

# Imperial College London

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## **ASPECTS OF LUBRICATION IN PISTON CYLINDER ASSEMBLIES**

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## ABSTRACT

The main objective of the project is to characterise piston ring lubrication in reciprocating test rigs and engines in terms of film thickness as a function of oil properties, reciprocating speed and load. A purpose-built test rig, which consists of a moving liner and a fixed piston ring, is used to measure the oil film thickness employing the capacitance and laser induced fluorescence (LIF) techniques. The intensity of the fluorescent light measured as voltage is interpreted as film thickness through calibration. Ring profile fitting is being used as the standard calibration method, but also a dynamic technique has been proposed whereby a groove of known depth on a piston ring specimen is used to provide in-situ dynamic calibration, circumventing the uncertainties of oil film temperature and properties variation during the cycle. A micrometer based static calibration method is compared directly to the dynamic one. The power output of the fibres should be taken into account as well as background noise and resident voltage on the fluorescence detector.

Extensive oil parametric studies have been conducted and oil film thickness, LIF, friction and oil film pressure measurements are presented and compared for different speed, load, temperature, lubricant type and piston-ring geometry. The goal of enhancing the repeatability of the parametric study was achieved through the different positioning of the capacitance transducer and through the increased rigidity of the simulation rig. Useful results were drawn and the effect of properties such as viscosity, viscosity index, HTHS, piston – ring curvature and ring “effective width” on film thickness, film pressure and friction could be determined. Viscosity remains an important factor.

Oil flow visualisation using a CCD camera allowed observation of cavitation throughout the stroke through a glass liner. At mid-stroke the presence of string cavities was evident, whereas near the dead centres the initiation of cavitation was observed in the form of fern-shaped cavities at the diverging wedge of the piston ring specimen at all times. The cavities take the shape of ferns, grown ferns, fissures, strings and bubbles. The existence of the cavities, previously identified indirectly with the LIF technique, was further supported by measurements using a new aluminium liner which incorporates



a miniature pressure transducer used for the first time for oil film pressure measurements and an optical fibre. The oil film pressure measurements provided further support to the interpretation of cavitation at different parts of the stroke. The transition between different cavitating conditions (from  $P_{\text{vapour}}$  to  $P_{\text{atm}}$ ) previously identified in the minimum oil film thickness measurement signal was further supported by the addition of the glass liner and the pressure transducer. Eventually, an insight was obtained into the onset and development of cavitation in the piston assembly via the simulation test rig, which will encourage further efforts in modelling cavitation under a much wider range of operating conditions than those examined here.

Engine testing under motoring conditions was conducted in a modified single cylinder engine with parts of the liner cut and fitted with quartz windows for visualisation experiments. Comparable cavity forms to the test rig were identified and an approach towards their formation is presented. It became clear that the numerous factors affecting the lubrication between the piston-rings and cylinder liner in the engine are giving a different picture compared to the single-ring test rig where the structures are more uniform and the transition from one cavitation type to another is much clearer. The unavailability of the oil film pressure measurements for the engine, though, does not support as strongly the findings from imaging. The top compression ring operates under starved lubrication conditions. Oil droplets were captured by the camera emerging from the ring side clearances that contribute to oil transport. Higher magnification imaging gave further evidence and was compared with previous engine research on oil transport.

The engine study was limited by the unclear images obtained from the CCD camera and the window size which on one hand has the required size for the future firing tests and on another is quite small to provide adequate information about the residence time of the cavities that were identified. Another limiting factor of the study regarding the simulation test rig is that it can only provide data for the case of atmospheric conditions.

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# NOTATION

## ABBREVIATIONS

|                          |                                                               |
|--------------------------|---------------------------------------------------------------|
| <b>2-D</b>               | Two-dimensional                                               |
| <b>3-D</b>               | Three-dimensional                                             |
| <b>A/D</b>               | Analogue to digital                                           |
| <b>AFR</b>               | Air to fuel ratio                                             |
| <b>BDC</b>               | Bottom dead centre                                            |
| <b>CCD</b>               | Charge coupled device                                         |
| <b>CF</b>                | Cavitation factor<br>High magnification lenses for CCD camera |
| <b>CI</b>                | Compression ignition                                          |
| <b>CNC</b>               | Condensation nucleous counter                                 |
| <b>CNG</b>               | Compressed natural gas                                        |
| <b>CO/CO<sub>2</sub></b> | Carbon monoxide/dioxide                                       |
| <b>DAQ</b>               | Data acquisition                                              |
| <b>DELIF</b>             | Dual Emission Laser Induced Fluorescence                      |
| <b>DI</b>                | Direct injection                                              |
| <b>DISI</b>              | Direct injection spark ignition                               |
| <b>DWF</b>               | Dual wavelength fluorescence                                  |
| <b>EGR</b>               | Exhaust gas recirculation                                     |
| <b>EHL</b>               | Elasto-hydrodynamic lubrication                               |
| <b>FC</b>                | Fibre channel                                                 |
| <b>FFT</b>               | Fast Fourier transform                                        |
| <b>GDI</b>               | Gasoline direct injection                                     |
| <b>HC</b>                | Hydro carbons                                                 |
| <b>IDI</b>               | Indirect injection                                            |
| <b><sup>40</sup>K</b>    | Radioactive isotope of potassium                              |
| <b>LED</b>               | Light emitting diode                                          |
| <b>LIF</b>               | Laser induced fluorescence                                    |
| <b>MOFT</b>              | Minimum oil film thickness                                    |
| <b>NO<sub>x</sub></b>    | Nitrogen oxides                                               |
| <b>OCR</b>               | Oil control ring                                              |
| <b>OFT</b>               | Oil film thickness                                            |
| <b>PAH</b>               | Polycyclic aromatic hydrocarbons                              |
| <b>PDA</b>               | Phase Doppler anemometry                                      |
| <b>PDF</b>               | Probability density function                                  |
| <b>PM</b>                | Particulate matter                                            |
| <b>PMT</b>               | Photomultiplier tube                                          |
| <b>ppm</b>               | Parts per million                                             |
| <b>ppr</b>               | Pulses per revolution                                         |
| <b>Psia</b>              | Pounds per square inch absolute                               |
| <b>RMS</b>               | Root mean square                                              |
| <b>SI</b>                | Spark ignition                                                |



|             |                                                                   |
|-------------|-------------------------------------------------------------------|
| <b>SMPS</b> | Scanning mobility particle sizer                                  |
| <b>SOF</b>  | Soluble organic fraction                                          |
| <b>TDC</b>  | Top dead centre                                                   |
| <b>TLOC</b> | Twin land oil control ring                                        |
| <b>TML</b>  | Transducer Measurement Limit                                      |
| <b>TTL</b>  | Transistor-transistor logic                                       |
| <b>UV</b>   | Ultra-violet                                                      |
| <b>YAG</b>  | Yttrium Aluminium Granat ( $\text{Y}_3\text{Al}_5\text{O}_{12}$ ) |

## ROMAN SYMBOLS

|                           |                                                                 |
|---------------------------|-----------------------------------------------------------------|
| <b>A<sub>MAX</sub></b>    | Fibre acceptance angle                                          |
| <b>ATDC</b>               | After top dead centre                                           |
| <b>b</b>                  | Ring width                                                      |
| <b>B</b>                  | Bearing length                                                  |
| <b>CA</b>                 | Crank angle                                                     |
| <b>CF</b>                 | Cavitation factor                                               |
| <b>d<sub>ns</sub></b>     | Number of molecules                                             |
| <b>F</b>                  | Friction force                                                  |
| <b>g</b>                  | gravity acceleration                                            |
| <b>h<sub>ex</sub></b>     | Excitation energy                                               |
| <b>h</b>                  | Path length                                                     |
| <b>HTHS</b>               | High temp high shear viscosity                                  |
| <b>I<sub>a</sub></b>      | Intensity of absorbed light                                     |
| <b>I<sub>o</sub></b>      | Intensity                                                       |
| <b>K<sub>mean</sub></b>   | Average static calibration coefficient                          |
| <b>k</b>                  | constant                                                        |
| <b>m<sub>h</sub></b>      | Mass applied to the hanging point                               |
| <b>NA</b>                 | Numerical apperture                                             |
| <b>n</b>                  | Refractive index                                                |
| <b>n<sub>s</sub></b>      | Molecular density                                               |
| <b>P</b>                  | Oil film pressure                                               |
| <b>P<sub>MAX</sub></b>    | Maximum pressure at combustion chamber                          |
| <b>P<sub>back</sub></b>   | Back pressure, pressure at the diverging wedge of ring          |
| <b>P<sub>comp</sub></b>   | Combustion chamber pressure at the converging wedge of the ring |
| <b>P<sub>atm</sub></b>    | Atmospheric pressure                                            |
| <b>P<sub>vapour</sub></b> | Vapour pressure of working liquid                               |
| <b>P<sub>VL</sub></b>     | Spectral fluorescence power yield                               |
| <b>R<sub>a</sub></b>      | Average roughness                                               |
| <b>R<sub>q</sub></b>      | RMS roughness                                                   |
| <b>R<sub>crank</sub></b>  | Crank radius                                                    |
| <b>S<sup>*</sup></b>      | Excited electronic singlet-state energy                         |
| <b>S<sub>p</sub></b>      | Relaxed singlet excited state                                   |
| <b>S<sub>o</sub></b>      | Ground state                                                    |
| <b>t<sub>res</sub></b>    | Residence time                                                  |

|                       |                                            |
|-----------------------|--------------------------------------------|
| <b>T</b>              | Temperature                                |
| <b>U</b>              | Axial piston velocity                      |
| <b>v<sub>ex</sub></b> | Wavelength of excitation                   |
| <b>V</b>              | Sample volume                              |
| <b>VI</b>             | Viscosity index                            |
| <b>W(θ)</b>           | Load acting on the ring                    |
| <b>x</b>              | Molar concentration of fluorescing species |

**GREEK SYMBOLS**

|                        |                                          |
|------------------------|------------------------------------------|
| <b>ε</b>               | Molar absorptivity                       |
| <b>λ</b>               | Beam wavelength                          |
| <b>ν</b>               | Kinematic viscosity                      |
| <b>ν<sub>40</sub></b>  | Kinematic viscosity at 40°C              |
| <b>ν<sub>100</sub></b> | Kinematic viscosity at 100°C             |
| <b>ρ</b>               | Lubricant density                        |
| <b>σ</b>               | Standard deviation                       |
| <b>σ<sub>κ</sub></b>   | Standard deviation for K <sub>mean</sub> |
| <b>Φ<sub>A</sub></b>   | Radiant power of absorbed energy         |
| <b>Φ<sub>F</sub></b>   | Fluorescence efficiency                  |
| <b>Φ<sub>L</sub></b>   | Quantum yield of fluorescence            |
| <b>ω(θ)</b>            | crank angular velocity                   |

**CHAPTER 4**

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## CHAPTER 6

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# CHAPTER 1

## INTRODUCTION AND LITERATURE REVIEW

### 1.1 INTRODUCTION

The success of the passenger car, powered by the internal combustion engine, as means of transport has led to a conflict between the wishes of the car owner and the impact of increasing number of vehicles on the environment. This is evident in the legislation used worldwide to reduce exhaust emissions, reduce fuel consumption and restrict the use of private cars in congested areas. Inevitably, car manufacturers are forced to reduce the impact of their vehicles on the environment whilst meeting consumer requirements for comfort, durability, performance and low cost of ownership. Environmental targets include reduced fuel consumption; reduced emissions (gaseous and particulate) and extended lubricant drain intervals. These targets are also applicable to manufacturers of large diesel engines as used in heavy duty vehicles and buses.

The lubricant used in an internal combustion engine must operate under extremely hostile conditions, such as high temperatures and pressures and in contact with compounds that cause it to degrade. Under such conditions, it must efficiently perform a number of vital roles such as lubrication of moving surfaces, cooling of engine components and removal of debris from tribological interfaces. It is also expected to maintain this performance for ever increasing periods of time before it is replaced. To meet these expectations requires the base oil to be combined with a number of additives, which must be carefully selected to provide the final lubricant with its desired performance.

Effective lubrication of the piston ring/cylinder liner interface is critical in ensuring that the engine operates efficiently. Overall, the piston assembly is the largest contributor to friction losses in the engine, particularly at the interfaces between the liner and the piston rings and skirt (Wakuri et al, 1995). These losses can be reduced by the use of lower viscosity lubricants but the consequent increase in fuel efficiency could be

offset by increased hydrocarbon emissions if the lower viscosity lubricant has an increased volatility. The lubricant can also contribute indirectly to hydrocarbon emissions through absorption/desorption of fuel.

The primary function of the piston ring is to prevent the leakage of gases from the combustion chamber to the crankcase and to create a heat path for the cooling of the piston crown. The piston assembly is perhaps the most complex tribological system in the internal combustion engine and is required to fulfil a number of important roles. One of these is to provide a seal between the combustion chamber and the crankcase. Without an effective seal, the gas pressures in the combustion chamber would be reduced during combustion and expansion, reducing the efficiency of the engine. The sealing function is performed by the piston ring pack, which effectively forms a labyrinth seal. By design, a small amount of gas passes through the ring pack, adding to the sealing force against the cylinder in a self-regulatory manner and discouraging oil transport towards the combustion chamber.

The control of particulate emissions, especially from diesel engines, also leads to a need to minimise piston deposits. Deposits on the top land of the piston in large diesel engines have been implicated in the formation of polished areas of the liner having a partial or total loss of the honing pattern. This bore polishing leads to increased oil consumption and blow-by thereby causing further combustion chamber deposits, increased particulate emissions and reduced engine power. Thus there is a need to understand and control the transport and consumption of lubricant on the cylinder liner and around the piston rings.

The crankcase lubricant plays an important role in internal combustion engines. Wear protection, deposit formation inhibition and heat transfer are amongst the most important functions of the lubricant. Part of the lubricant, however, will be unintentionally burned during the combustion process. Lubricant can enter the combustion chamber via:

- the piston ring pack
- the valve guides and
- the positive crankcase ventilation system

It is therefore essential to quantify the lubricant travelling through the piston-ring pack. For this purpose, different oil film measuring techniques have been implemented in engines and test rigs that simulate and simplify the lubrication inside the engine cylinder. The issues, though, arising with difficulties such as calibration, measurement accuracy, sensor durability and oil degradation in the engine environment, have been studied by numerous research groups worldwide for many years in succession.

The main subject of this project, is to study further the cavitation process between the piston-ring and the cylinder liner. Both in engines as well as test rigs cavitation in piston-ring lubrication is a subject studied by many researchers in the past. Although there is not sufficient evidence of cavitation erosion in the surface of the liner, this phenomenon is studied as part of the lubricant transport process.

Oil consumption is primarily affected by engine design factors and operating conditions. However, the significant reduction in oil consumption of most modern engines has increased the relative contribution of lubricant related facts.

During the transport process, the lubricant enters the combustion chamber as already stated above. The combustion products of consumed oil might contribute to the exhaust gas emissions (Van Diemen and Kleiser, 1995) and, with the emission legislation becoming more stringent, it is important that oil consumption is reduced to the lowest possible level.

The volume of lubricant present in, and transported by, the piston-ring pack is therefore very important from the point of view of increasingly stringent exhaust emission legislation. Its higher availability in the ring pack zone may lead to increased oil consumption and, in turn, to higher levels of emissions. Starved lubrication conditions, on the other hand, could lead to excessive engine wear. Higher engine operating temperatures, increases in engine power output, introduction of new engine technology like GDI with de-NO<sub>x</sub> catalysts and the use of alternative fuels will create even more demands on the formulation of engine oils. In order to provide engineers with experimental data on the performance of new formulations in cylinder measurements of minimum oil film thickness lubricant availability and its flow will be required.

Despite the wealth of research papers into the subject, a representative portion of which is described below, there are still many areas that require further understanding due to the large number of variables that influence piston-ring lubrication.

## **1.2 LITERATURE REVIEW**

### **1.2.1 OIL FILM THICKNESS MEASUREMENT TECHNIQUES**

The oil film thickness measurement techniques can be divided into two major categories: electrical and optical (Sherrington and Smith, 1985). The main technique used in the present research programme is Laser Induced Fluorescence (LIF), a method very useful for point measurements.

#### **1.2.1.1 ELECTRICAL TECHNIQUES**

The electrical methods used to measure the oil film thickness between piston rings and the cylinder liner can be divided into four categories according to their principle of operation: resistance, inductance, capacitance and eddy current.

##### **1.2.1.2 RESISTANCE METHOD**

The resistance method determines the thickness of the oil film from the resistance between two electrodes. In this case, one electrode may be the piston ring. The piston ring is electrically insulated, whole or part of the piston-ring from the rest of the piston and current passes through the ring to the liner. Eventually, changes in resistance of the ring/liner junction during operation would correlate with changes in thickness with the intervening oil-film. The value of oil film thickness, though, could not be determined due to the absence of smooth changes in the resistance. Instead, many individual short circuits occurred between ring and liner. Furuhashi and Sumi (1961) applied the technique to an idealised rig which incorporated a stationary, loaded piston ring and a reciprocating liner. The entire ring was used as an electrode which measures

the lower resistance around the ring. The need to get a better picture of the lubrication led the researchers to introduce three insulated electrodes, to examine the lubrication at specific points. The electrodes were placed at different axial positions, and at three positions around the circumference, opposite the ring gap and either side of it at 60 degrees apart. The circumferential variation was not significant and little electrical breakdown was recorded by the upper probe. An improved resistance technique was introduced by Furuham and Sumi (1961). In this method, one end of the piston was pinned whilst the free end was connected to the anode of a moveable anode valve. As the piston reciprocates, changes in the oil film thickness cause small changes in the diameter of the ring which were detected by the valve. After first measuring the static ring gap throughout the stroke and then deducting the variation from the dynamic results (variation in the bore circumference necessitated this step) the average oil film thickness could be deduced from geometrical considerations.

### **1.2.1.3 INDUCTANCE METHOD**

The evaluation of the distance between the piston rings and the liner is based on the principle that the inductance of a wire coil changes as its separation from a magnetic material varies. Wing and Saunders (1972) used self inductance transducers for the indirect measurement of the oil film thickness. It was impossible to fit the transducers in the ring due to their size and they were placed instead in the rear of the piston ring grooves. The variation in oil film thickness was assumed to be identical to the variation in the gap behind the ring.

Dow et al (1983) fitted an inductance probe to one end of the ring and monitored the variation in the ring gap during the engine cycle. The variation of oil film thickness could be calculated and the results taken were shown to be in general agreement with theory.

The inductance method has the advantage of measuring the oil film thickness as an average parameter and is less susceptible to local variations than a single point measurement. In addition, a continuous film thickness measurement is available for the whole stroke. However, the introduction of significant errors to the measurements due to

inaccuracies in the bore circularity and the fact that it is temperature sensitive have limited the applicability of the technique.

#### 1.2.1.4 CAPACITANCE

The capacitance technique has been for years the most popular technique to measure the oil film thickness. It is based on the principle that the capacitance of two parallel metal plates varies in inverse proportion to their separation.

Hamilton and Moore (1974) measured successfully the lubricant film thickness by using capacitance transducers mounted in the cylinder liner. Films up to 10  $\mu\text{m}$  could be measured, above this changes in capacitance were too small to be determined accurately. Initially it was hoped that the oil film thickness could be deduced from parallel plate capacitance theory but this proved to be impossible due to the difficulty of getting the screen close enough to the central electrode. Instead, a combination of slip blocks and a Talysurf was used. Although the gauges were small enough to plant in the rings the researchers decided it would be difficult to establish an electrical connection between the ring and the outside of the engine. Instead, three gauges were planted in the thrust side of the liner, with the consequent advantage of being able to observe the ring profile. After honing, no protrusion was detected, even when the liner temperature was increased. The results demonstrated different compression ring profiles for all three gauges, clearly indicating varying tilt. Owing to the small diameter of the probes, little electrical breakdown was observed until the film dropped below 0.5  $\mu\text{m}$ . Thinner oil films than expected (0.4 to 2.5  $\mu\text{m}$ ) indicated the presence of starved lubrication. The results though, were not repeatable.

Parker et al (1975) inserted capacitance probes into the rings of a firing engine. Thus, they were able to investigate the oil film thickness throughout the cycle. Shin et al (1983) solved the problem of short-circuiting by insulating the sensor electrode. In order to study piston tilt, they inserted two probes at different locations.

The capacitance technique was also used to validate models in simplified experimental set-ups (Hamilton and Moore, 1974b). Myers et al (1990) studied the circumferential variation of the oil film. Grice and Sherrington (1993) investigated

cavitation by comparing the output of several other probes implanted at various points in the liner with the measured ring profiles. Due to the ring twist, though, this proved to be difficult. The researchers while trying to observe the effect of speed noticed that with a rise in speed, came an unexpected initial reduction in oil film thickness although at higher speed the film thickness begun to increase again. However, repeatability was rather poor. Later on Mattson (1995) investigated the piston movement using diametrically opposite probes.

The capacitance technique proved to be a useful tool. Capacitance transducers offer good thermal stability, are quite robust as well and have been successfully used to investigate cavitation, ring tilt, piston movement and circumferential variation, especially in motored tests because of the difficulty to make accurate calculations of the thermal expansion of the ring in a firing engine.

#### **1.2.1.5 EDDY CURRENT SENSORS**

The sensors are equally large relative to the inductance probes and they are based on the same principle. The measurements are independent of the medium between the transducers and the rings. Sanda and Someya (1987) measured the oil film thickness with eddy current sensors. The relative displacement between the ring and the liner was measured and from this the minimum oil film thickness could be calculated. Shengua et al (1996) installed eddy current sensors on the liner, at TDC, mid-stroke and BDC. Unfortunately, the TDC probe failed whilst the location of the probes (too close to each other) made the data difficult to interpret. The researchers observed that the minimum oil film thickness under motoring conditions was similar for all four strokes, as well as oil starvation.

#### **1.2.1.6 OPTICAL TECHNIQUES**

The optical techniques can be divided into three categories:

- two-dimensional examination of the oil film distribution
- laser induced fluorescence point measurements

- interferometric methods

#### 1.2.1.7 VISUALISATION

In order to apply the first technique, researchers used cameras and appropriate illumination to take pictures of the lubricant film. The engines used have transparent cylinders and Perspex windows. Greene (1969) used a transparent cylinder which had the same roughness as the original liner. He used two optical methods: an ultra-violet light source was illuminating the oil film and a fluorescent compound mixed with the oil emitted an intense blue fluorescence. The resulting picture could be recorded at 800 frames per second in colour. The other method applied was using scattered light, to determine the oil in contact with the cylinder walls. The results showed that the piston-ring appeared to be separated by a continuous film of oil but the film thickness showed circumferential variation and sometimes was non-existent at top-dead centre. Squeeze action was dominant at the dead centres while at mid-stroke hydrodynamic lubrication was prevalent. During the downstroke, oil could either be scraped down, bypass the rings or collect in the lands. In the region of the ring gap, the film was observed to be broken down and gas flow was apparent.

Sanda et al (1993) used LIF to take two-dimensional images of the oil film. Inagaki et al (1995) used a Xenon-flash lamp and developed a two-dimensional system. The oil used had a fluorescing dye. Two cylinders were used for the experiments - a glass liner for the motoring tests and a cast iron liner with a pair of tapered slit holes in the axial direction for the firing tests.

The researchers observed from the firing tests that oil spouts were moving up through the top-ring gap into the combustion chamber.

Kim et al (1995) manufactured a rig which simulates piston ring lubrication. The rig consisted of a reciprocating flat plate with a straight bar test piece loaded on top of it. The researchers tried to investigate cavitation with this optical technique. Nakashima et al (1995) inserted a 5 mm thick transparent quartz cylinder. A red dye was added to the lubricant and a video camera was used to record the characteristics of the oil flow through the ring pack. When the piston ring gaps were in line greater oil flow



was observed. In a later paper (Nakashima et al, 1996) the oil flow was increased by a higher pressure difference, greater oil temperature and lower oil ring tension and engine speed. The main oil flow paths were observed to be through the ring gaps and along the lands. In 1998, the same researchers found that a triangular step joint in the top piston-ring reduced the oil flow. Yoshida et al (1996) investigated the oil flow to the combustion chamber through a v-shaped groove into the second land. They found that it was greatly reduced. Thirouard et al (1998) introduced a quartz and a sapphire window in a firing diesel and spark ignited engine and used the LIF method and a CCD camera to photograph the oil transport in both cases. The results showed that the oil flow was affected by the gas flow through the lands and rings. More details about this investigation are provided in Chapter 5.

#### **1.2.1.8 INTERFEROMETRIC METHODS**

The interferometric technique is based on the interference pattern generated by the path difference between light beams. Wakuri et al (1979) used thin film interferometry to measure the oil film thickness between a ring specimen and a glass plate to investigate cavitation. Schlesinger (1986) utilised a Mach-Zender interferometric arrangement in the study of post withdrawal drainage of viscoelastic suspensions. This configuration requires free optical access on both sides of the liquid film. Keely et al (1989) utilised the Haidinger interferometric arrangement which although it requires optical access from only one side of the film it suffers from the drawback inherent to all interferometric-based methods where only the film thickness variation can be measured and not the absolute thickness. Ozdemir and Whitelaw (1992) have calculated the theoretical resolution of single colour Haidinger interferometry using frequency counter to measure the film thickness variation.

#### **1.2.1.9 LASER INDUCED FLUORESCENCE (LIF) POINT MEASUREMENTS**

The LIF technique is based on excitation of the molecules of a fluorescent material by a light source. The main advantage of this technique is that it has the

potential to quantify the lubricant film thickness throughout the cycle. Similarly to all the other optical techniques, it has this major advantage compared to the electrical techniques where the oil film can be measured only under the piston-rings.

The first application of the technique in order to study oil films was by Smart and Ford (1974) who studied oil film formation on a spinning cylinder. They used a UV light source to illuminate the film and the range of the films measured was 0.001 to 1 mm. Later Ford and Foord (1978) decided to replace the UV light source with a He-Cd laser (441 nm, 18 mW) which gave better light source stability and measured liquid films on a spinning shaft. The introduction of a laser source for the experiments resulted in less chemical degradation of the oil and greater response linearity.

Ting (1980) applied the LIF technique to a research engine with a transparent liner and presented qualitative results of the lubricant film thickness. Hoult et al (1988) demonstrated one of the first in-situ calibrations of the LIF technique. The researchers measured the oil film thickness in a cylinder of a research engine, fitted with a quartz glass window. In order to interpret the results they used both static and dynamic calibration. The bench top measurements of the LIF signal obtained from oil samples of known thickness were compared with the ring face profiles discerned in the fluorescence signal acquired from the engine.

The calibration of the LIF signal is by itself a major issue because it still remains a source of uncertainty. Later on in Chapter 2 the attempts to calibrate the signal are presented. Accuracy is compromised when bench static calibration is used to interpret in-engine dynamic measurements, as fluorescence is temperature, viscosity and oil degradation dependent.

## 1.2.2 LUBRICATION AND EMISSIONS

As far as emissions are concerned, according to regulations for 2005, both  $\text{NO}_x$  and particulates need to be significantly reduced which is difficult due to the well known “trade-off”. The following schematic shows the technologies introduced to achieve exhaust emission reductions.

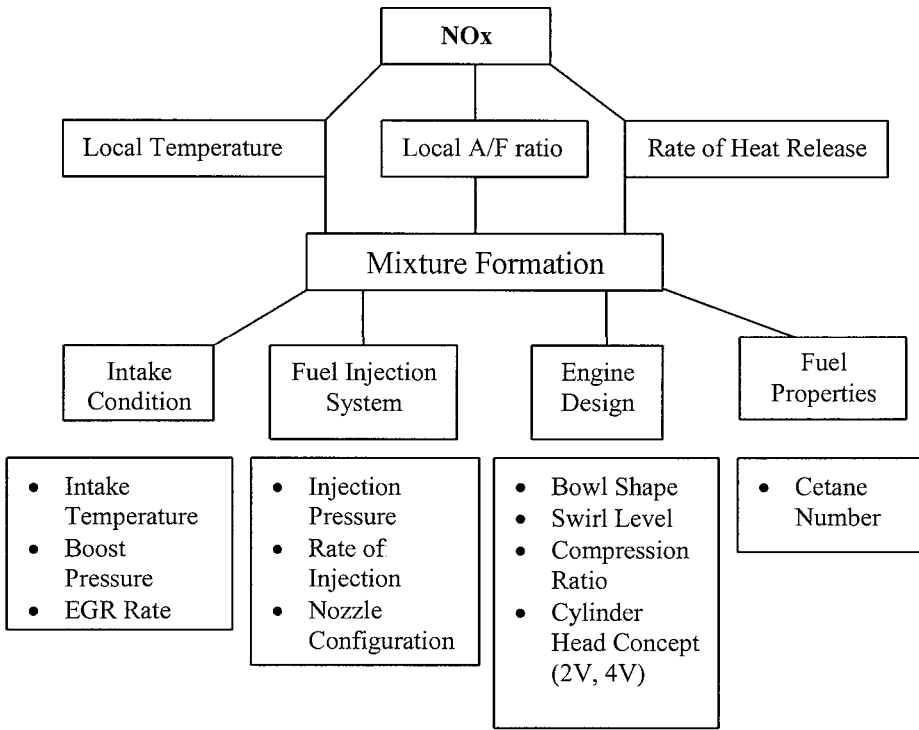
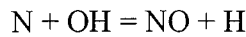
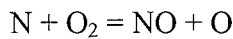
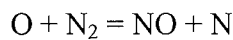


Figure 1.1 Main Parameters Affecting NO<sub>x</sub> Formation

The principal source of oxides of Nitrogen (NO<sub>x</sub>) is oxidation of atmospheric N<sub>2</sub> at high temperatures (thermal NO). If fuel contains nitrogen, the oxidation of fuel N containing compounds is an additional source of NO (fuel NO). There is also prompt NO but has a small contribution. The mechanism of NO formation from atmospheric nitrogen is being described by the Zeldovich mechanism. In combustion of near-stoichiometric fuel-air mixtures the principal reactions governing the formation of NO from molecular nitrogen are:



NO forms in both the flame front and the postflame gases; the latter mechanism dominates. It also depends on the maximum cylinder temperature.

The particulates formation is based on unoxidised or partially-oxidised fuel molecules. The principal mechanism is imperfect mixing (locally over-lean or locally over-rich) and flame quenching at combustion chamber walls. Their composition is

unburnt hydrocarbons, lubricating oil and various aromatic compounds. An example of the particulate composition of a heavy duty DI engine for USA test cycle is as follows:

|                                 |     |
|---------------------------------|-----|
| compounds of lube oil origin:   | 40% |
| carbon:                         | 31% |
| sulphates:                      | 14% |
| other compounds of fuel origin: | 7%  |
| unidentified:                   | 8%  |

The morphology of the particulates is spheres, 0.01  $\mu\text{m}$  diameter for the primary particulates and for the secondary particulates which are agglomerates of the primary ones is 0.1  $\mu\text{m}$ , with varying shapes.

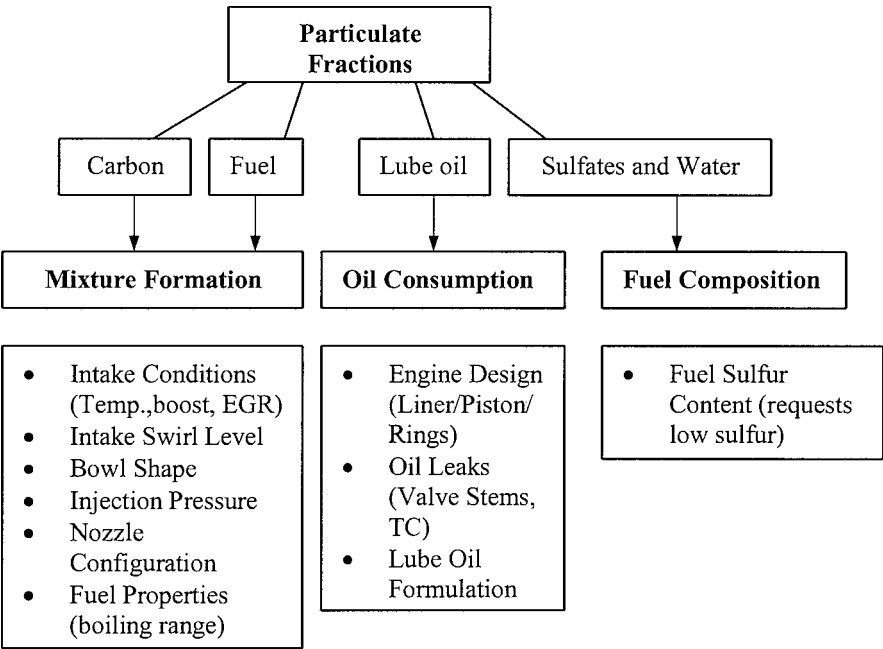


Figure 1.2 Main Parameters Affecting Particulate Formation

Oil consumption increases when the oil which fails to be scraped off the piston rings and remains in the combustion chamber evaporates and burns as it is exposed to high temperatures during the combustion phase. Therefore, one way to reduce the oil consumption is to improve the piston and piston ring designs to improve the oil scraping effect off the rings so that the oil evaporation can be eliminated.

The physical characteristics of oils can influence the mechanical efficiency of the internal combustion engine and, as a consequence the specific fuel consumption. The variation in the quantity of fuel consumed at constant power output can influence the combustion process thus affecting the exhaust emissions of NO<sub>x</sub>, HC, CO and particulate matter (Laurence et al, 1996; Schiemann et al, 1995).

The physical characteristics of the oil influence the oil consumption as well (Manni and Ciocchi, 1992; Van Dam and Kleiser, 1995). The quantity of lubricating oil drawn through the piston and the valves towards the combustion chamber is another important parameter, depending on the oil rheology, which has an effect on exhaust emissions.

Direct injection diesel engines have in principle better prospects than the traditional gasoline engines due to their lower fuel consumption and CO<sub>2</sub> emissions, even though the problem of particulate and NO<sub>x</sub> emissions has to be solved. Emissions and consumption are not only influenced by engine technology but also by the formulation of fuels and lubricants. So engine manufacturers and oil companies are actively investigating the control of emissions contributed by these products. An investigation has been carried out that showed that the use of low viscosity oils leads to a better mechanical efficiency and a lower specific fuel consumption than high viscosity oils within the examined viscosity range (Manni et al, 1997). Also some synthetic basestocks gave a lower contribution to particulate emissions.

The chemical characteristics of the oil can influence its “combustibility” (Manni et al, 1996; Dowling 1992), expressed as the tendency of the oil to burn into the combustion chamber. For instance the chemical nature of different basestocks could give a different behaviour of the oils into the combustion chamber, which consequently affects in a different way the exhaust emissions (Laurence et al, 1996). As a result of the different solubility and diffusion of the fuel into the oil (Manni and Ciocchi, 1992), different basestocks could give a different interaction with the injected fuel thus affecting in particular the emission of the soluble organic fraction of the particulate.

Lower viscosity oils than the traditional ones lead to higher AFR and that explains the good behaviour of the synthetic low viscosity oil. The lower viscosity and volatility would support the hypothesis of a smaller contribution of the synthetic

lubricant to oil consumption. Lately, though, there is a view that lower viscosity oils lead to thicker films due to increased oil availability in the piston-ring pack.

The lubricant composition can influence the vehicles exhaust emissions especially the particulates, in the short term and that the impact of viscosity, volatility and sulphur content of the lubricant on engine operation may partially explain this effect.

The issue of particulate emissions from internal combustion engines (both spark and compression ignition) has generated a great deal of interest. This is primarily due to recent evidence which suggests that there may be negative health effects associated with increased levels of ambient particulate matter (Graskow et al, 1999). There is particular concern about the emission of nanoparticles (particles smaller than 50 nm diameter) which have the ability to penetrate deep into the interstitial tissue of the lung (alveoli), where they may cause severe respiratory inflammation and acute plumonary toxicity.

#### **1.2.2.1 DIESEL AND SPARK IGNITION ENGINE PARTICULATE EMISSIONS**

The vast majority of effort in the exhaust emissions community has been focused on the characterisation and control of particulate emissions from diesel engines. This is due to the fact that mass-weighted emissions from spark ignition (SI) engines have traditionally been insignificant when compared to those from diesel engines. However the recent trend has been to shift focus away from particle emissions mass-weighting (where diesels dominate) to number-weighting; this has resulted in a renewal of interest regarding particulate emissions from SI engines.

According to recent studies the steady-state particulate number emissions from modern port fuel injected (PFI) SI engines are generally several orders of magnitude lower than modern diesel engines. However, since SI engines emit a large fraction of their particles in the sub-50 nm size range and because of the far greater number of SI engines on the road, particulate emissions from SI engines may still be overall significant. Furthermore, particle number emissions from SI engines have been shown to increase dramatically under high-load, transient and cold-start operating

conditions (Graskow et al, 1999). In fact, recent estimates have attributed up to 2/3 of the total mass of vehicle particulate emissions in the United States to SI engines. The difference in the combustion process between the PFI and direct injection spark-ignition engines may result in significant higher particulate emissions from DISI engines under part load conditions.

The fact is that some of the older SI engines could even exceed future diesel particulate emission regulations. However, most modern SI engines equipped with a catalyst can easily meet any future diesel particulate emission standards, although their particulate emissions maybe a significant fraction of these regulations. The highest emissions are generated during transient operation.

Laurence et al (1996) studied the effect of lubrication system parameters on Diesel particulate emission characteristics. The oil film thickness increases with decreasing viscosity. Therefore, lower viscosity oil will remain on the liner during the engine expansion stroke, exposing more oil to flow entrainment and combustion gases. Additional exposure is manifested as higher particulate rate, as more oil participated but was not totally oxidized in the combustion process. Similarly, the total particulate rate increased with enlarged ring gap. An enlarged ring gap allows a larger puddle of oil to pass into the combustion chamber when the cylinder pressure drops below the inter-ring pressure during expansion. At the high load condition, the predominant change in particulate composition is the non-soluble fraction. Due to oil viscosity and ring distortion, any changes in particulates would be due to a change in oil-derived soluble oil fraction (SOF), as it is the combustion of additional oil that causes the change. In the high load condition, however, this was not the case. With the expected higher in-cylinder temperatures the additional oil was partially oxidized and formed carbon. Such a process would cause an increase in carbonaceous or non-soluble particulates.

#### **1.2.2.2 PARTICULATE COMPOSITION AND SIZE DISTRIBUTION**

The chemical analysis of the particulates (Andrews and Ahamed, 1999) has shown that the bulk of the mass is ash and that second largest fraction is unburned lubricating oil.

Particulate rates are higher with lower viscosity oil and wider top-ring gap (Laurence et al, 1996). At high loads, the difference in particulate rate is due to changes in the non-soluble organic fraction while, at medium and low loads, any change in particulate rate is due to differences in the lubricant-derived portion of the soluble organic fraction (SOF). The percentage contribution of the lubricant to the total particulate matter varies predominantly with engine condition, reaching as high as 70% of total particulates (Cartellieri et al, 1987; Dowling, 1992) and 90% of the soluble organic fraction under some operating conditions. The problem is less severe at high loads, as most of the consumed oil is oxidised due to the higher temperatures. Zelenka et al (1990) and Essig et al (1990) have shown that the emission rate of oil derived particulates is directly proportional to the oil consumption rate. They further concluded that reduction of cylinder wall oil consumption, which represents the main source of lubricant derived particulates, should be the primary focus in reducing overall particulate mass.

Researchers have postulated several sources of oil consumption. These include:

- the release of the accumulated oil on the top piston land into the combustion chamber due to inertial forces
- the piston ring-pack dynamics and flow driven release of oil through the top ring and
- volatility effects

The verification of these assumptions, though, has been lacking for many years due to the difficulty in measuring the oil behaviour in the piston ring-pack region. The better understanding of oil consumption mechanisms will help the development of additional strategies for particulate reduction.



### 1.2.2.3 HYDROCARBONS

Parks II et al (1997) used LIF spectroscopy to investigate possible hydrocarbon emission sources within the engine. In particular, an effort was made to quantify the concentration of fuel absorbed in the oil on the cylinder wall of a small engine. Fluorescence provides an ideal spectrometric method for distinguishing the concentration of individual chemical components in a simple mixture.

Measuring the fuel content in the oil film is important to the auto industry for several reasons. Unburned fuel is suspected to be the primary contributor to hydrocarbon emissions during cold starts. Measurement of the fuel content in oil provides explanations of the fuel diffusion rate in oil for different engine conditions. And by knowing the amount of fuel absorbed in oil and the capacity of oil to store fuel, one can determine the significance of unburned fuel released from oil relative to emissions and engine performance. Because unburned fuel in the cylinder-wall oil film is a suspected major emission source during engine start-up (cold start), the amount of fuel in oil is of particular interest for determining hydrocarbon emission sources and the relevant detailed mechanism.

### 1.2.3 ENGINE OIL TRANSPORT

As the rings are sliding along the liner and the groove sides, lubrication is needed to minimise friction and wear. Consequently, a sufficient lubricant supply must be allowed to the contact regions whose most critical ones, namely the top ring running surface and groove, constitute direct links to the combustion chamber. The key to minimising oil consumption is to optimise oil transport within the piston ring pack. Because the regions above the oil control ring are only partly flooded with oil (Casey, 1998; Tian, 1996), the rate at which oil flows from the upper skirt region to the combustion chamber is the result of numerous oil transport processes between the various regions defined by the ring pack geometry (piston lands, ring grooves and liner).

Oil may flow from one land to another through the ring grooves, the gaps and the ring liner interface. Furthermore, due to the motion of the rings in their grooves,

connections between ring grooves and piston lands are repeatedly opened and closed. Eventually the gas and oil flows are highly coupled with the ring dynamics (Tian et al, 1997). The potential driving forces for oil transport between the different part of the piston ring pack are pressure gradients, shear stress from the gas flow, ring/liner and ring/groove relative motion, and inertia forces. Because of the great variation of engine operating conditions, oil transport is expected to exhibit similar variations. Oil consumption and thus oil transport were found to be significantly affected by the detailed geometry of the ring pack components such as ring and groove relative angles, ring running surfaces (Yoshida et al, 1991; Tian, 1997), piston land geometry (Rabuté and Tian, 2001; Yoshida et al, 1996) and bore thermal and mechanical deformations (Schneider et al, 1993; Ishizuki et al, 1991).

### 1.2.3.1 INTER-RING GAS DYNAMICS

In the piston cylinder assembly the ring-pack is primarily a gas seal and as such its principal task is to prevent blow-by or flow of gases from the combustion chamber to the crank case. However, blow-by across the ring does take place, especially during the latter part of the expansion stroke when the pressure differential across the rings is at its maximum. Inter-ring pressures along with the axial and radial position of the rings within their grooves determine the gas pressure forcing the rings against the liner. Inter-ring pressures also determine the film pressures at the boundaries which in turn affect film pressures and, hence, the ring dynamics within their grooves.

The gas flow and pressures within the ring pack are dependent on the boundary pressures in the combustion chamber and crankcase, the gas properties and the geometry of the available passageways through the piston assembly. The available passageways and their shapes vary owing to the deformation and dynamics of the piston and rings. The gas composition and temperature may vary throughout the engine cycle and will also be dependent on the location within the ring pack.

The gas pressures determine the lubricant boundary pressures, contribute to the loading on the ring and also affect the ring dynamics which affect the geometry of

the lubricated ring/liner conjunction. The boundary pressures of the inlet and outlet of the ring are equal to the gas pressures of the neighbouring lands. The gas pressure in the ring grooves acts on the back of the rings, forcing them against the liner and thereby improving the sealing action. In the case of either starvation or cavitation (external cavities) the land pressures which act on the running surface of the ring are in the opposite direction to the groove pressure. The land and groove pressures all contribute to the ring dynamics.

Much research has been carried out on this topic as the gas flow is also responsible for blow-by and oil consumption due to blow-back (reverse blow-by). The results are engine dependent but provide some general indications of the flow paths that can be expected. The main gas flow is through the ring gaps, with secondary paths occurring behind the ring (when the ring is not in contact with either the top or bottom of the groove), and in front when the ring does not conform to the liner. The ring gaps restrict the gas flow, a consequence being that the profiles of the inter-ring gas pressures lag behind that of the combustion chamber (Parker et al, 1975). The second land pressure usually rises above the combustion chamber pressure late in the expansion stroke, as the combustion chamber pressure drops quickly. This leads to blow back through the top ring gap, a cause of increased oil consumption if entrained oil/mist and/or liquid oil around the gap is carried into the combustion chamber (De Petris et al, 1996). Burnett et al (1995) measured the groove pressure behind the top ring and discovered that it corresponded to the cylinder pressure until shortly after this dropped below the second land pressure. A discontinuity then occurred and the groove pressure rose to the second land pressure. This result provided direct evidence of ring lift and showed the importance of the pressure drop over the ring in determining the ring dynamics. In large engines when the two ring gaps are separated by a large phase difference, significant flow restrictions can occur in the lands (Ruddy et al, 1981b). Kuo et al (1989) showed gas flows to be affected by the surface temperature of the liner, piston and rings.

The piston shape and dynamics influence the ring dynamics, the gas flow within the ring gap and the lubricant flow through the ring pack, particularly when side forces press the piston against the liner. The piston dynamics include secondary motions

both axial and transverse as well as the reciprocating axial motion. The axial secondary dynamics are due mainly to tolerances in the piston assembly components whereas those in the transverse direction, result from a combination of the clearance between the cylinder and bore and the side forces exerted by the crank.

Ring dynamics are dependent on the gas pressures in the lands and groove, the lubrication in the ring/liner conjunction and the lubrication between the ring and piston ring groove sides. They also depend on the ring inertia and its resistance to bending. Around the circumference, there are variations due to the ring gap and liner distortion. These forces lead to axial, transverse, twisting and circumferential movement (ring rotation). The majority of research (experimental and theoretical) has assumed uniformity around the circumference.

Force is transmitted to the ring by interaction with the flanks of the ring groove; the conjunction is lubricated by oil in the groove. The degree of lubrication is very hard to predict owing to uncertainties in the oil transport through the groove and hence the extend of oil availability. Generally any axial motion is modelled as squeeze action only and any transverse moment is assumed to result in boundary friction owing to the low velocity. The groove shape changes with thermal and elastic loading and is therefore difficult to measure or predict.

Miyachika and Fuse (1991) investigated the effect of second land dimensions on the oil consumption and showed both size and shape to be important. They determined that much of the oil consumption was due to reverse gas flow through the ring gap. The flow is related to the pressure drop across the ring gap, which in turn is affected by the pressure in the second land, a function of its dimensions.

Axially the velocity of the rings is approximately that of the piston. However, the secondary motions of the piston combined with possible ring flutter within its groove may complicate the matter. The dominant forces that affect the ring axial motion within its groove are the ring inertia, the gas pressure drop and the friction due to the ring/liner lubrication. Burnett et al (1995) used two inductive proximity sensors implanted in the groove flank, to determine ring lift. Two distinct modes of lift were identified; the first was axisymmetric, for which good agreement between measurements and theory was obtained, while the second was non-axisymmetric and showed the top

ring to lift preferentially on the anti-thrust side. The authors speculated that the ring tilt could be influenced by both piston secondary motion and ring gap location; the latter affects the gas flow and hence the gas pressures within the ring pack.

Radially, the dominant forces acting on the ring are the groove gas pressure, the pressure developed in the ring/liner film combined with any asperity loading and the ring tension. The friction between the ring side and the groove flank is generally thought to be insignificant.

Twisting moments may result from any of the forces acting on the ring, causing the ring/liner geometry to vary. Furuhashi and Hiruma (1975) investigated ring twist in a motored rig by attaching a lever to the ring's upper surface and extending it vertically to a dial gauge. The twist changed throughout the engine cycle.

Since engine liners are not perfectly circular and ring conformity is not uniform, the lubrication at any point around the circumference of the liner depends on the position of the ring gap. Therefore if the ring rotates or oscillates, the lubrication also varies cycle by cycle. Min et al (1998) investigated the ring rotation and correlated it with the measured oil consumption. Prior to the investigation, a point of each of the rings was irradiated by neutron beam in an atomic reactor. By measuring the intensity of radiation outside the cylinder, the positions of the ring gaps could be deduced. The authors discovered that the top ring rotated at a rate of 0.3 to 0.4 rpm whereas the second ring just oscillated. As the load increased, the top ring ceased rotating and begun to oscillate. The rotational and oscillatory characteristics were thought to be highly engine dependent. On varying the speed, it seemed to be impossible to predict the characteristics purely from engine operating parameters-the pattern of rotation did not seem to be particularly repeatable.

The shape of the profiles affects the lubrication. For example, a barrel faced ring may be designed with a steep or shallow profile. The former generates thicker films at mid-stroke but the latter gives better squeeze action at the dead centres. The best compromise is thus sought when designing the rings so as to give the thickest possible films at the dead centres. Ring twist may vary around the circumference; for example scraper rings often have varying static twist (although over time the profile presented to the liner should become more uniform due to wear).

Static and elastic distortions and wear contribute to an axially and circumferentially varying ring profile. Static and elastic distortions and wear contribute to an axially and circumferentially varying liner profile. Static distortions are a result of clamping forces, manufacturing errors and thermal effects during engine operation. Dynamic distortions occur due to varying pressure loading over the engine cycle. Wear particularly at the dead centres, results in an increased bore diameter and thus a larger ring gap (Ting and Myer, 1974).

The flow of gas through the piston assembly is a vital part of the degradation process in this region. It contains products that contribute to oxidation and it is likely to be an essential driver for transport of the lubricant. The influence of the ring pack design and piston motion on gas flow in the ring pack is an important consideration. Piston ring rotation is another factor that may play an important role, through the change in relative positions of the piston ring gaps. This has been seen to influence both the blow-by and oil consumption behaviour in the piston assembly.

