematic viscosity is 25% to reach a wear depth of 4 µm. A considerable gain is achieved when the piston ring is coated giving an increased sliding distance of 11 times to reach wear depth of 4 µm and 3 times to reach wear depth 10 µm. The result for a composite surface roughness $R_q = 0.6$ or $0.3$ µm with a kinematic viscosity of 43 cSt is very similar to the one of $R_q = 0.1$ µm because that a boundary lubrication regime is dominant. This model doesn’t show any significant advantage of decreasing the composite surface roughness $R_q$ from 0.6 to 0.1 µm.

*Figure 54: Calculated wear depth versus sliding distance.*
7 CONCLUSION

This research has added to our knowledge of how piston rings function in modern hydraulic motors. This knowledge will hopefully contribute to future tribological solutions as applied in hydraulic motors. The most important findings of the study are as follows:

- Wear occurs mainly on the top of the face and on the flank of the piston ring.

- Wear on the piston ring is dominated by mild wear, but with a minor contribution of abrasive wear.

- Model tests and simplified component tests of wear on piston rings indicate slightly poorer performance than do tests of an actual hydraulic motor.

- A component, when boundary sliding, displays better properties (such as low wear and friction) on a polished surface rather than a textured surface.

- A component, when boundary sliding, displays the most stable friction on a textured rather than an untextured surface.

- A coated component sliding on a lapped surface, when boundary sliding in a lubricant containing an anti-wear additive, displays a 500-times lower wear coefficient than an uncoated one sliding on a roller burnished does.

- A cylinder bore with roller-burnished surface displays the lowest wear coefficient for the piston ring, i.e., \(4 \times 10^{-19} \text{ m}^2/\text{N}\).

- During hydraulic motor operation, a coated piston ring will have a 11-times greater calculated sliding distance than will an uncoated piston ring, for a composite surface roughness of \(R_q = 0.1 \mu\text{m}\).

- The measured oil film thickness is in the range of 0.7 to 1.3 \(\mu\text{m}\) at velocities of 6 to 12 \(\text{mm/s}\); the calculated oil film thickness is 0.5 \(\mu\text{m}\) at a velocity of 6 \(\text{m/s}\).
8 FUTURE WORK

Future research into the lubrication, friction, and wear behavior of a piston ring in a hydraulic motor should address the following:

- Develop the use and interpretation of the results of ultrasonic oil film thickness measurements.

- Evaluate the results of the capacitive oil film thickness measurement method, applying it to the measurement of the squeezed oil film at the end positions of the piston stroke where the squeezing motion takes place.

- Determine the hydrostatic pressure change over time at the end positions, determine other fluctuations during the stroke, and formulate a theoretical model.

- Evaluate a piston ring with an isotropic surface and also a customized coating having high wear resistance and low friction.

- Extend the wear model to include the time-dependent wear coefficient, surface roughness, oil film build-up according to Reynold’s equation, and hardness dependent on the wear depth. Thus each increment of a stroke, from first to last, will have a unique history.

- Further refine the wear model by introducing initial cleanliness and ingested particles, as well as in-situ generated wear debris.

- Conduct evaluations so that the piston ring can be made to function with environmentally acceptable fluids.

- Measure and formulate a friction model of both piston ring contacts.

- Measure and formulate a model of the piston and piston ring kinematics.

- Measure and formulate a model of the leakage occurring during the piston cycles over time; distinguish the leakage occurring from each contact.

- Measure and evaluate under what condition the face contact is experiencing cavitation.
9 REFERENCES