### **1.2.3.2 LUBRICANT AVAILABILITY**

Lubricant starvation and lack of lubricant availability at the leading edge of the ring has an important influence on film thickness, frictional losses and oil transport. Through its effect on oil film thickness, the lubricant availability determines the onset of transition from hydrodynamic to mixed lubrication and thus should be expected to affect the position and extent of wear. Whereas the leading edge boundary of the film is simply determined by oil availability, the position of the trailing edge cavitation boundary is determined by the operating conditions, the gas pressures at the film boundaries, as well as the pressure gradient at the exit.

### **1.2.3.3 LUBRICANT RHEOLOGY-MODELLING**

Herbst and Priebsch (2000) describe a simulation method which predicts lube oil consumption due to evaporation from liner wall and mechanical losses of oil which go past the top ring. The applicability of the simulation is underlined by the comparison

of predicted and measured inter-ring gas pressure, axial ring motion and blow-by. They have noted that the top edge scraping of the first compression ring can be influenced by the second land pressure. This top side sealing of the second ring may also cause a radial collapse and lead to a higher remaining amount of oil. In addition, the magnitude of the inter-ring gas pressure is shown to be significantly different between the piston major and minor thrust side. This is due to secondary movement of the piston causing ring fluttering. The authors summarised the ways the total mass transport is determined as follows:

- Evaporation from the liner wall.
- Throw-off accumulated oil above the top ring.
- Oil blow through the top ring end into the combustion chamber due to a reverse gas flow.
- Oil scraping of the piston top land edge.

According to Gamble (2002), the complex tribology of the piston assembly has been divided into two mathematical models, one for the piston and one for the piston-ring pack. One of the ways in which models may impact on each other is through secondary motion of the piston. Hoult and Shaw (1994) identified piston ring gaps as one of the key factors which influence oil consumption and gas flow for a piston assembly. Inclusion of tilting motion into ring pack lubrication models may have implications for the ring gap size and hence the ring pack lubrication. Small amount of gas flow passes through the ring pack, adding to the ring sealing force against the cylinder, thus discouraging oil transport towards the combustion chamber.

Piston secondary motion is governed by a number of factors.

- |                                               |                                                                    |
|-----------------------------------------------|--------------------------------------------------------------------|
| a) gas and inertia forces                     | } produce lateral motion<br>and rotation around the<br>gudgeon pin |
| b) piston-cylinder clearances                 |                                                                    |
| c) offset of the gudgeon pin                  |                                                                    |
| d) hydrodynamic friction of the skirt and pin |                                                                    |
| e) elastic deformation of the piston skirt    |                                                                    |

This has an influence on engine friction and engine noise caused by the piston slap.

A basic requirement for modelling the lubrication is that the gas pressures between the piston rings and the flow past each ring be calculated as an intermediate step. This is achieved by using the “orifice and volume” method. Gas flows between each volume via the piston ring gaps because it is assumed that lubricant fills the clearance between the ring face and the cylinder and that there is no flow of gas around the back of the ring. Pressure above the top ring is assumed that of the combustion chamber while the crankcase to be at a fixed pressure, local atmospheric. Average value is assumed for the diameter of each of the piston lands giving the same clearance between each land and the cylinder wall.

The piston motion model developed by Chittenden and Priest (1993), includes the effect of hydrodynamic lubrication of the piston skirt but neglects surface roughness, bore distortion and elastic deformation of the piston skirt. Piston is subject to lateral motion across the cylinder and tilting around the piston pin.

Effect of piston secondary motion is included therefore the clearance is not constant around the circumference of the piston. The orifice area for each ring fluctuates depending upon the circumferential position of the ring gap. The model also incorporated the effect of ring axial motion in the groove on orifice area. Orifice area depends on whether the ring is in contact with the top or bottom face of the groove.

For the model conditions the secondary motion caused the piston to be forced to one side of the cylinder around the whole of the engine cycle (reduced clearance at thrust side-increased clearance at anti-thrust side).

The secondary motion of the piston can produce a large variation in the gas flow through the ring gaps. Circumferential position of piston ring gaps is also significant.

Reduced gas flow rates may increase the time oil spends in the piston assembly but would reduce the amount of combustion products.

In general, piston-ring lubrication models calculate the oil flow on the cylinder wall beneath each of the piston rings. Most take no account of any build up of oil that may occur ahead of a piston-ring. If all the oil film ahead of the ring does not



flow beneath the ring face as the piston moves along the cylinder oil will begin to accumulate at the leading edge of the ring.

A considerable amount of oil flow can take place in the piston assembly via a number of different transport mechanisms. For all the transport mechanisms the inclusion of secondary piston motion is only of minor significance for the predicted lubricant flow rates. Significant oil transport occurs in the piston assembly by a number of mechanisms.

The shape of piston rings and configuration of piston ring pack have a significant impact on the amount and direction of oil transport. The condition of piston assembly is likely to have a major influence on the length of time that lubrication remains in this region and hence its degradation.

#### **1.2.3.4 OIL TRANSPORT MECHANISMS IN THE PISTON ASSEMBLY**

Oil may flow through the ring pack along two routes: the liner and the piston. These paths are geometrically connected by the rings and thus oil may be transported between the liner and the piston. Oil transport along the piston involves oil flow across the lands, through the ring grooves and gaps. The oil flow through the ring grooves require oil from the piston lands to be supplied to the ring/groove side clearances and are therefore highly related to oil displacements on the lands. Tian (1997) has modelled the oil flows out of the ring groove side clearance which are mostly determined by the ring/groove interactions and the resulting oil squeezing process. Due to lack of direct experimental observations and theoretical efforts, the oil mechanical behaviour on the piston lands has never been thoroughly investigated. Although well developed one point Laser Induced Fluorescence have had success in quantifying the oil film thickness between the rings and the liner, as pointed out in the OFT measurement section of the literature review, it was not sufficient for studying oil transport processes on the piston as oil moves both axially and circumferentially.

Thirouard (2001) describes in his thesis a two-dimensional visualisation system, based on multiple-dye LIF technique, which was designed to conduct direct real-time observation of the oil distribution in the piston ring pack during engine operation. The setup was implemented both in a diesel and a spark ignition single

cylinder engine. First, each type of oil transport (through the grooves, through the gaps and between the lands and the liner) were identified and characterised. Then the physical mechanisms governing each oil transport process were proposed. Dividing the ring pack in three zones, oil transport process were analysed region by region emphasising the effect of the ring design and engine operating conditions on the ability to control the oil supply to the critical areas that are the top ring groove and the crown land.

It was observed that oil accumulated on the piston lands moves up and down in the axial direction-the direction of the piston primary (reciprocating) motion. The observed axial displacement of the lubricant with the piston motion implied that the oil motion was driven by the body force resulting from the piston alternative accelerations and decelerations.

- Oil was observed to move axially on the piston lands and the inertia force resulting from the piston motion was identified as the driving force.
- The amplitude of the oil displacement was found to depend on the engine speed and the oil accumulation volume.

Before the inertia force switches direction, oil will accumulate against the ring. Once oil reaches the ring at the end of the land some oil may be transported out of the land region, through the clearance between the ring and its groove. As far as the oil transport from the land to the ring groove is concerned, the rate at which oil is transported into the groove is greatly affected by the crank angle at which the oil reaches the ring and the duration of the period during which oil accumulates against the ring.

The effect of changing the gas flow rate on the oil flow is of particular interest because it represents a sudden change of engine running conditions. Simulation results show that in this condition the average flow of blow-by gases was going from the crankcase to the combustion chamber. In images, the oil accumulated on the second land was flowing from left to right, which according to the fixed ring gap positions means that the oil was being transported from the second to the top ring gap. Therefore, in this particular case where the flow of blow-by gases was reversed the circumferential oil flow was observed to be reversed as well.

LIF observations showed that the oil layer covering the piston lands could be very non-uniform. The thickness of the oil film was found to have significant variations

in both the axial and the circumferential direction. Experiments showed that the combination of the dragging effect of blow-by gases and the inherent non-uniformity of the piston land oil distribution may result in the formation of steep fronts on the oil surface.

In none of the experimental visualisations conducted with non-pinned rings the oil distribution on the piston land was uniform. In experiments where the top two rings were pinned  $180^\circ$  from each other the gas dragging action was consistent and significant on both sides of the piston. The oil layer was observed to be very uniform on both thrust and anti-thrust sides. This confirms the fact that if the gas dragging effect is steady, it can overcome the effect of non-uniform oil supply and maintain a uniform oil distribution on the lands. Moreover it shows that even though the action of gas dragging tends to evenly distribute the oil on the lands, due to the ring rotation uniform oil distribution is very unlikely to exist on the piston lands of a real engine.

Yu et al (1999) presented an analytical solution of piston ring pack lubrication for two cylinder truck air brake compressor. A detailed analysis of the cavitation occurring at the ring/bore interface is conducted. To describe the lubrication phenomena of the piston ring pack for both fully flooded and starvation conditions a system of non-linear equations with seven variables was developed. It was concluded that a flat face ring has a stronger load capacity than the one with the curved profile. Except for the lower rail of the oil control ring, each remaining ring works in a severely starved lubrication condition during each compression cycle. Each ring suffers a cavitation bubble, enclosed or open. The lower rail of the oil control ring has a significant effect on the performance of the complete ring pack and any measures to enhance the lubrication in the rail will result in power loss reduction and an increase in upward oil transport of the complete ring pack and vice versa.

Ishihara et al (1999) have utilised a modified engine with a transparent glass cylinder for motoring. Oil was prevented to flow through the drain back slots to the oil ring groove. Upper side rail had a smaller end clearance and lower side rail standard end clearance. Thus, it was difficult for the oil to flow through the upper side rail gap onto the third land and it is easy for the oil to return through the lower rail gap to the peripheral surface of the skirt. Using the top ring with a special joint (triangle step joint)

it is difficult for the oil to flow through the top ring gap onto the top land. Combination of the above, reduces the oil flow through the ring pack.

Tian and Wong (2000) developed a theoretical model to study the lubrication, friction, dynamics and oil transport of twin land oil control rings (TLOCR). Results showed that significant oil can pass under one land of the TLOCR along the bore, although the other land perfectly seals the bore. The MOFT between two lands may result in oil accumulation inside the OCR and further oil flow to the upper regions. Piston dynamic tilt is able to create oil accumulation in different regions that may result in irreversible oil transport to upper piston regions.

Rabuté and Tian (2001) considered the high levels of blow-by often encountered under high speed and low load conditions. Their model predicted top ring flutter under these conditions, thus providing an extra gas flow path through the ring groove. The vast majority of blow-by was attributed to this route. This gas transport mechanism would only become significant at high speed/low load conditions, due to the high inertia forces and low pressure acting on the top ring. Top ring radial collapse was also proposed as a route for gas flow in the piston assembly. Here, gas pressure forces the ring away from the cylinder wall providing an additional flow path. This was predicted around TDC firing for high speed and low load conditions, however the mechanism was not as significant as ring flutter.

Seki et al (2000) used the LIF technique to study oil film thickness on the cylinder wall in the region of a diesel engine ring pack. This demonstrated that the oil control ring scraped a large amount of oil down the cylinder wall on the downstroke, while accumulated oil was evident ahead of the scraper ring on the same stroke. This was also seen for the compression ring, though to a lesser degree. Comparison of measured and theoretical minimum oil film thickness for the rings showed good correlation for the top ring; however the comparisons for the second and oil control ring were not so close. The oil film thickness for the oil control ring was found to be up to four times higher than the calculated values, though no explanation for this was given. For the second ring, larger films were measured than predicted on the downstrokes, with the opposite being the case on the up strokes. This was attributed to the differing levels of oil available on the liner to lubricate the rings.

Another detailed lubricant transport model for the piston assembly was developed by Logan (1998) as part of a lubricant degradation study. This included flow of an ideal mixture of gas and oil from the cylinder wall to the ring grooves during ring axial motion from one side of the groove to the other. Build up of oil on the upper face of the rings was also included, being attributed to gravity driven flow down the cylinder and transport of oil in the honing marks on the cylinder wall. This seems unlikely considering engine operating speeds and hence the time available for oil to flow down the cylinder. Oil flow across the ring and oil mist from the accumulated oil on the cylinder to the piston was also included. However, no consideration of lubricant on the piston was made. By varying the efficiency of relative importance of the transport mechanisms considered, the residence time of the oil in the upper region of the cylinder was predicted to be within the range of 10 to 1000 seconds.

Evaporation as an oil consumption mechanism has been considered in the ring pack analysis. Audette and Wong (1999) presented a coupled model, the link was unidirectional, as vaporisation did not impact on oil film thickness on the liner. When applied to a heavy duty diesel engine lubricated with SAE 15W40 oil and operating at 2200 rpm full load, oil accumulation ahead of the top ring detaching at TDC, accounted for a consumption rate of 14g/hour. A 26% reduction in this was predicted if all ring tensions were doubled, while downward scraping of the second ring increased by 50% in this case. In this scenario, the vaporisation of oil from the cylinder was reduced by 7% overall. In the upper region of the cylinder a 23% decrease was predicted, suggesting a correlation between the amount of oil in the upper cylinder region and the level of vaporisation.

A similar model was presented by Wahiduzzaman et al (1992), who predicted oil consumption due to evaporation of oil from the cylinder wall, as part of a ring pack lubrication model. Again the analysis was applied to a heavy duty diesel engine, operating at full load and a speed of 1900 rpm. When lubricated with SAE 15W40 oil, evaporation contributed 2-5% of the overall oil consumption. A lower average consumption rate due to evaporation was predicted with SAE 30 oil. The largest factor determining the evaporation rate for a given oil was unsurprisingly shown to be the liner

temperature. An increase of 40°C produced an 85% increase in evaporation, while a similar reduction in temperature lead to a predicted five-fold decrease.

The contribution of bore distortion to oil consumption has also been studied by Green, Hayward and Tyrell (1989). Their model predicted that the majority of the blow-by passes through the ring gaps and that the increase in blow-by was not observed with the introduction of circumferential bore distortion. However, the reduced conformity between the ring and the liner caused by distortion did lead to higher levels of oil consumption than for a non-distorted bore.

According to Gamble (2002), oil transport mechanisms taking place in the piston assembly are:

- gas driven flow on piston surfaces
- oil mist entrainment in gas flow around the piston assembly and on the cylinder wall
- accumulation of inter-ring volumes
- flow on the cylinder wall through the piston ring gaps
- oil mist entrainment in gas flow through the ring gaps

In general, piston-ring lubrication models calculate the oil flow on the cylinder wall beneath each of the piston rings. Most take no account of any build up of oil that may occur ahead of a piston-ring. If all the oil film ahead of the ring does not flow beneath the ring face as the piston moves along the cylinder oil will begin to accumulate at the leading edge of the ring.

A considerable amount of oil flow can take place in the piston assembly via a number of different transport mechanisms. For all the transport mechanisms the inclusion of secondary piston motion is only of minor significance for the predicted lubricant flow rates.

Shape of piston rings and configuration of piston ring pack have a significant impact on the amount and direction of oil transport. Condition of piston assembly is likely to have a major influence on the length of time that lubrication remains in this region and hence its degradation.

The following conclusions were drawn about oil transport through the ring pack:

⇒ Piston assembly tribology is influenced by piston secondary motion through increase or decrease of ring pack gas flow and inter-ring pressures.

⇒ Knowledge of film thickness is crucial. Lubricant transport on the piston was influenced by inertia and gas flow.

⇒ MOFT is reduced by fuel dilution in lubricant, especially for top compression ring.

The summary for each area covered is as follows:

**a) piston secondary motion**

- Ring pack gas flow is increased or decreased by piston secondary motion and inter-ring pressures. Ring axial dynamics are also altered.

- Ring rotation should be taken into account

**b) lubricant transport mechanism**

- Significant lubricant transport is caused by gas flow and due to inertia.
- OFT is crucial to know
- Oil mist entrained via gas flow over oil films on the piston lands or over oil films at the ring gaps did not produce significant oil transport

- The assumed shape for oil accumulation ahead of the piston ring is an important consideration

- Change of piston rings profile has an impact on the amount of lubricant in the piston assembly and direction of oil transport.

- Piston assembly design and changing component surfaces have an influence on the length of time that lubricant remains in this region and hence its degradation.

**c) influence of degradation on piston assembly tribology**

- Fuel dilution influences lubricant transport in the piston assembly. When this was taken into account reduced viscosity led to lower predictions of oil consumption.

- When the effect of fuel dilution on lubricant viscosity was considered MOFT was significantly reduced (this applies for top ring).

- Oil mist production: Oil can be moved between piston assembly and cylinder wall and vice versa in this manner.

### 1.2.4 THE ROLE OF CAVITATION IN PISTON-RING LUBRICATION

The phenomenon of cavitation in fluids occurs when the local pressure falls below atmospheric. Cavitation in a fluid is the formation of a cavity or cavities of gas or vapour within the body of the fluid owing to the inability of the fluid to sustain large and continuous negative pressures (Priest et al, 2000). The formation of cavities and their subsequent disposition affects the pressure generated in the continuous thin lubricant film, and hence, any integrated quantities such as the load capacity of bearings. There are two forms of cavitation:

- **Gaseous**, in which gas (air) cavities arise in a lubricant film whenever subambient pressures occur and
- **Vaporous**, in which the pressure in a lubricant film falls to its vapour pressure and the lubricant may boil locally at local temperature.

When the pressure becomes sub-atmospheric, cavities are formed by the release of gas present in the solution. Mineral oils normally contain 8-10% by volume of dissolved air which nucleates on pits, cracks and other surface irregularities on the sliding surfaces (heterogeneous nucleation) and also on dirt particles (homogeneous nucleation) (Stachowiak and Bachelor, 1993). The liquid becomes metastable when its saturation pressure is reached. At this point the role of contaminants, gas nuclei etc, becomes important for the formation of cavities (Jones et al, 1999).

Lubricants are usually rich in wear particles and are regularly aerated by churning (Floberg, 1965). The presence of hydrophobic surfaces causes heterogeneous nucleation and leads to much reduced tensile strength. Homogeneous nucleation occurs when bubbles nucleate at particles as stated above. According to Rastogi and Gupta (1986) the cavitation phenomenon may be mitigated by the presence of polymeric additives in the lubricant. The primary purpose of the additives is to reduce the dependence of lubricant viscosity on temperature but the polymer addition also renders the oil mildly viscoelastic and slightly shear-thinning (Rastogi and Gupta, 1986; Arcoumanis et al, 1995). Another publication negotiated the same subject (Deshimaru, 1994) where the influence of the base oil quality and the viscosity enhancing additive on the cavitation behaviour has been presented. According to Dowson and Taylor (1979)



cylinder liners are prone to cavitation erosion. It should be noted that cavitation erosion in bearings has not been frequently observed, and in the few cases that it has occurred its effects have not been serious.

With a convergent-divergent geometry in a liquid film, the occurrence of gaseous cavitation is a natural consequence under all circumstances (Taylor, 1973). The mechanisms responsible for cavitation are:

- 1) reverse squeezing in a converging-diverging geometry
- 2) shear-thinning of the fluid by modifying film thickness
- 3) reciprocating speed, varying load and temperature (variable film thickness)

The mechanism by which a fluid in a film may sustain a negative pressure is through cohesion of adjacent molecules and adhesion of the fluid to the solid boundary. When the applied pressure exceeds either the cohesive or adhesive strength, rupture of the film may occur. In reciprocating piston rings, superambient pressures are generated in the converging wedge and subambient or negative pressures at the diverging wedge of the ring. In general, two forms of film breakdown can exist: the first depends on the balance between viscous shear and surface tension forces and the second is due to vaporous cavitation which may occur if the first type of rupture allows the formation of sufficiently low pressures.

The formation of gas cavities in the diverging region of the lubricant film in a converging-diverging wedge influences the shape of the hydrodynamic pressure profile in the entrainment direction (Priest et al, 2000). Dowson et al (1979) state that the lubricant film cavitates according to the Reynolds criterion and then reforms to ensure that the lubricant pressure is equal to the gas pressure at the trailing edge of the ring. The base of hydrodynamic lubrication was given by Reynolds (1886). He derived the differential equation for the pressure build-up in a thin lubricating oil film but he did not study the cavitation problem despite being aware of the fact that oils cannot withstand negative pressures. Sommerfeld (1904) published a solution of the pressure distribution in the infinitely wide 360° journal bearing with a full oil film all around the circumference and he stated cavitation as a limitation of his theory.

In 1975, Brown and Hamilton installed a small piezo-electric pressure gauge in the cylinder liner of an engine. The results clearly showed the build up of pressure in the inlet region and cavitation in the outlet. Impressive agreement was obtained between one dimensional theory (using the Reynolds boundary condition) and the pressure measurements for the fully flooded condition. The technique was subsequently applied to a running engine although problems were encountered with the gauge due to high temperatures. It was suggested that further work is needed to show whether the negative pressures recorded in the region after the minimum film thickness, showing the effect of cavitation, are meaningful.

In a later publication (Brown and Hamilton, 1978) they measured large negative pressures in the oil film lubricating piston-ring. In a journal bearing a negative pressure can help in supporting the load whereas in a piston-ring the reverse is true. The effect of the negative pressure is to reduce the minimum film thickness and to extend the application of the squeeze film to a larger fraction of the stroke. Kim et al, (1995) have also published oil film pressure measurements; they performed oil film visualisation in a single-ring test rig. Both groups of researchers have observed negative oil film pressures in the diverging section of the piston-ring.

Cameron (1981) states that as oil cannot stand large continuous negative pressures it is likely that they will cause the oil to rupture. Consequently, valid modelling solutions can be derived with the implementation of boundary conditions to connect the oil film region with the cavitating region (Floberg, 1965). In his publication he presented a numerical solution for the pressure distribution in a cylinder-plane geometry. Floberg's model ignored surface tension and assumed that the air cavity spanned the full width of the gap between cylinder and block thus forcing all the fluid into the narrow channels separating adjacent air fingers. He obtained an alternative film rupture boundary condition.

Olsson (1965) who also studied cavitation in dynamically loaded bearings reported two types of cavitation that appeared in his experiments:

- **Finger cavitation**, which is the type of cavitation occurring in steady running bearings. In this case the oil is divided into strips and the gas fingers between them always extend to the trailing edge of the journal bearing used

in the test apparatus, thus having free connection with the atmosphere. Subsequently, pressure in the finger cavitation region can only show minor deviations from atmospheric pressure.

- **Bubble cavitation**, which occurs when the pressure reaches absolute vacuum. Bubbles are gradually transformed into fingers. The position of the bubble zone never reached the trailing edge.

The numerical results have been validated by a flow visualization experiment. The results showed good agreement with Floberg's theory.

Pearson (1960) sought an analytic solution on the basis that the air fingers arose owing to the presence of a secondary disturbance on the main flow. The agreement over a wide range of the relative effect of viscous ( $\tau$ ) and surface tension forces was encouraging. A critical value of  $\tau$  did exist above which the two dimensional flow was stable. Savage (1977) published a mathematical model for describing steady perturbations to a steady and uniform flow of a viscous fluid in which the perturbation to the cavity fluid interface is represented by a small amplitude harmonic wave. A boundary value problem was formulated involving the homogeneous Reynolds equation.

Coyne and Elrod's (1971) theory assumes the presence of a continuous cavity-fluid interface beneath which all the fluid flows to form a uniform layer of a certain thickness.

An elastohydrodynamic lubrication (EHL) algorithm is described by Yang et al (1995). This algorithm combines a compressible fluid model and a pressure velocity relation with cavitation boundary conditions. According to the authors, they acquired more realistic results for piston-ring lubrication which in turn when using the Reynolds boundary condition is considered to be valid only when the pressure acting on the trailing edge of a piston is negligible. Dowson et al (1983) had confirmed that EHL effect is a strong factor in piston-ring lubrication. For a typical top compression ring, the calculated minimum cyclic film thickness more than doubled by taking account of the effect of pressure on viscosity. Also accounting for elastic deformation of the ring and the liner, further doubled the minimum film thickness. This means that elastohydrodynamic action enhances the balance of the load carrying capacity in favour

of the fluid film lubrication in the mixed lubrication regime, and it is therefore of significant importance to friction and wear in the vicinity of the dead centres.

Previous work within the lubrication project at the Internal Combustion Engines Group at Imperial College involved the proposal of a numerical method for prediction of lubricant film parameters which allows the Swift-Stieber, separation, Coyne and Elrod and Floberg boundary conditions to be employed. A delayed cavitation mechanism has been identified (Ostovar, 1996). The Laser Induced Fluorescence measurements in the simplified piston-ring assembly of Pyke (2000) gave an insight of the cavitation occurring at mid-stroke.

The work in the present project concentrates on the development of cavitation throughout the stroke in a idealised test rig. Observation of cavitation in the lubricant film between the piston-ring and the cylinder liner can be a matter of great importance as it is directly related to fluid film rheology phenomena. Understanding of the flow in a simplified single ring rather than in an ordinary ring pack provides information of practical importance and leads to characterisation of the lubricant transport phenomena in the piston-ring pack of production engines.

The simplest solution to the Reynolds equation is obtained by assuming that there is no lubricant film rupture, such that the only boundary conditions are the bounding gas pressures at the inlet and outlet. Sommerfeld proposed this solution for journal bearings and it is generally referred to as the Full Sommerfeld condition. This very simple approach is, however inadequate as it unrealistically requires the fluid to sustain continuously very large and negative pressures and as a consequence underpredicts the hydrodynamic load carrying capacity. Gümbel recognised the inability of the fluid to sustain large and continuous negative pressures and modified the full Sommerfeld condition by proposing that the negative pressures should simply be neglected (i.e zero gauge or absolute atmospheric) effectively producing a cavitated region.

In the cavitated region obtained with the Reynolds cavitation and reformation condition and the modified Reynolds condition (Figure 1.3 b) and d)), it is assumed that a series of gas cavities with oil flowing around them is formed. Therefore there is still hydrodynamic shear stress and a resulting piston force acting on the piston ring.

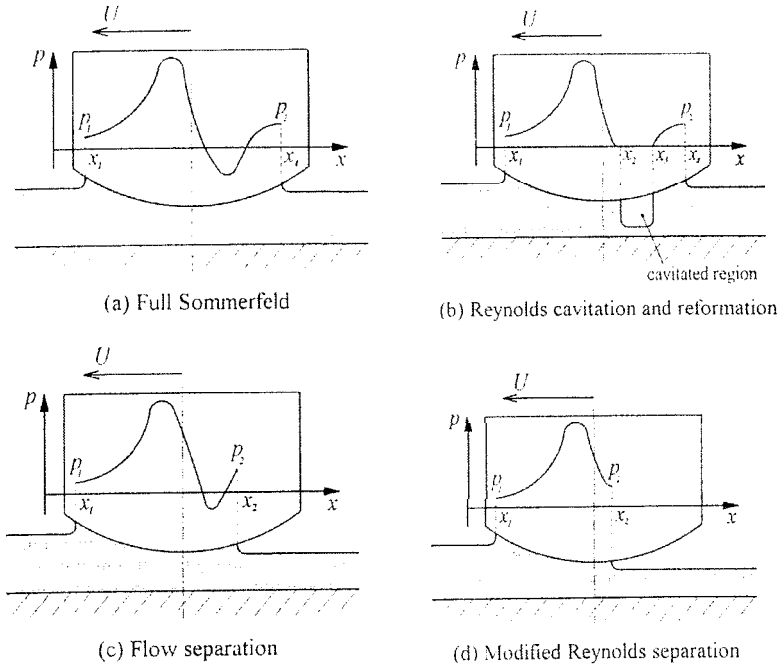


Figure 1.3 Hydrodynamic pressure profile and film shape with alternative cavitation models (Priest et al, 2000)

The actual separation of the two bearing surfaces varies with time. In this case superambient pressures are generated in the lubricating film as the two surfaces move together, while separation of the surfaces results in subambient pressures which may cause cavitation.

Another group of researchers, Bolander et al (2005), modelled the lubrication of an experimental test rig with Reynolds equation. Boundary and mixed lubrication were implemented using the Greenwood-Tripp stochastic approach. The Elrod cavitation algorithm was used to investigate the effects of fluid film rupture and reformation at the top and bottom dead centres. OFT and friction were calculated for a piston-ring operating in the mixed lubrication regime. The assumptions that were made for the modelling were: lubricant is Newtonian fluid with constant viscosity, the pressure remains constant across the film, the flow is laminar, inertia forces were neglected and the lubricant cavitates when the film pressure falls below the lubricant vapour pressure. The mode of cavitation is closed form and the lubrication was considered fully flooded. Actual piston-ring and liner segments were used for the experimental test rig. In their results they presented the transition through the various lubrication regimes with the aid of friction force as well as the speed effect on OFT and

load effect on friction force. Attention is paid to the surface roughness which is used to denote the onset of asperity interaction in the MOFT curve. The shift of MOFT curve at the dead centres is due to squeeze-film effect. A good correlation between the experimental and numerical results was also found. The results give a detailed insight of the entire range of lubrication regimes experienced by the piston-ring from boundary to full-film hydrodynamic lubrication.

### 1.2.5 CALIBRATION ATTEMPTS IN ENGINE LIF APPLICATIONS

As already mentioned before, the calibration issue of the LIF signal has been a major challenge. Smart and Ford (1974) used light by a mercury lamp to illuminate the lubricant film on a spinning shaft. For the calibration of the signal, they used a known volume of oil over a known surface area. The calibration was found to be nearly linear with the exception of thicker lubricant films.

Ting (1980) suggested grooves on piston-ring for the calibration of the LIF signal. A single cylinder engine with transparent sleeve was used for the fluorescence. It was the first LIF engine application, no calibration of LIF signal was presented however.

Hoult et al (1988) made the first attempt to calibrate in-situ the measurement of oil film thickness in the piston-ring zone of a firing engine fitted with quartz glass window.

Two independent methods were used A bench calibration was performed using lubricant films of constant, known thickness. Shims were placed between optically flat microscope slides. Lubricant was drawn between the slides by capillary action and the sides were clamped and edges sealed. Dynamic calibration which involves discerning a ring contour from the fluorescent intensity measured as a voltage versus time, during engine operation as the ring passes through the beam. This voltage profile contour is compared to the ring measured beforehand. Piston-ring profiles were measured before installation with a Talysurf profilometer.

They concluded that there is agreement of in-situ dynamic with the bench calibration and that the wear pattern on the cylinder liner demonstrates that the window inserted into the liner does affect the lubrication of the cylinder and piston.

Calibration was performed with marks on piston by Lux et al (1990). For their experiments they used a fired single cylinder diesel engine with quartz window. The identifying marks on the piston were low amplitude periodic trace directly behind the largest film thickness readings. The point directly behind this corresponds to the start of the piston skirt area. Machine tool marks are on the piston starting at this point. A digitized profile of the skirt tool marks matches the periodic signal observed in the film thickness trace. The agreement between the start of the periodic trace and the tool marks verifies that the pistons features have been located correctly on the film thickness trace. The voltage is linearly proportional to oil film thickness up to approximately 200  $\mu\text{m}$ .

They concluded that large differences were found between the statically and dynamically obtained calibration coefficients. The dynamically obtained coefficients varied with speed and load owing to the varying temperature.

Wong and Hoult (1991) used two identical small production IDI diesel engines and five different ring configurations. For the calibration three chemically etched marks of known depths were created on the piston to provide distinguishing marks for calibration in addition to the scraper ring profile. Two grooves were etched on the piston skirt and one on the piston land. The calibration marks are installed to provide a dynamic reference for converting the photomultiplier voltage output to oil film thickness. The etched marks on the piston skirt should provide almost the same calibration and their repeatability and consistency is checked by comparing the two local calibration coefficients. Depth of etched grooves in piston skirt: 11.1  $\mu\text{m}$  and 19.6  $\mu\text{m}$ .

Richardson and Borman (1991) suggested that the method of ring profile fitting has to deal with problems arising from oil starvation and ring twist. They proposed the two following calibration methods:

- Dynamic Calibration: Calibration steps were ground into the third piston-ring. The third ring was pinned to ensure that the steps passed over the measurement region. The steps were ground to depths of 15 and 20  $\mu\text{m}$ . These were the smaller sized steps that the machinists could grind into the ring face but proved to be too large-sized

to provide a calibration coefficient. A solution was proposed with smaller steps; a coating on the ring or an etching process might be used to produce the small steps required (plating or etching).

- Static calibration: A bench top fluorescence calibration cell was used which had the facility to continuously vary the film thickness.

Brown et al (1993) proposed an in-situ calibration method with grooves in the piston skirt and top compression ring with depths that are readily machined or spark eroded (i.e.  $>25\text{ }\mu\text{m}$ ). Their filling behaviour was stochastic in nature and that generally there was insufficient oil in the contact to support complete filling of the grooves. Thus, when time averaged, the intensity of the fluorescent signal suggested that the grooves were continuously under-filled, rendering them unsuitable for calibration purposes. Therefore, to quantify properly the relationship between oil film fluorescence intensities and oil film thickness and the temperature dependence of the fluorescing quenching effect, a bench-top calibration procedure was chosen.

Phen et al (1993) proposed ring profile fitting for calibration but it is subject to serious error due to variation of reflectance over the ring surface. There was also inconsistency when comparing the actual measured profiles of the rings versus what was measured by the fibres as the ring passed by. As a result, a calibration coefficient was chosen for each fibre such that the profiles matched. The method of using the ring profiles for a calibration is inherently inaccurate because it is based on the judgement of the person doing the comparison. Static calibration was used as well; the static test rig showed that the calibration was linear.

Scanning LIF was applied to a transparent and a normal cylinder with quartz window from Konomi et al (1993). A special gauge with nickel-plated steps of  $2.2\text{ }\mu\text{m}$  in the range of  $2\text{--}10\text{ }\mu\text{m}$  was used for calibration of the LIF signal. It is suggested that the absolute value of lubricant film thickness can be obtained from the data at a groove of known depth produced on a piston wall where oil is sufficiently applied because of the linearity between the photomultiplier output and oil film thickness in high temperature as in firing operating condition.

Dearlove and Cheng (1995) used a reciprocating test rig to measure friction of a single piston ring against a liner segment. A chemically etched calibration groove on



the ring, 25  $\mu\text{m}$  deep 150  $\mu\text{m}$  long was used. The width of the groove was approximately 10% of the ring thickness. The assumption was that this groove would not substantially alter the oil flow under the ring and the frictional behaviour. The calibration procedure was to adjust the gain of measurement system (oil film thickness to photomultiplier voltage ratio) to match the signal trace and the physical dimension of the groove, which has been determined by a Talysurf instrument. While the groove height matches, the calculated profile has a ring thickness narrower than the actual ring thickness. Then the groove depth was underestimated by  $\sim 40\%$ . This allowed the use of an unmodified ring in the apparatus and there was no uncertainty as to whether the pressure of the calibration groove would change the frictional behaviour. With the use of the grooved ring it was found that the groove wore out in use and did not provide a useful calibration throughout the test period. The uncertainty of the LIF film thickness measurement is estimated to be of the order of 40%.

Measurements of minimum oil film thickness with the capacitance technique were presented by Stiyer and Ghandi (1997). At the minimum oil film thickness the capacitance technique provides an accurate measurement of the ring-wall distance and this value is used as a reference for the photomultiplier voltage, giving a calibration coefficient. Again the same ring fitting method problems were noticed: the term good fit and the method by which it is achieved is highly subjective. The location where the oil attaches to the ring face is not precisely known and as the vertical scaling is altered this location changes.

Also the exact profile of the piston ring at the location where the ring crosses in front of the fibre cannot be known. The ring profile is measured with a profilometer before the piston ring is installed in the engine. Once the piston is used in the engine, the profile will begin to change due to normal wear. If the piston-ring was brand new when measured, the profile will change even more due to greater wear. Additionally, the piston-ring profile is not constant at all circumferential locations around the ring. It would be very difficult to match the correct measured profile with the actual profile that is seen by the fibre. Lastly the effect of ring twist will also alter the actual ring profile during engine operation.

Froelund et al (1997) used a 4 cylinder spark ignition engine, LIF focusing probe at 83% down from TDC. For the calibration procedure, toolmarks on the piston skirt found by surface roughness measurements are compared with the raw LIF signal from the same piston skirt region. If the height of the tool marks corresponds with the same height on the LIF signal, the right calibration factor has been found, assuming that the toolmarks are fully flooded with oil.

Nakayama et al (2003) proposed both a static and dynamic calibration method used for their measurements of oil film thickness in an engine bearing. Optical fibres sensors were installed within the bearing. The static calibration method is based on a test rig that has a known oil film thickness that is formed by two thickness gauges. On the other hand, the dynamic calibration is based on a step created by two gauges of known thickness. It was made sure that it was filled with oil and that the film thickness was measured exactly. The measurement value was in accordance to the one derived from the static calibration test rig.

Arcoumanis et al (1998) proposed a static calibration method for the Lister-Petter single cylinder diesel engine LIF results. The method consists of a micrometer of enhanced resolution and makes use of an optical fibre mounted flush with the anvil of a micrometer. This calibration method enables measurement of continuous variation of the oil film thickness.

In his PhD thesis, Duszynski (1999) performs a quantitative study of the temperature effect on the calibration coefficients of all the engine oils tested with a custom made calibration apparatus. It consists of 20 x 20 x 100 mm block of fused silica on which the calibration standard was in the form of grooves of known depth. The selection of the material was made on the base of its physical properties and in particular its low thermal expansion. The grooves etched into the surface of the block were intended to be incremental in depth between 2 and 10  $\mu\text{m}$ , in steps of 2  $\mu\text{m}$ . From the early engine tests, it was found that the oil film thickness under the piston rings varied within that range. This range proved the most difficult during calibration of the engine oils using the micrometer set-up described in Arcoumanis et al (1998). For the thinnest oil film thicknesses the lowest repeatability of the calibration procedure was observed. Therefore, obtaining calibration coefficients of high accuracy in the region between 0

and 10  $\mu\text{m}$  is vital for improving the accuracy of the LIF system. Calibrating the LIF signal in this range of oil film thickness might also compensate for some non-linear effects in the fluorescence signal which can be observed for very thin lubricant films. Prior to the process of etching the grooves, one of the 20 x 100 mm sides of the silica block was polished flat to achieve accuracy of  $1\lambda$ , where  $\lambda=632\text{ nm}$ . The flat surface was then coated with a 500 nm protective layer of chromium and a uniform UV-sensitive photomask. A micrometer anvil as the one used in the ambient calibration apparatus was used in this arrangement to create oil films of known thicknesses. This was achieved simply by placing the anvil on top of the flat surface of the calibration block and across a groove of desired depth. Thus the anvil bridged the gap between two consecutive uprights of the crenellated surface of the standard. Holding both parts tightly together, and pumping the oils through the hypodermic needle into the space between the anvil and the bottom of the groove provided an oil film of known thickness. The whole apparatus was submerged in an oil bath of controlled temperature. A K-type thermocouple built into the anvil immediately next to the optical fibre was used to monitor the temperature of the oil flowing through the interrogation zone. Sampling the LIF signal while changing the temperature of the whole apparatus enabled calibration coefficients for the engine oils at elevated temperatures to be obtained. Data were sampled for the temperature range 20 to 100°C. The upper limit represented the temperature of the liner measured during firing engine tests. These temperatures were measured by the thermocouples built into the LIF cylinder probes. Before any calibration could commence, the quality of the bottom surface of each groove had to be examined. In order to ensure the repeatability of the calibration for any arbitrary placement of the micrometer anvil across a groove, the surface of the bottom of the groove used had to be flat and parallel to the interface surface between the anvil and the top surface of the block. Prior to performing the calibration, it was also confirmed that for the micrometer anvil resting flat on a clean surface of the calibration block, with any oil residue between the anvil and the block removed, the PMT voltage response detected was the same as that of a closed calibration micrometer. Since the oil film thicknesses under the piston rings had been found to be below 10  $\mu\text{m}$ , it was decided that the deeper calibration groove (9.1 $\mu\text{m}$ ) would be used to obtain the second data point. This

calibration methodology was employed to obtain calibration coefficients at elevated temperatures. The data acquired during the firing engine tests were post-processed using the calibration coefficients obtained at the temperature of 100°C.

### 1.3 THESIS OBJECTIVES

The interface between the piston, piston – rings and the cylinder liner represents one of the most hostile environments the crankcase lubricant experiences in reciprocating engines. Very thin films of lubricant, typically of a few microns are exposed to extreme thermal and mechanical stresses as well as to labile by – products of combustion. Rapid changes in the piston sliding speed, the temperature of the metal surfaces and the in – cylinder gas pressure impose demanding operating conditions on the piston – rings. The end result is that the lubrication regime of the piston – rings and the piston skirt oscillates between fully flooded hydrodynamic lubrication at mid-stroke and boundary/elastohydrodynamic lubrication which prevails near the piston reversal points. Thus, the operation of the piston ring pack has direct impact on engine performance. The demand for more engine power output, urges the tribology between the sliding surfaces of the liner wall and ring running face and the piston dynamics towards more detailed investigations. Oil transport on the cylinder walls can affect not only consumption but emissions. To meet the upcoming emissions standard and beyond, advances need to be made in the understanding and control of oil consumption and oil transport. Reduction of oil consumption is required to satisfy the factors that currently govern the development of automotive engines: utilisation of hydrocarbon resources, protection of the environment and customer satisfaction. According to estimates, more than 50% of the engine oil consumption in automotive engines is from the piston – ring pack (Thirouard et al, 1998). The ring pack controls the lubricant flow into the combustion chamber as well as transfers heat from piston towards the liner or cylinder wall. There should be a compromise between supplying a sufficient quantity of oil to lubricate the piston assembly while minimising how much oil will reach the combustion chamber. Oil transported to this region may be burnt thus contributing to the level of emissions from the engine.

The design of piston and piston – rings as well as the physical and chemical properties of the lubricant determine the frictional power loss, the rate of wear of the cylinder liner and scuffing of the piston – rings, the oil consumption and its degradation as well as the level of emissions of unburned hydrocarbons and particulates. The governing factors in oil flow at the interface between the piston and the liner are the ring and groove geometry along with temperature and pressure conditions. The improvement of the friction loss without an increase in oil consumption is strongly required. For resolving this issue, detailed investigations of the behaviour and thickness distribution of a lubricant film between a piston, piston rings and cylinder liner is essential. Despite its importance, the understanding of the complex phenomena associated with the lubrication of piston – rings still remains incomplete. To understand fully the effect of these parameters on oil transport in the piston – ring pack, plenty of experimental data are essential. The ability to determine the lubricant availability at the inlet and outlet of the piston – ring, through OFT measurement is of great importance (Arcoumanis et al, 1998).

The investigation of lubricant cavitation highlighted the deficiency of LIF single – point measurements for obtaining a clear image of piston – ring lubrication as there was uncertainty in the variation of the cavitation along the ring (Pyke, 2000).

Following the previous work within the Internal Combustion Engines Group, this thesis takes the previous findings a step further and keeping as a basic concept the characterisation of OFT between the piston assembly and the cylinder liner over a range of operating conditions and lubricants. This work concentrates on getting a clearer insight of the cavitation mechanism on piston – ring lubrication which was previously identified in the LIF results of the single – ring test rig. The convergent – divergent passage between the piston ring profile and cylinder liner suggests that the oil film may experience cavitation. For a fully – flooded piston – ring the enclosed cavitation is very likely to occur whereas for a starved piston – ring, where there is no sufficient oil to support the piston – ring, the frequently occurred flow pattern is the open cavitation one (Yu et al, 1999).

So, modifying the liner segment and installing a quartz window section will provide the images necessary for the identification and interpretation of cavitation that

takes place at the interface between the piston-ring and the cylinder liner at the reciprocating test rig. Introducing the miniature pressure transducer for the oil film pressure measurements is a novel technique. The size of the transducer is critical for these measurements and prior to applying similar measuring techniques to the engine, it had to be certain that the oil film pressure measurements were repeatable and the results were correctly interpreted in the case of the single ring test rig.

The important issue of LIF calibration is being encountered with the introduction of a grooved piston-ring specimen to be fitted in the single ring test rig. This proposal has the inherent advantage of using the simulation test rig instead of a real engine to produce the dynamic calibration results and thus, one can concentrate on finding possible reasons behind any signal calibration inconsistency by means of a much easier way. Another objective is to use the test rig to provide consistency between experimental and theoretical results through the acquired signals at different testing conditions. Eventually, a closer study on friction results and oil film thickness at increased temperature will be interpreted in a much safer way and conclusions can be drawn about the piston ring behaviour in the real piston-cylinder assembly.

The basic limitation of the simulation test rig study lies in the geometrical restrictions imposed by the rig design itself. Ring rotation, piston secondary motion, gas flow, ring groove and piston-land interaction is not accounted for. Since in an engine the liner and piston-ring configuration are reversed relative to the examined test rig, any direct extrapolation of the conclusions should be treated with some extra caution.

Imaging and oil film pressure measurements are considered the necessary steps towards understanding fully the onset and development stages of cavitation during the stroke. The knowledge of the driving forces behind oil transport and the effect of fluid film lubrication (fully flooded or starved) in oil availability under the piston-rings of a ring pack is essential. A clear picture of the physics behind the cavitation phenomena will bring more light towards understanding the lubricant transport in the piston – ring pack. The large scale model of the single – ring test rig provides the necessary information and abundance of parametric OFT, friction and oil film pressure results without the interference of the uncertainties and complexities involving the engine ring pack operation. Another objective is to compare the simulation test rig

visualisation results with these that are derived from the motoring engine testing and where possible to establish a correlation between the similar findings. Thus, having the single ring test rig as a base for understanding and interpreting the piston – ring lubrication related phenomena, the real size scale of the engine can be studied in a similar manner and the emerging results can be directly compared and explained by means of a much safer way. So, after the installation of the pressure transducer the following objectives were set:

- Get pressure readings and analyse the effect of speed, load, stroke variation, ring-face geometry and support the visualisation findings
- Implement same technique in the engine and compare the effect of the above parameters

## 1.4 SUMMARY AND CONTENT OF THESIS

The present thesis consists of six chapters including Chapter 1 where previous work on oil film transport, cavitation, oil film measurement techniques, lubrication and emissions as well as Laser Induced Fluorescence calibration is being summarised and reviewed.

Chapter 2 reviews thoroughly one of the latest relevant publications in the area of static and dynamic calibration of the Laser Induced Fluorescence signal. Another dynamic method is proposed and a comparison with static calibration experimental results is given. Reasons for the inconsistency of their comparison are being presented and attention is paid on the fibre power output measurement between different fibre optic installations. The capacitance technique is also analysed as well as the concept of LIF.

Extensive parametric lubricant study is given in Chapter 3. Capacitance, LIF and friction measurements are acquired from an idealised single ring test rig and their variation with speed, load, temperature and oil type is being discussed.

Chapter 4 presents experimental results from the single ring test rig, this time introducing new techniques. Oil film is visualised during the stroke and various transient shapes of cavities were identified during the stroke. Oil film pressure measurements

complement the imaging experiments and provide a physical statement towards their inception and development throughout the stroke. Additionally, a parametric study of the effect of speed, load, temperature and piston ring curvature on oil film pressure is explained.

Engine application of the visualisation experiments occupies Chapter 5. The modification of the engine block to allow for camera imaging is described and an approach through the introduction of cavitation factors and calculation of residence time for the test rig and engine is described.

Chapter 6 concludes the thesis with the summary of major findings and it also provides recommendations for further work in the area of engine lubricant testing.

A list of relevant publications which were reviewed and referenced in the thesis, is given in the *REFERENCES* section.



## CHAPTER 2

### DIAGNOSTICS-EXPERIMENTAL TECHNIQUES-CALIBRATION

#### INTRODUCTION

This chapter presents the two techniques of Laser Induced Fluorescence and capacitance that has been used for the oil film thickness measurements. Previous static and dynamic LIF calibration methods are briefly discussed and an approach to dynamic calibration is presented. It is directly compared to the micrometer based static calibration technique and its variation with temperature is considered and discussed.

#### 2.1 LASER INDUCED FLUORESCENCE

The combination of fibre optics and laser induced fluorescence for measuring the oil film thickness is a non-intrusive method. The concept of the LIF method is based on the fact that when laser light shines onto an oil film, the oil will fluoresce. Fluorescence arises when molecules in the oil film which are electronically excited due to photon absorption, subsequently relax to their ground electronic state. This relaxation process occurs virtually instantaneously (within a few nanoseconds) after the initial excitation and results in the emission of photons of lower energy than the excitation light. Spectral discrimination of this emission can be readily achieved by appropriate optical filtering so as to remove the scattered light which originates from the excitation source. The intensity of the fluorescence is related to the thickness of the oil film. Since this method is measuring actual oil volume, oil films on the cylinder wall can be examined throughout the cycle.

Once optical access to the lubricating contact of interest has been achieved, then the application of fluorescence techniques to the measurement of lubrication phenomena does offer several advantageous features. The signal yield is exclusively due to fluorescence from the oil within the contact apart from a negligibly small contribution due to fluorescence excited on the surface of metallic components within the engine.

More importantly, the rapid electronic relaxation time of the oil under the fluorescence excitation means that the temporal resolution of the technique will be limited by the frequency bandwidth of the detector and the sampling rates of the data acquisition electronics. The spatial resolution will be governed by the effective area illuminated by the excited source. Using conventional optics it is relatively easy to produce an interrogation spot with a diameter (50-200  $\mu\text{m}$ ) which is significantly less than the width of a typical piston-ring (2-3 mm).

An air-cooled argon-ion laser operating at a wavelength of 488 nm provided blue excitation light. The laser power was adjusted to 50 mW at which the laser stabilising system became active. A wavelength selective beam splitter (dichroic mirror) reflecting blue laser light and transmitting green fluorescent light emitted by the measured lubricant film was used to direct the laser beam towards the fibre coupler. The low back reflection fibre coupler is capable of accommodating beams between 0.6 and 1.2 mm. The laser light is then launched into the optical fibres via the coupler's FC receptacle and a fibre adaptor. The optical fibre was a graded-index multimode fibre with a 30  $\mu\text{m}$ -thick aluminium protective coating capable of withstanding temperatures of up to 400°C. The diameter of the core of the fibre is 50  $\mu\text{m}$ , the cladding 125  $\mu\text{m}$  and the outer diameter of the coating 175  $\mu\text{m}$ .

### 2.1.1 THE FLUORESCENCE PROCESS

Fluorescence is the result of a three-stage process that occurs in certain molecules called fluorophores or fluorescent dyes. The process is explained by the following schematic:

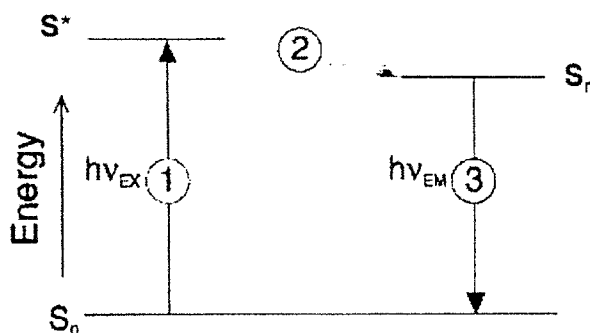


Figure 2.1 Fluorescence process

- Stage 1: Excitation

A photon of energy  $h_{\text{ex}}$  (wavelength  $\nu_{\text{ex}}$ ) is supplied by an external source such as a flash lamp or a laser and is absorbed by the fluorescent dye, creating an excited electronic singlet-state  $S^*$ .

- Stage 2: Excited State lifetime

The excited state exists for a finite time (typically 1 to 10 nsec). During this time, the fluorescence undergoes conformational changes and is also subject to a multitude of possible interactions with its molecular environment. These processes have two important consequences:

- The energy of  $S^*$  is partly dissipated yielding to a relaxed singlet excited state ( $S_r$ ) from which fluorescence emission originates.
- Not all the molecules initially excited by absorption return to the ground state ( $S_0$ ) by fluorescence emission. Other processes such as collisional quenching and fluorescence energy transfer may also depopulate  $S_r$ . The fluorescence quantum yield, which is the ratio of the number of fluorescence photons emitted to the number of photons absorbed, is a measure of the relative extent to which these processes occur.

- Stage 3: Fluorescence Emission

A photon of energy  $h_{\text{em}}$  (wavelength  $\nu_{\text{em}}$ ) is emitted as the fluorophore returns to its ground state  $S_0$ . Due to the energy dissipation during the excited state lifetime, the

energy of this photon is lower and therefore its wavelength longer than the excitation one.

The difference between the wavelength of the excitation source and that of the maximum of the resulting fluorescence is known as Stokes shift. The Stokes shift allows emission photons to be isolated from excitation photons. The entire fluorescence process is cyclical. Unless the fluorescent dye is irreversibly destroyed in the excited state (photobleaching), the same dye molecule can be repeatedly excited and detected.

#### The fluorescence Spectra:

For polyatomic molecules in solution, the discrete electronic transitions of Figure 2.1 are replaced by rather broad energy spectra called the fluorescence excitation spectrum and the fluorescence emission spectrum (Figure 2.2). The bandwidth of these spectra is a parameter of particular importance for applications in which two or more different fluorescing dyes are simultaneously used. Under the same conditions, the fluorescence emission spectrum is independent of the excitation wavelength due to partial dissipation of excitation energy during the excited state lifetime.

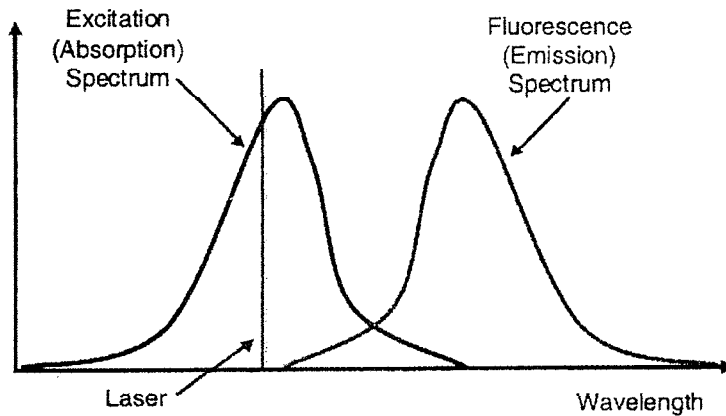


Figure 2.2 Excitation and absorption spectrum

#### Theory:

To estimate the intensity of the emitted fluorescence, the amount of photons absorbed must be evaluated first. The Beer-Lambert law expresses the amount of transmitted light as a function of the absorbing medium thickness.

$$I_a = I_0(1 - 10^{-\epsilon ch}) \quad (2.1)$$

where  $I_a$  is the intensity of the absorbed light,  $I_o$  the intensity of the incident light source,  $\epsilon$  the molar absorptivity,  $h$  the path length (lubricant film thickness) and  $x$  the molar concentration of the fluorescing species. Both the molar absorptivity and the molar concentration of the fluorescing species may vary with temperature and also with the lubricant composition, which changes in the top ring zone due to lubricant degradation by the extreme conditions associated with combustion.

The quantum yield of fluorescence ( $\Phi_L$ ) is defined as the ratio of quanta emitted to absorbed and is normally less than 100% ( $0 \leq \Phi_L \leq 1$ ). The spectral fluorescence power yield  $P_{VL}$  over a frequency band  $\nu + d\nu$  is defined as the ratio of the radiant power of fluorescence, measured over that frequency interval, to the radiant power of absorbed

$$\text{energy, i.e. } P_{VL} = \frac{\Phi_{VL}}{\Phi_A} \quad (2.2)$$

The fluorescence power yield will normally decrease with an increase in the excitation frequency as a greater fraction of the absorbed energy will be converted to heat.

Because internal conversion is much more likely than emission directly from electronic states higher than the lowest one, fluorescence emission typically occurs from the singlet excited state  $S^*$ . This is the reason why fluorescence quantum yield is related to absorption and emission from the first excited state and strongly depends on the number of molecules with fluorescent properties (fluorophores). The fluorescence quantum efficiency is thus defined as the ratio of fluorescent radiant power to the absorbed radiant power:

$$\phi_F = \frac{\Phi_F}{\Phi_A} \quad (2.3)$$

The rate of change of the number of molecules populating the first excited singlet state can be expressed by the following equation:

$$\frac{dn_{S_1}}{dt} = k_A n_{S_0} - (k_F + k_{NR}) n_{S_1} \quad (2.4)$$

where  $n_{S_0}$  and  $n_{S_1}$  are the densities of the ground and first excited state in molecules/mm<sup>3</sup> and  $k_A$ ,  $k_F$  and  $k_{NR}$  are constants of absorption, fluorescence and non radiative deactivation respectively. If a sample of a fluorophore is irradiated with light of constant intensity, a steady state population of the first excited singlet state will be achieved almost instantaneously. Applying this condition to the above equation gives

$$n_{S_1} = \frac{k_A n_{S_0}}{k_F + k_{NR}} \quad (2.5)$$

This population density of the first excited state is directly proportional to the rate of absorption and fluorescence emission can be expressed by:

$$\Phi_A = k_A n_{S_0} V \quad (2.6)$$

$$\Phi_F = k_F n_{S_1} V \quad (2.7)$$

Where  $V$  is a sample volume. In the case of the multi-mode fibre-optic probes mounted inside the cylinder liner, this volume is equal to the volume of the frustum of the cone formed by the 50  $\mu\text{m}$  core of the fibre, the circular area on a piston ring or the piston itself illuminated by the laser beam diverging from the fibre and the side surface of the truncated cone at an angle equal to the fibre acceptance angle. The fibre acceptance angle is defined as:

$$A_{\max} = \arcsin\left(\frac{NA}{n_o}\right) \quad (2.8)$$

where  $NA$  the numerical aperture of the fibre and  $n_o$  is the refractive index of the external medium-the lubricant present between the cylinder wall and the piston and piston rings. Combining equations 2.3, 2.5, 2.6 and 2.7 gives another expression of the fluorescence efficiency:

$$\Phi_F = \frac{k_F}{k_F + k_{NR}} \quad (2.9)$$

which shows the effect of non-radiative deactivation of fluorescence quantum yield. Using this equation together with the substitution of equation 2.5 for the density  $n_{S_1}$  in equation 2.7 results in the following expression for fluorescence emission

$$\Phi_F = n_{S_0} k_A \phi_F V \quad (2.10)$$

The above expression shows that fluorescence intensity is a function of the density of the fluorescence species present in the ground electronic state, the coefficient of absorption, the fluorescence quantum yield and the volume of a sample. Assuming that the lubricant present inside the cone of laser light between the tip of the fibre and the piston is continuously replenished and the rate of absorption is the same, the intensity of the emitted fluorescence is proportional to the thickness of the oil film. The volume of the frustum of a cone of oil is not a linear function of the oil film thickness, but, due to the divergent laser beam, it is governed by a cubic relationship.

The fluorescent quantum yield of engine oils is predominantly caused by the presence of polycyclic aromatic hydrocarbons (PAH). The intense fluorescence of these molecules is attributed to the low energy levels of their excited singlet state. Because PAH's require a relatively small amount of absorbed energy for the electronic transmissions to occur, the rate of band disruption is low. High excitation energy is one of the reasons for bond rupturing occurring after absorption (dissociation) causing fluorescence quenching.

Environmental factors such as temperature can bring about significant changes in the viscosity of engine oils and their fluorescence spectra. The main condition for fluorescence to occur is the stability of the excited electronic state. In liquid oils, where the probability of collisional quenching is high, an increase of temperature will normally disturb the stability of molecules in the excited state and will have a marked effect on the characteristics of fluorescence emission. A higher temperature of the fluorophore encourages conversion to the triplet state or other energy transfer reactions in which electrons from non-fluorescent molecules fill the ground electronic state thus blocking the return of the original electron whose energy, in turn, is deactivated through a radiationless transition such as dynamic quenching. Aromatic hydrocarbons present in engine oils will normally undergo increased dynamic quenching with increasing temperature, or decreasing viscosity. These intermolecular quenching mechanisms are controlled by the rate of diffusion in the oil and thus the fluorescence quantum yield will depend on viscosity and temperature.

As a consequence, increasing the temperature decreases fluorescence emission. This statement, however, is a very general one as there might exist

circumstances in which the effect of temperature could be found to be quite the opposite; increased temperature could bring about increased molecular fluorescence quantum yield. The specific conditions under which such phenomena could take place are related to both the type of fluorescent oil molecules populating the ground electronic state and the configuration of the experimental apparatus used for quantitative fluorescence studies. Amongst the possible circumstances conducive to increasing the fluorescence emission of engine oils with increasing temperature are:

- Dimmer formation
- Indirect excitation
- Spectral shift

The complex mixture of molecules in engine oils will almost certainly undergo dimerization. Dimmers can be defined as molecular structures of two similar molecules combined by forces other than chemical bonds (electrostatic interactions, van der Waals bonding enhanced by electronic transitions) or very weak chemical bonds, such as hydrogen bonds. Dimmers can be formed between one molecule in the ground electronic state and another molecule in an excited state. Since the excitation energy is always divided equally between the two molecules forming a dimer, dimerization will lead to deactivation of the absorbed energy through internal conversion and reduction of fluorescence quantum yield. With the increase in temperature, the level of dimer formation will be lower and, thus, dynamic quenching of fluorescence reduces. Disrupted dimer formation by increased diffusion at elevated temperatures results in monomer fluorescence of higher intensity.

In the process of indirect excitation, the reverse singlet transition is brought about by thermal energy. The temperature assisted populating of the first electronic excited state can give rise to delayed fluorescence.

The receiving optics of the LIF experimental system could also be held responsible for detecting a higher level of fluorescence intensity with increasing temperature of engine oils. A slight increase in the fluorescence signal could occur. This apparent increase may not necessarily be the result of higher fluorescent intensity, but may simply be the result of a spectral shift of the fluorescence peak. The phenomenon of blue shift in the fluorescence emission maxima could occur with an increase in



temperature. The phenomenon can be associated with a Franck-Cordon excited state at lower temperatures. This effect, however, could be compensated for either by the bandwidth of the interference filter or by employing a monochromator in the receiving optics of the LIF system.

## 2.2 TEMPERATURE EFFECT ON FLUORESCENCE

Fluorescence is strongly dependent on temperature. The fluorescing properties of dyes can be permanently or temporary affected by temperature. First of all, for any dye if the temperature is elevated high enough, the chemical structure of molecules will be modified and the dye will lose its ability to fluoresce. This phenomenon, which is called thermal bleaching, must be avoided by choosing dyes that can stand the temperature variations experienced by the dye/solvent solution. Secondly, below the thermal bleaching temperature, the absorptivity  $\epsilon_{\text{dye}}$ , and the quantum yield efficiency  $\phi$  and the absorption and emission of a dye are function of temperature. Absorptivity usually decreases as temperature increases. According to Schäfer (1973), the decay of absorptivity results from a broadening of the absorption spectra induced by an increased molecular vibration level. The quantum yield efficiency  $\phi$  usually decreases with temperature. This is also related to the increased level of molecular vibration (Guilbault, 1990; Schäfer, 1973; Duarte and Hillman, 1990). According to Schäfer (1973), the non radiative decay (relaxation) from the excited state to the ground state is enhanced by molecular vibrations. According to Duarte and Hillman (1990) and Guilbault (1990), the decay of the quantum yield efficiency can be explained by the increasing propability of collisional quenching as the temperature rises. Guilbault (1990) additionally observed that the quantum yield efficiency was decreasing if the viscosity of the solvent was reduced. In many solvents and certainly in engine oil, the viscosity dependency of the fluorescence efficiency will result in an additional temperature decay of fluorescence. Eventually, as long as the chemical structure of the dye molecule is not altered, the temperature effects are reversible.

All the temperature related phenomena that affect the fluorescence emission are strongly related to the chemical structure of the molecule and are thus very dye

specific. Moreover, due to the complexity of the phenomena involved, the amplitude of the temperature effect on the dye fluorescing properties are very difficult to predict and must therefore be experimentally studied.

2.2.1 TEMPERATURE VARIATION ON THE PISTON

The temperature of the oil/dye solution lying on the piston can be assumed to be equal to the piston surface temperature. It is well known that there is a significant temperature gradient along the piston when the engine is firing (Figure 2.3). This gradient, which increases with the engine load, can be such that the crown land surface temperature is above 250°C while the skirt temperature is only 80°C. The fluorescence efficiency of any dye is strongly affected over such a wide range of temperature. This was found to be a problem in previous oil film thickness measurement using the LIF technique (Hoult, 1988; Saeki and Hart, 2001). In Thirouard (2001) thesis, the experiments conducted showed that, at 170°C, the fluorescence emission of some commonly used dyes could be decreased to an extent that it was not detectable. Consequently, LIF images of the oil distribution on the piston lands are very likely to be greatly altered by the oil temperature distribution.

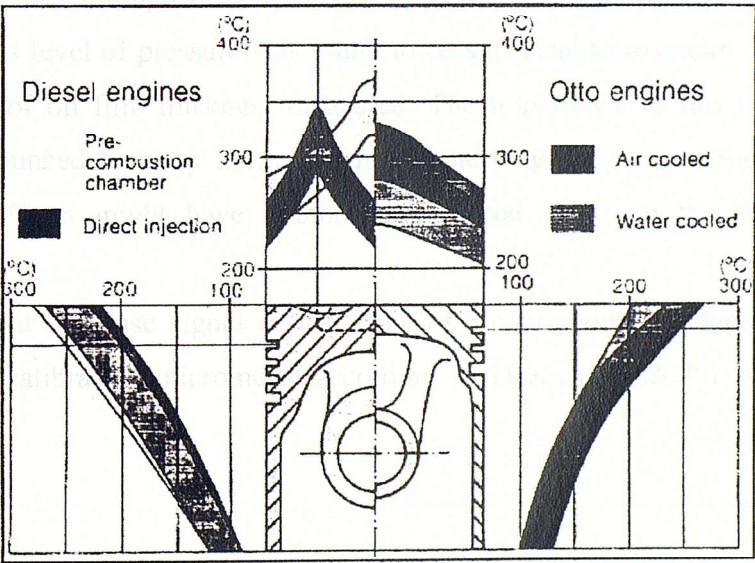


Figure 2.3 Piston operating temperatures in vehicle engines at full load (Source: Mahle Mini Piston Manual)

## 2.3 LIF CALIBRATION-GENERAL

The LIF calibration issue has been presented in paragraph 1.2.5. Different proposals were submitted which covered both the issue of static and dynamic calibration. In the following paragraphs more details about the LIF calibration techniques previously applied at the internal combustion engines group will be presented, a recently published application and the dynamic calibration results achieved with a groove etched on a piston ring specimen built to specifications for the single ring test rig (described in Chapter 3).

### 2.3.1 LIF STATIC CALIBRATION

Duszynski (1999) and Pyke (2000) performed the static calibration with a high resolution micrometer (Figure 2.5). Modifications to the anvil of the micrometer were made to accommodate an optical fibre and to supply a continuous oil flow between the anvil and the spindle. The optical fibre was polished flush with the surface of the micrometer anvil. Thus, the thickness of the oil film flowing through the interrogation zone could be read from the scale of the micrometer. The oil for calibration is pumped from a small reservoir through the micrometer gap, with a pressure of 50 kPa (Figure 2.6 a and b) . This level of pressure was found to be sufficient to maintain oil flow for the whole range of oil film thickness measured. The importance of this condition is particularly pronounced for very thin oil films. In the range of 0-5  $\mu\text{m}$  for which the photobleaching effects might have a more pronounced effect on the fluorescence quantum yield.

A typical response signal of the PMT to a continuously changing oil film thickness in the calibration micrometer according to Duszynski (2000) is shown in Figure 2.4.

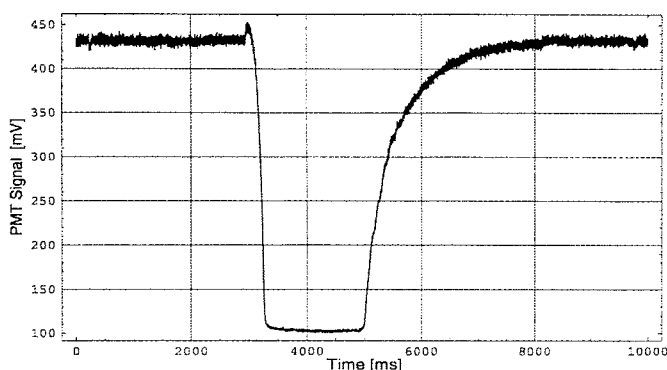


Figure 2.4 Performance of the calibration micrometer-LIF signal (Duszynski, 2000)

Figure 2.4 shows the resulting LIF signal in mVolts first at a micrometer gap of 500  $\mu\text{m}$ , then at its closure and reopening above the initial gap width. For the oil tested, widening the micrometer gap further did not cause any increase in the voltage signal. An obvious conclusion is that this LIF system cannot be used for oil film measurements thicker than 500  $\mu\text{m}$ . The precise upper limit of the measurable range is a function of the wavelength, laser power and physical and chemical properties of the oil tested (viscosity, polymer contents and oil additives). The limited range, however, has no effect because according to Duszynski (1999) engine experiments the lubricant film thickness between the piston rings and the cylinder liner was found to be within the range 0-10  $\mu\text{m}$ . The thickness of the oil films between the piston lands or piston skirt and the cylinder liner were found to be below 150  $\mu\text{m}$ .

The static calibration method showed very low repeatability for the data points corresponding to an oil film thickness of 10  $\mu\text{m}$ . Additionally, for zero film thickness the photomultiplier would detect a signal of very low level. This offset is present in the LIF signal throughout the experimental work. The factors contributing to this inherent photomultiplier behaviour are listed in paragraph 2.3.2.3. Since the lowest repeatability of the calibration procedure was observed for the thinnest oil film thicknesses, obtaining calibration coefficients of high accuracy in the region 0 to 10  $\mu\text{m}$  is vital for improving the accuracy of the LIF system. To overcome this, more experiments with the silica block already mentioned in the literature review were done. According to paragraph 1.2.5 the temperature calibration of the signal could be performed.



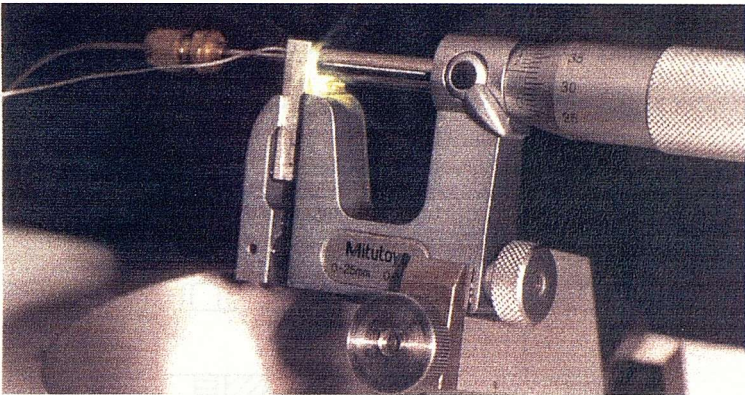
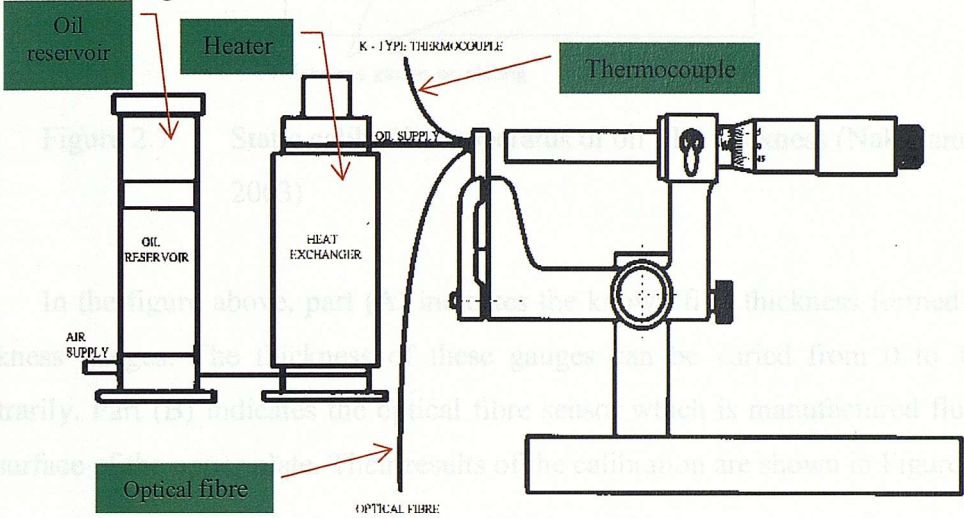
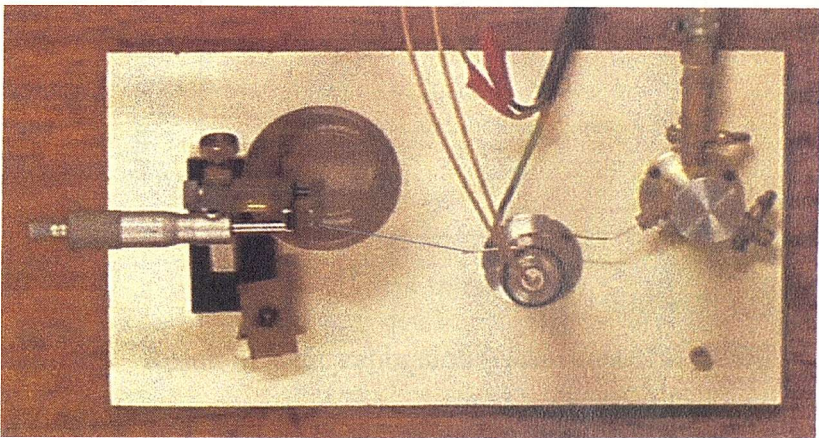


Figure 2.5 The static calibration micrometer



(a) Schematic



(b) photo

Figure 2.6 Layout of the static calibration micrometer based test rig

Nakayama et al (2003) in their work about measuring the oil film thickness on engine bearings using LIF presented a static calibration apparatus (Fig 2.7).

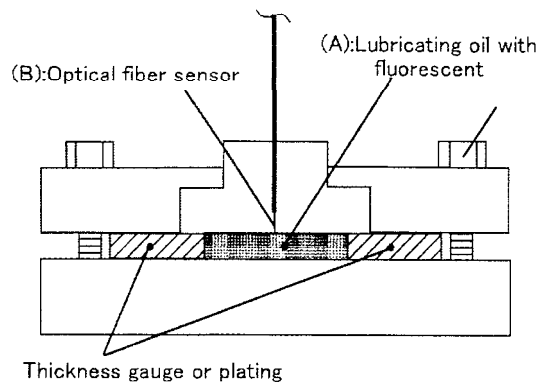


Figure 2.7      Static calibration apparatus of oil film thickness (Nakayama et al, 2003)

In the figure above, part (A) indicates the known film thickness formed by two thickness gauges. The thickness of these gauges can be varied from 0 to 100  $\mu\text{m}$  arbitrarily. Part (B) indicates the optical fibre sensor which is manufactured flush with the surface of the upper plate. Their results of the calibration are shown in Figure 2.8:

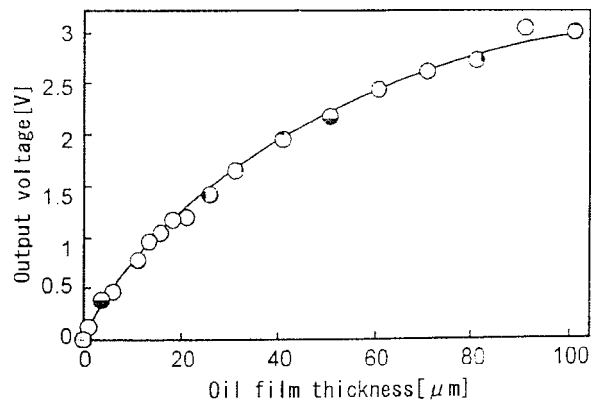


Figure 2.8    Result of static calibration (Nakayama et al, 2003)

It was found that output characteristics depended on the oil film thickness in the range from 0 to 100  $\mu\text{m}$ .

2.3.1.1 RESULTS FROM STATIC CALIBRATION TEST RIG

To further enhance the certainty of static LIF calibration a series of tests were carried out to compare the static calibration coefficient with the dynamic calibration ones that will follow on the next paragraph. Static calibration tests were carried out with the micrometer based test rig following the method described at Duszynski (1999). The following matrix (Table 2.1) shows the results taken at an average oil temperature of 40° C. The test conditions for oil 3B were set at 50 mW laser power, 0.39 kV:

| Gap (μm) | 1 <sup>st</sup> set (mV) | 2 <sup>nd</sup> set (mV) | 3 <sup>rd</sup> set (mV) | 4 <sup>th</sup> set (mV) | Average (mV) |
|----------|--------------------------|--------------------------|--------------------------|--------------------------|--------------|
| 0        | 10.96                    | 9.66                     | 9.45                     | 9.37                     | 9.86         |
| 1        | 10.73                    | 9.62                     | 9.46                     | 9.38                     | 9.80         |
| 2        | 10.56                    | 9.63                     | 9.45                     | 9.40                     | 9.76         |
| 3        | 10.47                    | 9.65                     | 9.48                     | 9.42                     | 9.76         |
| 4        | 10.29                    | 9.70                     | 9.52                     | 9.44                     | 9.74         |
| 5        | 10.40                    | 9.73                     | 9.56                     | 9.53                     | 9.81         |
| 6        | 10.27                    | 10.03                    | 9.65                     | 9.59                     | 9.89         |
| 7        | 10.43                    | 10.04                    | 10.25                    | 9.83                     | 10.14        |
| 8        | 10.46                    | 10.43                    | 10.43                    | 9.75                     | 10.27        |
| 9        | 10.62                    | 14.09                    | 12.62                    | 10.22                    | 11.89        |
| 10       | 10.95                    | 14.50                    | 12.43                    | 13.42                    | 12.83        |
| 20       | 25.65                    | 30.44                    | 27.67                    | 28.76                    | 28.13        |
| 30       | 42.94                    | 51.64                    | 44.64                    | 45.28                    | 46.13        |
| 40       | 60.25                    | 60.61                    | 61.48                    | 61.31                    | 60.91        |
| 50       | 74.60                    | 72.32                    | 76.36                    | 75.41                    | 74.67        |
| 60       | 82.11                    | 85.49                    | 89.04                    | 87.41                    | 86.01        |
| 70       | 91.74                    | 98.73                    | 100.66                   | 99.79                    | 97.73        |
| 80       | 104.54                   | 107.66                   | 110.89                   | 111.15                   | 108.56       |
| 90       | 112.20                   | 119.96                   | 117.36                   | 120.20                   | 117.43       |
| 100      | 121.40                   | 127.43                   | 127.50                   | 129.49                   | 126.46       |

Table 2.1 Static LIF data matrix at 40°C, 50 mW laser power, 0.39 kVolts.

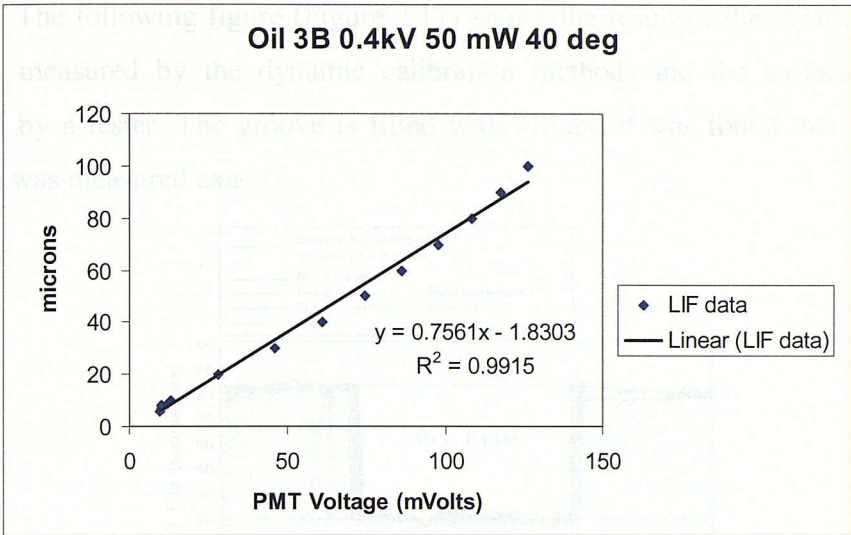


Figure 2.9 Static LIF calibration for Oil 3B

The calibration coefficient derived is 756  $\mu\text{m}/\text{Volts}$ .

2.3.2 LIF DYNAMIC CALIBRATION

Nakayama et al (2003) have also presented a dynamic calibration method for their LIF experimental results. A shaft is used for the dynamic calibration. Two sheets are pasted on the surface of the shaft which have a known thickness of 48  $\mu\text{m}$ . Thus a step of the oil film thickness is devised on the shaft surface for the calibration (Figure 2.10).

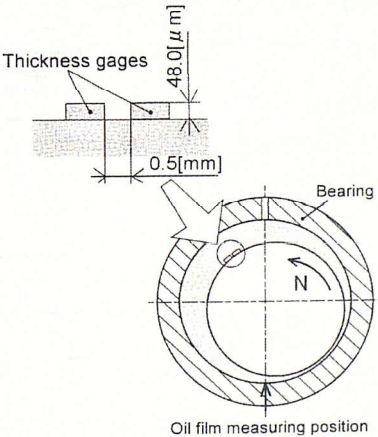


Figure 2.10 Dynamic calibration by using thickness gauges (Nakayama et al, 2003)



The following figure (Figure 2.11) shows the result of the measured oil film thickness measured by the dynamic calibration method, and the surface roughness measured by a tester. The groove is filled with oil and it was found that the oil film thickness was measured exactly.

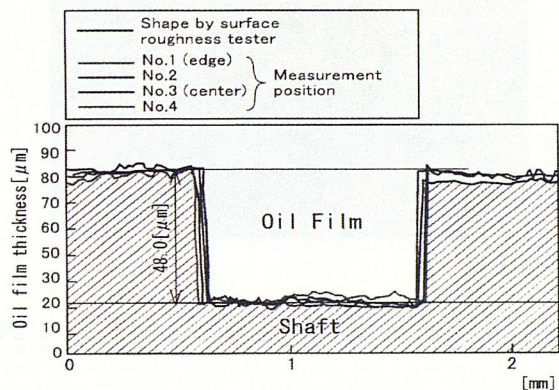


Figure 2.11 Results of measurement of oil film thickness by dynamic calibration method (Nakayama et al, 2003)

For the purpose of LIF dynamic calibration, a new piston-ring specimen was manufactured to be used for the experimental set-up of the single ring test rig. The dynamic calibration is based on etching a groove or grooves of known depth on the surface where the optical fibre travels above. The drawing of the piston ring specimen with the groove can be seen in Figure 2.12.

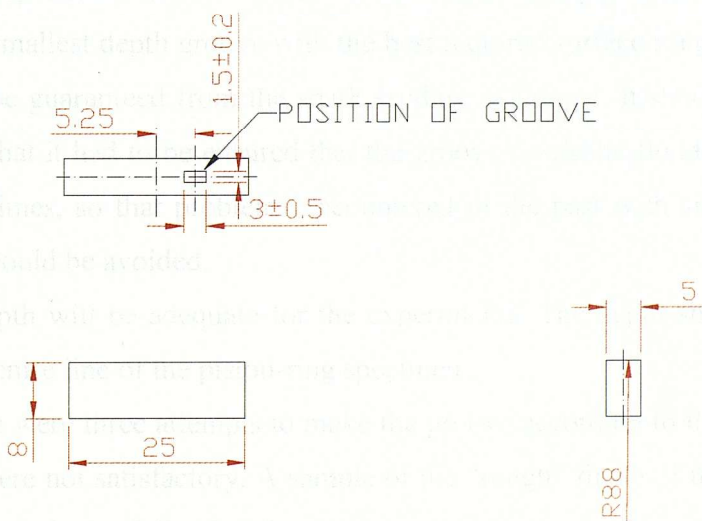


Figure 2.12 Drawing of the piston ring specimen used for dynamic calibration

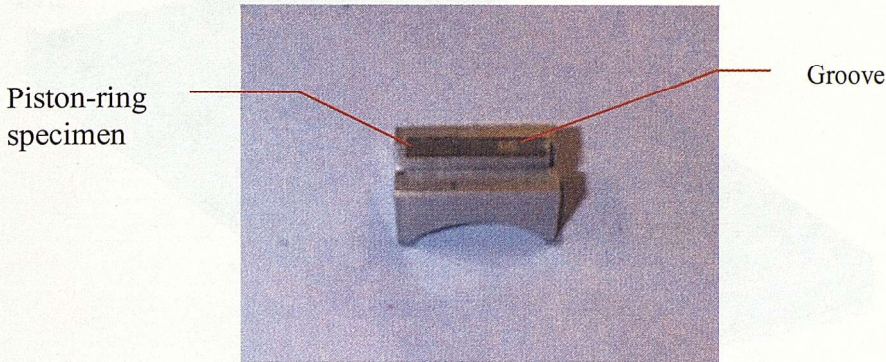


Figure 2.13 The groove on the surface of the piston-ring

This piston-ring is rectangular shaped with a curved upper surface and is used in the idealised test rig for the oil film thickness experiments. The groove was spark eroded on the specimen.

The requirements for the groove are:

- Surface finish of the groove: the best possible the spark eroding machine can provide (ideally  $R_a < 1 \mu\text{m}$ ).
- Depth of the groove: It should be around  $15 \mu\text{m}$ . The groove depth depends on the ability of the manufacturing equipment and  $15 \mu\text{m}$  is the smallest depth groove with the best required surface roughness that could be guaranteed from the spark eroding technique. It should also be noted that it had to be ensured that the groove would be flooded with oil at all times, so that problems encountered in the past with similar techniques could be avoided.

This groove depth will be adequate for the experiments. The depth should be uniform and along the centre line of the piston-ring specimen.

There were three attempts to make the groove according to the specifications. The first two were not satisfactory. A sample of the “rough” finish of the unsatisfactory groove can be seen in the following figure.



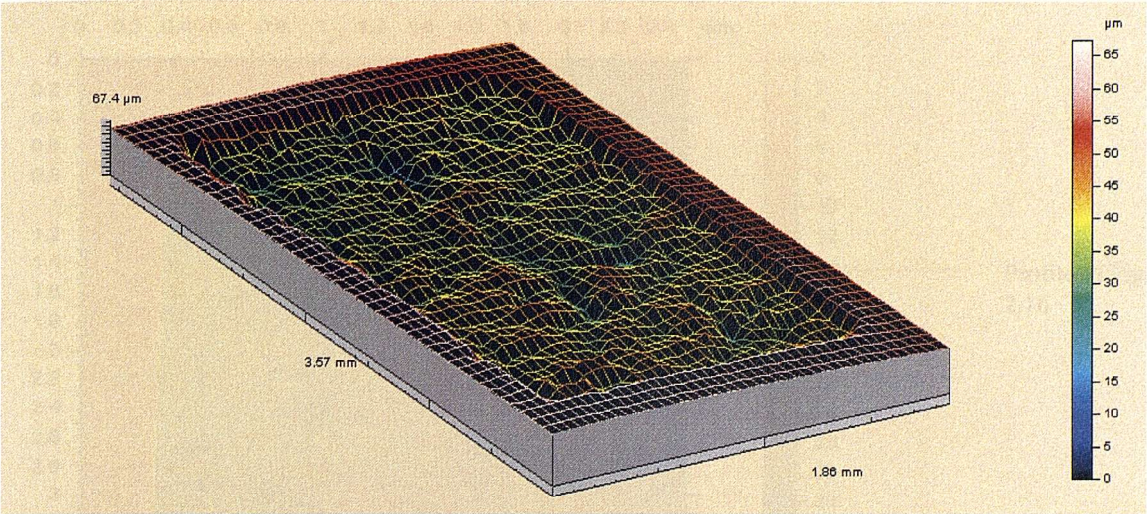


Figure 2.14 First unsuccessful attempt to etch groove on the surface of the piston-ring

The Talysurf surface profilometry equipment showed that the surface of the groove is quite rough. Its depth varies from a minimum of 0  $\mu\text{m}$  to 65  $\mu\text{m}$ . The surface finish was not according to the specification, therefore, a new groove had to be machined, with the best surface finish possible.

The final attempt was made in a specially made piston ring specimen the material of which is most suited for spark erosion machining. The surface profile of the specimen was measured in the Talysurf surface profilometry and a satisfactory surface finish was achieved as the following figures show. A radius of 89.0513 mm for the upper curved surface was measured. The following pictures are showing the quality and the depth of the groove.

The variation of the groove depth is shown in Figures 2.15 and 2.16.



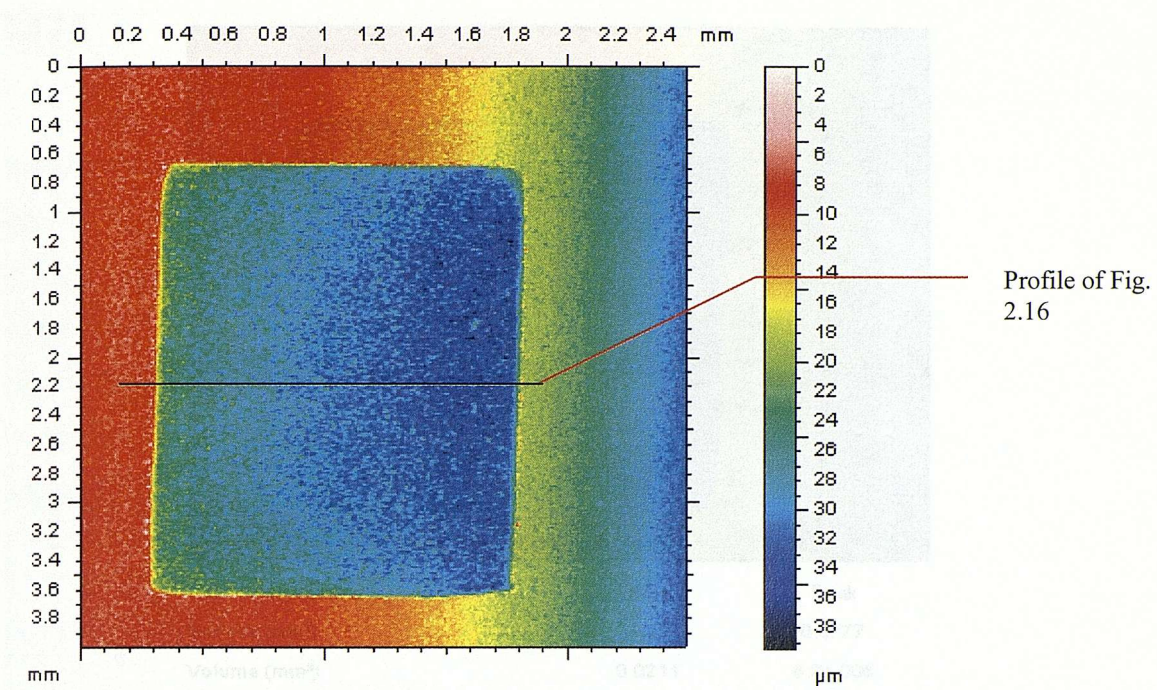


Figure 2.15 2-D coloured surface roughness of the groove.

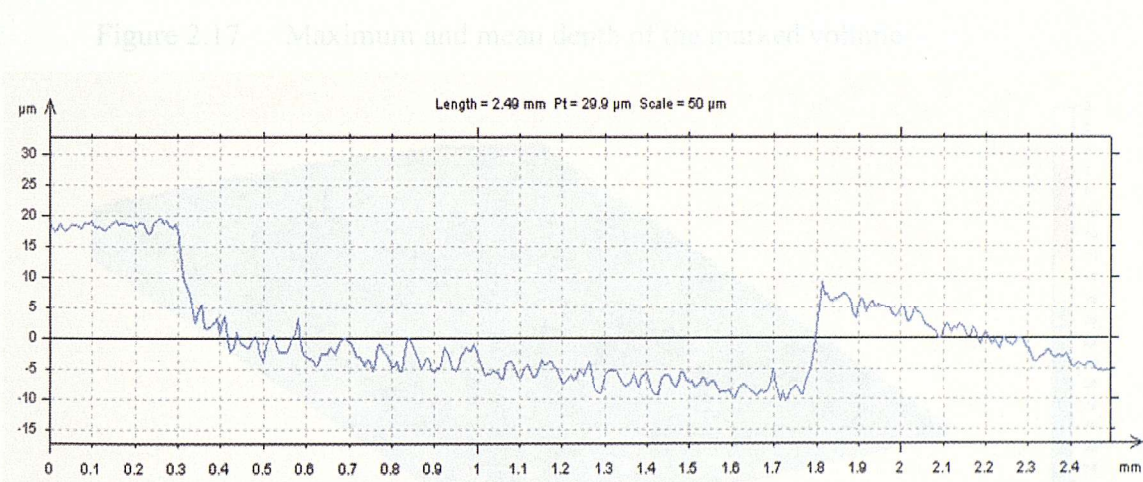
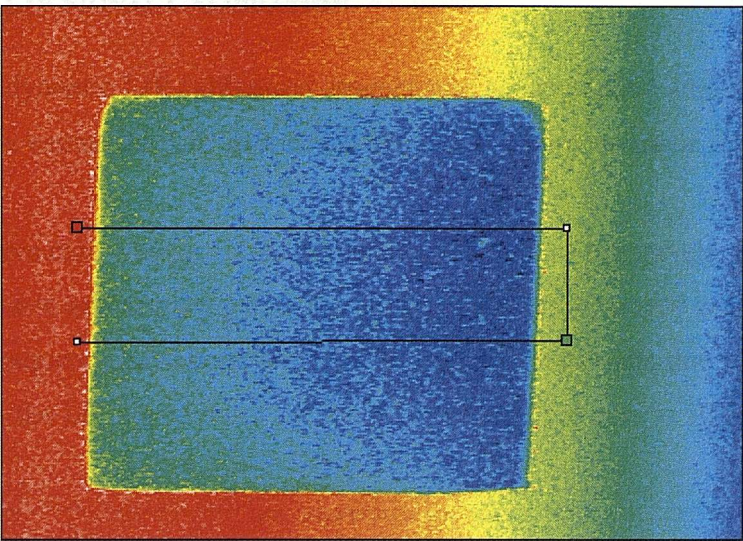


Figure 2.16 Marked groove profile from Figure 2.15

The above graphs show that the requirements for the surface roughness were met with the best possible results.





|                        | Hole   | Peak     |
|------------------------|--------|----------|
| Surface (mm²)          | 1.35   | 0.0777   |
| Volume (mm³)           | 0.0211 | 8.9e-005 |
| Max. depth/height (µm) | 23.3   | 6.66     |
| Mean depth/height (µm) | 15.7   | 1.14     |

Figure 2.17     Maximum and mean depth of the marked volume

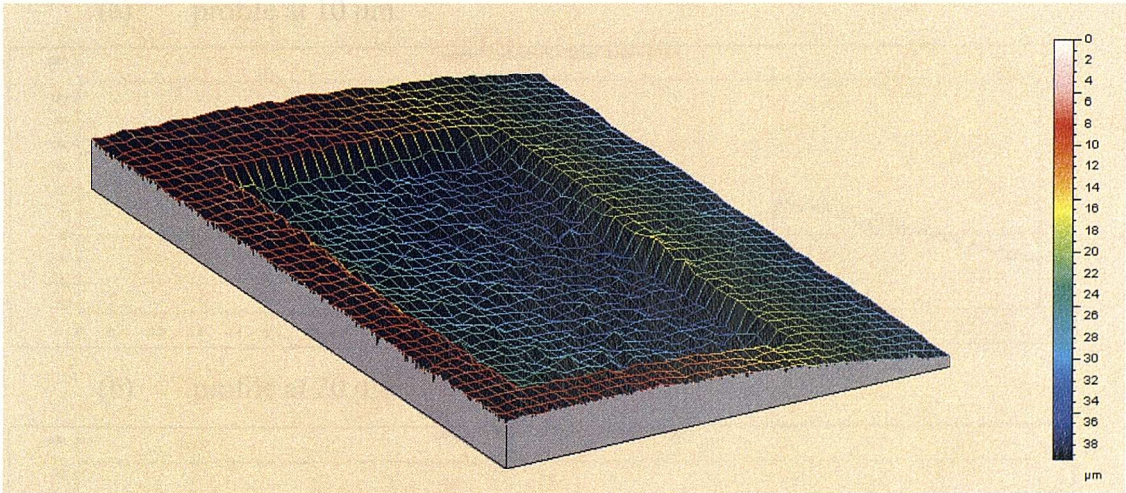


Figure 2.18     3-D meshed axonometric of groove.

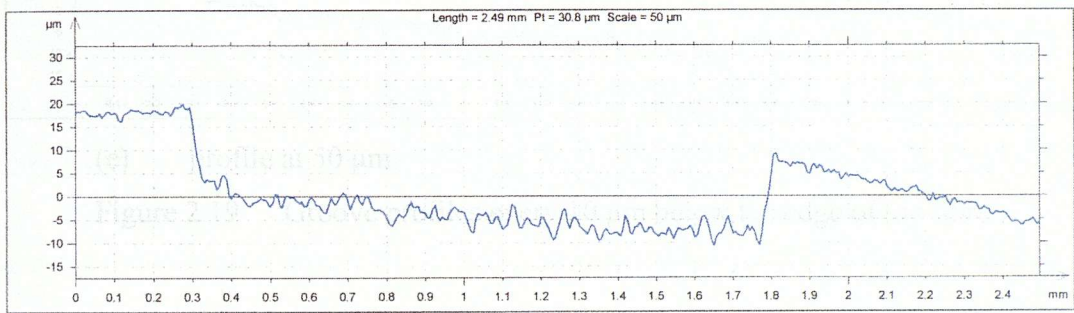
The above graphs show that the requirements set with the original drawing were met with the best possible results.



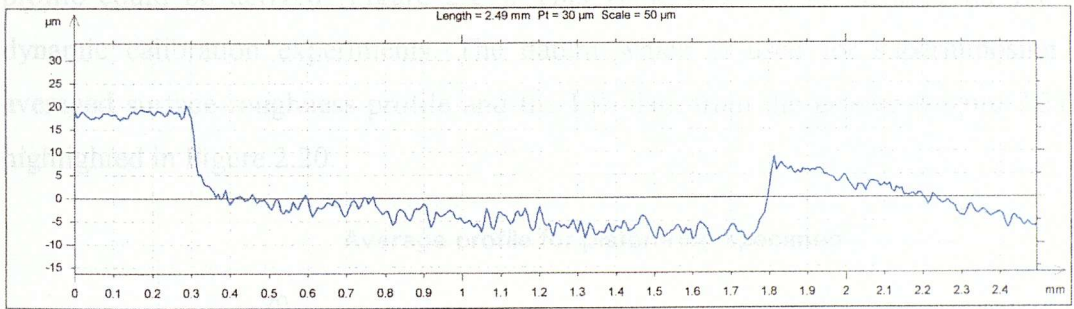
2.3.2.1 LIF SIGNAL CALIBRATION

The dynamic LIF calibration attempt was accomplished with the matching of the groove data versus the LIF data acquired during a purpose made experimental set-up.

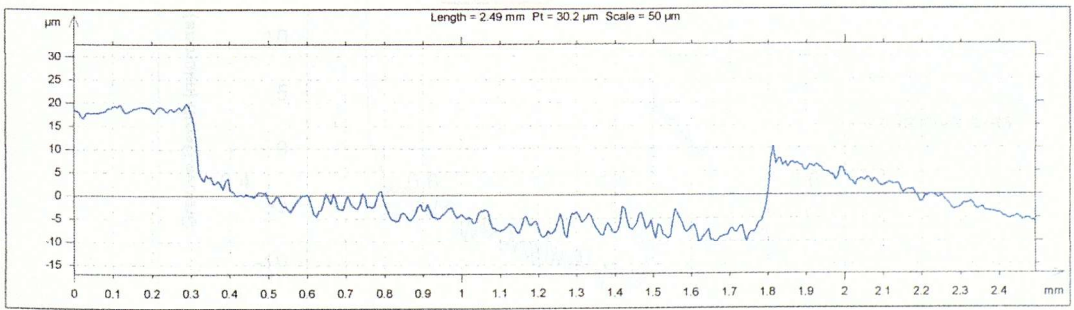
Groove data were acquired at the area where the fibre travels. Since the stylus of the profilometer was acquiring data for the groove every 10  $\mu\text{m}$  of groove length, five (5) profiles in sequence have been averaged in order to get the data that were going to be compared with the LIF signal since the diameter of the fibre core is 50  $\mu\text{m}$  (Figure 2.19 a to e).



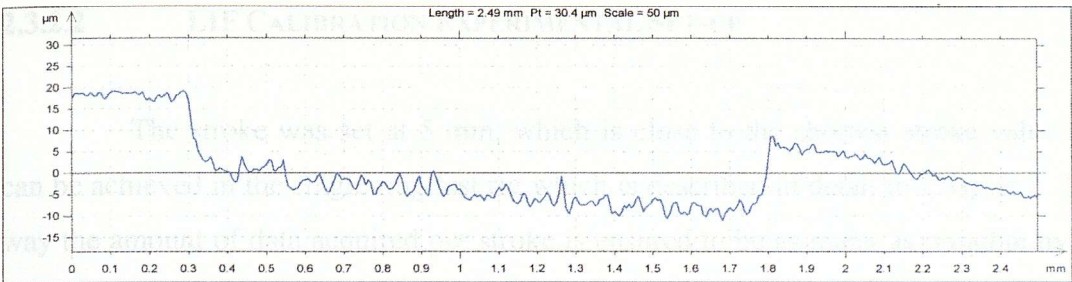
(a) profile at 10  $\mu\text{m}$



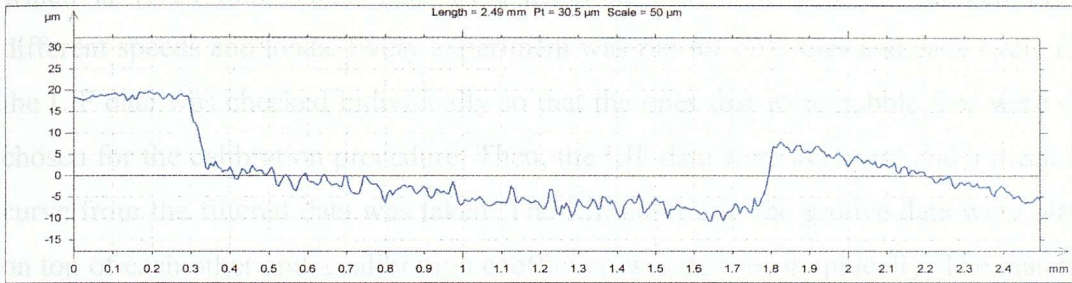
(b) profile at 20  $\mu\text{m}$



(c) profile at 30  $\mu\text{m}$



(d) profile at 40  $\mu\text{m}$



(e) profile at 50  $\mu\text{m}$

Figure 2.19 Groove profiles every 10  $\mu\text{m}$  below the edge of the fibre

The five (5) profiles above were averaged so that a mean value of the groove profile could be derived (Figure 2.20). This value is going to be used for the LIF dynamic calibration experiments. The datum which is used for superimposing the averaged surface roughness profile and the LIF data from the groove (Figure 2.21) is highlighted in Figure 2.20.

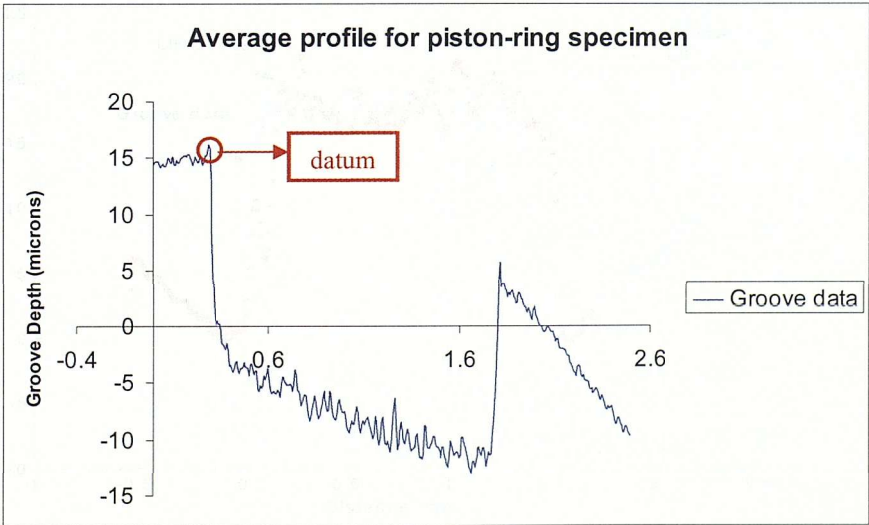


Figure 2.20 The averaged profile over which the LIF signal is calibrated



2.3.2.2 LIF CALIBRATION EXPERIMENTAL SET-UP

The stroke was set at 5 mm, which is close to the shortest stroke value that can be achieved in the single ring test rig which is described in detail in Chapter 3. This way the amount of data acquired per stroke is ensured to be as many as possible by the data acquisition system when the fibre travels on top of the piston groove, the width of which is 1.525 mm. Then, data were acquired with a 37-40°C oil temperature, at different speeds and loads. Every experiment was run for 20 cycles and each cycle from the LIF data was checked individually so that the ones that were bubble free were only chosen for the calibration procedure. Then, the LIF data were averaged and a mean LIF curve from the filtered data was taken. The LIF curve and the groove data were placed on top of each other and a calibration coefficient was derived graphically. The matching was repeated for every experimental condition that was chosen from the beginning of the tests and in the end the calibration coefficient was taken ( $K_{mean}$ ) as the statistical average of each individual matching measurement ( $K_{i=1,...,24}$  from Table 2.2). Experiments were accomplished at different speeds and loads and statistical averages were taken as the mean calibration coefficient values.

The appearance of bubbles in the groove is a major source for uncertainties, because it interferes in a negative manner to the LIF signal as the groove is not entirely filled with lubricant.

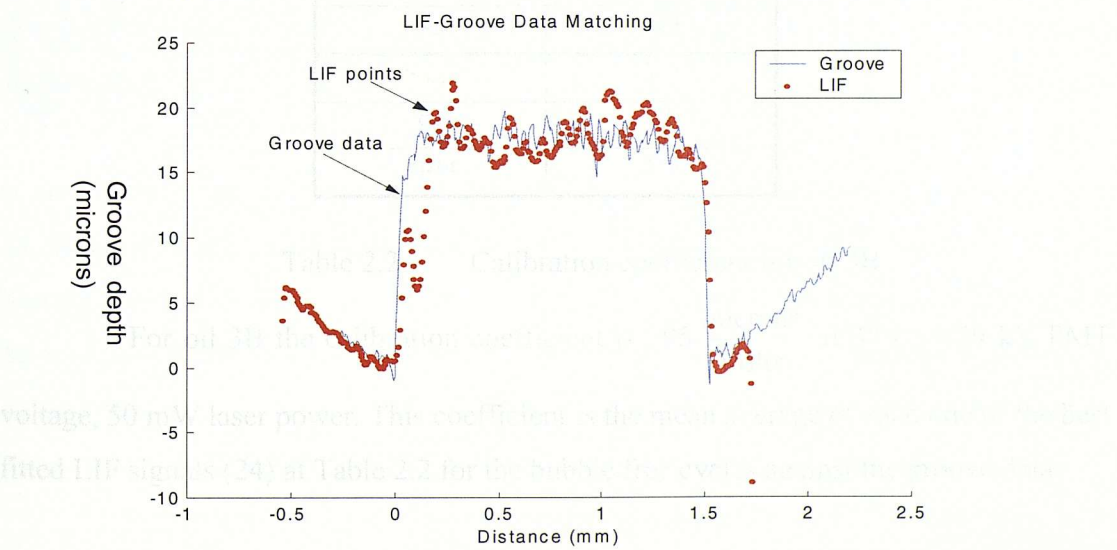


Figure 2.21 Matching of groove data and averaged LIF data



|                   |     |
|-------------------|-----|
| K <sub>1</sub>    | 367 |
| K <sub>2</sub>    | 366 |
| K <sub>3</sub>    | 372 |
| K <sub>4</sub>    | 308 |
| K <sub>5</sub>    | 285 |
| K <sub>6</sub>    | 302 |
| K <sub>7</sub>    | 277 |
| K <sub>8</sub>    | 282 |
| K <sub>9</sub>    | 279 |
| K <sub>10</sub>   | 280 |
| K <sub>11</sub>   | 286 |
| K <sub>12</sub>   | 281 |
| K <sub>13</sub>   | 287 |
| K <sub>14</sub>   | 285 |
| K <sub>15</sub>   | 291 |
| K <sub>16</sub>   | 290 |
| K <sub>17</sub>   | 295 |
| K <sub>18</sub>   | 311 |
| K <sub>19</sub>   | 300 |
| K <sub>20</sub>   | 302 |
| K <sub>21</sub>   | 300 |
| K <sub>22</sub>   | 302 |
| K <sub>23</sub>   | 301 |
| K <sub>24</sub>   | 354 |
| K <sub>mean</sub> | 295 |
| σ <sub>K</sub>    | 14  |
| Error             | 5%  |

Table 2.2 Calibration coefficient for oil 3B

For oil 3B the calibration coefficient is  $295 \frac{\text{microns}}{\text{Volts}}$  at 38°C, 0.39 kV PMT voltage, 50 mW laser power. This coefficient is the mean average of each one of the best fitted LIF signals (24) at Table 2.2 for the bubble free cycles against the groove data.

### 2.3.2.3 BACKGROUND NOISE CONSIDERATIONS

When the PMT is switched on, a shift on the PMT voltage is noticed. This shift should be taken into account when calibrating the LIF signal in order to get the correct result. The PMT tube detects radiation which is derived from background ambient radioactivity. Such sources can be traces of long lived gamma-emitters in the bricks of the building or the concrete floor,  $^{40}\text{K}$  in the glass of the photomultiplier and the sample vials, cosmic rays, traces of uranium in lead shielding, etc. Much of such background sources, if the source is external to the detector, can be absorbed and prevented from interfering with the detection process by the use of dense shielding material. There are also spurious counts which are not radioactive background and for which shielding does no good. They arise from PMT noise. Typical noise rates can be between 200 and 1000 pulses per second.

Light production is more prevalent in large PMT's rather than the small ones. The greatest problem is the  $^{40}\text{K}$  content of the glass envelope. When a beta-particle, such as that from a  $^{40}\text{K}$ -decay event ( $E_{\text{max}}=1.34 \text{ meV}$ ), passes through a medium of high refractive index faster than the speed of light in that medium, it gives rise to Cerenkov radiation, light photons propagated perpendicular to the direction of the particle motion. This light falls within the wavelength region where the PMTs are most sensitive. In addition to this beta,  $^{40}\text{K}$  is a hard gamma emitter (1.46 meV). These gammas activate the photocathodes by counting coincidences. The larger the tube, the more  $^{40}\text{K}$  and more such counts.

There are still other volume related sources of spurious counts. Though there is an extremely high vacuum within the PMT, there still remain many gas molecules. These occasionally interact with the electron avalanche in the region of the anode and become ionized. The ions migrate to the cathode where they are discharged, sometimes producing light. In general, the smaller tubes are to be favoured over the larger ones. The advantage is not only in the lower mass, but also in their lower operating voltage which makes for less noise and less residual gas ionisation.

Repeated tests using the reciprocating test rig at slow speed without any lubricant present between the two surfaces, showed that the noise level that the PMT

detects has a mean value of 0.0275 Volts. The following graph (Figure 2.22) shows this trend.

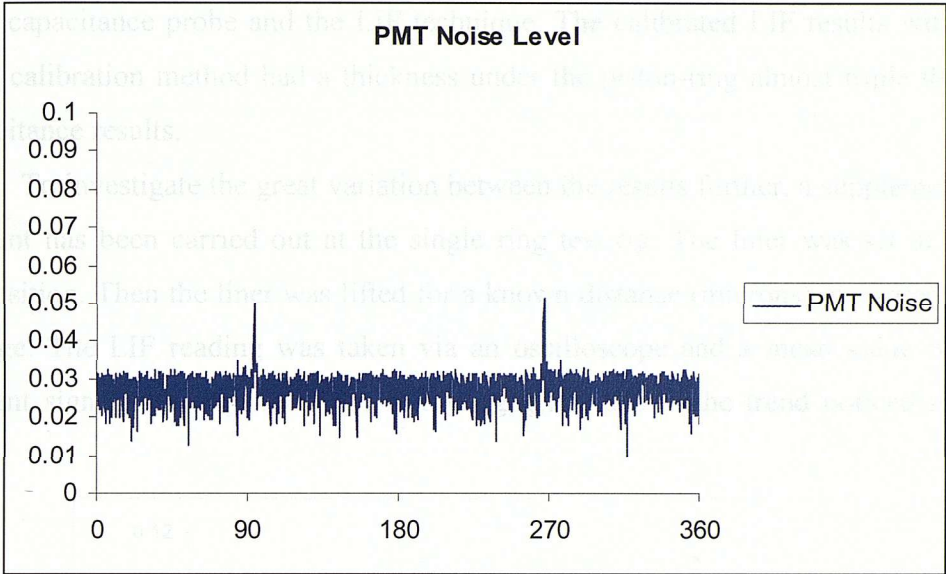


Figure 2.22 PMT noise-background radiation level

The peaks recorded at 90° and 270° are reflections of the piston ring specimen. This mean value has to be subtracted from the final LIF signal that will be calibrated. According to the dynamic calibration coefficient found for oil 3B at ambient conditions, this noise level that needs to be subtracted represents a total thickness value added on the film of 8 microns.

2.3.2.4 OIL FILM COMPARISON (CAPACITANCE-LIF MEASUREMENTS)

A comparison has been made between the capacitance and the LIF results for the same oil so that the two methods can be directly compared. The oil chosen is the 3B for 973 N/m load at 300 rpm. The OFT measurements were taken with the capacitance technique using the probe described in paragraph 2.4. LIF and capacitance data were acquired simultaneously at the single-ring test rig described in Chapter 3. Then the dynamic calibration coefficient derived from Table 2.2 ( $K_{mean}$ ) was used to calibrate the



LIF intensity data and the capacitance OFT data were calibrated according to the capacitance probe calibration method described in paragraph 3.4.3.1. The initial comparison showed that there is a big difference between the measurements acquired with the capacitance probe and the LIF technique. The calibrated LIF results with the dynamic calibration method had a thickness under the piston-ring almost triple that of the capacitance results.

To investigate the great variation between the results further, a supplementary experiment has been carried out at the single ring test rig. The liner was set at mid-stroke position. Then the liner was lifted for a known distance (microns) measured by a dial gauge. The LIF reading was taken via an oscilloscope and a mean value of the fluorescent signal was recorded. The following graph shows the trend noticed in the results:

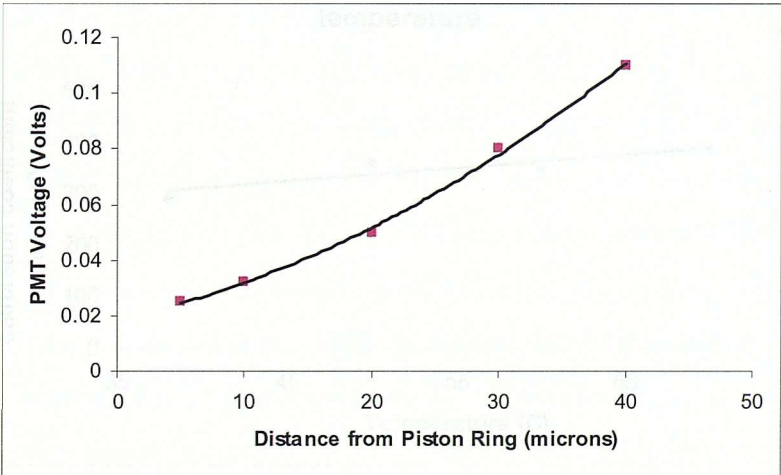


Figure 2.23 LIF signal linearity over the measured distance

There is no linear correlation for the data points. If the measurement is less than 5  $\mu\text{m}$ , which is the case for the minimum oil film thickness according to the capacitance results, it is not accurate because the ratio between the voltage and the thickness is too small. For more than 10  $\mu\text{m}$ , the calibration curve can be used for the results.

2.3.2.5LIF DYNAMIC CALIBRATION VARIATION WITH TEMPERATURE

In this paragraph the effect of temperature on the dynamic calibration coefficient is studied. For this experiment, similar set-up to paragraph 2.3.2.2 was used and 4 different temperatures were chosen, 38, 50, 60 and 70 °C. For every experimental set, 24 different experimental conditions were chosen, varying the speed and load each time so that data from different conditions can be acquired and compared before a mean value is derived.

The following graph (Figure 2.24) represents the observed trend; the coefficient increases respectively with temperature. The temperature results are viewed in Table 2.3.

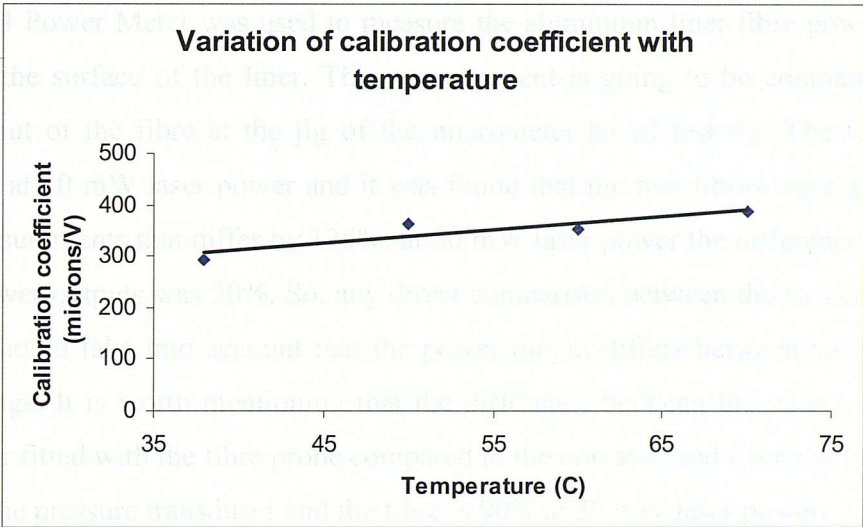


Figure 2.24 The effect of temperature on fluorescence emission

| Temperature<br>(°C) | Calibration Coefficient<br>(µm/Volt) | Standard Deviation<br>(µm/Volts) | Error (%) |
|---------------------|--------------------------------------|----------------------------------|-----------|
| 38                  | 295                                  | 14                               | 5         |
| 50                  | 364                                  | 26.9                             | 7         |
| 60                  | 353                                  | 21                               | 6         |
| 70                  | 389                                  | 55.37                            | 14        |

Table 2.3 Temperature results for the calibration coefficients

### 2.3.3 COMPARISON BETWEEN LIF STATIC AND DYNAMIC CALIBRATION

The calibration coefficient derived from the static calibration tests (756  $\mu\text{m/Volts}$ ) for the same oil at the same testing conditions as the dynamic method (295  $\mu\text{m/Volts}$ ) provides less than encouraging results. The above comparison enhances the uncertainty for the calibration methods used and gives ground to multiply the reasons behind the big difference between the calibration factors. Possible reasons are the different laser power output from the optical fibres (the one on the liner surface and the one on the micrometer based test rig) and the temperature variation. Since different fibre is used for the experiments at the static calibration test rig, the use of this technique should be accompanied by the fibres' laser power output measurements. A Spectra Physics 404 Power Meter was used to measure the aluminium liner fibre power output as it exits the surface of the liner. This measurement is going to be compared to the power output of the fibre at the jig of the micrometer based test rig. The tests were carried out at 50 mW laser power and it was found that the two fibres were giving out power measurements that differ by 125%; at 30 mW laser power the difference between the two power outputs was 20%. So, any direct comparison between the two calibration methods, should take into account that the power output differs between various fibre optics fittings. It is worth mentioning that the difference between the power output of the old liner fitted with the fibre probe compared to the one acquired from the liner fitted with both the pressure transducer and the fibre is 90% at 30 mW laser power.

### 2.4 CAPACITANCE TECHNIQUE

A parallel plate capacitance sensor was used to measure the distance between the arbitrary fixed point in space and the liner from where the minimum oil film thickness could be inferred. The probe contained a central metal core surrounded by a plastic insulation cylinder. This in turn was surrounded by a metal cylinder acting as the other electrode. A further layer of insulation surrounded the outer electrode and for protection the whole probe is wrapped in a steel cylinder. The output from the capacitor was directed to a Capacitec unit which converted the signal and output to a voltage

recorded by the data acquisition system. The electrical circuit associated with the purpose built capacitance probe generated a signal which is proportional to its capacitance. The sensor is fixed at a distance around 30  $\mu\text{m}$  below the flat surface which is specially manufactured for the capacitance transducer. The calibration graphs of the capacitance sensor presented in Chapter 3 show that the sensor response is linear, at least for the region of thicknesses that are going to be measured. The top surface of the custom made sensor is completely flat as opposed to the one used by Pyke (2000) who used a curved top surface sensor to obtain the results.

## 2.5 CONCLUSIONS

A novel dynamic LIF calibration technique is proposed and a direct comparison to the static calibration method used by Duszynski (1999) and Pyke (2000) is attempted. The static technique is based on data derived by a micrometer based test rig whereas the dynamic calibration data are derived from a groove of known depth machined on the surface of a custom made piston-ring specimen. The ring in turn is set at the simulation test rig described in Chapter 3. Direct comparison of the techniques showed great differences in the calibration factors derived and the issue of different Laser power output emanating from different fibre fittings is introduced and needs to be taken into account when LIF point measurements are involved. Another static LIF calibration experiment using the fibre optic setup at the test rig showed that for oil films less than 5  $\mu\text{m}$  thick, which is the case for MOFT according to the capacitance results, the technique is not accurate as the ratio between the voltage and the thickness is too small. For OFT measurements more than 10  $\mu\text{m}$  the static calibration can be used to obtain results.

## CHAPTER 3

### TEST RIG

#### INTRODUCTION

A single-ring test rig which simulates under idealised conditions the lubrication process between the piston ring and the cylinder liner in reciprocating engines (Arcoumanis et al, 1995), has been used for parametric lubricant studies. It uses a fixed single-barrel-faced piston ring and a reciprocating flat liner, reversing the respective roles in engines. Six different oils were tested and capacitance, LIF and friction results are being presented. Oil film thickness behaviour is being examined under high temperature testing for oils with viscosity extremes. Friction results are discussed with a focus on their peaks shift during oil flow reversal and HTHS effect on their magnitude. Effect of high temperature on friction is also discussed.

#### 3.1 DESCRIPTION OF TEST RIG

The aim behind this design was the simulation of realistic stroke lengths under speeds typical of engine idling. In Figure 3.1 a and b the photo and schematic of the simulation test rig can be seen. The liner, a flat rectangle surface, is made from polished steel gauge plate which reciprocates over a section of piston ring which is fixed in a ring holder. The single ring test rig itself is capable of providing valuable information about piston ring lubrication under simplified conditions, by eliminating the in-cylinder complications resulting from ring dynamics, starved lubrication, circumferential variations and lubricant degradation. Measurements of oil film thickness in the test rig can be accomplished with both capacitance and LIF.

A rectangle of polished steel gauge plate reciprocates over a fixed section of the piston ring. The speed, load, temperature and stroke length can all be adjusted. Jets supply lubricant to the ring/liner interface, ensuring fully flooded lubrication at all times. Measurements of minimum oil film thickness, oil film thickness across the ring and total axial friction can be recorded. To drive the liner, an electric motor is coupled to the main



drive train; a rubber coupling was fitted to avoid vibrations disrupting the system. A crank mechanism translates the circular motion of the shaft into the linear reciprocating motion of the liner. This results in a sinusoidal velocity for the liner- not exactly simulating the engine piston motion. The stroke length is adjustable up to 50 mm. The steel liner, fixed in an aluminium holder, is held in the assembly by two high precision linear bearings.

The liner holder is connected to the drive mechanism by a removable connecting joint. One end of the liner assembly is pivoted. The LIF sensor is located in the middle of the liner specimen. In order to allow the mid-stroke position to be varied over the desired range, three links of varying lengths were manufactured so as the whole length of the stroke can be scanned.

The lubricant is supplied by a pressurised heated tank and fed to the ring/liner interface by ten jets, issuing holes drilled in two parallel tubes either side of the ring. An electric motor drives a pump which recirculates the oil through an oil filter fitted in the pressurised oil tank. A K-type thermocouple inserted in one of the tubes is monitoring the oil temperature that is flooding the ring with oil. The oil temperature can be increased from room temperature to 80°C. During operation the oil temperature has a variation of  $\pm 0.5^\circ\text{C}$ . This variation is low enough not to significantly affect the lubrication but it does produce significant drifts in the output from the capacitance transducer.

This test rig was designed such that minimum lubricant film thickness and friction could be measured continuously throughout the cycle. The drive mechanism for the reciprocating assembly was powered by an electric motor which was mechanically isolated from the rig in order to minimize any vibrations. Isolation at the shaft was achieved by using a flexible neoprene coupling, while the rig was isolated from the lab floor using vibration inhibiting cushions. In order to maximize the speed stability of the system and reduce vibrations, provision was made to increase and adjust the system's inertia using a pair of contra-rotating weights which were driven from the shaft and coupled together by a pair of low noise nylon helical gears. The centre of gravity of the weights could be radially adjusted to optimize the inertia of the reciprocating system.

Removing this joint permitted full access to the ring support and sensor systems by allowing the liner block to be lifted by rotating it about the pivot axis. The

liner block was made out of aluminium except for the surface of simulating the liner which was made out of heat treated gauge plate. In order to minimise the reciprocating mass, most of the material block was removed by drilling deep wide bore holes in it. Calibrated weights could be hung from the end of the assembly in order to increase the cyclic load acting on the ring. Oil is delivered locally in the form of 4 jets on either side of the ring issuing through holes drilled in the circular cross sectioned sealed tubes, thus ensuring fully flooded lubrication at all times.

The piston ring assembly is supported on two legs which were minutely deflected due to friction and connected to the friction transducer by a rigid arm.

### **3.2 PARAMETRIC LUBRICANT TESTING**

The test rig (Figure 3.1) designed by Ostovar (1996) and modified by Pyke (2000) is initially used for parametric lubricant testing. Six different oils are tested. Capacitance, laser induced fluorescence (LIF) and friction results are obtained. The initial position of the capacitance probe is modified. All the experiments are carried out with the probe mounted just beneath a flat surface, which is bolted to the liner specimen and moves with it. Results of the film thickness under the piston ring are also obtained with the LIF method. These results are used for validation purposes of the capacitance measurement method.

### **3.3 DESCRIPTION OF TEST RIG – EXPERIMENTAL SET-UP**

According to the old design the capacitance probe is mounted next to the piston-ring holder, hence it is filled with lubricant from the oil jets used to flood the ring with oil.

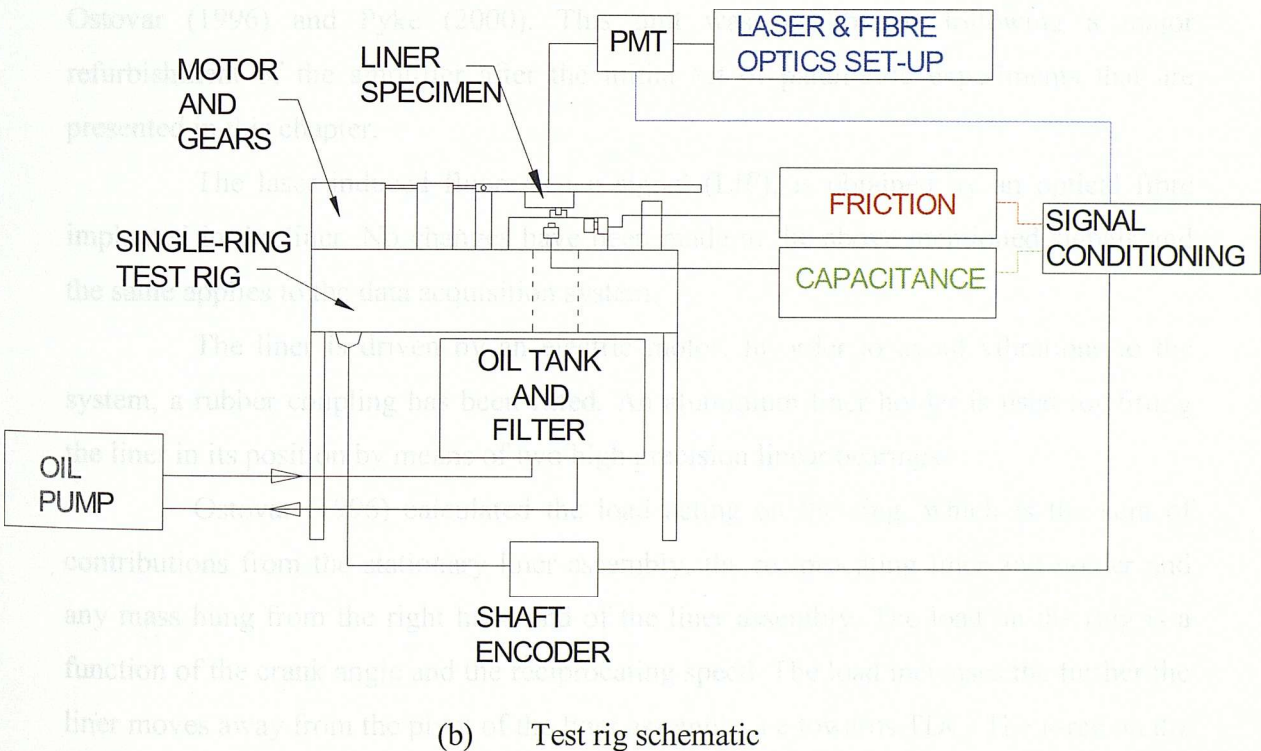
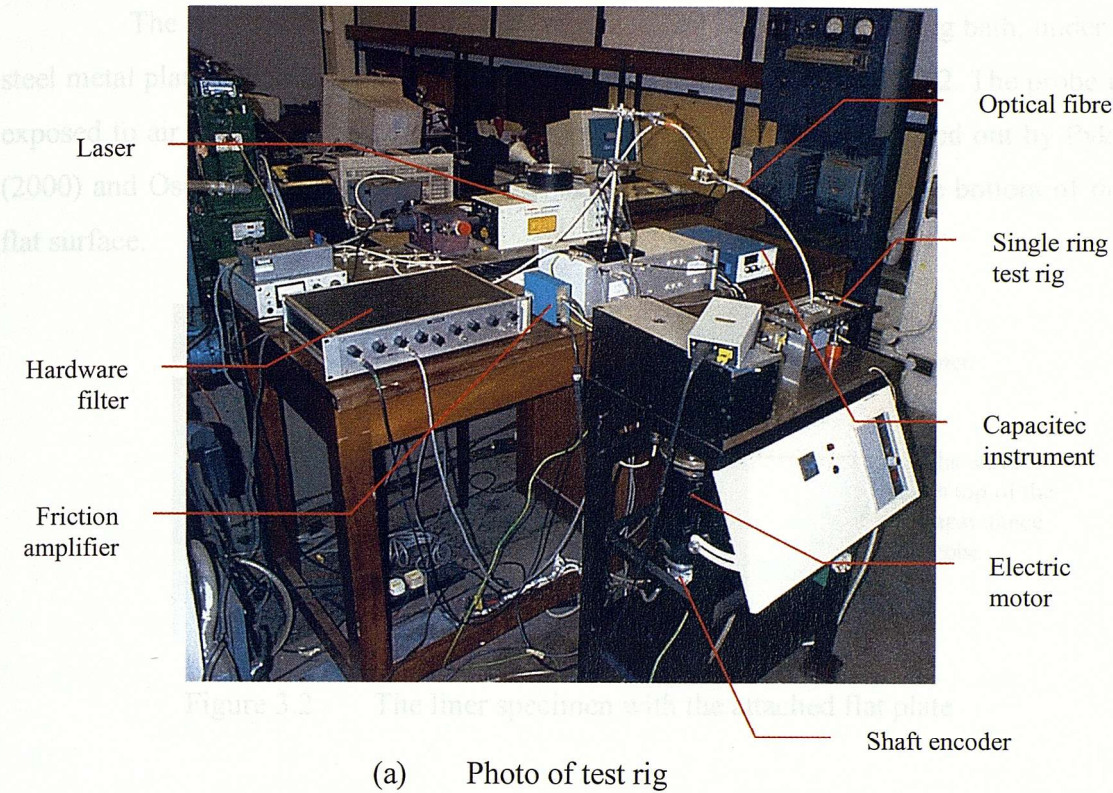


Figure 3.1 Simulation test rig - experimental set-up



The new proposal has the probe mounted just outside the oil ring bath, under a steel metal plate which is bolted to the liner specimen as seen in Figure 3.2. The probe is exposed to air instead of oil which was the case in previous testing carried out by Pyke (2000) and Ostovar (1996) and “reads” the vertical liner movement at the bottom of the flat surface.

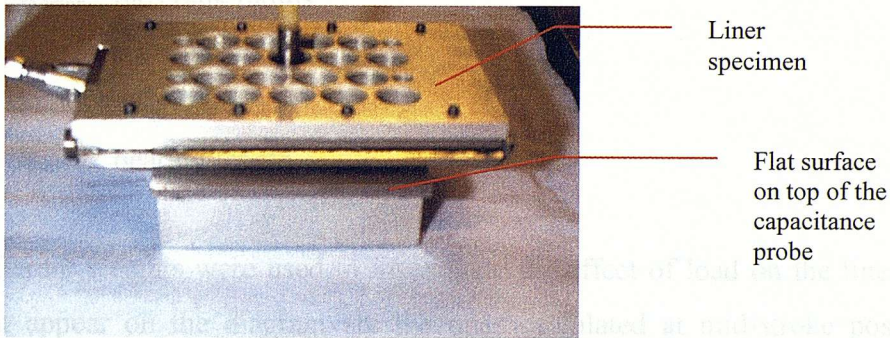


Figure 3.2 The liner specimen with the attached flat plate

Total axial friction is measured with a force transducer same as the one used by Ostovar (1996) and Pyke (2000). This unit was recalibrated following a major refurbishment of the amplifier after the initial set of parametric experiments that are presented in this chapter.

The laser induced fluorescence signal (LIF), is obtained by an optical fibre implanted in the liner. No changes have been made to the above mentioned signals and the same applies to the data acquisition system.

The liner is driven by an electric motor. In order to avoid vibrations to the system, a rubber coupling has been fitted. An aluminium liner holder is used for fitting the liner in its position by means of two high precision linear bearings.

Ostovar (1996) calculated the load acting on the ring, which is the sum of contributions from the stationary liner assembly, the reciprocating liner and holder and any mass hung from the right hand end of the liner assembly. The load on the ring is a function of the crank angle and the reciprocating speed. The load increases the further the liner moves away from the pivot of the liner assembly, i.e towards TDC. The force on the ring can be calculated by the formula:

$$W(\theta) = \frac{0.1830g + 0.25gm_h + 0.5024g(0.089 - R_{crank} \cos\theta) + 5.78 \times 10^{-3} R_{crank} \cos\theta \omega^2(\theta)}{(0.088 + 0.5b)B} \quad (3.1)$$

where:

g: gravity acceleration

$m_h$ : mass applied to the hanging point

$R_{crank}$ : the crank radius

$\omega(\theta)$ : crank angular velocity

b: ring width

B: bearing length

Different weights were used to investigate the effect of load on the liner. All the loads that appear on the diagram are the ones calculated at mid-stroke position. Relatively large masses were used in order to minimise the variation on the signal.

During all the experiments the stroke length was set to 50 mm.

In order to achieve good conformity between the length of the ring and the width of the liner, the ring holder is set free to tilt only in the transverse direction. To ensure the free tilt the ring holder has a groove and sits on a knife edge. Consequently, self-alignment of the ring with the surface of the liner is achieved. In order to minimise vibration problems generated by the increase in speed, the inertia of the system is increased through a pair of contra-rotating weights, which are driven from the shaft by a pair of low noise nylon helical gears.

### 3.4 MEASUREMENT TRANSDUCERS

#### 3.4.1 FRICTION SENSOR

The friction sensor is a PCB model 208B force sensor and is amplified using a PCB F482A unit. This sensor is very sensitive to small displacements or deformations of the ring assembly caused by the axial friction. Ostovar (1996) had calibrated the friction sensor using weight hangers. The calibration curve is shown in Figure 3.3.



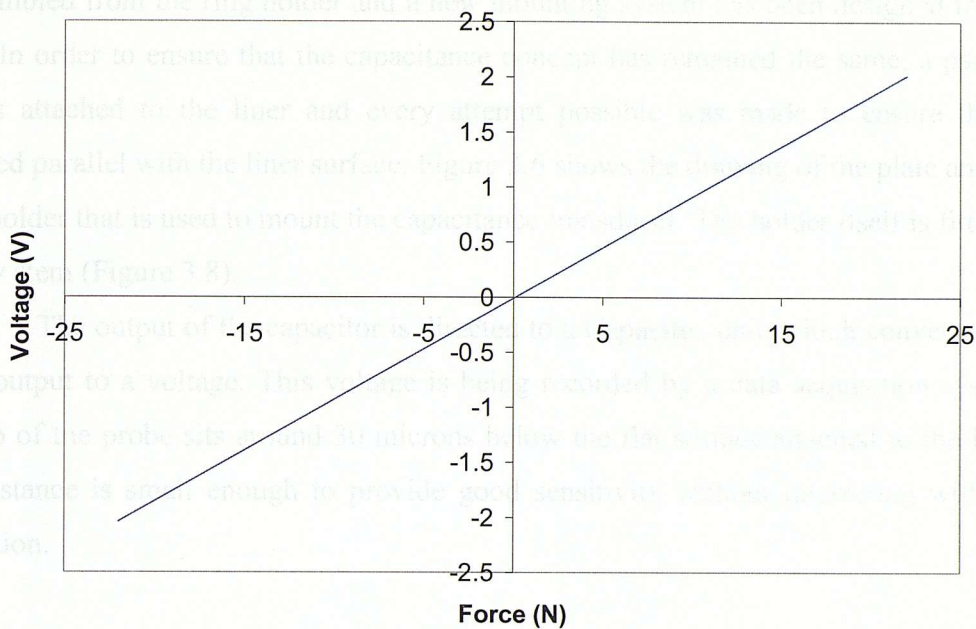


Figure 3.3 Calibration graph of PCB 208B friction sensor

3.4.2 SHAFT ENCODER

The shaft encoder is fitted to the base of the electric motor and triggers the data acquisition system to sample the data two thousand (2000) times a cycle. At each point, capacitance, LIF and friction signals are recorded. Meanwhile, the position of TDC is identified by a second pulse emitted by the encoder. The signals are digitised using a DT 2839 Data Translation acquisition unit.

3.4.3 CAPACITANCE TRANSDUCER

The capacitance transducer, based on the parallel plate theory is used for the minimum oil film thickness measurement. As it has been mentioned before, the major modification is to change its initial position. The main purpose of this idea is to avoid the problems occurring when the probe is flooded with oil as it was fitted next to the piston-ring specimen (Figure 3.4). Oil bubbles appearing at any high speed and inadequate oil supply to the probe cause ambiguity regarding the results. The probe has been



disassembled from the ring holder and a new mounting system has been designed for the probe. In order to ensure that the capacitance concept has remained the same, a parallel plate is attached to the liner and every attempt possible was made to ensure that it remained parallel with the liner surface. Figure 3.6 shows the drawing of the plate and the probe holder that is used to mount the capacitance transducer. The holder itself is fitted to the new stem (Figure 3.8).

The output of the capacitor is directed to a Capacitec unit, which converted the signal output to a voltage. This voltage is being recorded by a data acquisition system. The top of the probe sits around 30 microns below the flat surface attached to the liner. This distance is small enough to provide good sensitivity without interfering with the lubrication.

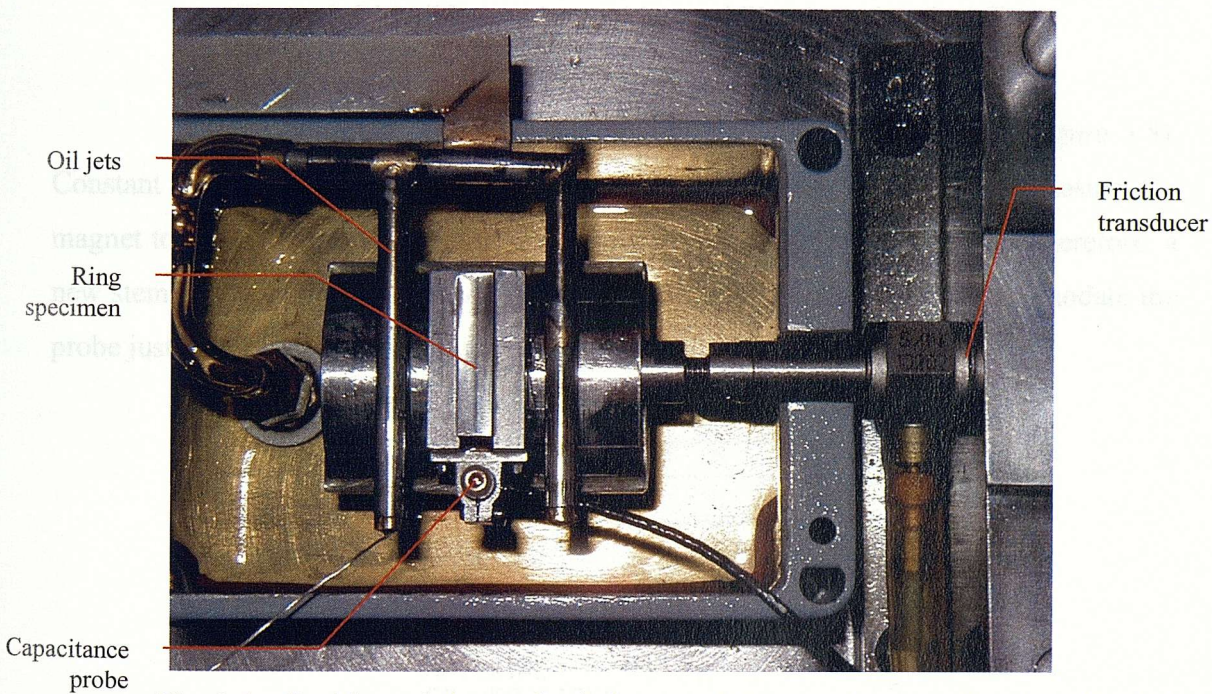


Fig. 3.4 Position of the probe before modification (next to the ring, flooded with oil during testing)



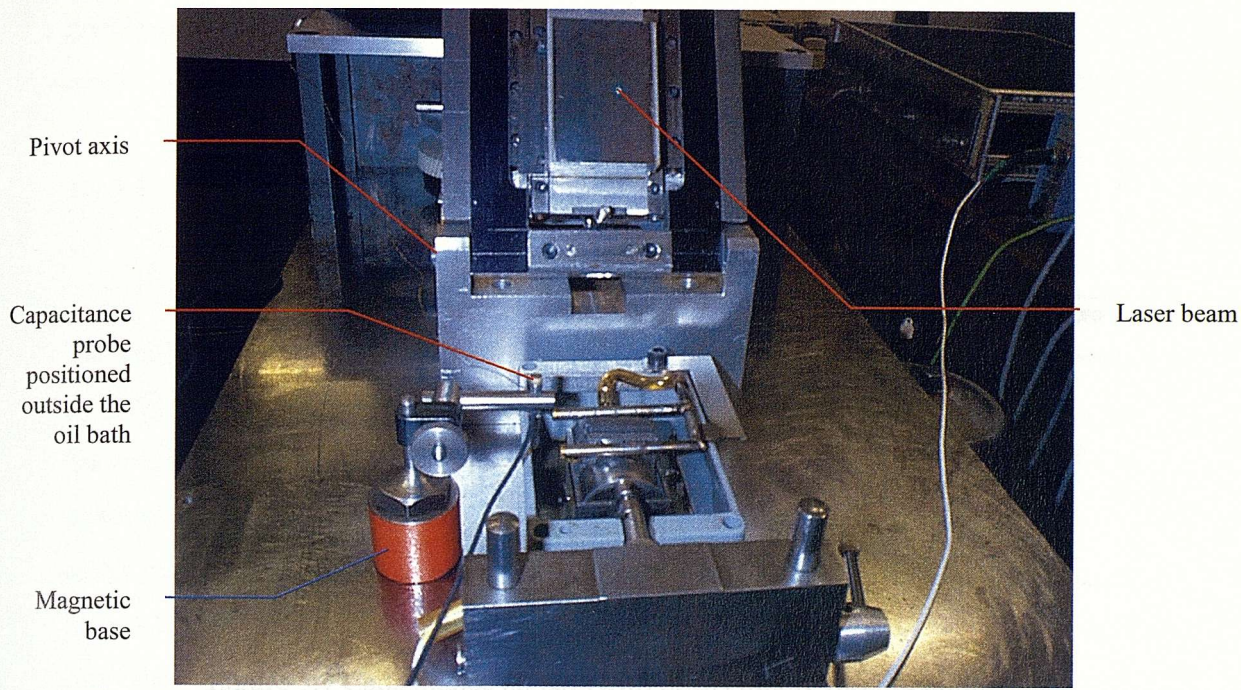


Figure 3.5 Modified probe position and illuminated optical fibre

The probe holder is bolted in a powerful Eclipse E909 magnet (Figure 3.5). Constant and big vibrations, though, coming from the electric motor and causing the magnet to oscillate, introduce a large amount of error in our measurements. Therefore, a new stem has been designed, made of high tensile steel, tall enough to accommodate the probe just under the flat plate (Figure 3.8).

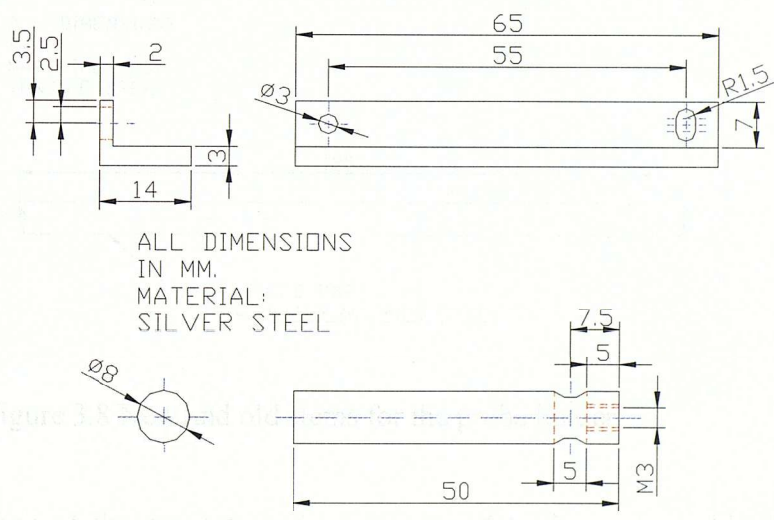


Figure 3.6 Flat plate and probe holder drawings



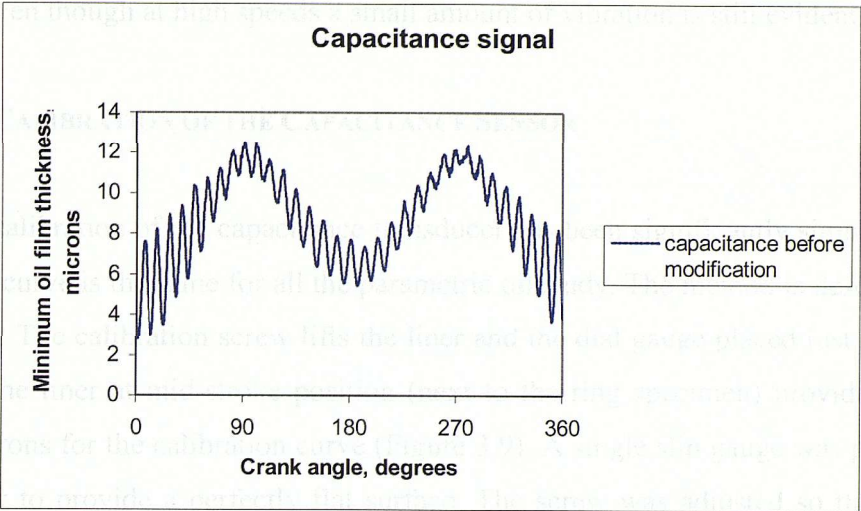


Figure 3.7 Capacitance signal (with error) before modification

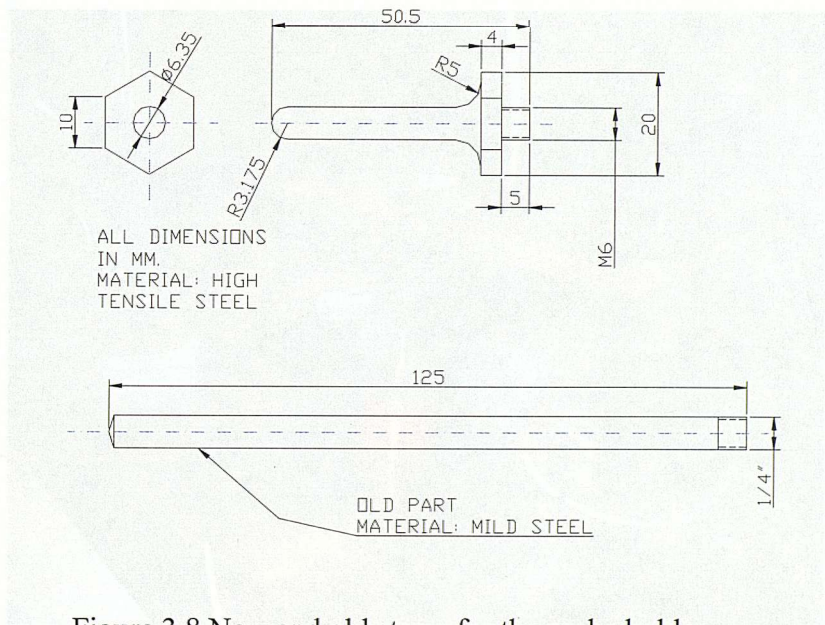


Figure 3.8 New and old stems for the probe holder

The surface of the magnet used to accommodate the probe holder has been made completely flat in a lathe and then the stem has been bolted. Eventually the signal is

greatly improved compared to Figure 3.7 which shows the capacitance output before the modification, even though at high speeds a small amount of vibration is still evident.

3.4.3.1 CALIBRATION OF THE CAPACITANCE SENSOR

The calibration of the capacitance transducer has been significantly simplified. The calibration curve is the same for all the parametric oil study. The method is described by Pyke (2000). The calibration screw lifts the liner and the dial gauge placed just under the middle of the liner at mid-stroke position (next to the ring specimen) provides the readings in microns for the calibration curve (Figure 3.9). A single slip gauge was placed under the screw to provide a perfectly flat surface. The screw was adjusted so that the liner was raised in two (2) microns increments - the resolution of the dial gauge. Since the capacitance probe is fit solid and it does not tilt with the piston ring as it did in the previous setting, there was no need to ensure that its position remained the same during the calibration procedure.

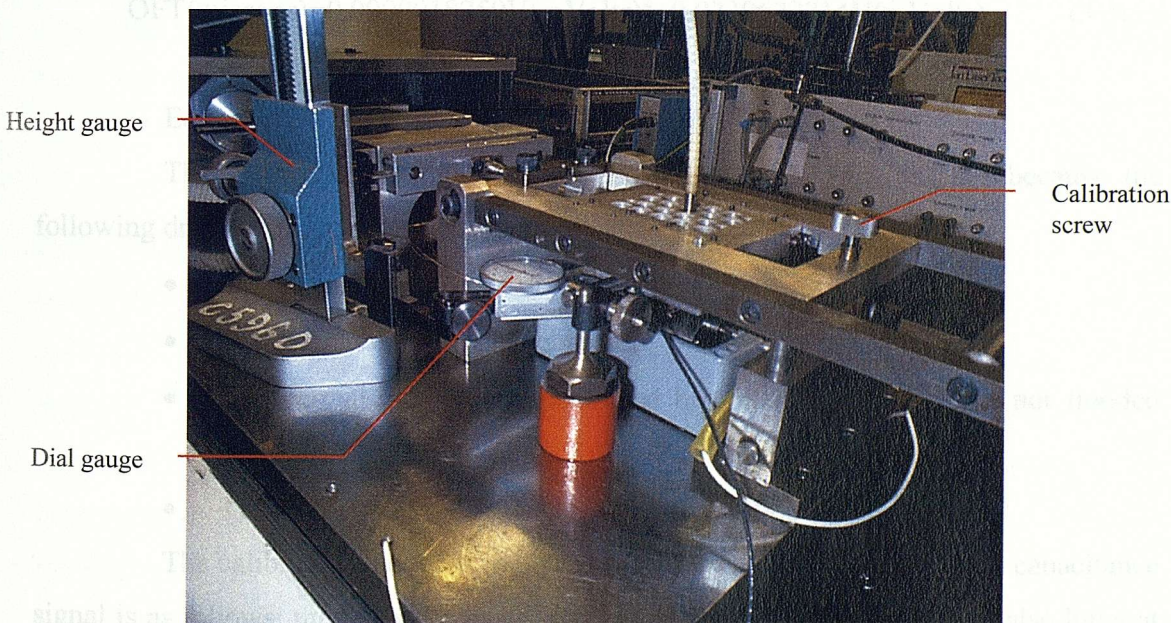


Figure 3.9 Instruments used for the capacitance probe calibration

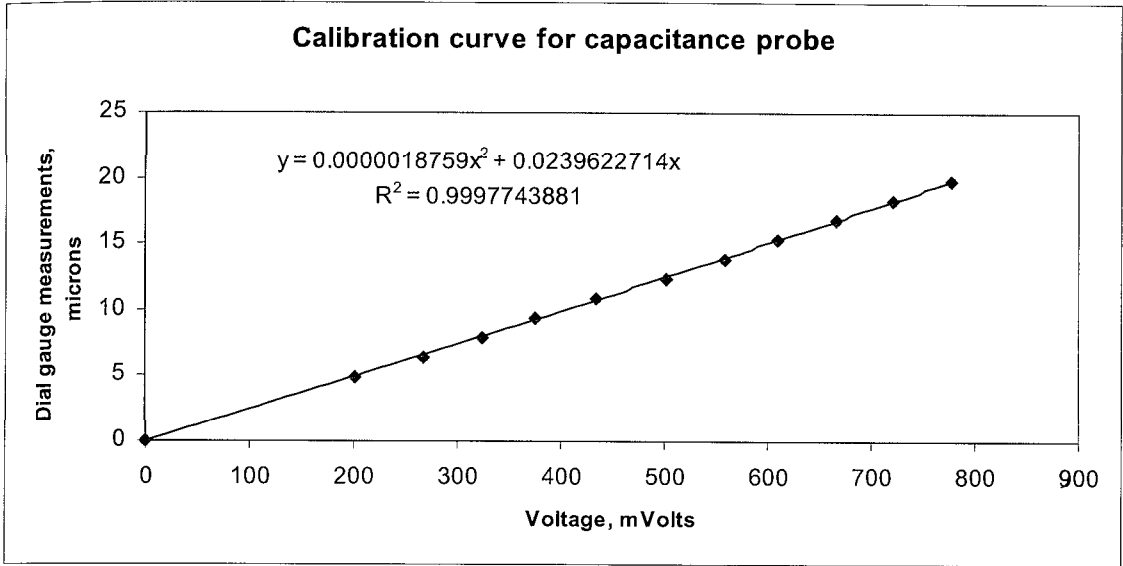


Figure 3.10 Typical calibration curve for the capacitance probe

The calibration curve for the capacitance probe is shown in Figure 3.10. Equation derived for the capacitance sensor calibration:

$$\text{OFT}(\text{microns}) = 0.0000018759V(\text{mVolts})^2 + 0.0239622714V(\text{mVolts}) \quad (3.2)$$

Equation (3.2) is used for all the oils tested.

The calibration method has proved to be quick and reliable because the following drawbacks have been avoided:

- No oil feeding is necessary
- No thermal expansion is observed
- No thermal drifts are experienced because the transducer is not flooded with oil.
- The piston-ring specimen need not be positioned solid (glued)

The calibrated signal has to be error free. The error analysis for the capacitance signal is as follows: the test rig runs at very low speed (30 rpm) and load is also hung at the edge of the liner holder (2823 N/m). The signal from this low speed experiment represents the error, i.e. what the probe measures when no oil film between the liner and the ring is developed. Eventually, this is the capacitance signal when the flat surface on



top of the probe is moving without any lubricant film in between the liner and the piston-ring. The probe itself is very sensitive and slight variations can be measured. Even if we have ensured that the plate is as flat as possible with reference to the liner surface, the dial gauge shows a small variation (20 microns) between top dead centre (TDC) and bottom dead centre (BDC). The technique employed to manufacture the flat plate is responsible for this variation. The grinding machine, has caused expansion and shrinkage to the surface resulting in quite big surface anomalies introducing a big error for the detailed and comparatively small scale oil film thickness measurements (several microns).

#### 3.4.4 LASER INDUCED FLUORESCENCE SIGNAL

Pyke (2000) installed an optical fibre in the middle of the liner surface to investigate further the oil film thickness measurement technique and validate them in parallel to the capacitance measurements. The implant of the fibre required the same techniques developed for the installation of the fibres in the engine described by Duszynski (1999). Figure 3.11 shows the arrangement of the optical set-up; the air-cooled Laser and the box (Figure 3.12) that contains the mirrors, filters, photomultiplier tube and FC fibre coupler.

The calibration of the signal is accomplished with a similar method to that Pyke (2000) used. Using the post-processed data obtained from the capacitance signal (i.e. error - free series of data) the ring profile is fitted to the LIF signal at maximum speed (90° and 270°). The minimum point of the ring is assumed to be that given by the capacitance probe and the calibration coefficient was then adjusted to give a good fit.

In order to have more accuracy for the calibration, after the LIF curve is fitted in the best way possible to the ring profile, the distance between the LIF curve edge and the ring profile is measured and recorded. When the calibrated LIF data are used for the graphs, the curves will be shifted by the recorded offset.

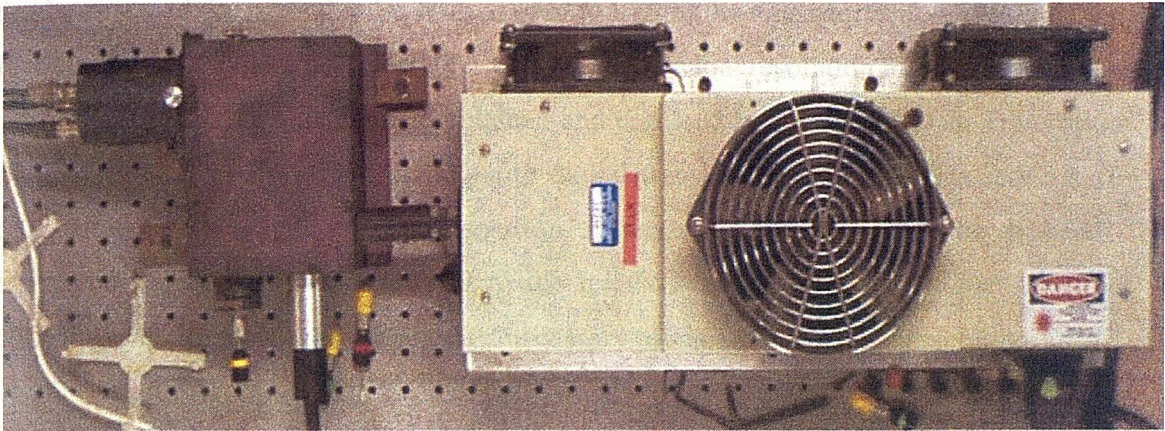


Figure 3.11 Optical Set-up

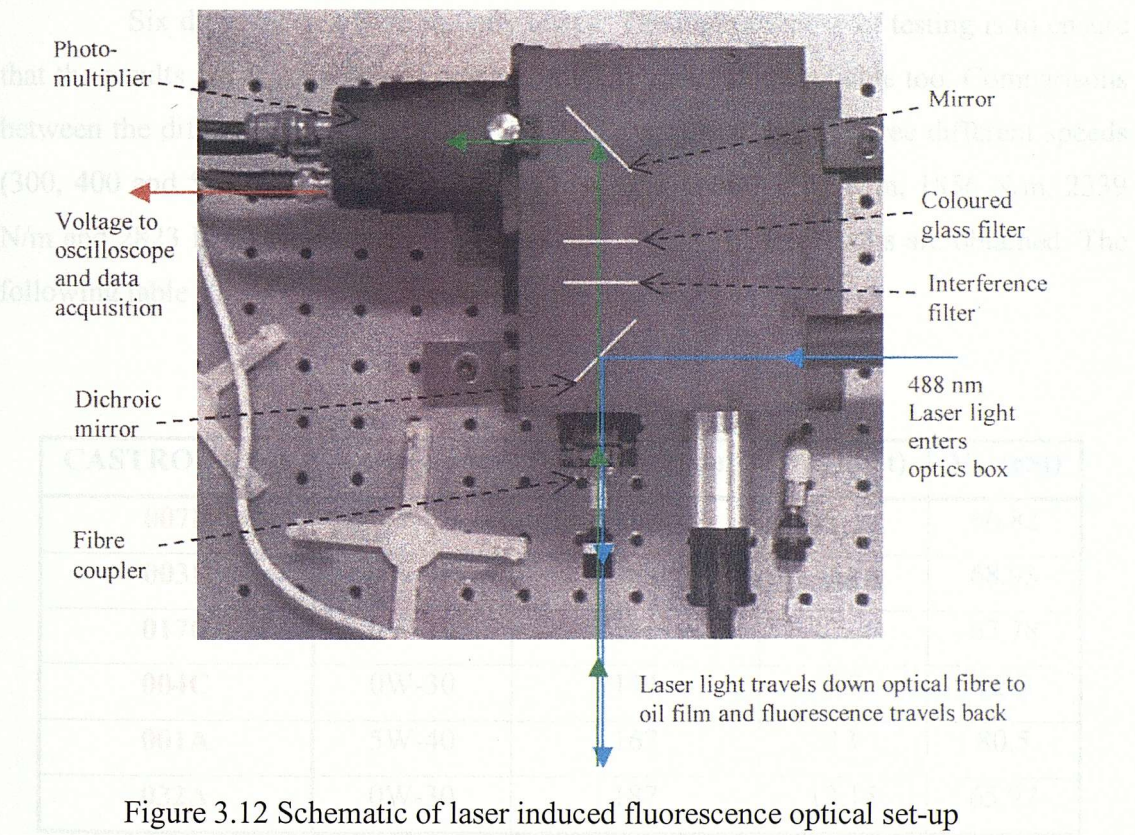


Figure 3.12 Schematic of laser induced fluorescence optical set-up

3.4.5 DATA ACQUISITION

The data acquisition is triggered by a shaft encoder, which gave two outputs. The first was a single marker pulse per revolution which allowed the identification of top



dead centre and the second pulse train is used to trigger the data acquisition to sample the first output. 2000 pulses per revolution allow a good spatial resolution to be obtained.

The analog signals were obtained and digitised by a DT2839 Data Translation acquisition card which allows analogue to digital translation rates of 1MHz through 16 differential channels with a 12 bit resolution. A maximum gain of 8 could be applied to the signal before digitisation. Owing to the relatively small voltages produced by the optical set-up, imposing the gain proved necessary to maintain adequate resolution.

3.5 RESULTS

Six different oils were initially tested. The main purpose of testing is to ensure that the results are repeatable and those of the LIF method are reliable too. Comparisons between the different oils were made. The tests are carried out for three different speeds (300, 400 and 500 rpm) and five different loads (971 N/m, 1372 N/m, 1856 N/m, 2339 N/m and 2823 N/m) for each oil. Capacitance, LIF and friction results are obtained. The following table shows the properties of the oils tested.

| CASTROL Code | SAE Grade | Viscosity Index | V <sub>100</sub> (cSt) | V <sub>40</sub> (cSt) |
|--------------|-----------|-----------------|------------------------|-----------------------|
| 007B         | 0W-30     | 187             | 12.18                  | 66.82                 |
| 003B         | 0W-30     | 182             | 12.16                  | 68.93                 |
| 017C         | 0W-30     | 183             | 12.26                  | 67.78                 |
| 004C         | 0W-30     | 174             | 9.8                    | 52.9                  |
| 001A         | 5W-40     | 167             | 13                     | 80.5                  |
| 032A         | 0W-30     | 187             | 12.15                  | 65.97                 |

Table 3.1 Tested Oil properties

3.5.1 OIL 7B

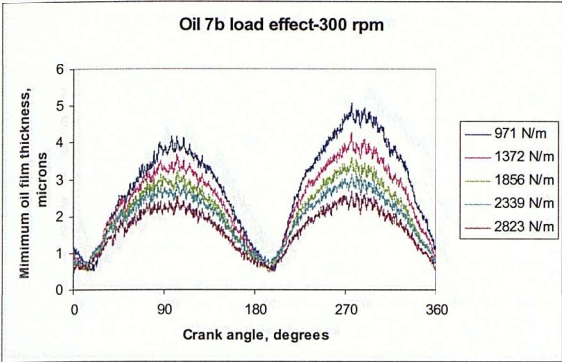


Fig. 3.13 Capacitance: Load effect at 300 rpm

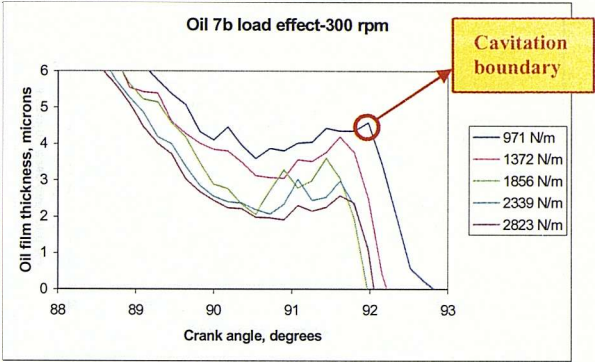


Fig. 3.14 (a) LIF: Load effect at 300 rpm

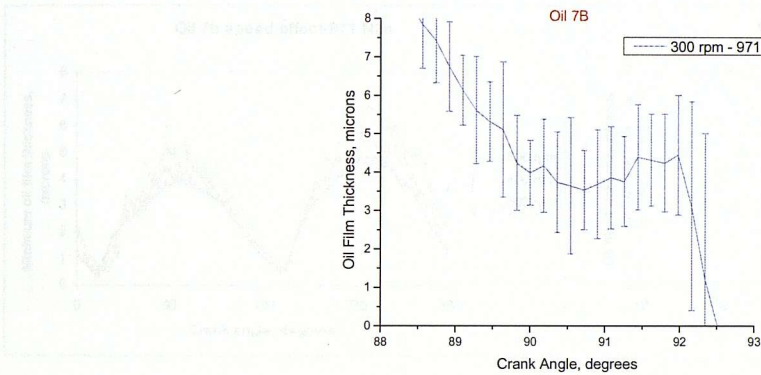


Fig. 3.14 (b) LIF: Error bars for 300 rpm, 971 N/m, based on 20 cycles

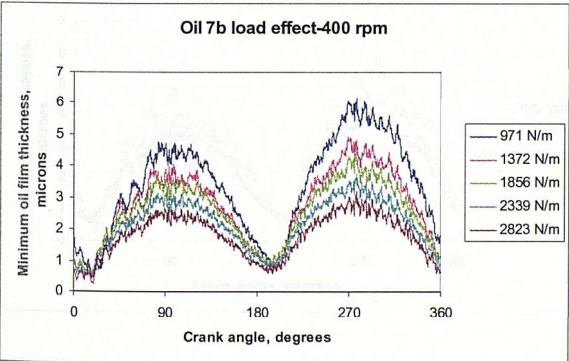


Fig. 3.15 Capacitance: Load effect at 400 rpm

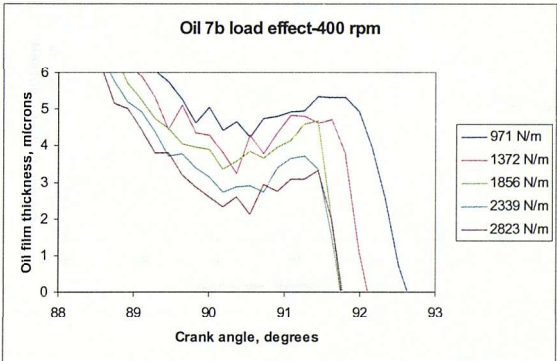


Fig. 3.16 LIF: Load effect at 400 rpm



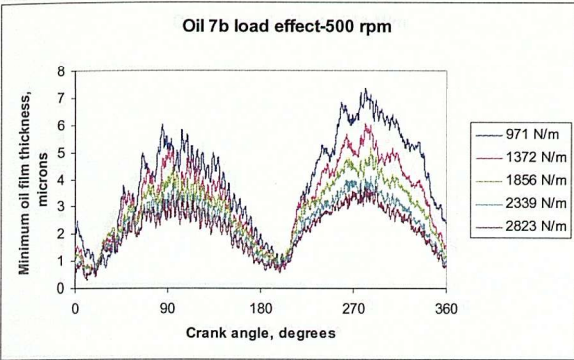


Fig. 3.17 Capacitance: Load effect at 500 rpm

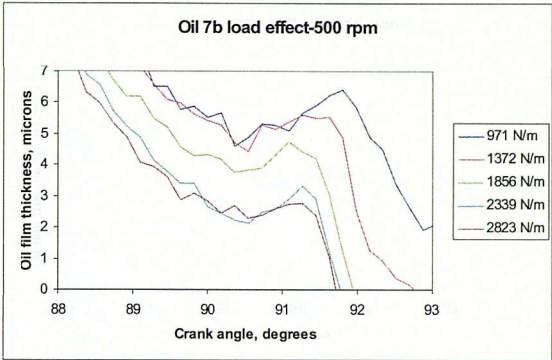


Fig. 3.18 LIF: Load effect at 500 rpm

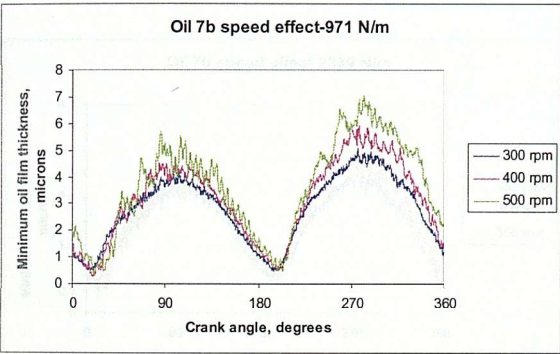


Fig. 3.19 Capacitance: Speed effect at 971 N/m

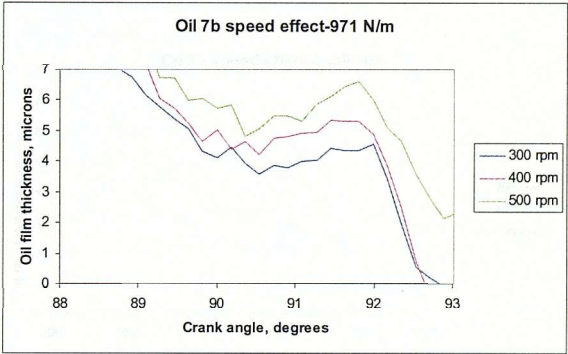


Fig. 3.20 LIF: Speed effect at 971 N/m

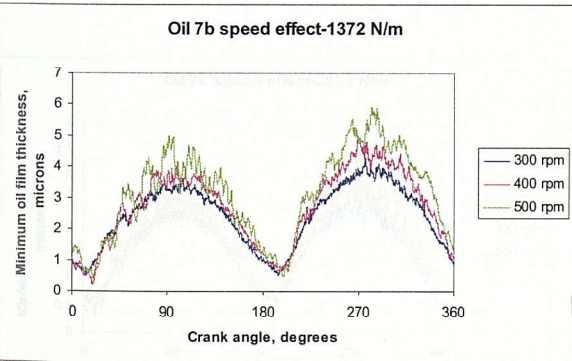


Fig. 3.21 Capacitance: Speed effect at 1372 N/m

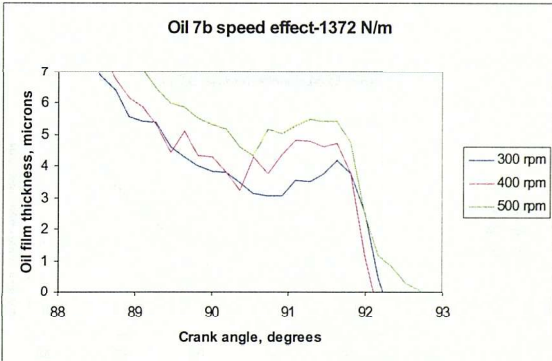


Fig. 3.22 LIF: Speed effect at 1372 N/m

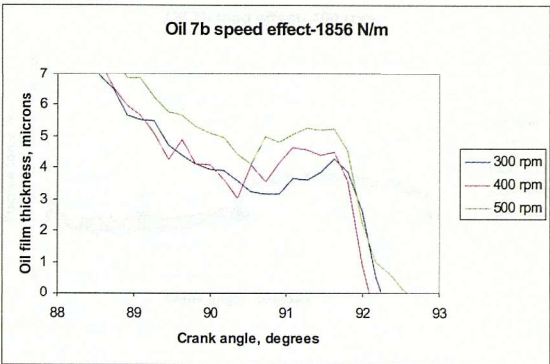
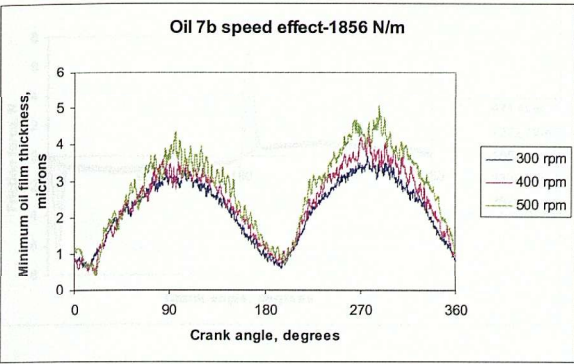


Fig. 3.23 Capacitance: Speed effect at 1856 N/m Fig. 3.24 LIF: Speed effect at 1856 N/m

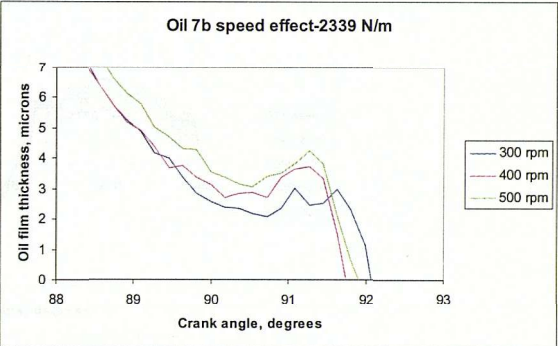
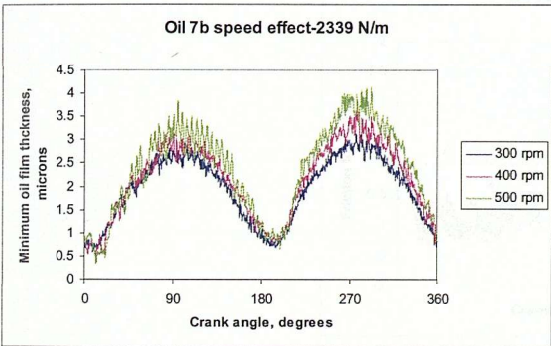


Fig. 3.25 Capacitance: Speed effect at 2339 N/m Fig. 3.26 LIF: Speed effect at 2339 N/m

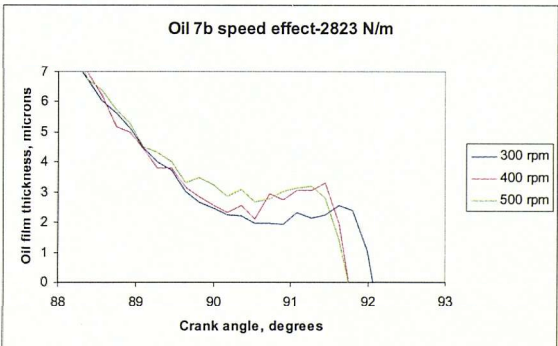
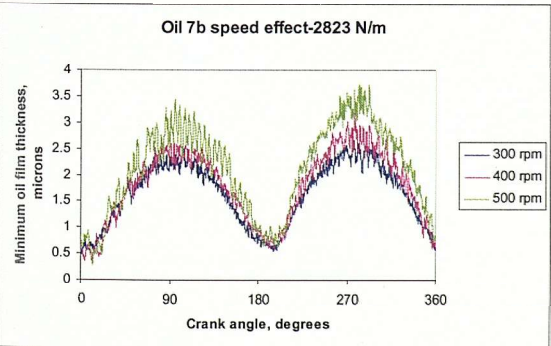


Fig. 3.27 Capacitance: Speed effect at 2823 N/m Fig. 3.28 LIF: Speed effect at 2823 N/m

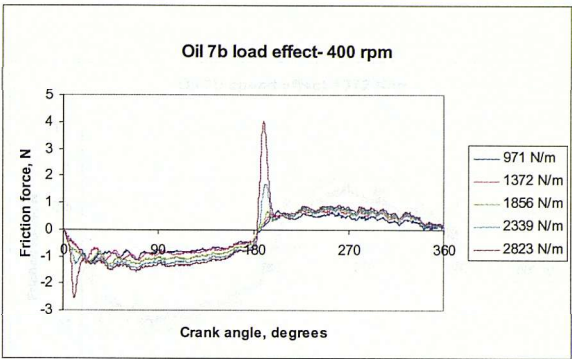
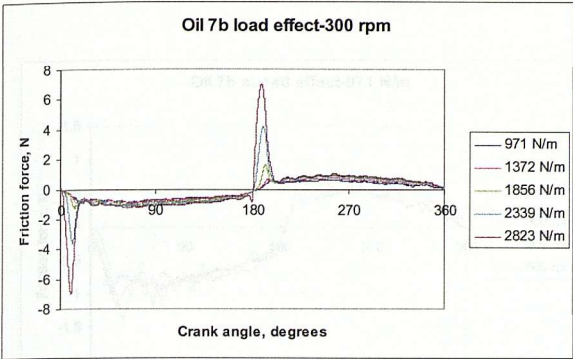


Fig. 3.29 Friction: Load effect at 300 rpm

Fig. 3.30 Friction: Load effect at 400 rpm

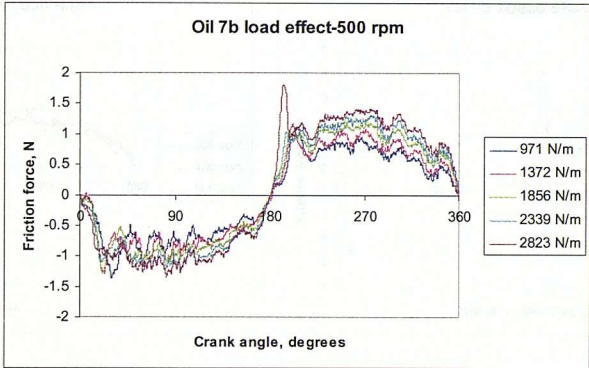


Fig. 3.31 Friction: Load effect at 500 rpm



3.5.2 Oil 4C

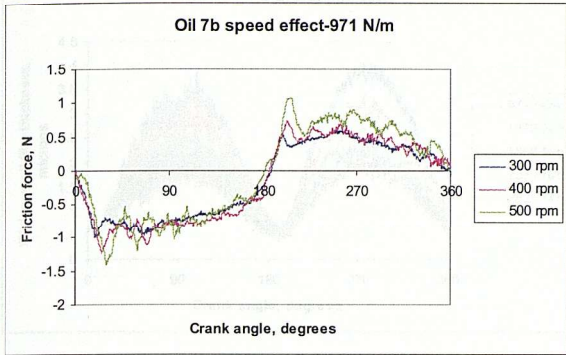


Fig. 3.32 Friction: Speed effect at 971 N/m

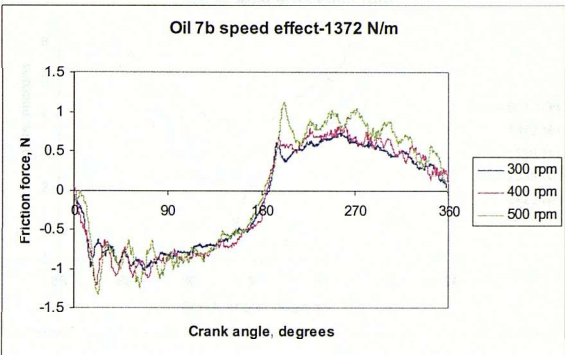


Fig. 3.33 Friction: Speed effect at 1372 N/m

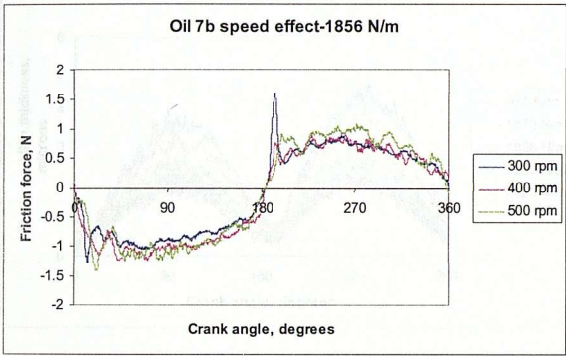


Fig. 3.34 Friction: Speed effect at 1856 N/m

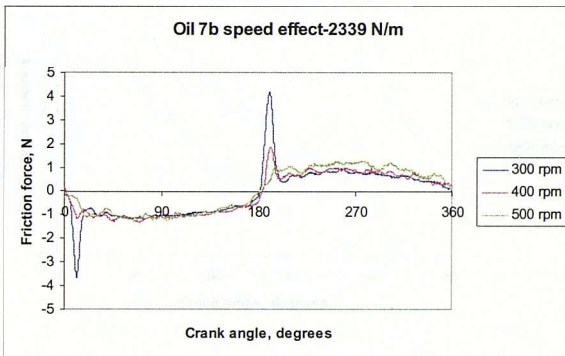


Fig. 3.35 Friction: Speed effect at 2339 N/m

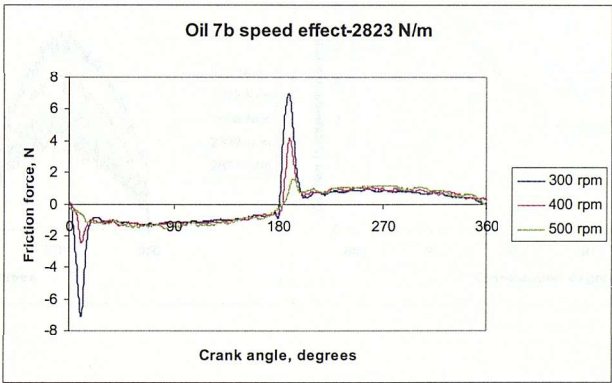


Fig. 3.36 Friction: Speed effect at 2823 N/m

3.5.2 OIL 4C

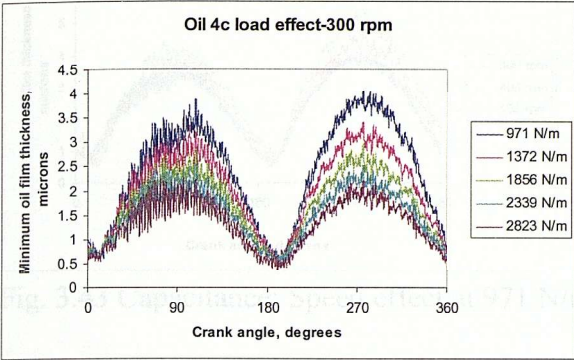


Fig. 3.37 Capacitance: Load effect at 300 rpm

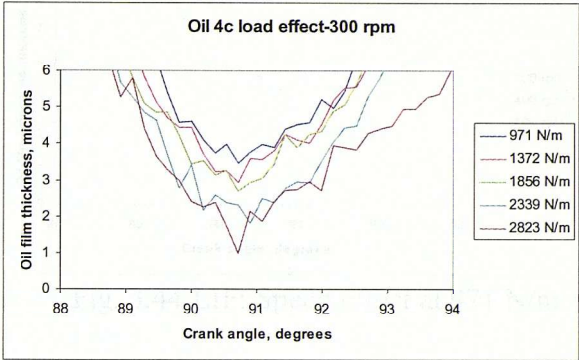


Fig. 3.38 LIF: Load effect at 300 rpm

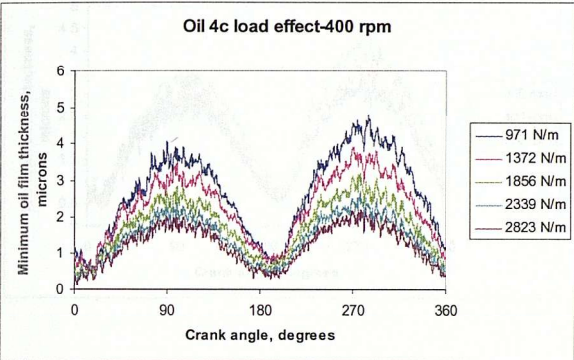


Fig. 3.39 Capacitance: Load effect at 400 rpm

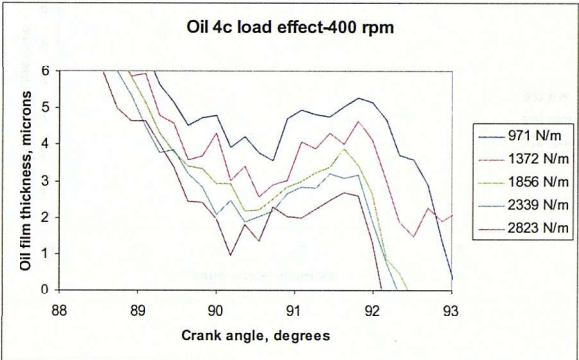


Fig. 3.40 LIF: Load effect at 400 rpm

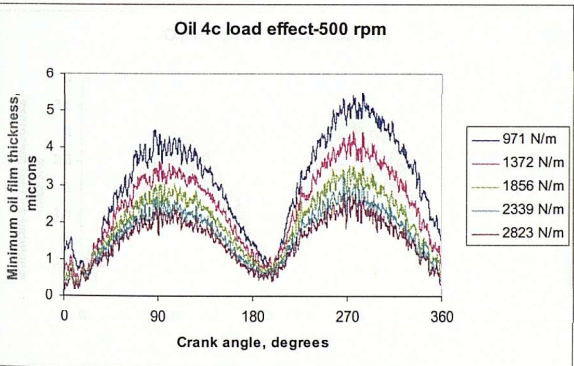


Fig. 3.41 Capacitance: Load effect at 500 rpm

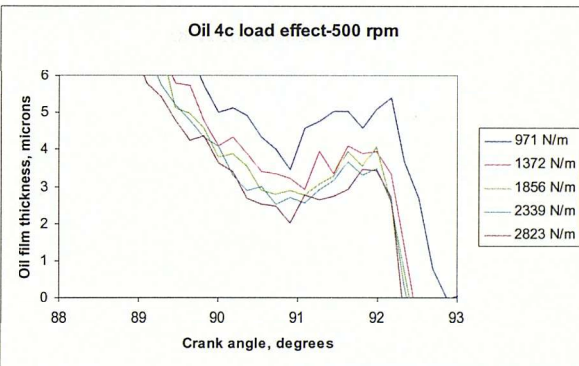


Fig. 3.42 LIF: Load effect at 500 rpm



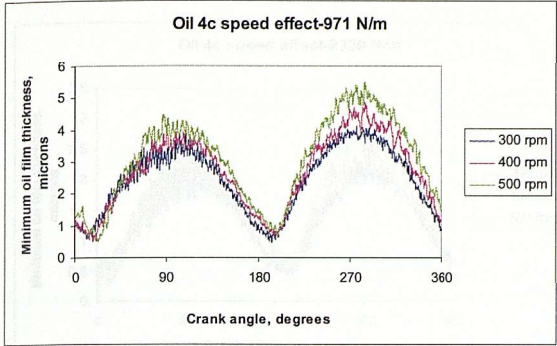


Fig. 3.43 Capacitance: Speed effect at 971 N/m

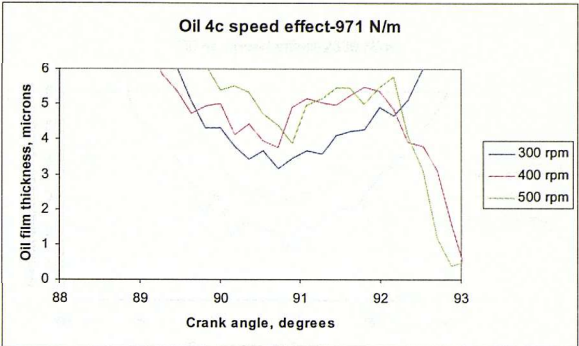


Fig. 3.44 LIF: Speed effect at 971 N/m

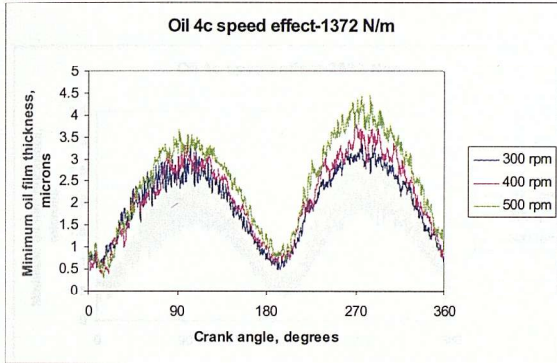


Fig. 3.45 Capacitance: Speed effect at 1372 N/m

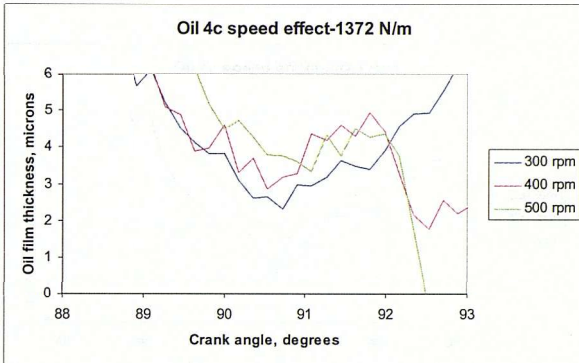


Fig. 3.46 LIF: Speed effect at 1372 N/m

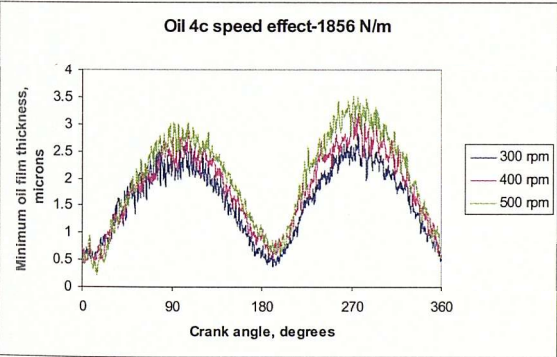


Fig. 3.47 Capacitance: Speed effect at 1856 N/m

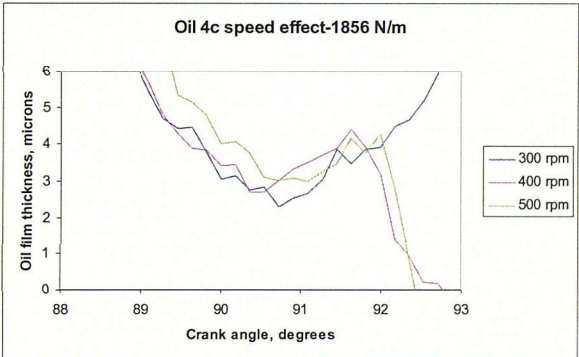


Fig. 3.48 LIF: Speed effect at 1856 N/m



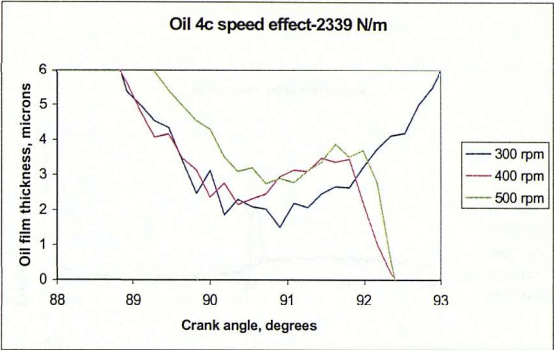
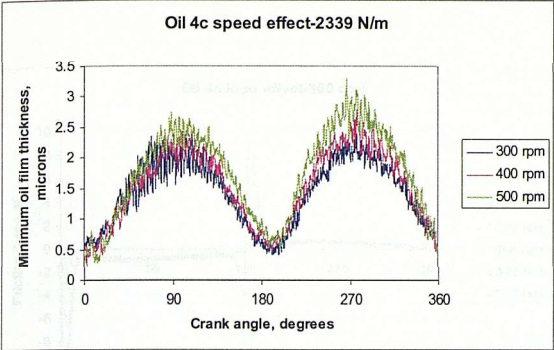


Fig. 3.49 Capacitance:Speed effect at 2339 N/m      Fig. 3.50 LIF: Speed effect at 2339 N/m

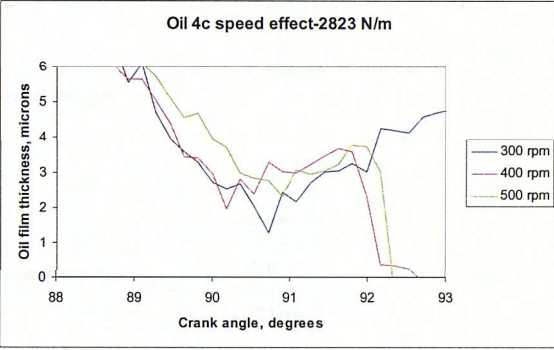
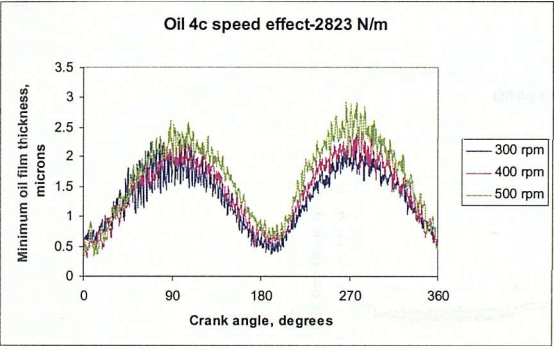


Fig. 3.51 Capacitance:Speed effect at 2823 N/m      Fig. 3.52 LIF: Speed effect at 2823 N/m

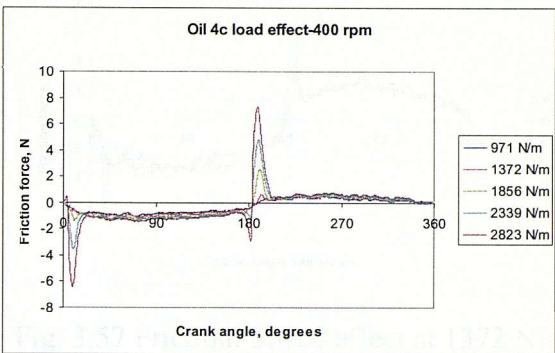
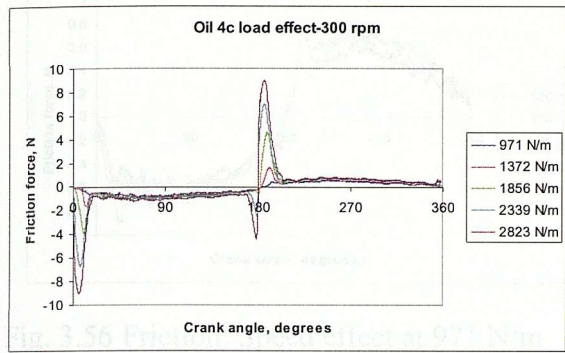


Fig. 3.53 Friction: Load effect at 300 rpm

Fig. 3.54 Friction: Load effect at 400 rpm

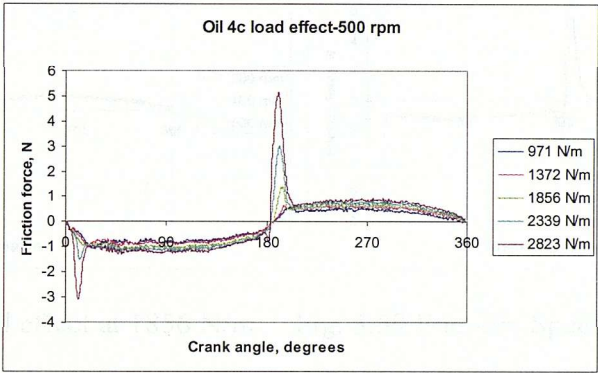


Fig. 3.55 Friction: Load effect at 500 rpm

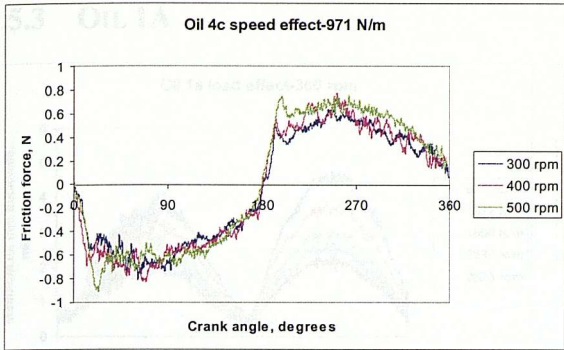


Fig. 3.56 Friction: Speed effect at 971 N/m

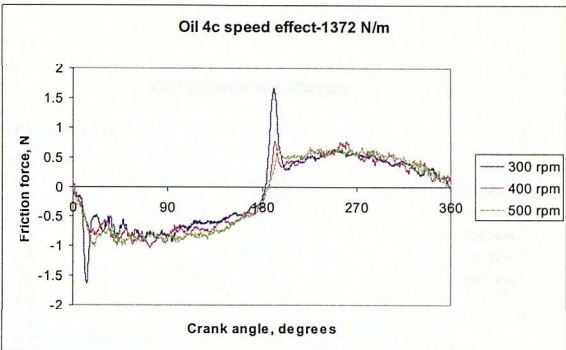


Fig. 3.57 Friction: Speed effect at 1372 N/m

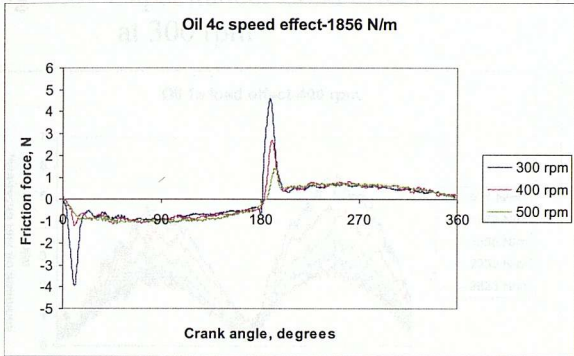


Fig. 3.58 Friction: Speed effect at 1856 N/m

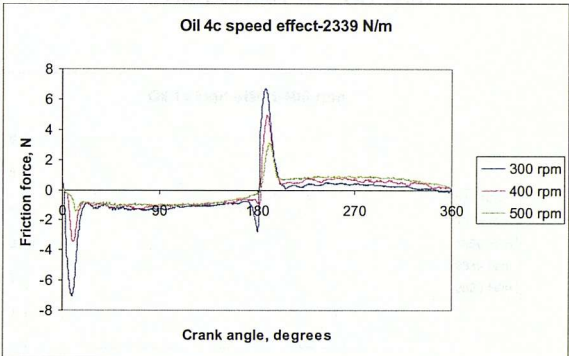


Fig. 3.59 Friction: Speed effect at 2339 N/m

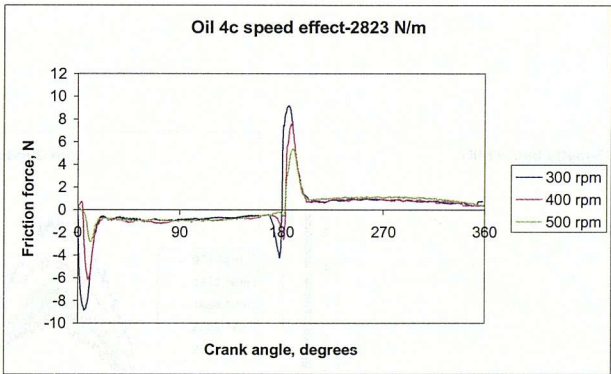


Fig. 3.60 Friction: Speed effect at 2823 N/m



3.5.3 OIL 1A

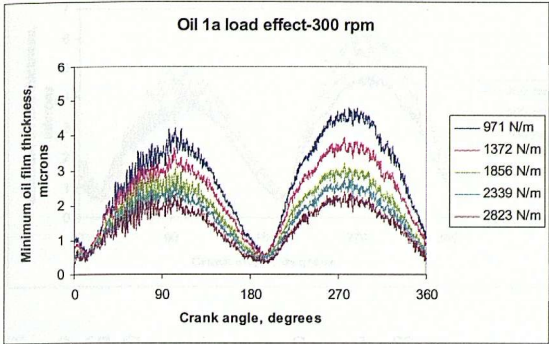


Fig. 3.61 Capacitance: Load effect at 300 rpm

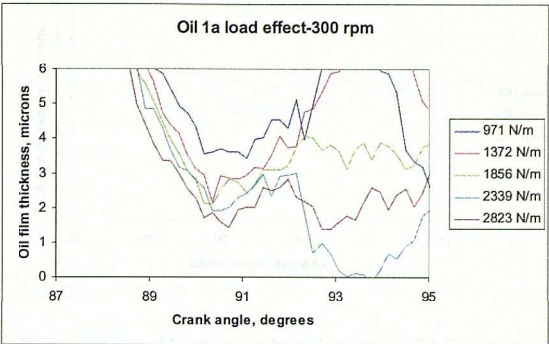


Fig. 3.62 LIF: Load effect at 300 rpm

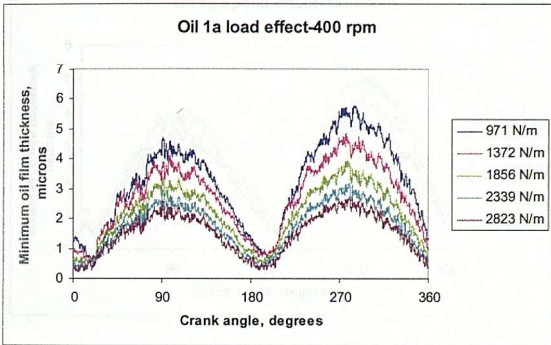


Fig. 3.63 Capacitance: Load effect at 400 rpm

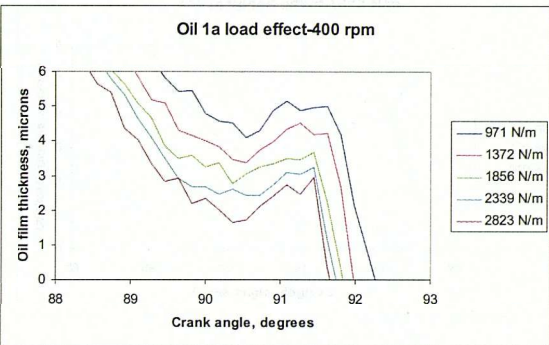


Fig. 3.64 LIF: Load effect at 400 rpm

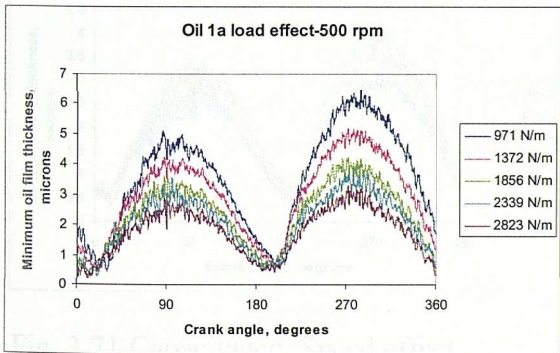


Fig. 3.65 Capacitance: Load effect at 500 rpm

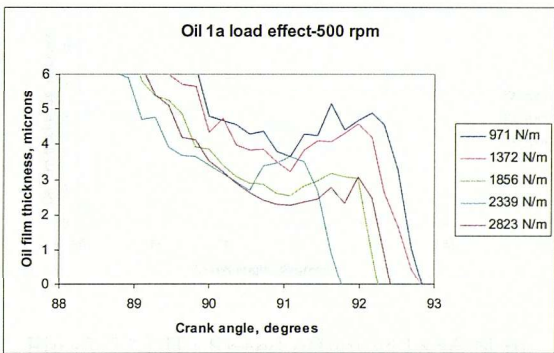


Fig. 3.66 LIF: Load effect at 500 rpm

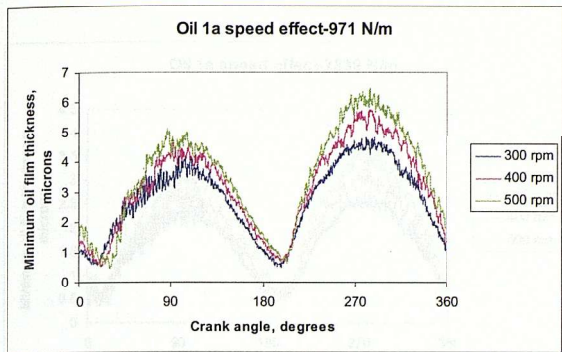


Fig. 3.67 Capacitance: Speed effect at 971 N/m

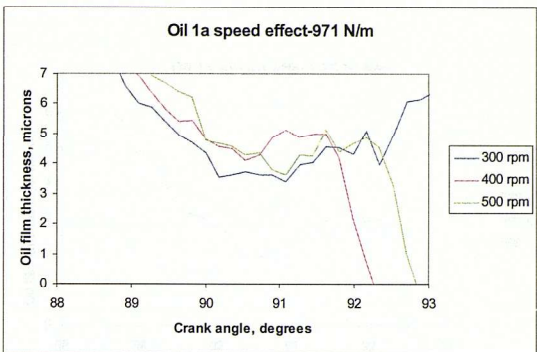


Fig. 3.68 LIF: Speed effect at 971 N/m

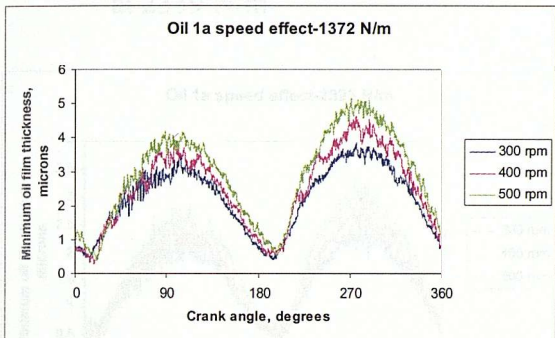


Fig. 3.69 Capacitance: Speed effect at 1372 N/m

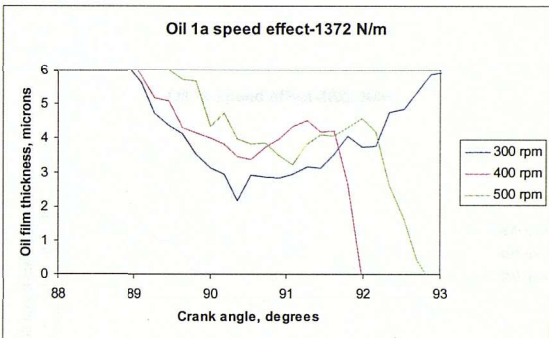


Fig. 3.70 LIF: Speed effect at 1372 N/m

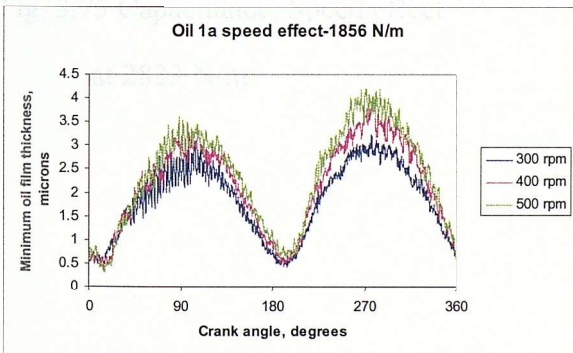


Fig. 3.71 Capacitance: Speed effect at 1856 N/m

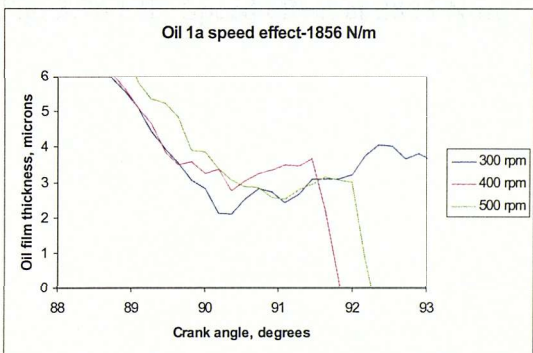


Fig. 3.72 LIF: Speed effect at 1856 N/m

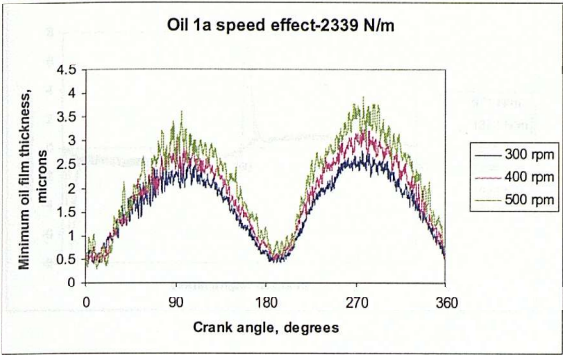


Fig. 3.73 Capacitance: Speed effect at 2339 N/m

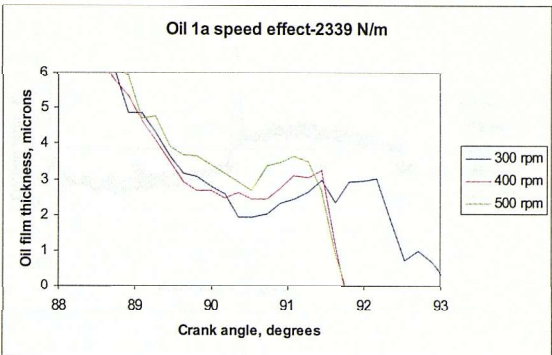


Fig. 3.74 LIF: Speed effect at 2339 N/m

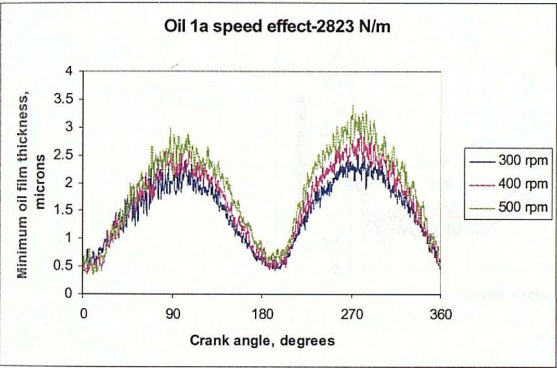


Fig. 3.75 Capacitance: Speed effect at 2823 N/m

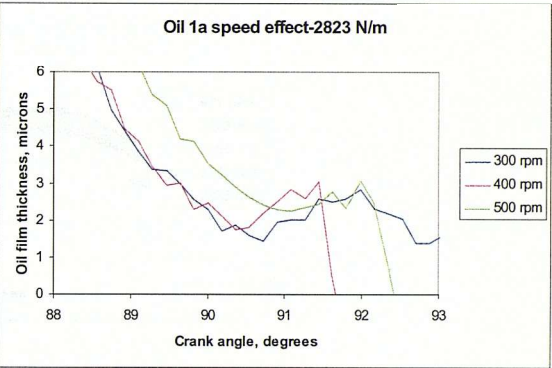


Fig. 3.76 LIF: Speed effect at 2823 N/m



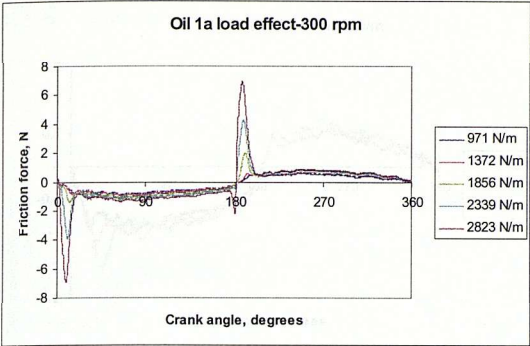


Fig. 3.77 Friction: Load effect at 300 rpm

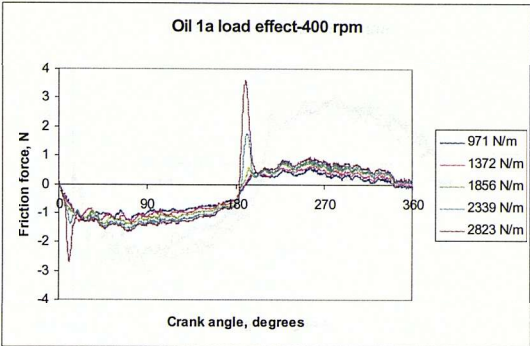


Fig. 3.78 Friction: Load effect at 400 rpm

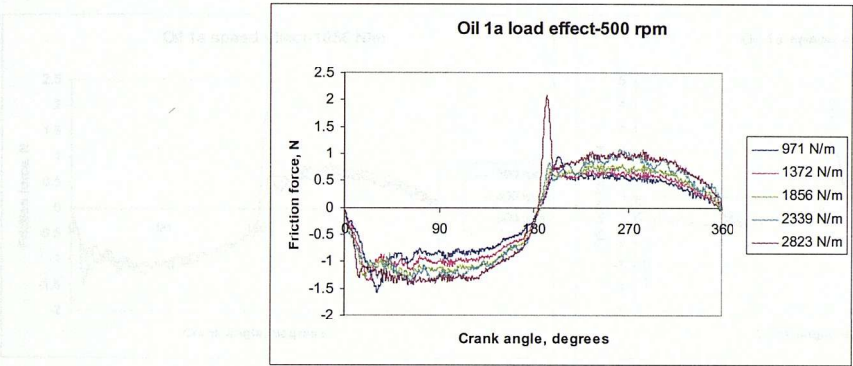


Fig. 3.79 Friction: Load effect at 500 rpm

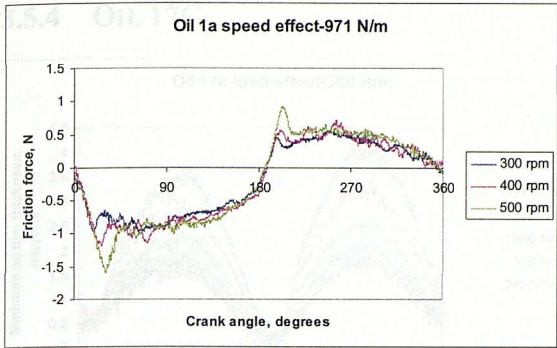


Fig. 3.80 Friction: Speed effect at 971N/m

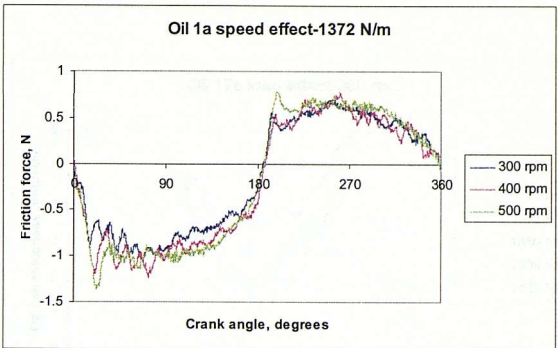


Fig. 3.81 Friction: Speed effect at 1372 N/m

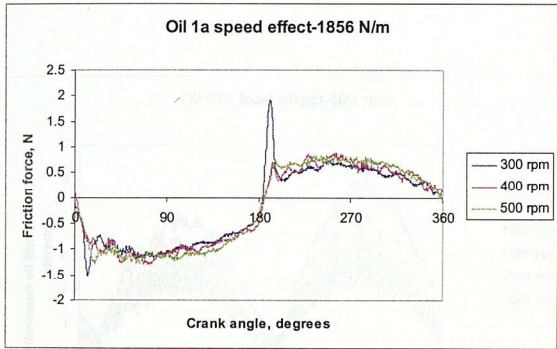


Fig. 3.82 Friction: Speed effect at 1856 N/m

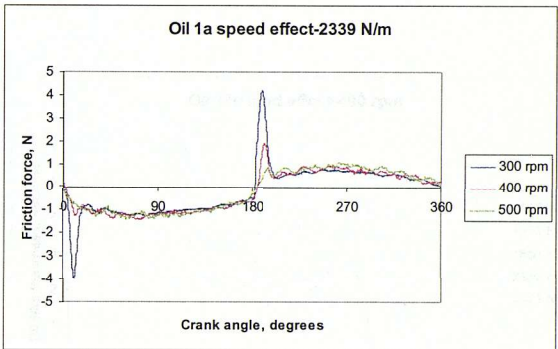


Fig. 3.83 Friction: Speed effect at 2339 N/m

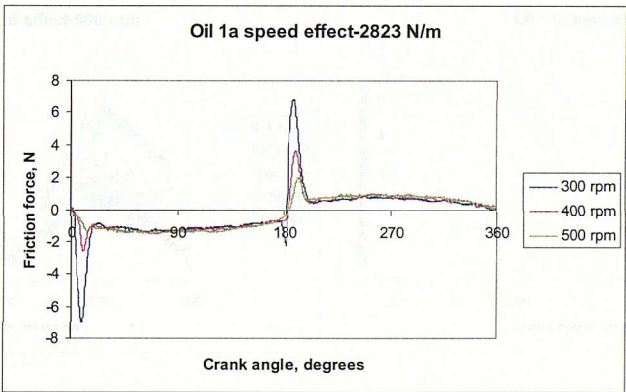


Fig. 3.84 Friction: Speed effect at 2823 N/m

3.5.4 OIL 17C

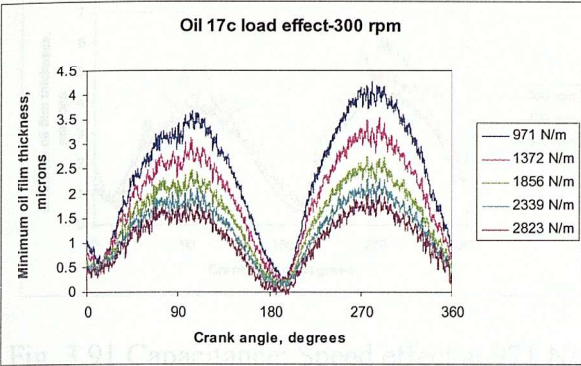


Fig. 3.85 Capacitance: Load effect at 300 rpm

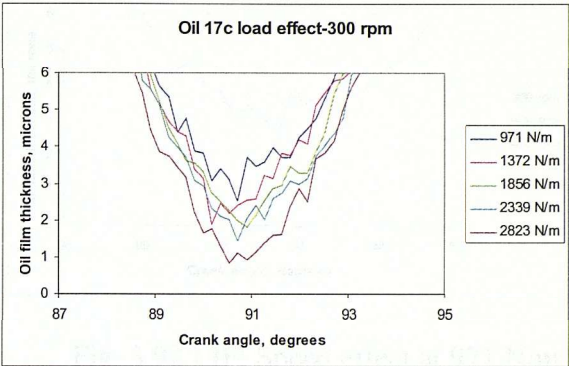


Fig. 3.86 LIF: Load effect at 300 rpm

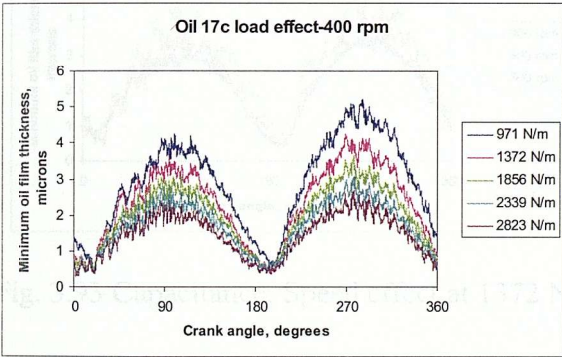


Fig. 3.87 Capacitance: Load effect at 400 rpm

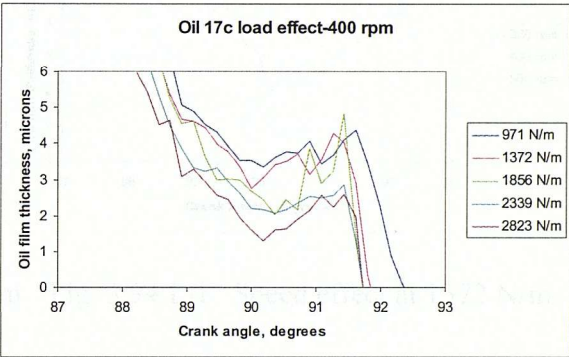


Fig. 3.88 LIF: Load effect at 400 rpm

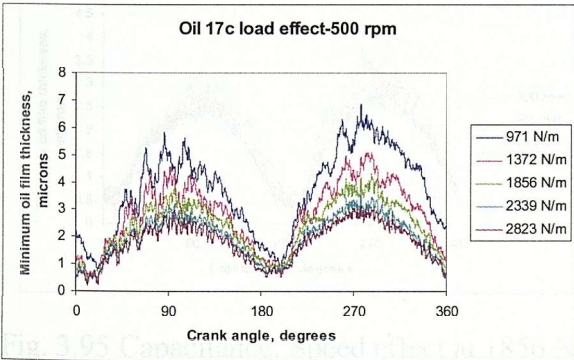


Fig. 3.89 Capacitance: Load effect at 500 rpm

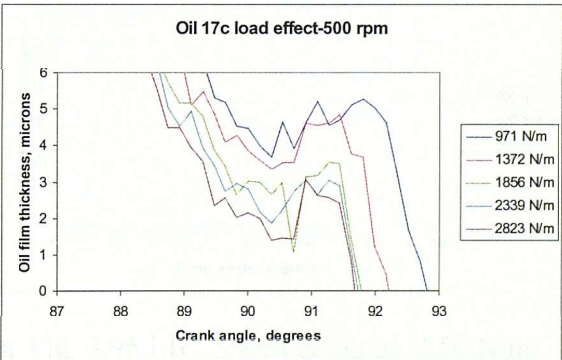


Fig. 3.90 LIF: Load effect at 500 rpm



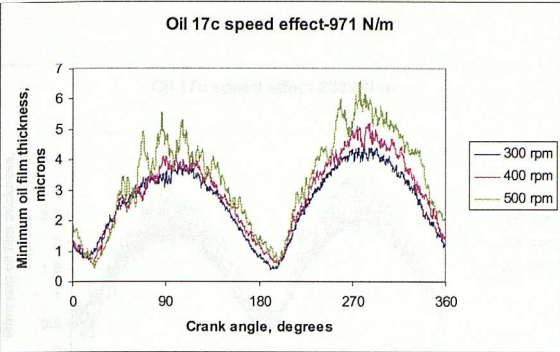


Fig. 3.91 Capacitance: Speed effect at 971 N/m

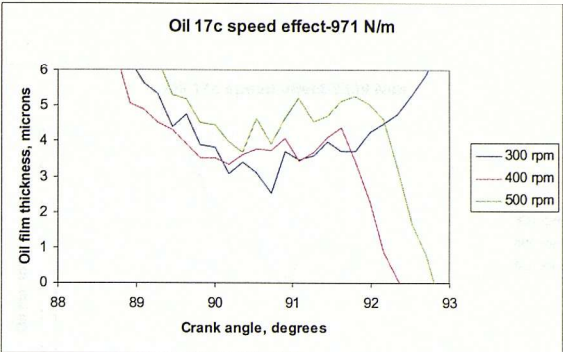


Fig. 3.92 LIF: Speed effect at 971 N/m

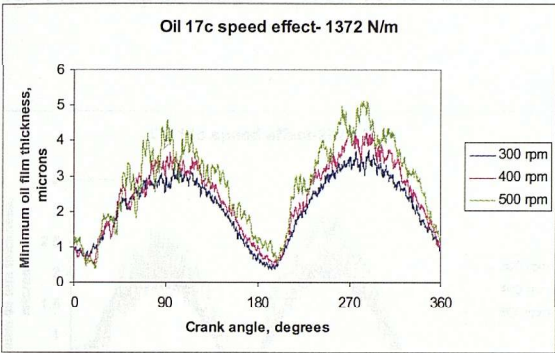


Fig. 3.93 Capacitance: Speed effect at 1372 N/m

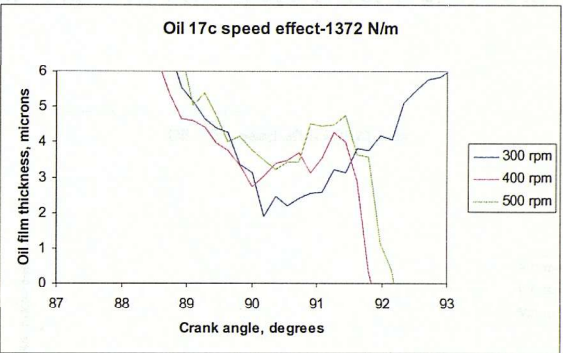


Fig. 3.94 LIF: Speed effect at 1372 N/m

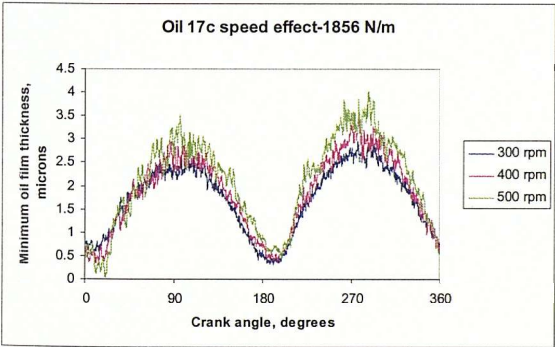


Fig. 3.95 Capacitance: Speed effect at 1856 N/m

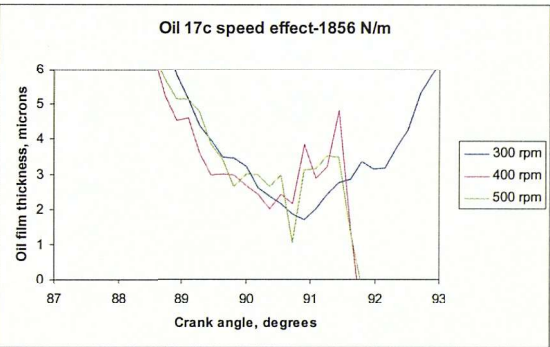


Fig. 3.96 LIF: Speed effect at 1856 N/m

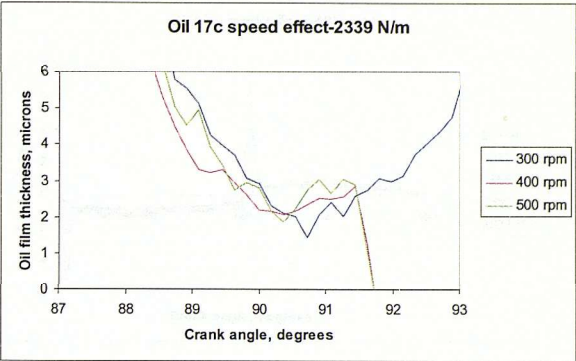
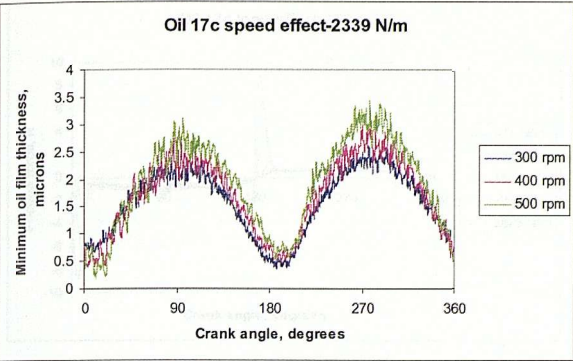


Fig. 3.97 Capacitance: Speed effect at 2339 N/m      Fig. 3.98 LIF: Speed effect at 2339 N/m

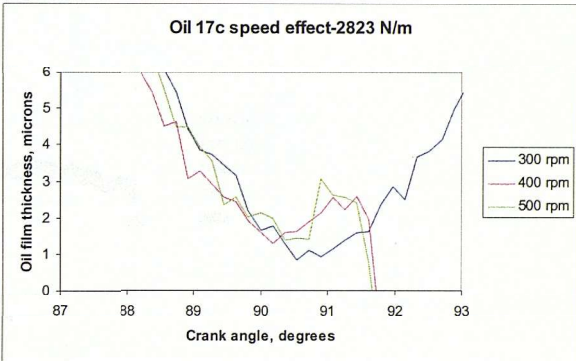
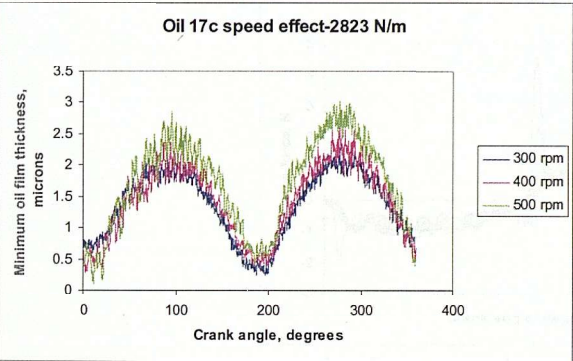


Fig. 3.99 Capacitance: Speed effect at 2823 N/m      Fig. 3.100 LIF: Speed effect at 2823 N/m

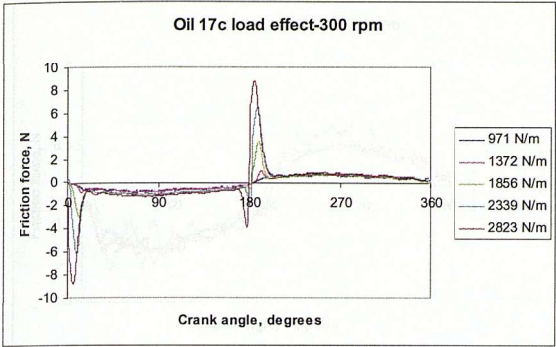


Fig. 3.101 Friction: Load effect at 300 rpm

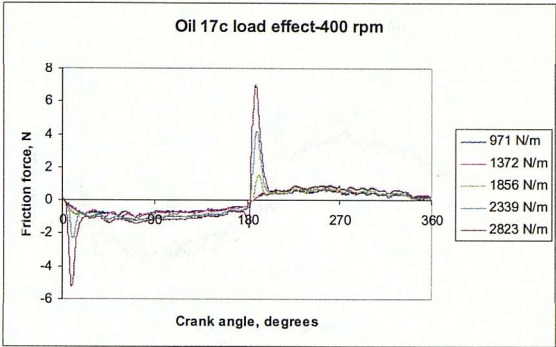


Fig. 3.102 Friction: Load effect at 400 rpm

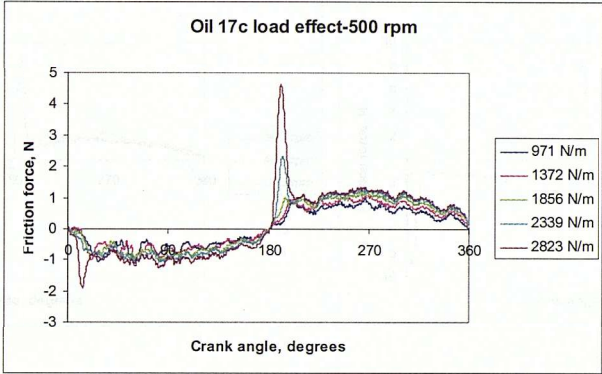


Fig. 3.103 Friction: Load effect at 500 rpm



3.5.5 OIL3B

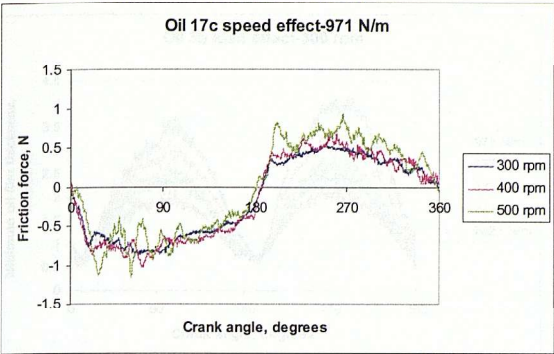


Fig. 3.104 Friction: Speed effect at 971 N/m

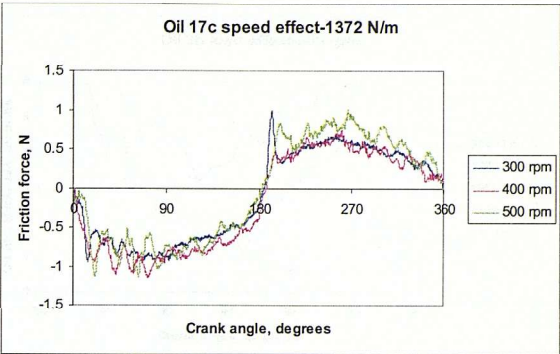


Fig. 3.105 Friction: Speed effect at 1372 N/m

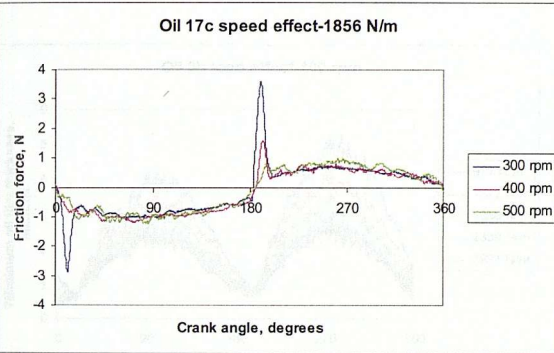


Fig. 3.106 Friction: Speed effect at 1856 N/m

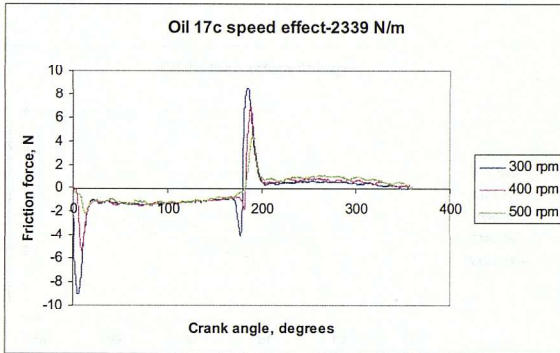


Fig. 3.107 Friction: Speed effect at 2339 N/m

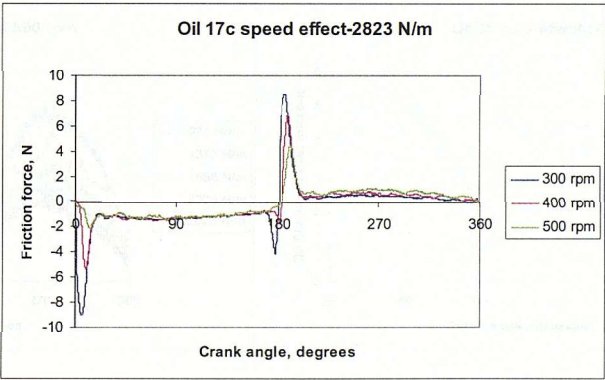


Fig. 3.108 Friction: Speed effect at 2823 N/m

3.5.5 OIL 3B

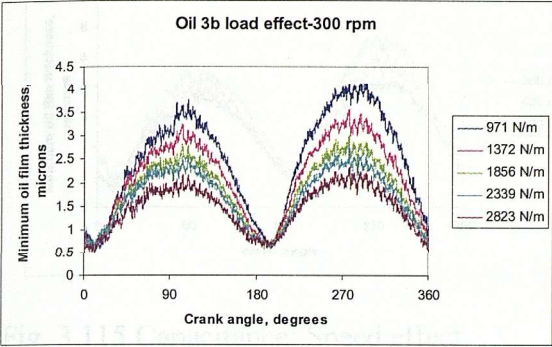


Fig. 3.109 Capacitance: Load effect at 300 rpm

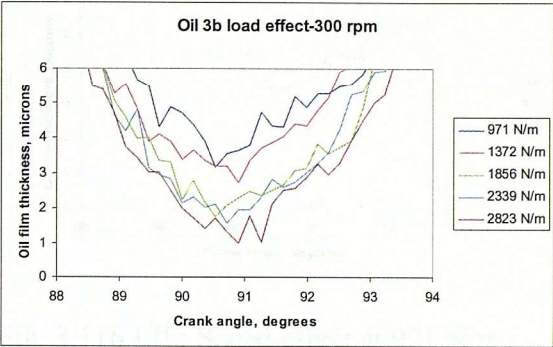


Fig. 3.110 LIF: Load effect at 300 rpm

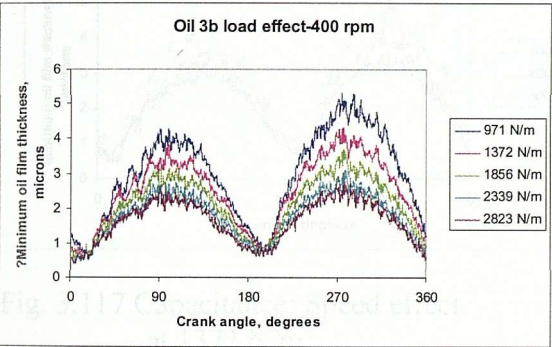


Fig. 3.111 Capacitance: Load effect at 400 rpm

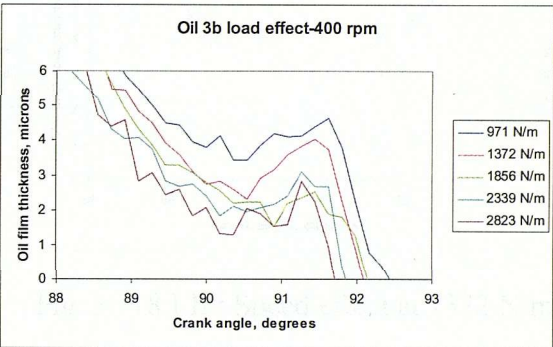


Fig. 3.112 LIF: Load effect at 400 rpm

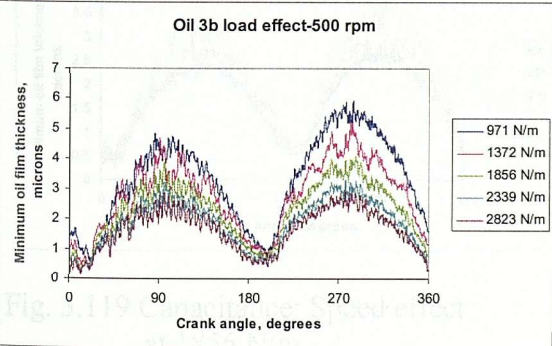


Fig. 3.113 Capacitance: Load effect at 500 rpm

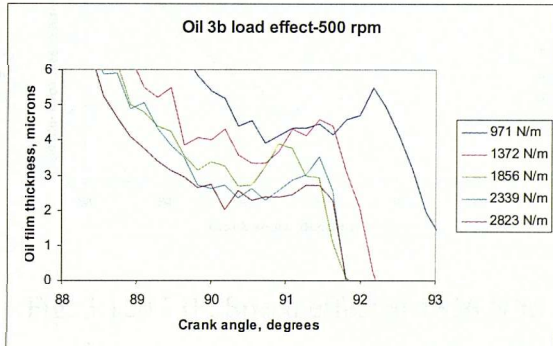


Fig. 3.114 LIF: Load effect at 500 rpm



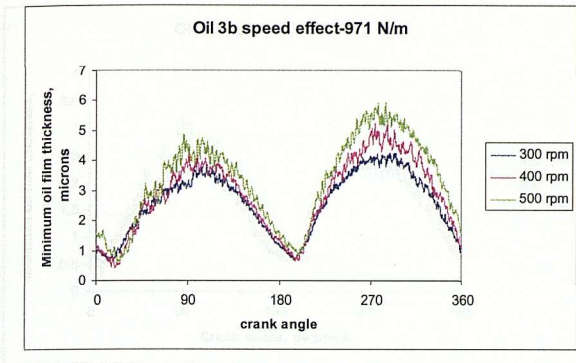


Fig. 3.115 Capacitance: Speed effect at 971 N/m

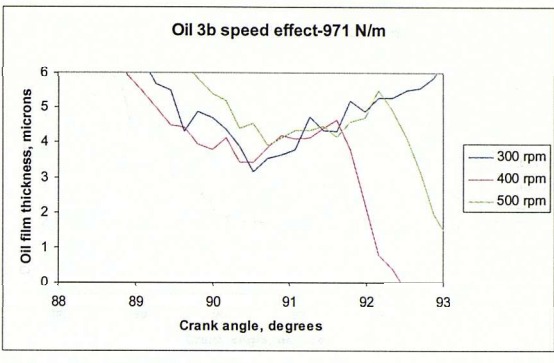


Fig. 3.116 LIF: Speed effect at 971 N/m

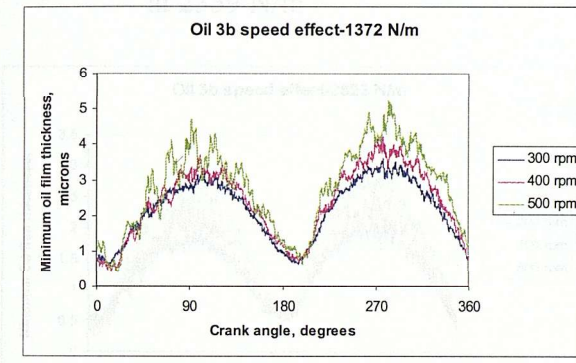


Fig. 3.117 Capacitance: Speed effect at 1372 N/m

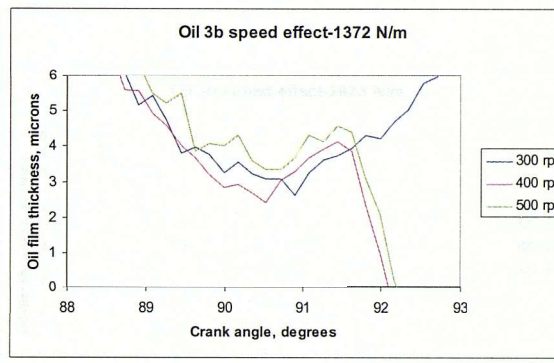


Fig. 3.118 LIF: Speed effect at 1372 N/m

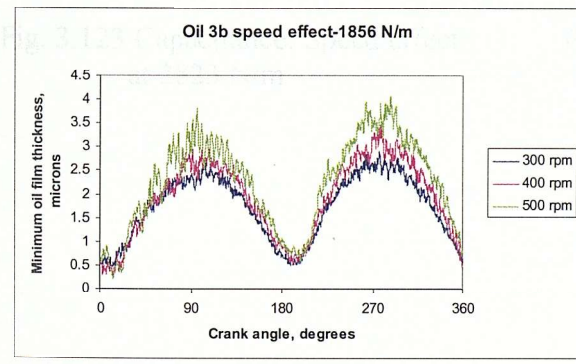


Fig. 3.119 Capacitance: Speed effect at 1856 N/m

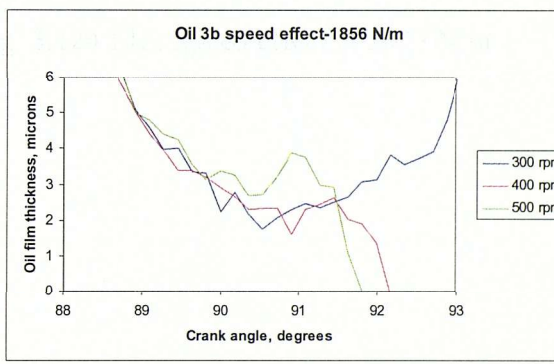


Fig. 3.120 LIF: Speed effect at 1856 N/m

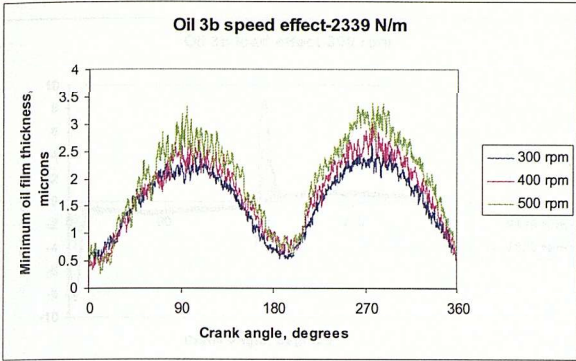


Fig. 3.121 Capacitance: Speed effect at 2339 N/m

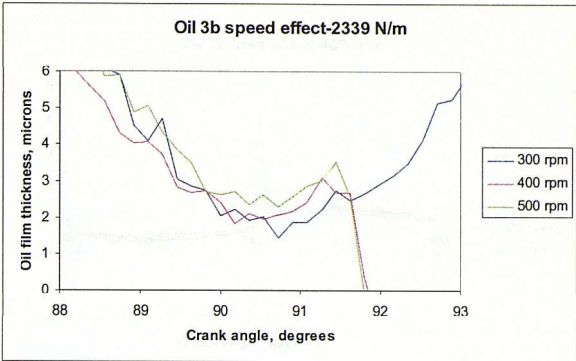


Fig. 3.122 LIF: Speed effect at 2339 N/m

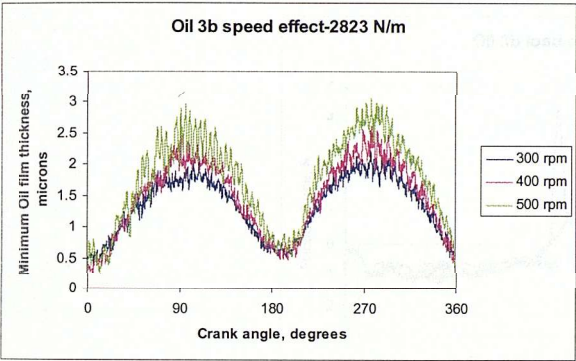


Fig. 3.123 Capacitance: Speed effect at 2823 N/m

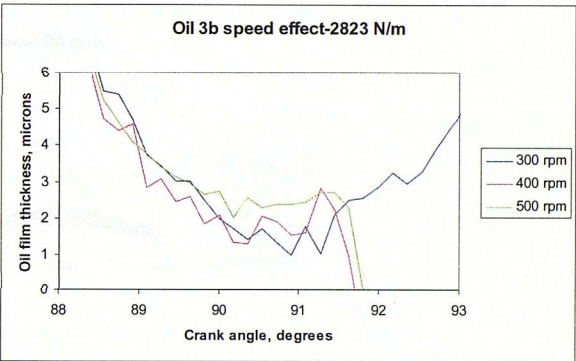


Fig. 3.124 LIF: Speed effect at 2823 N/m

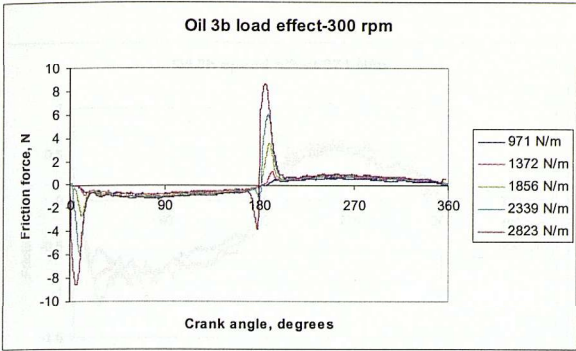


Fig. 3.125 Friction: Load effect at 300 rpm

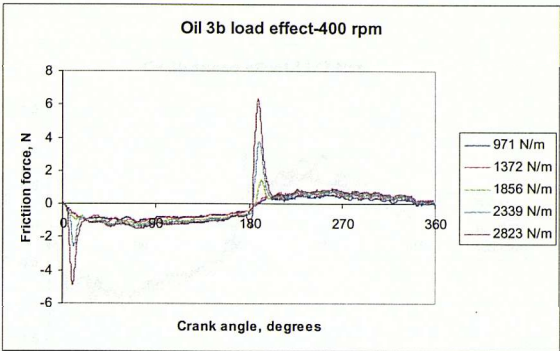


Fig. 3.126 Friction: Load effect at 400 rpm

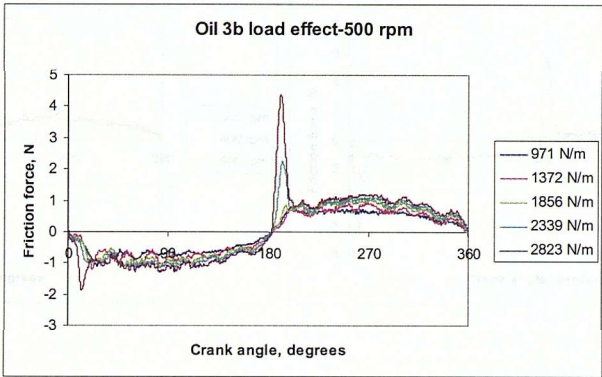


Fig. 3.127 Friction: Load effect at 500 rpm



3.5.6 Oil 32A

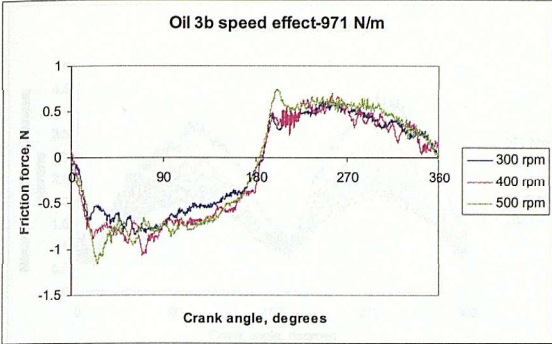


Fig. 3.128 Friction: Speed effect at 971 N/m

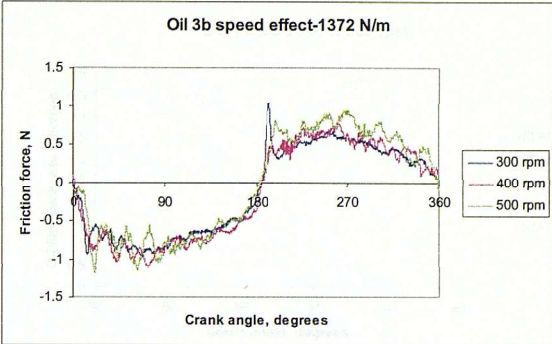


Fig. 3.129 Friction: Speed effect at 1372 N/m

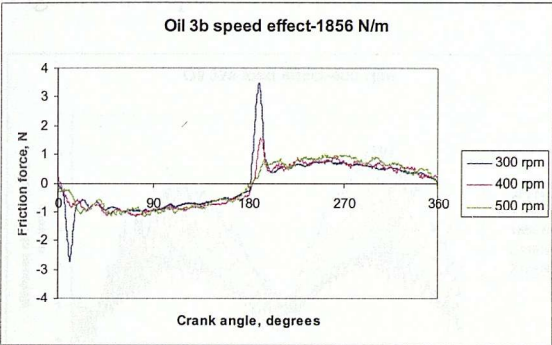


Fig. 3.130 Friction: Speed effect at 1856 N/m

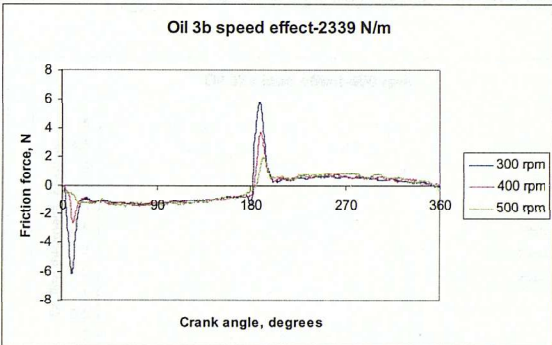


Fig. 3.131 Friction:Speed effect at 2339 N/m

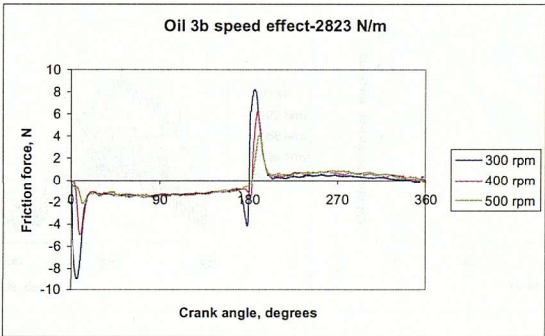


Fig. 3.132 Friction: Speed effect at 2823 N/m



3.5.6 OIL 32A

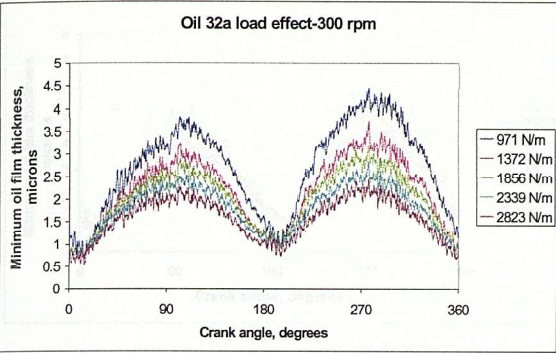


Fig. 3.133 Capacitance: Load effect at 300 rpm

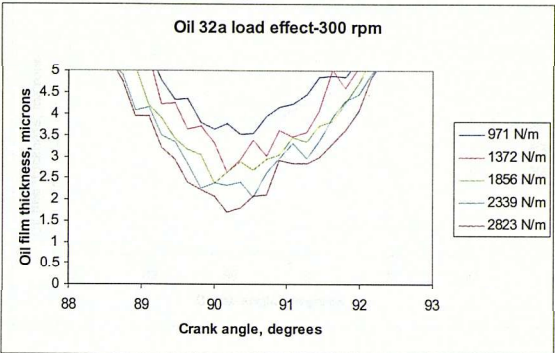


Fig. 3.134 LIF: Load effect at 300 rpm

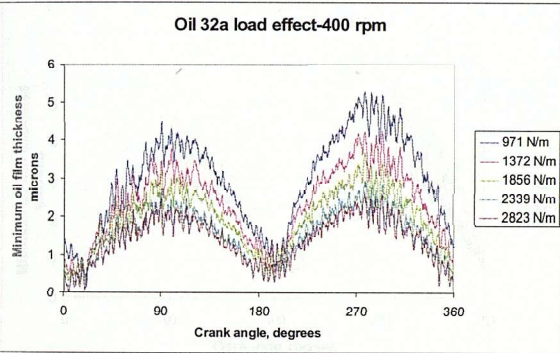


Fig. 3.135 Capacitance: Load effect at 400 rpm

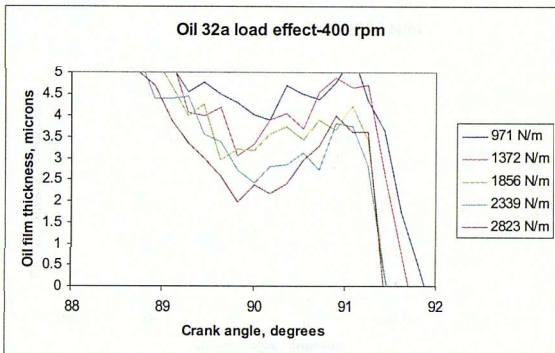


Fig. 3.136 LIF: Load effect at 400 rpm

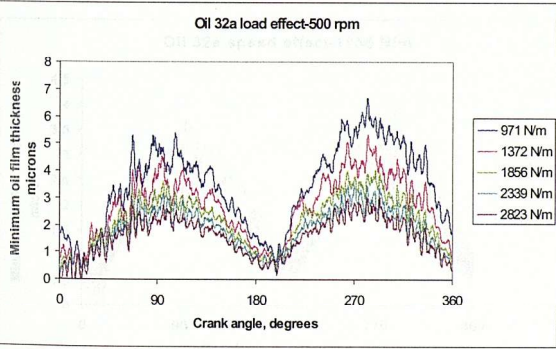


Fig. 3.137 Capacitance: Load effect at 500 rpm

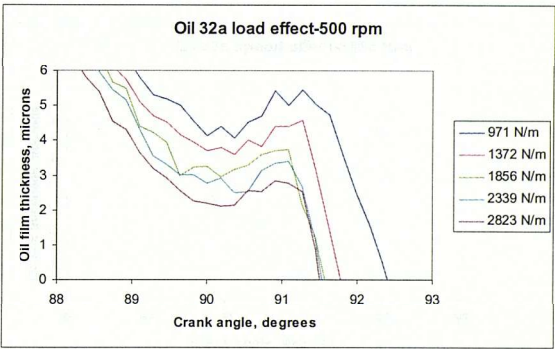


Fig. 3.138 LIF: Load effect at 500 rpm

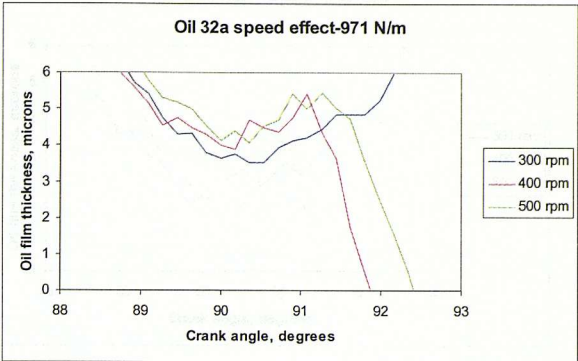
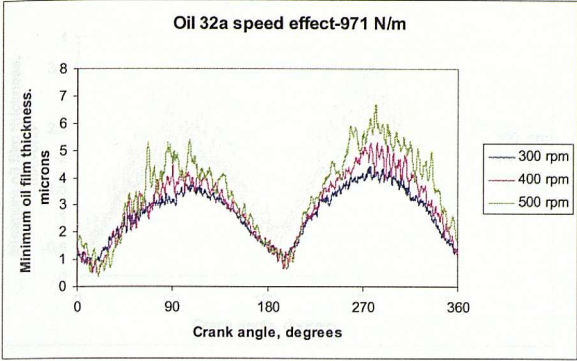


Fig. 3.139 Capacitance: Speed effect at 971 N/m Fig. 3.140 LIF: Speed effect at 971 N/m

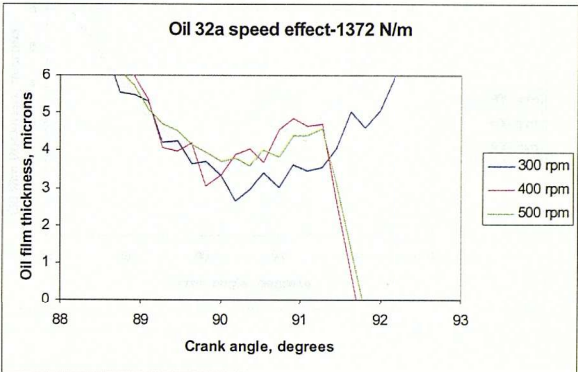
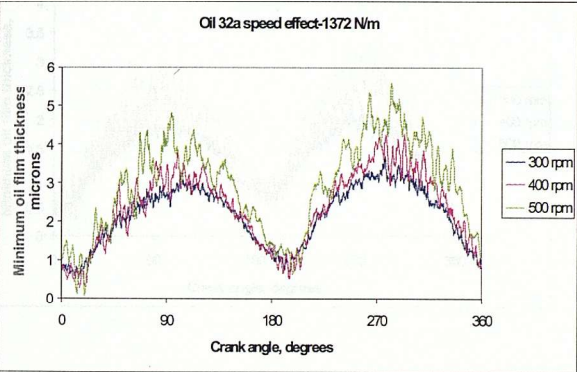


Fig. 3.141 Capacitance: Speed effect at 1372 N/m Fig. 3.142 LIF: Speed effect at 1372 N/m

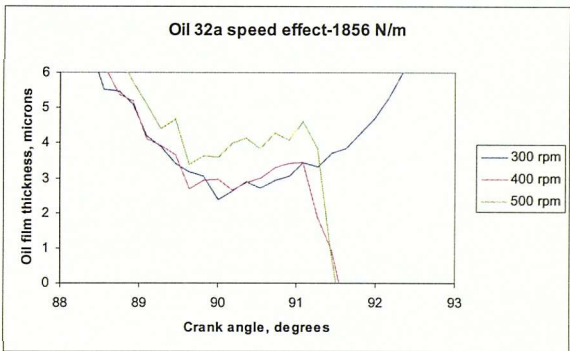
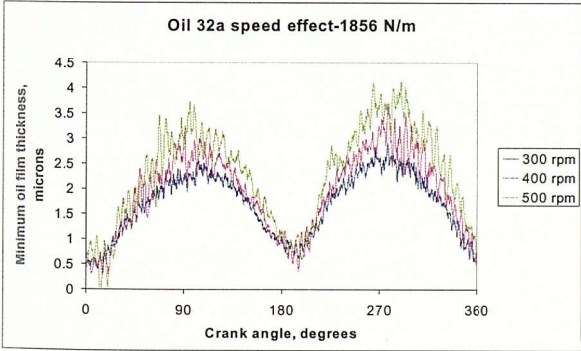


Fig. 3.143 Capacitance: Speed effect at 1856 N/m Fig. 3.144 LIF: Speed effect at 1856 N/m



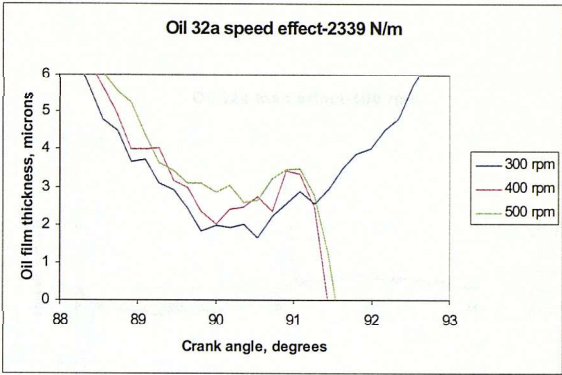
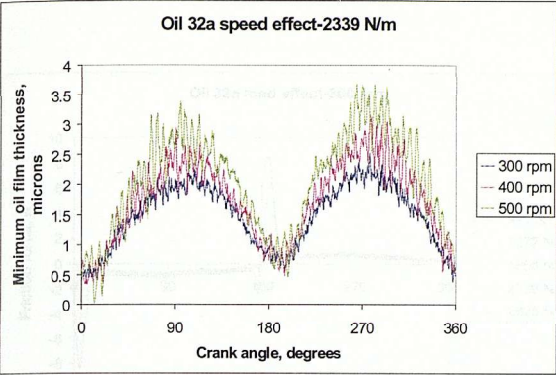


Fig. 3.145 Capacitance: Speed effect at 2339N/m Fig. 3.146 LIF: Speed effect at 2339 N/m

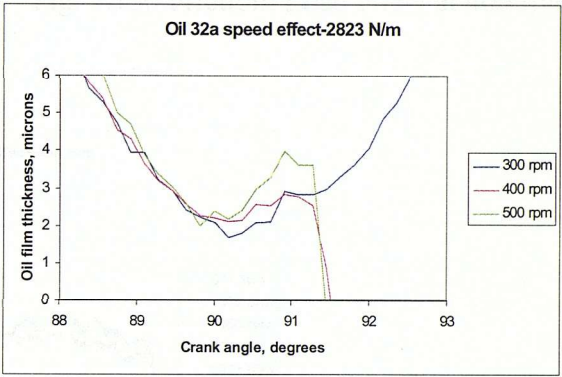
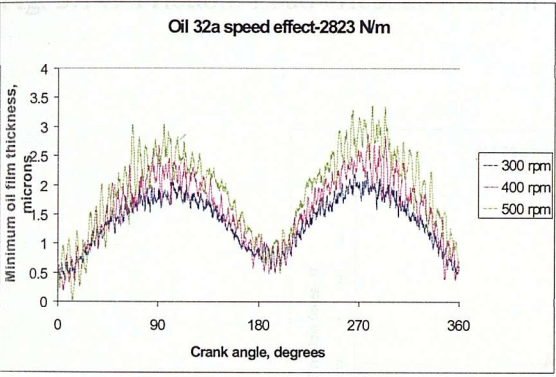


Fig. 3.147 Capacitance: Speed effect at 2823 N/m Fig. 3.148 LIF: Speed effect at 2823 N/m

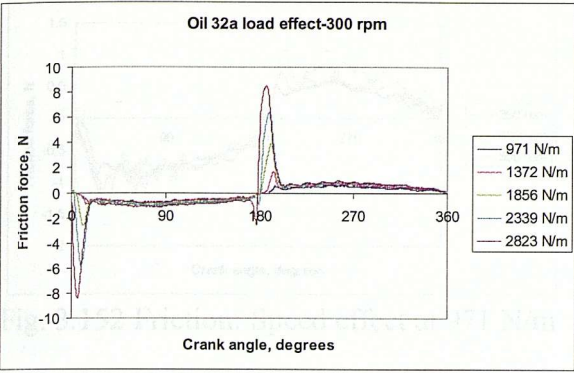


Fig. 3.149 Friction: Load effect at 300 rpm

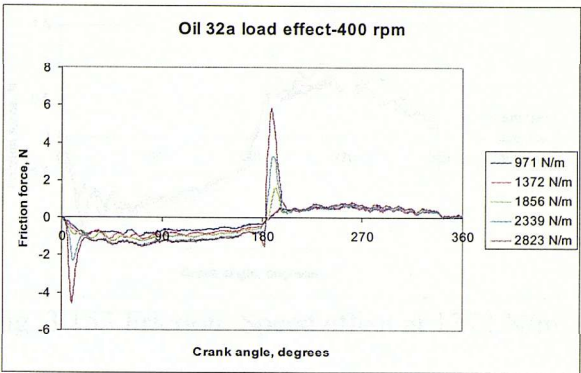


Fig. 3.150 Friction: Load effect at 400 rpm

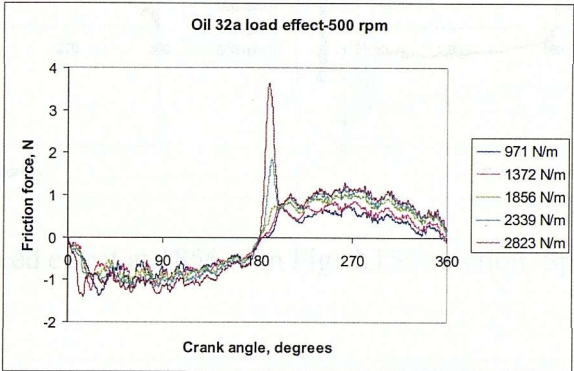


Fig. 3.151 Friction: Load effect at 500 rpm

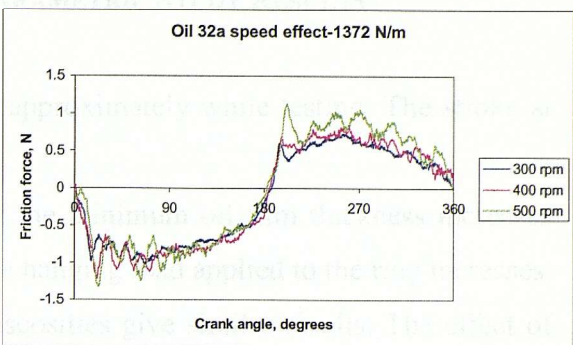
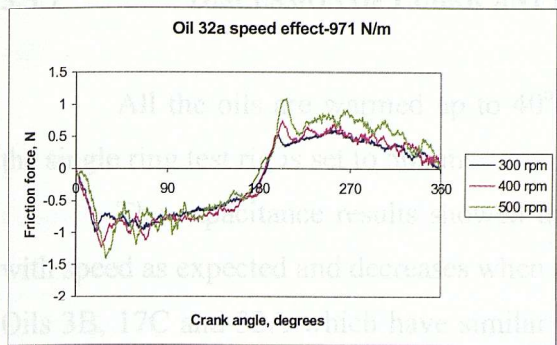


Fig. 3.152 Friction: Speed effect at 971 N/m      Fig. 3.153 Friction: Speed effect at 1372 N/m

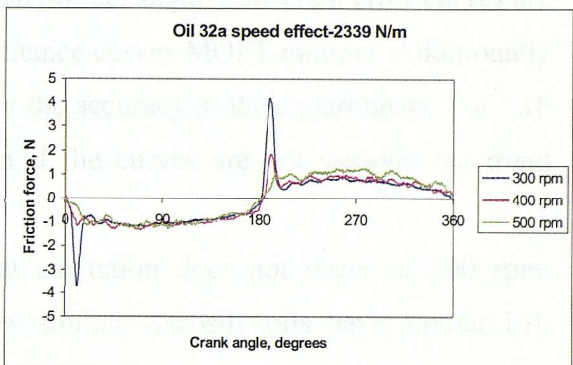
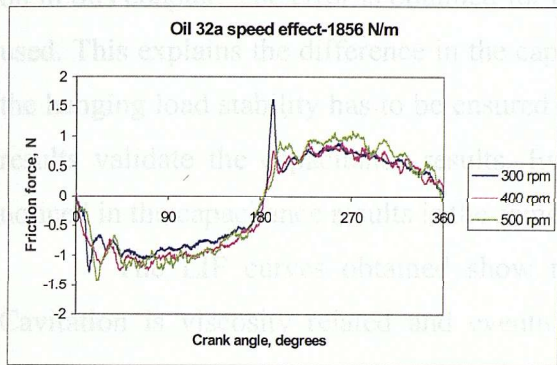


Fig. 3.154 Friction: Speed effect at 1856 N/m      Fig. 3.155 Friction: Speed effect at 2339 N/m

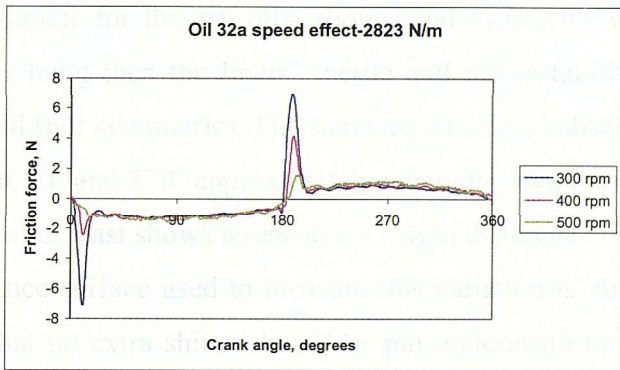


Fig. 3.156 Friction: Speed effect at 2823 N/m



### 3.5.7 DISCUSSION OF LUBRICANT PARAMETRIC STUDY RESULTS

All the oils are warmed up to 40°C approximately while testing. The stroke at the single ring test rig is set to 50mm.

The capacitance results showed that the minimum oil film thickness increases with speed as expected and decreases when the hanging load applied to the ring increases. Oils 3B, 17C and 32A which have similar viscosities give similar results. The effect of viscosity and viscosity index on oil film thickness is studied in a separate paragraph later on in this chapter. The error is obtained for each oil, i.e. slightly different error curves are used. This explains the difference in the capacitance curves MOFT minima. Additionally the hanging load stability has to be ensured for the accuracy of the experiments. The LIF results validate the capacitance results. Even if the curves are not smooth, the trend noticed in the capacitance results is the same.

The LIF curves obtained show that cavitation does not occur at 300 rpm. Cavitation is viscosity related and eventually similar viscosity oils have similar LIF curves in terms of cavitation. At 400 and 500 rpm all six oils have the same cavitation pattern (Figures 3.16, 3.18, 3.40, 3.42, 3.64, 3.66, 3.88, 3.90, 3.112, 3.114, 3.136, 3.138).

Increasing load provides good stability for the test rig results. Friction results, which again are very similar for the six oils, show good symmetry when the load is increasing. If no load is hung then the liners' inertia and the instability of the system gives an oscillating signal (not symmetric). The same asymmetry is observed between the two capacitance peaks at 90° and 270° approximately where the two maxima appear. The liner itself is quite level or at least shows a repeatable height difference between TDC and BDC because the reference surface used to measure this variation is not completely flat either. It was decided that no extra shims should be put underneath the ring to make it more level. The capacitance signal peaks (as shown in all capacitance curves) get more even when load is applied. At 971 N/m and 1372 N/m though, it is quite obvious that the 270° peak is slightly bigger than the one at 90° for all the capacitance MOFT curves obtained. A possible explanation is that the pivot axis of the liner holder is located closer to BDC, thus different load is applied from TDC to BDC and vice versa. Moreover, the asymmetric ring profile is causing the peak variation between upstroke and downstroke

along with the instability of the system at low loads. The difference in load has been calculated by Ostovar (1996), equation (3.1).

Friction results indicate that friction is maximum close to the dead centres. The peaks noticed in the friction curves are due to the high loads and they move closer to the dead centres with increasing load. The higher the load, the greater the contact and, thus, the friction. For lower loads, the cavitation delay is greater resulting in higher hydrodynamic friction peaks before cavitation occurs. The peaks occurring at high load are due to asperity interaction as the surfaces are forced together and establish contact and the lubrication between the two surfaces becomes boundary. The increase in the friction force mid-stroke measurements noticed for all the oils tested is due to the increase of the cavitating region between the piston-ring specimen and the liner. The bigger the cavitating region the smaller the contribution of the oil film to the ring's load carrying capacity.

Oils 4C and 1A which have different viscosities give different oil film thicknesses. Oil 1A which is more viscous gives thicker oil films than the other oils (compare oil 1A, Figures 3.61, 3.63, 3.65 with oil 7B Figures 3.13, 3.15, 3.17, oil 4C Figures 3.37, 3.39, 3.41, oil 17C Figures 3.85, 3.87, 3.89, oil 3B Figures 3.109, 3.111, 3.113 and oil 32A Figures 3.133, 3.135, 3.137). LIF results for oil 4C which is less viscous than 1A shows some similarity with the other oils of the same grade. No cavitation is observed at 300 rpm (Figure 3.38) but there is cavitation observed in the oil film at 400 (Figure 3.40) and 500 rpm (Figure 3.42). The same applies for oil 3B LIF results (Figures 3.110, 3.112 and 3.114).

The more viscous oil 1A at 300 rpm presents a unique cavitation pattern (Figure 3.62). At 90°, the LIF signal is the largest by far but cavitation is still not evident as in Figure 3.62. Friction results show similar trends. When all the friction Figures are compared the following trends are noticed: the lower the speed the higher the friction peaks while smoother curves are obtained with increasing load. The friction peaks appear at similar crank angles although the absolute value of the less viscous oil is greater than the value of the thinner oil (4C and 1A oils, Figures 3.53, 3.54, 3.55, 3.77, 3.78, 3.79). Additionally, at the lowest speed (300 rpm) the peaks of the friction curves appear to be more symmetric than at higher rpm (compare Figures 3.29, 3.53, 3.77, 3.101, 3.125,

3.149-300 rpm with 3.30, 3.31, 3.54, 3.55, 3.78, 3.79, 3.102, 3.103, 3.126, 3.127, 3.150, 3.151).

As the speed increases, the cavitation boundary (Fig. 3.14 a) moves further away from the centre of the ring. Pyke (2000) has extensively investigated cavitation over the entire stroke and has concluded that cavitation is not detected under the ring until mid-stroke. The whole mechanism is subject to uncertainties that will become clearer in the next chapter. Filling of the outlet region at the dead centres does not occur and this leads to starvation of the inlet boundary after ring reversal.

Oil 7B is tested in order to compare previous results taken by Pyke (2000). The capacitance results show a clearer trend of the minimum oil film thickness development throughout the stroke because of the new probe positioning.

The friction results are obtained without any changes. As far as the LIF is concerned, since it is calibrated from the capacitance signal, we expect to have different results from the beginning. The problem that has arisen, though, with the new series of experiments is that the LIF curves are not smooth as they used to be. A possible explanation is the removal and reinstallation of the liner to fit the flat plate for the capacitance probe but the most convincing is that the laser instrument used needs cleaning in order to produce the same power output as it did before the experiments were performed. The magnitude of the signal is five times smaller than that Pyke (2000) obtained with the same data acquisition system. Apart from the above mentioned reason, it should be noted that 50 cycles are averaged in order to take the mean value. In Figure 3.14 b the consecutive cycles are averaged and for the case of oil 7B 300 rpm – 971 N/m load the error bars for every averaged LIF value are shown.

Another point of interest is how the minima MOFT of the capacitance results (in all cases and for all oils) are compared to the surface roughness tests obtained by Pyke (2000). The measurements of ring and liner surface roughness (for the liner with the optical fibre) can be seen in the following table:

|               | R <sub>q</sub> (μm) |       | R <sub>a</sub> (μm) |       |
|---------------|---------------------|-------|---------------------|-------|
| Liner         | 0.213               | 0.286 | 0.162               | 0.224 |
| Piston - Ring | 0.396               | 0.469 | 0.332               | 0.370 |

Table 3.2 Surface roughness parameters for piston-ring specimen and liner used for oils 007B, 003B, 017C, 004C, 001A and 032A.

The liner surface roughness as well as the piston-ring specimen measurements refer to the longitudinal direction. At the minima of the MOFT curves, there is a measurement of oil film thickness in the range of 0.3 to 0.9 μm for oils 007B, 003B, 017C, 004C, 001A and 032A. In this case, the piston ring is operating in a boundary lubrication mode. The piston ring specimen and the liner are no longer separated by a fluid film. Contact occurs and the friction force results from the shearing of asperities and the films on the surfaces. This effect has been described as mixed lubrication regime in which the load applied to the lubricated contact is partially supported by the fluid film and partially supported by asperity contact, hence the surface roughness results are presented in table 3.2. Mixed and boundary lubrication occurs close to the dead centres. The minimum MOFT as well as the friction peaks appearance does not coincide with the liner's mechanical reversal at 0° and 180° CA. In all the capacitance and friction test cases (Figures 3.13, 3.15, 3.17, 3.19, 3.21, 3.23, 3.25, 3.27, 3.29-3.36, 3.37, 3.39, 3.41, 3.43, 3.45, 3.47, 3.49, 3.51, 3.53-3.60, 3.61, 3.63, 3.65, 3.67, 3.69, 3.71, 3.73, 3.75, 3.77-3.84, 3.109, 3.111, 3.113, 3.115, 3.117, 3.119, 3.121, 3.123, 3.125-3.132, 3.133, 3.135, 3.137, 3.139, 3.141, 3.143, 3.145, 3.147, 3.149-3.158) minimum MOFT and friction peaks are obtained a few degrees of crank angle after the dead centres. The phase difference between the flow reversal and the liner's mechanical movement is attributed to the effect of oil flow entrainment. According to the mentioned phenomenon, there is a residual OFT at the dead centres which is reduced later on in the stroke; this residual is further squeezed.

Another possibility is the appearance of stick-slip lubrication between the two surfaces. It is a question if this applies in the case of the single ring test rig or not as what



should be taken into account is the slight speed that the liner has from its mechanical movement when the minimum MOFT appears on the capacitance curves.

Following the removal and re-installation of the lubrication laboratory

### 3.6 FURTHER MODIFICATIONS ON THE SINGLE-RING TEST RIG

following series of capacitance experiments, including the ones presented in Chapter 2

The improvement of the capacitance signal was significant with the first set of the parametric study that was accomplished. The rigidity of the test rig itself was not considered adequate. Thus, further modifications were accomplished in order to improve the rigidity of the test rig. Intense oscillations observed especially at high speed were going to be eliminated. These modifications included:

- Welding of steel brackets positioned diagonally between the lower steel structure of the test rig
- Minimising electric motor vibrations by manufacturing peripheral clamp with damping material that holds the part of the motor which is hanging freely under the test rig bench
- Bolting of the test rig on the floor without any damping material in between

After these modifications, the comparison in Figure 3.157 showed that the capacitance signal was further improved, therefore allowing more accurate acquisition at high liner speeds:

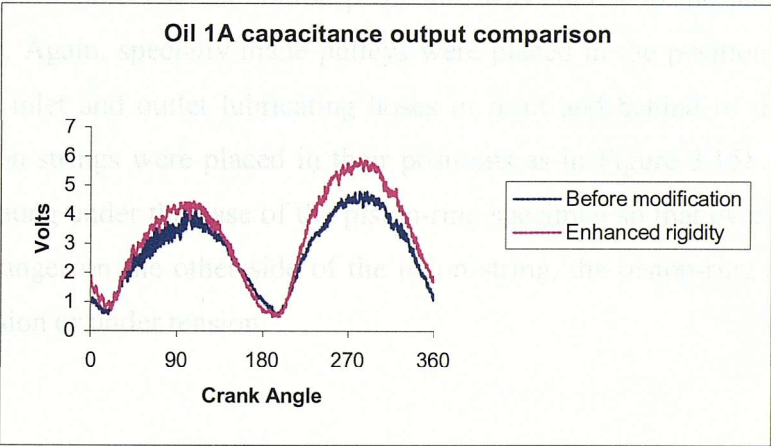


Figure 3.157 Comparison between the capacitance signals before and after the modifications



### **3.7 INSTRUMENT RE-CALIBRATION**

Following the removal and re-installation of the lubrication laboratory equipment, major components of the test-rig were re-calibrated and fine-tuned for the following series of capacitance experiments, including the ones presented in Chapter 2 about the LIF signal dynamic calibration.

#### **3.7.1 CAPACITANCE TRANSDUCER RECALIBRATION**

The Capacitec Unit was refurbished and recalibrated in order to make sure that the response of the instrument was not faulty under the circumstances specified in the User's Manual. The new settings for the Capacitec amplifier were according to the manufacturer's suggestion for optimum signal acquisition, that is offset and gain set in the middle of the adjusting knob range. Current is set to "high" to withstand the frequency response of the test rig measurement.

#### **3.7.2 FRICTION TRANSDUCER RECALIBRATION**

The PCB 208B friction transducer was recalibrated. The amplifier unit for the sensor was reconditioned and the basic amplifying circuit was changed for routine electrical safety reasons. The calibration procedure was similar to the one described in Ostovar (1996). Again, specially made pulleys were placed in the positions of the brass fittings for the inlet and outlet lubricating hoses in front and behind of the piston-ring specimen. Nylon strings were placed in their positions as in Figure 3.158. Each side of the string was hung under the base of the piston-ring specimen so that every time weight is put on the hanger on the other side of the nylon string, the piston-ring will be either under compression or under tension.

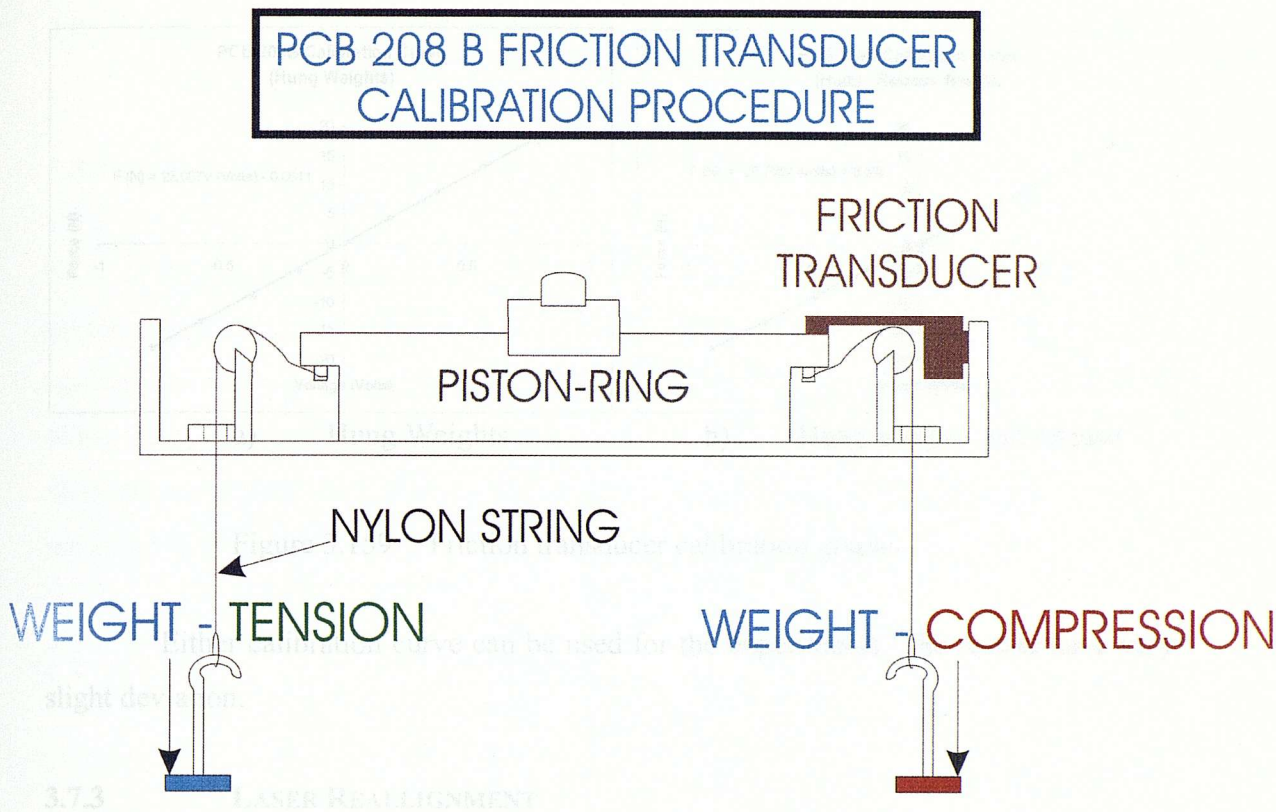


Figure 3.158 Friction Transducer calibration schematic

We used two calibration methods for comparison:

- a) Weight is hung on the lower side of the nylon string and at the time that is placed on the hanger we obtain the voltage reading on the multimeter simultaneously.
- b) Weight is hung on the lower side of the nylon string - this time we wait so that the reading on the multimeter goes to 0 (zero), which is the same reading when no weight is hung. This is due to the nature of the transducer itself (piezoelectric). Then the weight is removed with one movement and the reading is recorded. This time the reading will be representing the balancing force (reverse direction).

Both graphs are represented below (Figure 3.159a and b) and no significant differences in the calibration coefficients are obvious:

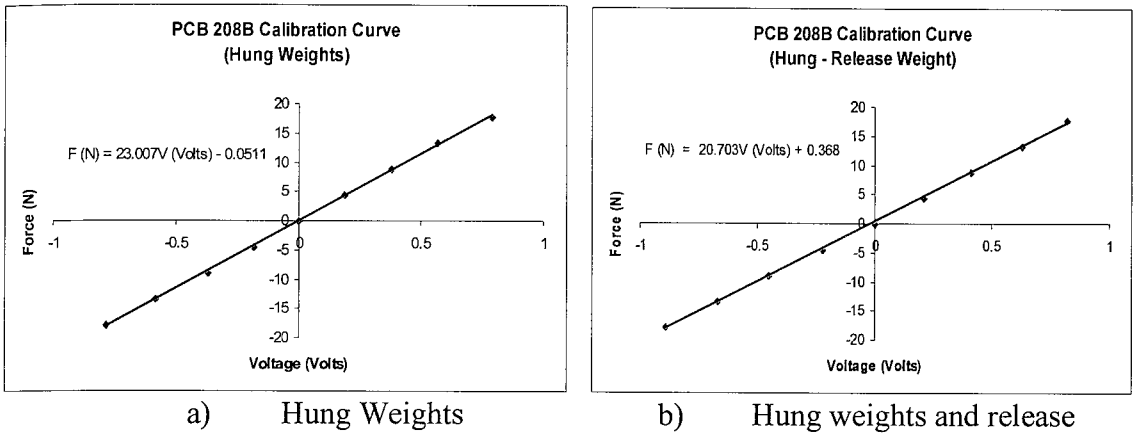


Figure 3.159 Friction transducer calibration graphs

Either calibration curve can be used for the experiments. The results have very slight deviation.

3.7.3 LASER REALIGNMENT

After moving and re-installing all the laboratory equipment, the LASER was re-aligned so that it will give its maximum power from the fibre coupling. The device was cleaned and the mirrors were re-adjusted to give the maximum of the blue light emitting efficiency. A new bare fibre adaptor was used as the fibre termination fitting to the fibre coupler. Changing the vertical position and the yaw angle of the LASER on the SPINDLER & HOYER optical table had a positive effect on maximising the power output from the fibre coupler towards the fibre probe. Afterwards, the fluorescence from the single ring test rig experiments could be observed with bare eye before it entered the photomultiplier tube, thus enabling less voltage applied on the tube and therefore less noise-background radiation amplification while testing. After the above modifications took place, new sets of experiments were acquired. This time the emphasis was on the effect of Viscosity Index on MOFT, friction comparison between the tested oils and temperature investigations.

3.8 CAPACITANCE EXPERIMENTS-TEMPERATURE AND LOAD EFFECT AT HIGH TEMPERATURES

3.8.1 VISCOSITY INDEX EFFECT ON MINIMUM OIL FILM THICKNESS

The viscosity index effect in oil film thickness is a factor that needs further investigation. The following graph shows the increase in oil film thickness as the viscosity increases. The results are for the same speed, 400 rpm, no external load (1159 N/m), 40°C and similar grade oils 0W-30 but with different additives that affect the viscosity index (VI) slightly. The oils tested are 032A, 003B, 004C and their properties are listed in the following table (Table 3.3):

| Blend Code             | 004C/02 | 032A  | 003B  |
|------------------------|---------|-------|-------|
| Grade                  | 0W-30   | 0W-30 | 0W-30 |
| HTHS (mPa s)           | 2.95    | 3.35  | 3.30  |
| V <sub>100</sub> (cSt) | 9.8     | 12.15 | 12.16 |
| V <sub>40</sub> (cSt)  | 52.9    | 65.97 | 68.93 |
| VI                     | 174     | 187   | 182   |

Table 3.3 Oil properties for the viscosity index effect experiment

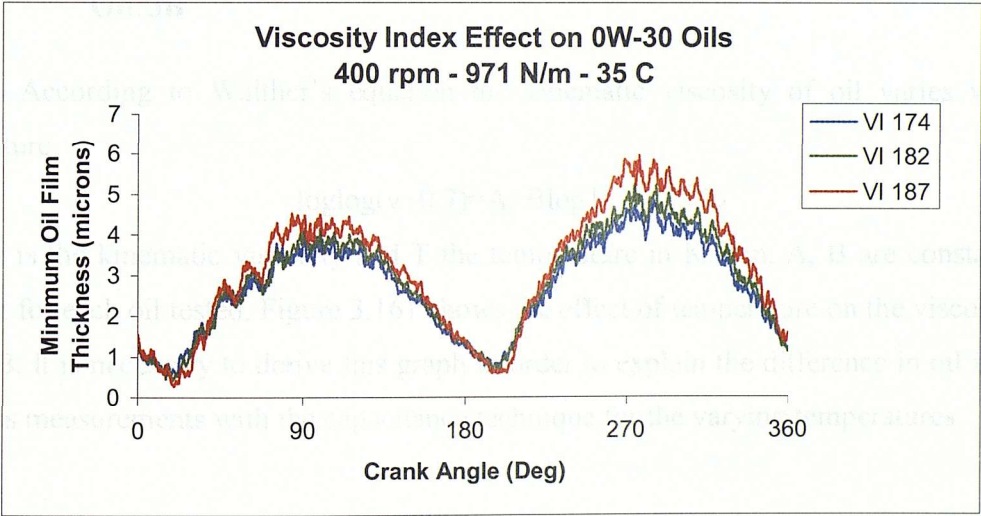


Figure 3.160 Viscosity index effect on oil film thickness for 0W-30 oils



The results in Figure 3.160 showed that the higher the viscosity index, the higher the MOFT for oils of the same grade.

3.8.2 TEMPERATURE INVESTIGATIONS

The temperature effect was studied in order to determine the oil behaviour under high temperature circumstances. The expected trend is that oil film decreases as oil temperature rises-similar behaviour as with the viscosity decrease. So, this study is going to be a viscosity related investigation on minimum oil film thickness measurements.

Table 3.4 shows the list of oils tested for the temperature investigations:

| Blend Code             | 003B  | 006E/02 | 005A/02 | 002A/02 |
|------------------------|-------|---------|---------|---------|
| Grade                  | 0W-30 | 0W-40   | 0W-20   | 10W-40  |
| HTHS (mPa s)           | 3.30  | 3.4     | 2.14    | 4.05    |
| V <sub>100</sub> (cSt) | 12.16 | 12.8    | 6.04    | 14.97   |
| V <sub>40</sub> (cSt)  | 68.93 | 66.8    | 31      | 97.8    |
| VI                     | 182   | 196     | 146     | 160     |

Table 3.4 Oils tested for temperature investigations

3.8.2.1 OIL 3B

According to Walther’s equation the kinematic viscosity of oil varies with temperature.

$$\log\log(v+0.7)=A+B\log T \qquad (3.3)$$

where v is the kinematic viscosity and T the temperature in Kelvin. A, B are constants different for each oil tested. Figure 3.161 shows the effect of temperature on the viscosity of oil 3B. It is necessary to derive this graph in order to explain the difference in oil film thickness measurements with the capacitance technique for the varying temperatures.



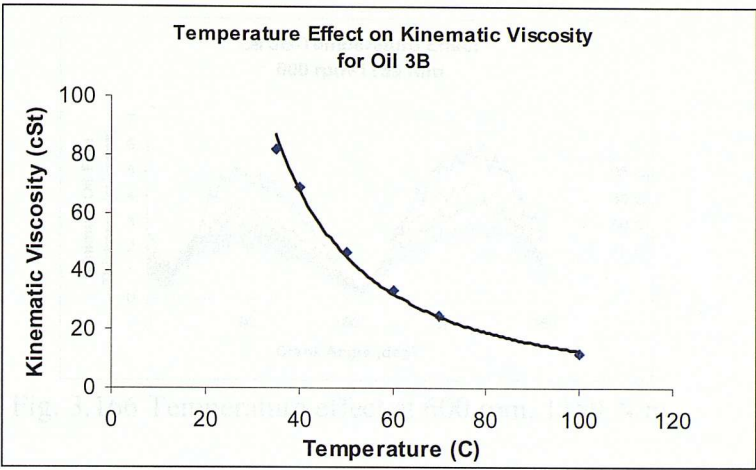


Figure 3.161 Temperature effect on kinematic viscosity, oil 3B

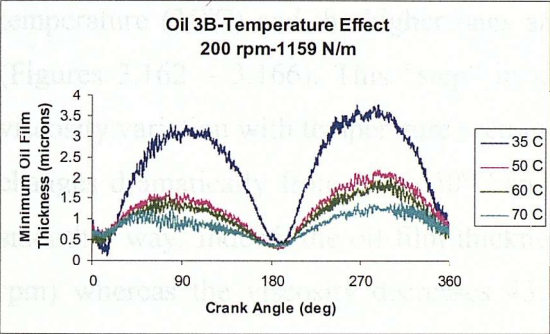


Fig. 3.162 Temperature effect at 200 rpm,  
1159 N/m

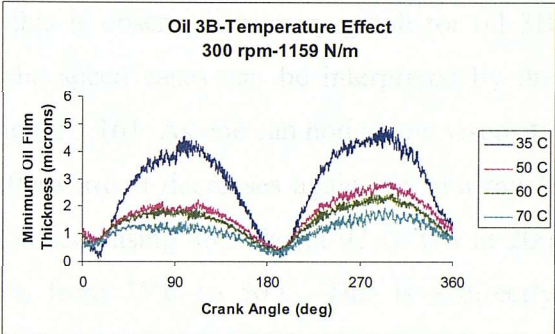


Fig. 3.163 Temperature effect at 300 rpm,  
1159 N/m

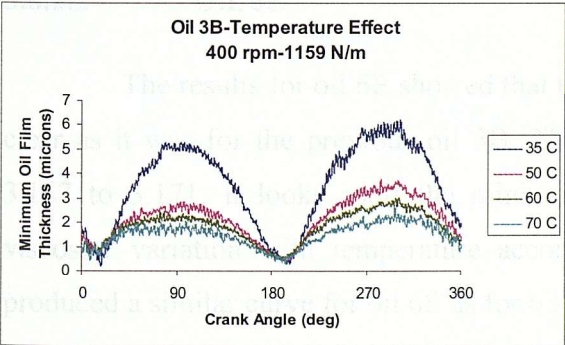


Fig. 3.164 Temperature effect at 400 rpm,  
1159 N/m

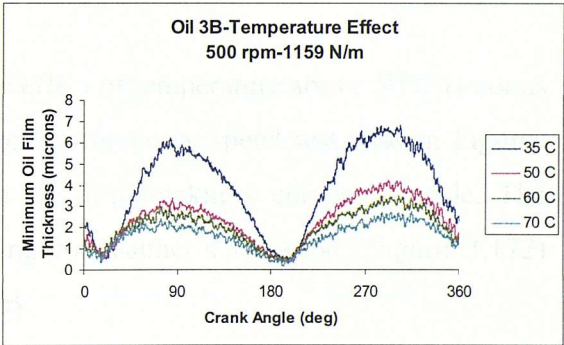


Fig. 3.165 Temperature effect at 500 rpm,  
1159 N/m

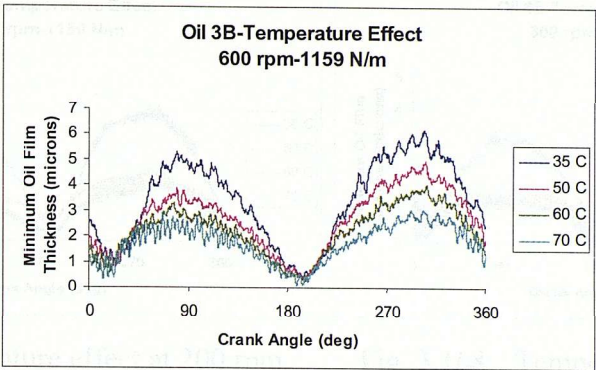


Fig. 3.166 Temperature effect at 600 rpm, 1159 N/m

From the above temperature results for oil 3B, the trend is that at higher temperatures the oil film thickness decreases. There is a big gap between the ambient oil temperature (35°C) and the higher ones and this is observed in every graph for oil 3B (Figures 3.162 – 3.166). This “step” in all the speed cases can be interpreted by the viscosity variation with temperature seen in Figure 3.161. As one can notice, the viscosity changes dramatically from 35 to 50°C and afterwards it decreases again but in a much smoother way. Indeed, the oil film thickness is decreasing 46.58% (at 92.16 CA at 200 rpm) whereas the viscosity decreases 43.35% from 35°C to 50°C. This is a directly comparable amount of decrease in the oil film thickness and eventually enables to draw the conclusion that oil film thickness is strongly viscosity dependent.

3.8.2.2 OIL 6E

The results for oil 6E showed that the effect of temperature above 50°C is not as clear as it was for the previous oil 3B. Therefore for every speed test case in Figures 3.167 to 3.171, it looks as if the minimum oil film thickness curves coincide. The viscosity variation with temperature according to Walther’s equation (Figure 3.172) produced a similar curve for oil 6E as for oil 3B.



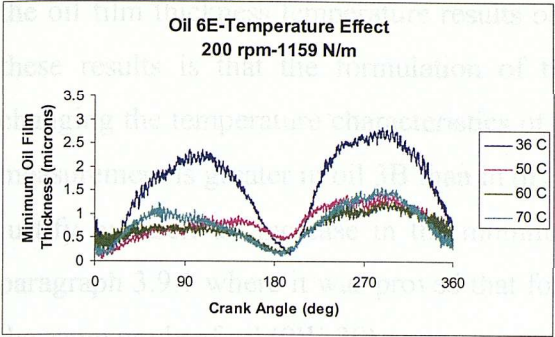


Fig. 3.167 Temperature effect at 200 rpm, 1159 N/m

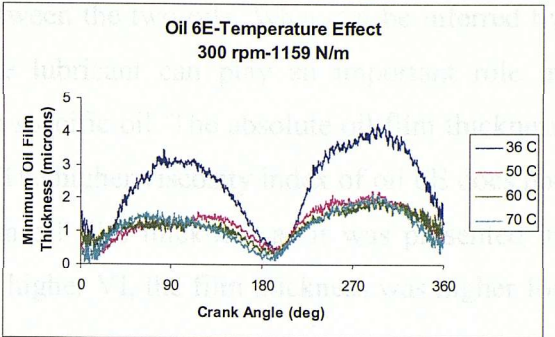


Fig. 3.168 Temperature effect at 300 rpm, 1159 N/m

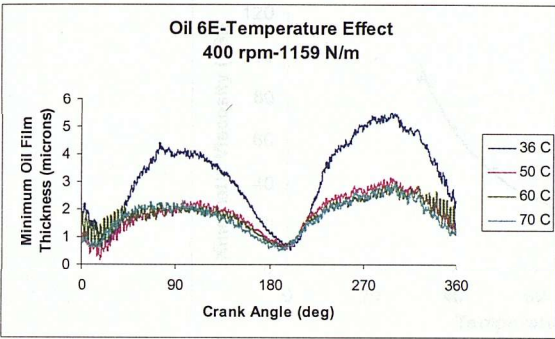


Fig. 3.169 Temperature effect at 400 rpm, 1159 N/m

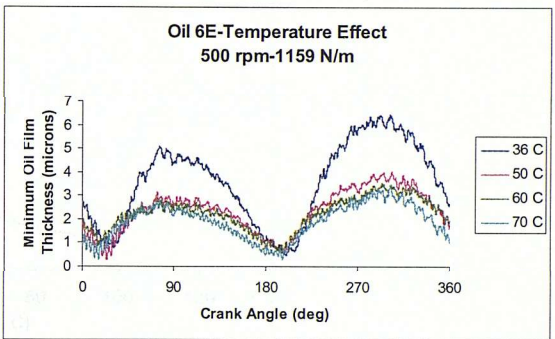


Fig. 3.170 Temperature effect at 500 rpm, 1159 N/m

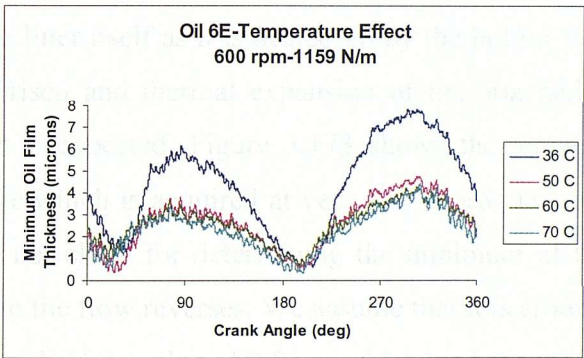


Fig. 3.171 Temperature effect at 600 rpm, 1159 N/m

In absolute numbers when the viscosities of the two oils (3B and 6E) are compared with rising temperature, it increases for oil 6E by 1% at 50°C, 2.6% at 60°C and 4.6% at 70°C. This slight increase by itself is not capable to justify the difference in

the oil film thickness temperature results between the two oils. What can be inferred by these results is that the formulation of the lubricant can play an important role in changing the temperature characteristics of a specific oil. The absolute oil film thickness measurement is greater in oil 3B than in 6E. The higher viscosity index of oil 6E does not justify by itself an increase in the minimum oil film thickness as it was presented in paragraph 3.9.1 where it was proved that for higher VI, the film thickness was higher for the same grade of oil (0W-30).

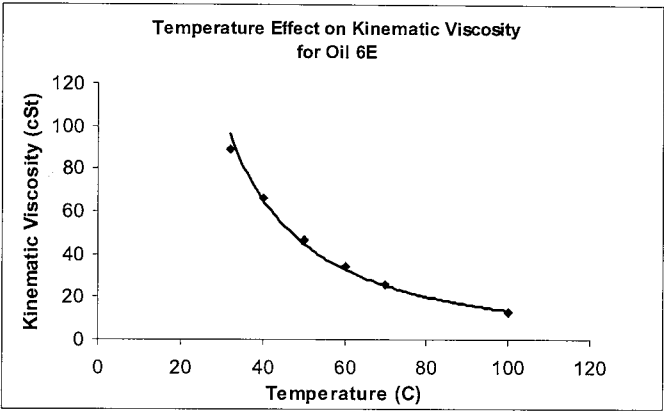


Figure 3.172 Temperature effect on kinematic viscosity, oil 6E

The experimental results of the temperature effect are affected by the temperature drift of the liner itself as it is heated up by the hot oil flow. The temperature of the liner itself has risen and thermal expansion of the attached plate on top of the capacitance transducer is expected. Figure 3.173 shows the temperature effect on the offset of the error curve which is acquired at very low speed. The error variation due to temperature drift does not allow for determining the minimum of the MOFT curve, i.e what is the MOFT when the flow reverses. We assume that it is around 0.3 or 0.3 microns in average, less than the absolute value of 0.5  $\mu\text{m}$  which we have measured for the MOFT measurement during the ambient temperature tests.

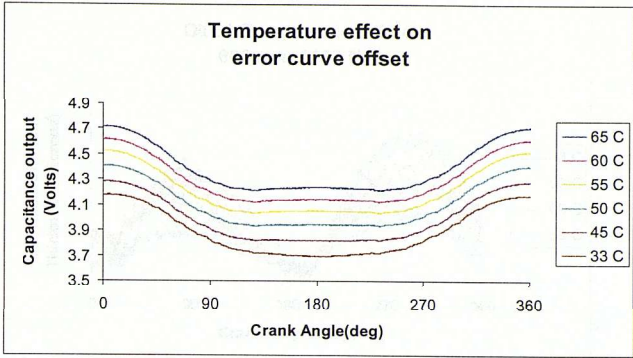


Figure 3.173 Temperature effect on error curve offset

3.8.2.3 OIL 5A

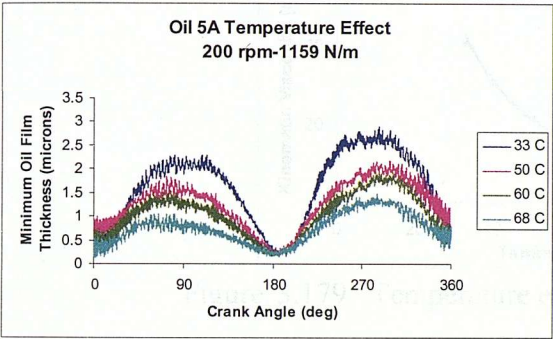


Fig. 3.174 Temperature effect at 200 rpm, 1159 N/m

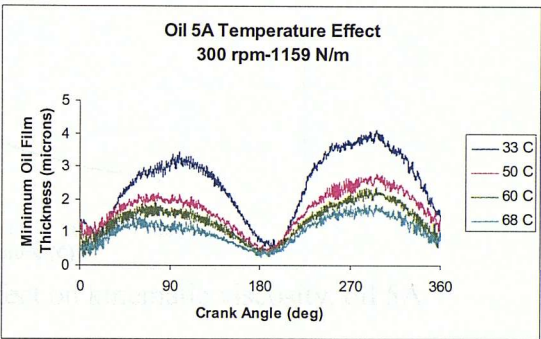


Fig. 3.175 Temperature effect at 300 rpm, 1159 N/m

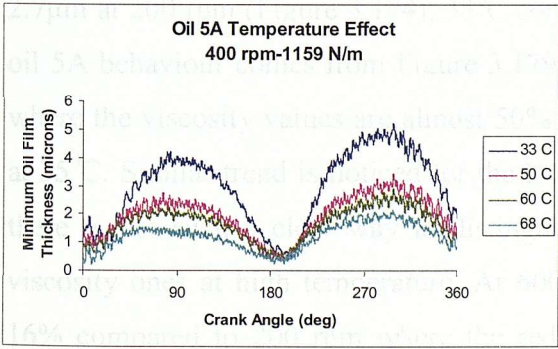


Fig. 3.176 Temperature effect at 400 rpm, 1159 N/m

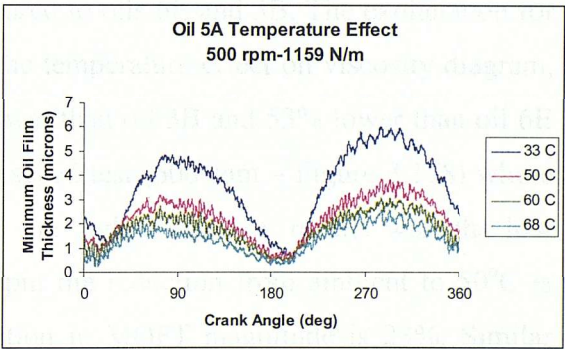


Fig. 3.177 Temperature effect at 500 rpm, 1159 N/m



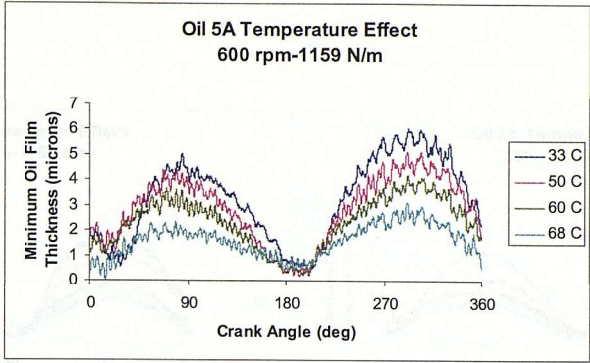


Fig. 3.178 Temperature effect at 600 rpm, 1159 N/m

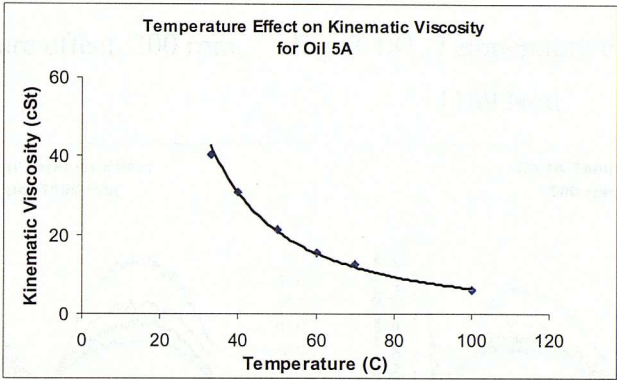


Figure 3.179 Temperature effect on kinematic viscosity, oil 5A

For oil 5A (Figures 3.174-3.178) a similar trend to the previous oils (6E and 3B) is noticed. This time though, at very low speeds, MOFT has a low absolute value, max  $2.7\mu\text{m}$  at 200 rpm (Figure 3.174),  $33^{\circ}\text{C}$  compared to oils 6E and 3B. The explanation for oil 5A behaviour comes from Figure 3.179, the temperature effect on viscosity diagram, where the viscosity values are almost 50% lower than oil 3B and 53% lower than oil 6E at  $35^{\circ}\text{C}$ . Similar trend is noticed for the high speed test (600 rpm – Figure 3.178) where there is not such a clear way to distinguish the higher viscosity results with the low viscosity ones at high temperature. At 600 rpm the reduction from ambient to  $50^{\circ}\text{C}$  is 16% compared to 200 rpm where the reduction in MOFT magnitude is 23%. Similar results were presented for oil 3B. At high speed (600 rpm – Figure 3.166) the temperature effect does not have such a strong effect as at low speeds, the percentages this time are at 600 rpm 34% and 53% reduction respectively at 200 rpm (Figure 3.162). Oil 6E on the other hand shows greater stability in this aspect. MOFT reduces at 600 rpm 47% (Figure 3.171) which is comparable to 55% reduction at 200 rpm (Figure 3.167).

3.8.2.4 OIL 2A

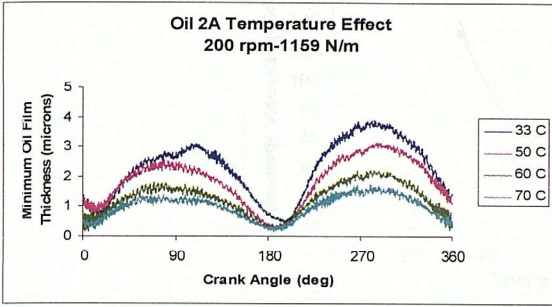


Fig. 3.180 Temperature effect, 200 rpm, 1159 N/m

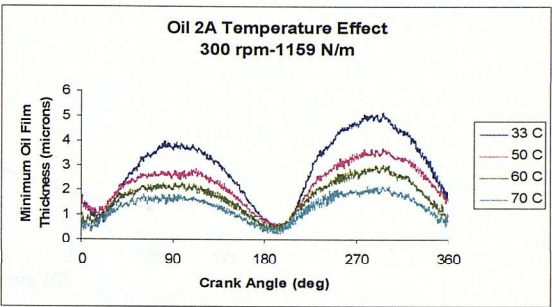


Fig. 3.181 Temperature effect, 300 rpm, 1159 N/m

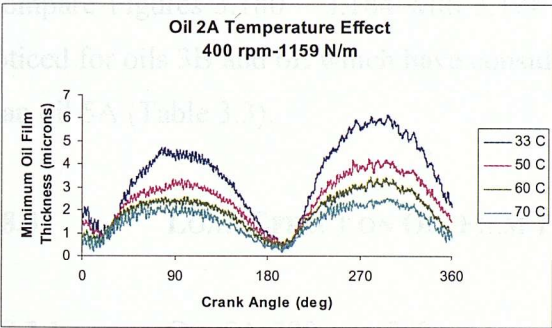


Fig 3.182 Temperature effect, 400 rpm, 1159 N/m

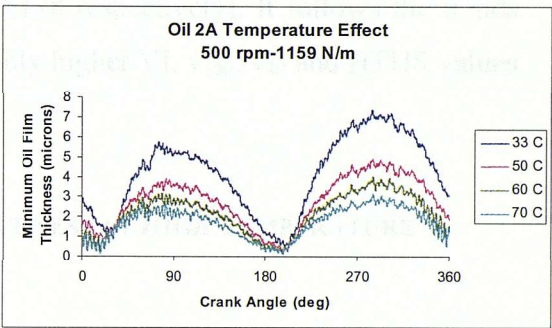


Fig. 3.183 Temperature effect, 500 rpm, 1159 N/m

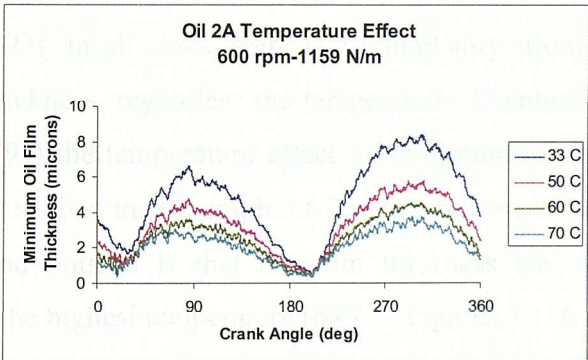


Fig. 3.184 Temperature effect, 600 rpm, 1159 N/m

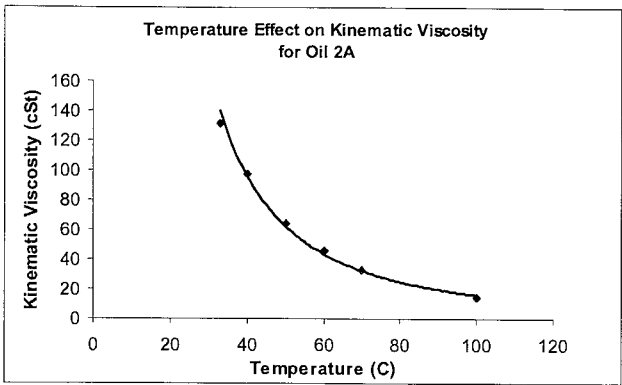


Fig. 3.185      Temperature effect on kinematic viscosity

The behaviour of this oil is not similar to 5A which was tested in 3.8.2.3 (compare Figures 3.180 – 3.184 with 3.174 – 3.178 respectively). It follows the trends noticed for oils 3B and 6E which have considerably higher VI,  $v_{100}$ ,  $v_{40}$  and HTHS values than oil 5A (Table 3.3).

**3.8.3                    LOAD EFFECT ON OIL FILM THICKNESS AT HIGH TEMPERATURE**

**3.8.3.1                OIL 5A-300 AND 500 RPM**

Oil 5A was tested for the effect of load on oil film thickness at high temperature. The results show the trend at 300 (Figures 3.186 – 3.189) and 500 rpm (Figures 3.190 – 3.193). In all cases, there is a remarkably strong effect of load on the minimum oil film thickness, regardless the temperature. Eventually, as load gets higher (Figures 3.189 – 3.193) the temperature effect on minimum oil film thickness becomes less than in the cases studied in paragraph 3.8.2 which represent the temperature effect at 1159 N/m. The trend noticed is that the film thickness has a certain value which decreases slightly at the highest temperature (68°C – Figures 3.186 – 3.193).



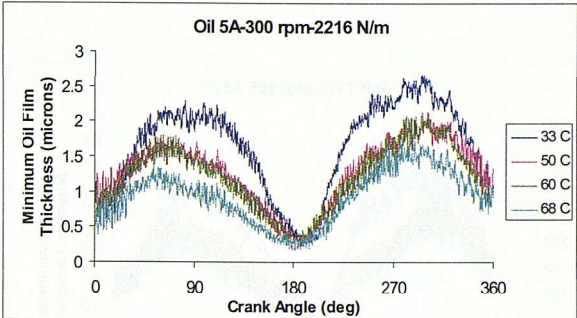
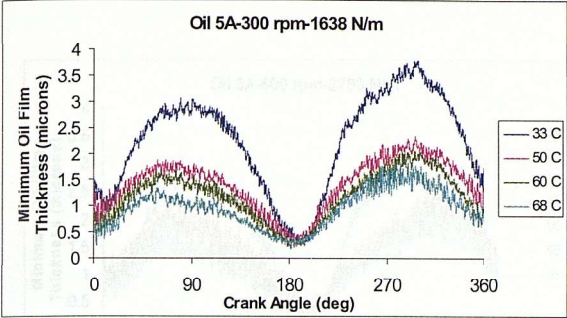


Fig. 3.186 Load effect at 300 rpm, 1638 N/m Fig. 3.187 Load effect at 300 rpm, 2216 N/m

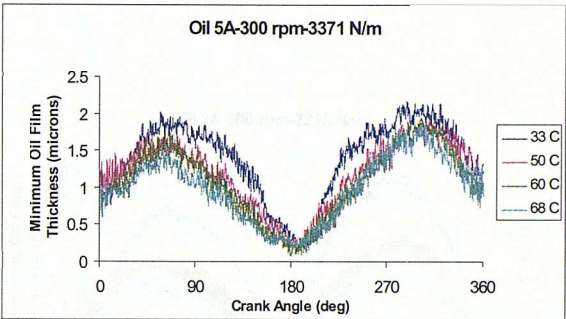
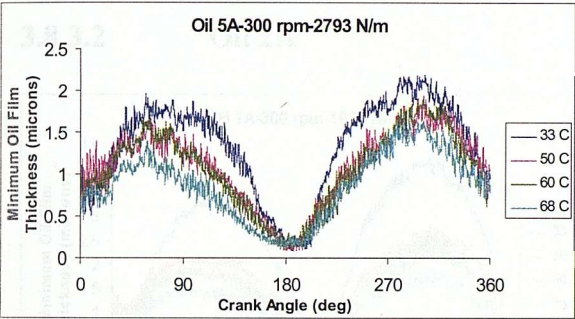


Fig. 3.188 Load effect at 300 rpm, 2793 N/m Fig. 3.189 Load effect at 300 rpm, 3371 N/m

The MOFT is smaller at BDC than at TDC. According to the friction measurements at higher loads, the friction is higher at BDC, which is forcing us to assume that the load effect on BDC side of the test rig is much more pronounced than at TDC.

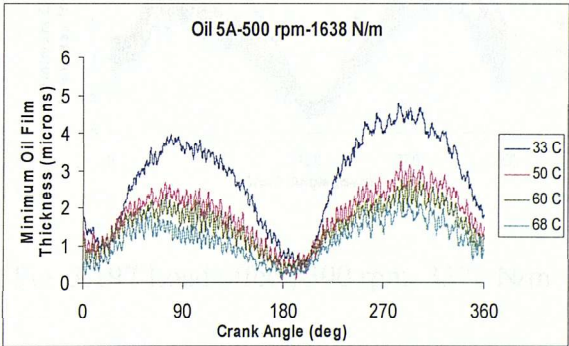
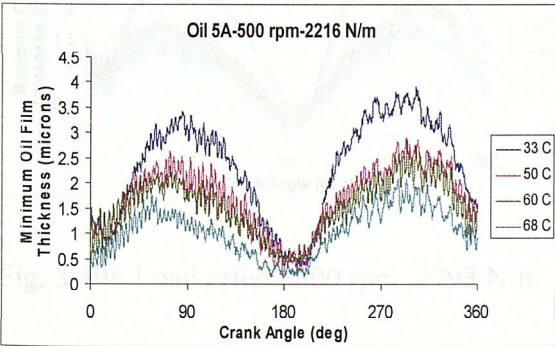


Fig. 3.190 Load effect at 500 rpm, 1638 N/m Fig. 3.191 Load effect at 500 rpm, 2216 N/m

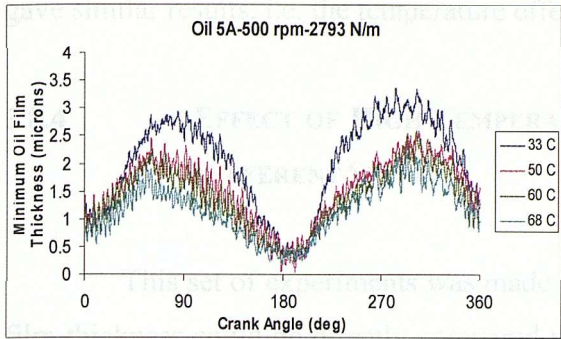


Fig. 3.192 Load effect at 500 rpm, 2793 N/m

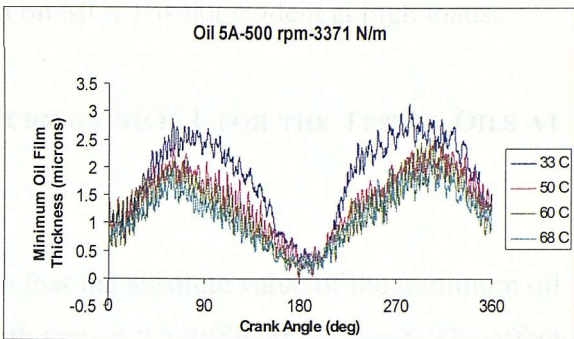


Fig. 3.193 Load effect at 500 rpm, 3371 N/m

3.8.3.2 Oil 2A

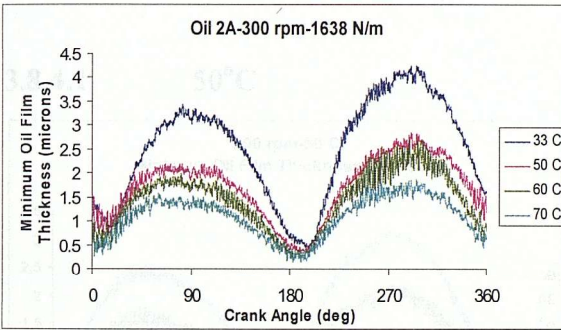


Fig. 3.194 Load effect, 300 rpm, 1638 N/m

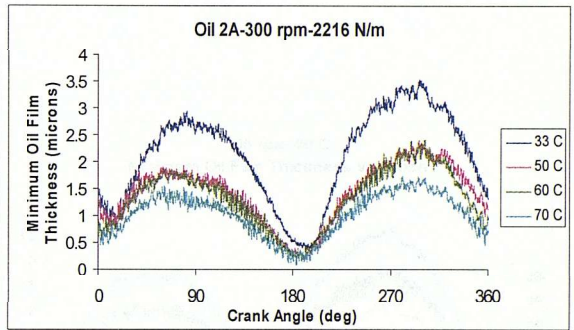


Fig. 3.195 Load effect, 300 rpm, 2216 N/m

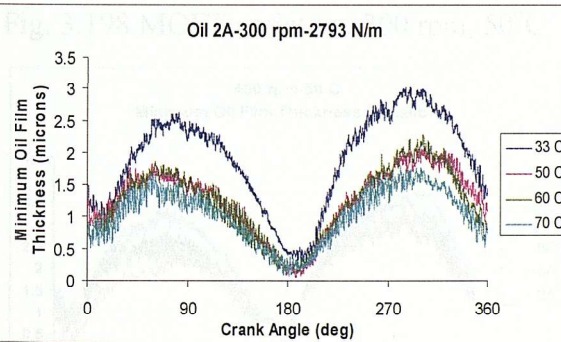


Fig. 3.196 Load effect, 300 rpm, 2793 N/m

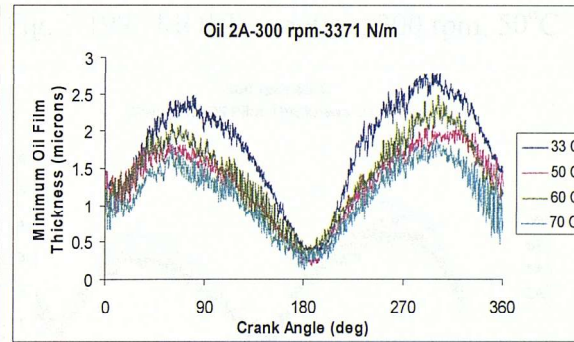


Fig. 3.197 Load effect, 300 rpm, 3371 N/m

Again, the above graphs (Figures 3.194 – 3.196) show how load is affecting the minimum oil film thickness for oil 2A as temperature varies. Oil 2A which has greater  $v_{40}$



and  $v_{100}$  than oil 5A for which the same comparison has been done in paragraph 3.8.3.1 gave similar results, i.e. the temperature effect on MOFT is not evident at high loads.

3.8.4 EFFECT OF HIGH TEMPERATURE ON MOFT FOR THE TESTED OILS AT DIFFERENT SPEEDS

This set of experiments was made so that the absolute value of the minimum oil film thickness could be directly compared with that of the different oils used. The effect of high temperature is taken into account for these comparisons and graphs are presented for all the oils tested at high temperatures, at 60 and 70°C.

3.8.4.1 50°C

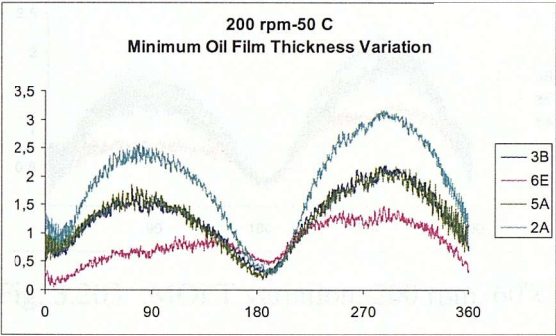


Fig. 3.198 MOFT variation, 200 rpm, 50°C

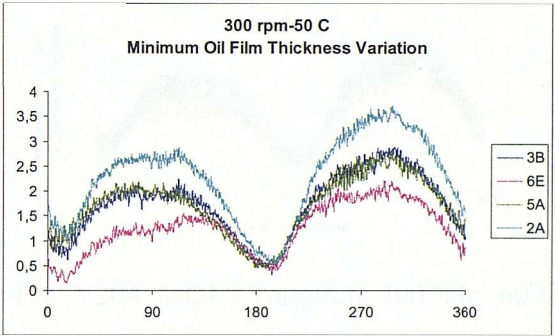


Fig. 3.199 MOFT variation, 300 rpm, 50°C

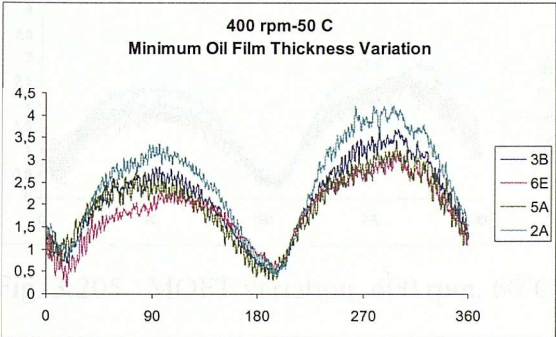


Fig. 3.200 MOFT variation, 400 rpm, 50°C

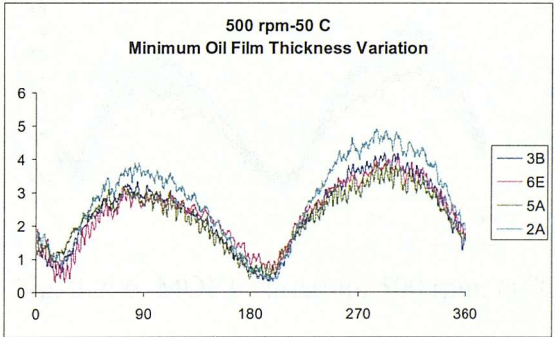


Fig. 3.201 MOFT variation, 500 rpm, 50°C

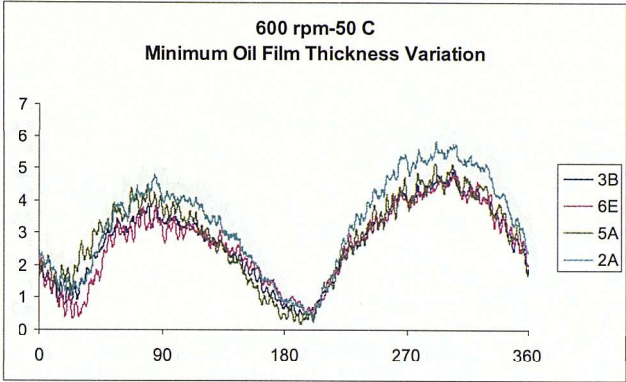


Fig. 3.202 MOFT variation, 600 rpm, 50°C

3.8.4.2 60°C

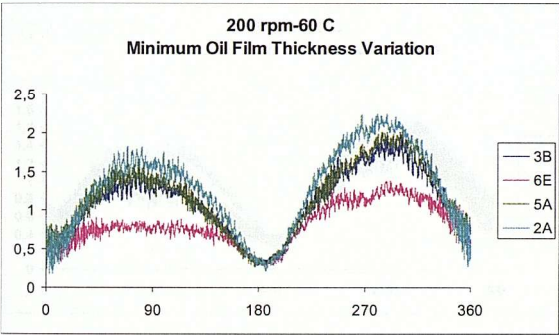


Fig. 3.203 MOFT variation, 200 rpm, 60°C

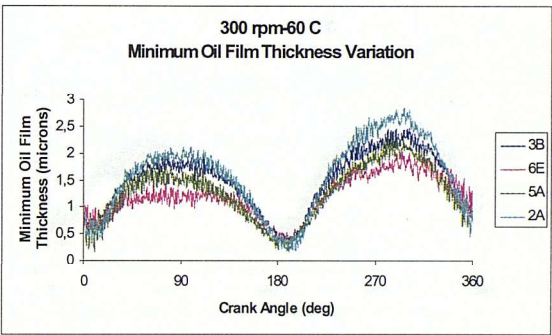


Fig. 3.204 MOFT variation, 300 rpm, 60°C

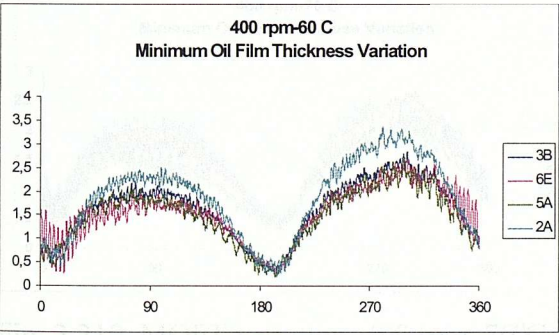


Fig. 3.205 MOFT variation, 400 rpm, 60°C

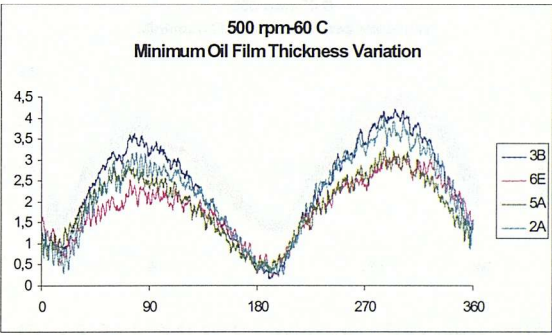


Fig. 3.206 MOFT variation, 500 rpm, 60°C



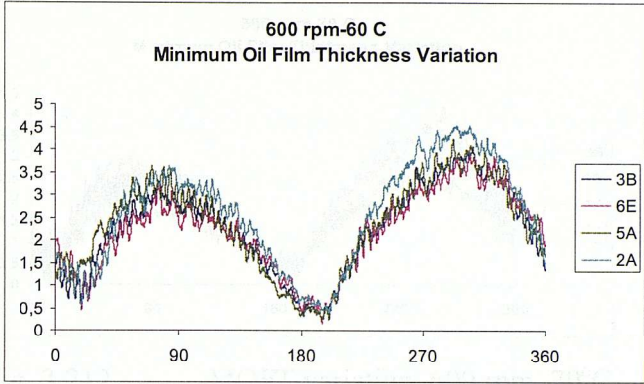


Fig. 3.207 MOFT variation, 600 rpm, 60°C

3.8.4.3 70°C

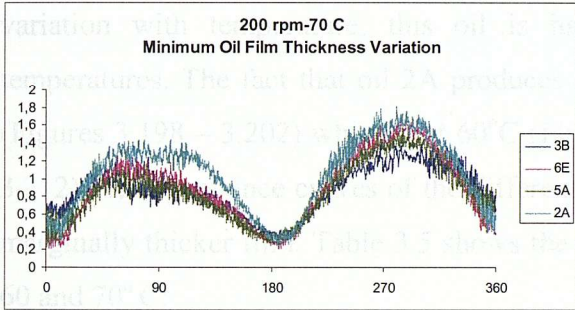


Fig. 3.208 MOFT variation, 200 rpm, 70°C

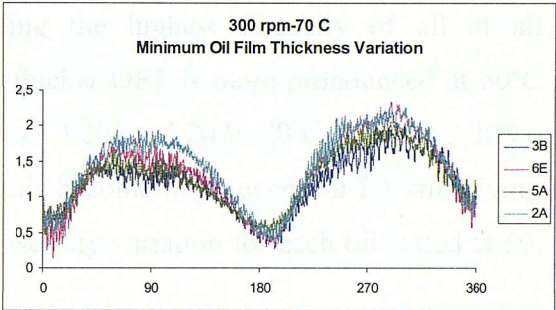


Fig. 3.209 MOFT variation, 300 rpm, 70°C

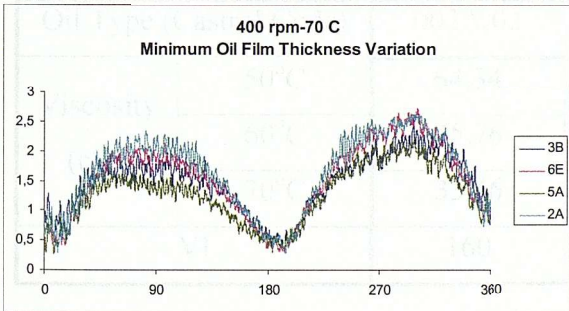


Fig. 3.210 MOFT variation, 400 rpm, 70°C

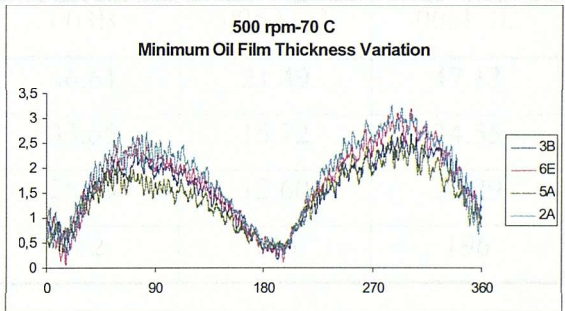


Fig. 3.211 MOFT variation, 500 rpm, 70°C

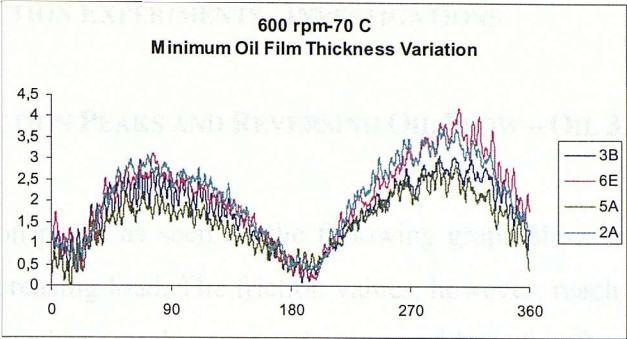


Fig. 3.212 MOFT variation, 600 rpm, 70°C

The majority of the testing showed that for the above series of experimental data (Figures 3.198 – 3.205 and 3.207 – 3.211), oil 2A had the thicker film as the temperature varies in comparison to the other oils. According to Walther’s equation for viscosity variation with temperature, this oil is having the highest viscosity of all at all temperatures. The fact that oil 2A produces a thicker OFT is more pronounced at 50°C (Figures 3.198 – 3.202) whereas at 60°C (Figures 3.203 – 3.207), 70°C (Figures 3.208 – 3.212) the capacitance curves of the different oils become more even; oil 2A still giving marginally thicker film. Table 3.5 shows the viscosity variation for each oil tested at 50, 60 and 70° C:

| Oil Type (Castrol Code) |      | 002A/02 | 003B  | 005A/02 | 006E/02 |
|-------------------------|------|---------|-------|---------|---------|
| Viscosity<br>(cSt)      | 50°C | 64.34   | 46.61 | 21.49   | 47.13   |
|                         | 60°C | 45.76   | 33.65 | 15.72   | 34.55   |
|                         | 70°C | 33.46   | 25.09 | 12.60   | 26.29   |
| VI                      |      | 160     | 182   | 146     | 196     |

Table 3.5 Viscosity properties for the temperature-load effect experiments and VI properties



3.9 FRICTION EXPERIMENTS - INVESTIGATIONS

3.9.1 FRICTION PEAKS AND REVERSING OIL FLOW – OIL 32A

The friction peaks as seen on the following graph show an increasing friction force value with increasing load. The friction values, however, reach their peaks after the dead centres and this happens because it became evident that flow reverses (and also cavitation is taking place) a few degrees of crank angle later in the stroke than at the dead centres. During flow reversal the flow decelerates and speed is getting towards zero resulting in the appearance of the minimum value of minimum oil film thickness (MOFT). At this point, the surface interaction between the piston ring specimen and the ring is becoming more pronounced, the lubrication becomes mixed (elastohydrodynamic and hydrodynamic) and boundary and therefore the friction reaches its peak.

Moreover, as can be noticed in the friction peaks region (Figure 3.213) for crank angles between 180°-220° the peaks move closer to the dead centre as the load increases.

This is in accordance to the visualisation experiments results, where the oil flow was reversing closer to the dead centres when the single ring test rig was subject to higher load.

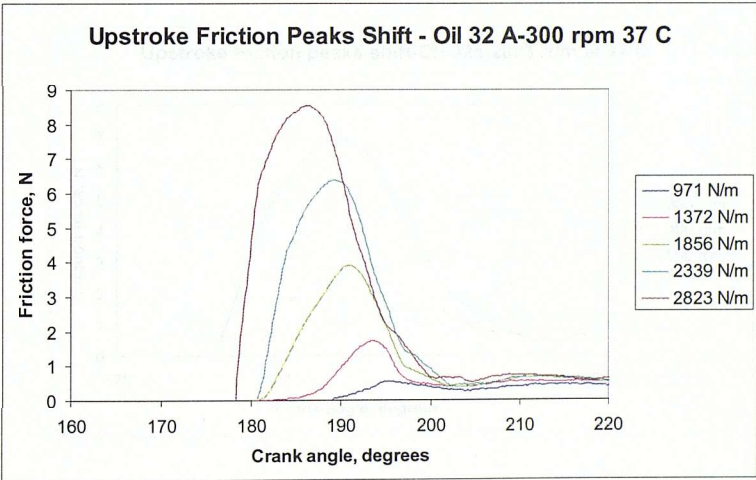


Figure 3.213 Upstroke friction peaks shift for oil 32A at 300 rpm, 37°C



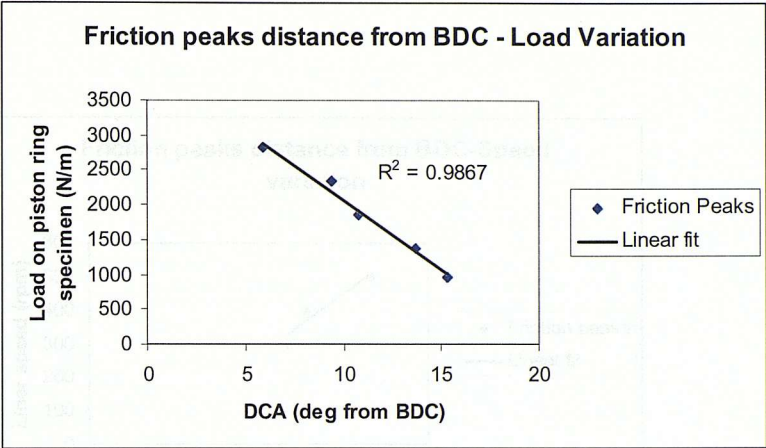


Figure 3.214 Friction peaks distance from BDC, variation with load - DCA is the Crank Angle where the friction peak appears minus  $180^\circ$

A similar trend is noticed for the friction peaks that are taken for oil 32A, 2823 N/m load when the speed is varied. The noticed trend, which is for higher speed the friction peaks are reversing later in the stroke, is in accordance, again, to the flow reversal visualisation experiments that are presented in the next chapter.

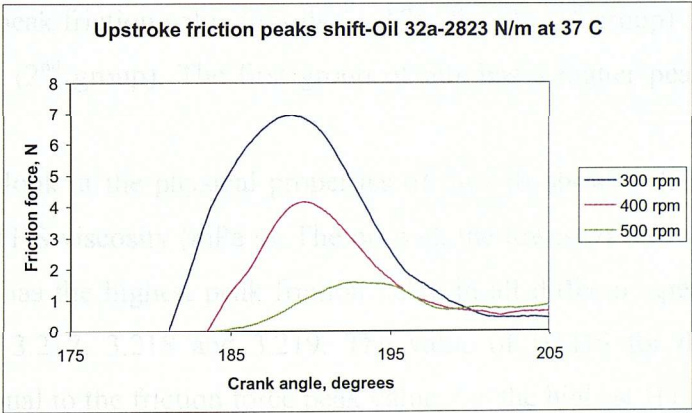


Figure 3.215 Upstroke friction peaks shift for oil 32A at 300 rpm,  $37^\circ\text{C}$

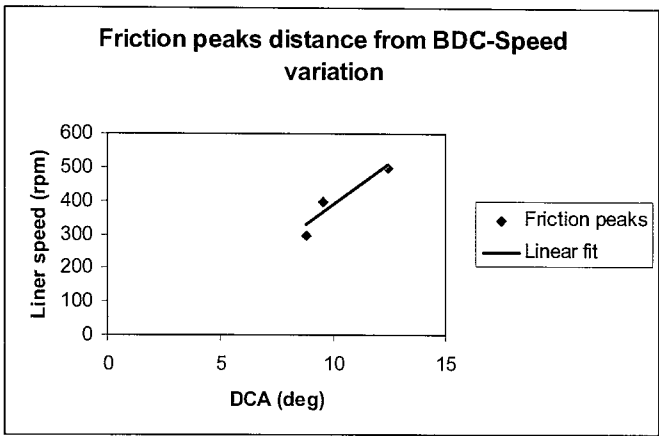


Figure 3.216 Friction peaks distance from BDC, variation with speed

**3.9.2 FRICTION FORCE COMPARISON BETWEEN ALL OILS TESTED – HTHS EFFECT AT AMBIENT TEMPERATURE**

The following figures show a friction force comparison between all oils that have been tested at 300 rpm, 400 rpm, 500 rpm for the highest load case, 2823 N/m. There is a similar peak friction value for oils 4c, 17c, 3b, 32a (1<sup>st</sup> group) and respectively for oils 7b and 1a (2<sup>nd</sup> group). The first group of oils has a higher peak friction force value.

A closer look at the physical properties of the oils showed that each group of oils has similar HTHS viscosity (mPa s). The oil with the lowest HTHS value, 2.95 mPa s, is oil 4c which has the highest peak friction value in all different speed test cases as shown in Figures 3.217, 3.218 and 3.219. The value of HTHS for the oil tested is inversely proportional to the friction force peak value, i.e. the highest HTHS gives lowest peak friction results.

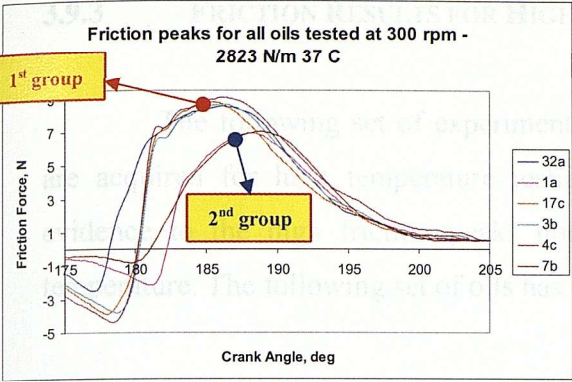


Figure 3.217 Friction peaks for all oils tested, 300 rpm, 37° C

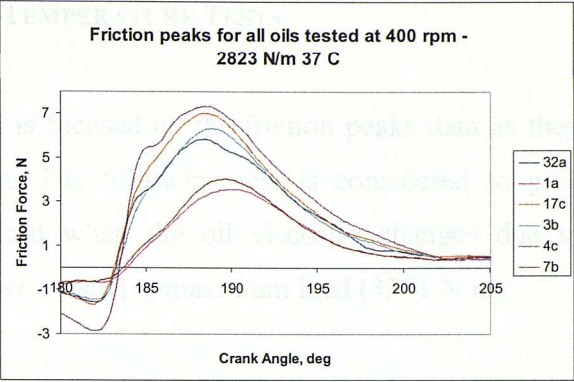


Figure 3.218 Friction peaks for all oils tested, 400 rpm, 37° C

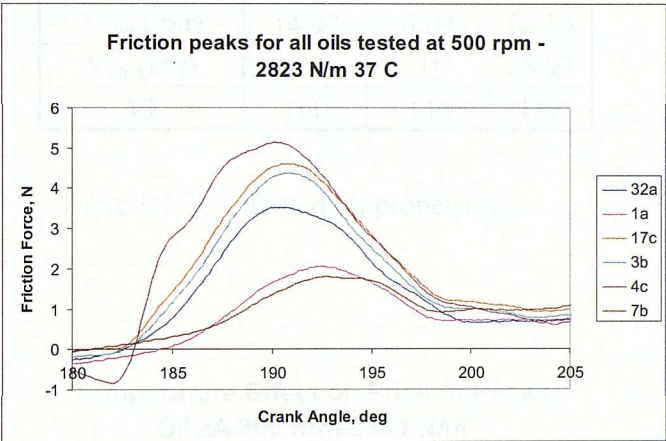


Figure 3.219 Friction peaks for all oils tested, 500 rpm, 37° C

| Oil Grade | HTHS Viscosity (mPa s) | Oil Group (Fig 3.217) |
|-----------|------------------------|-----------------------|
| 32a       | 3.35                   | 1 <sup>st</sup>       |
| 1a        | 3.60                   | 2 <sup>nd</sup>       |
| 17c       | 3.39                   | 1 <sup>st</sup>       |
| 3b        | 3.30                   | 1 <sup>st</sup>       |
| 4c        | 2.95                   | 2 <sup>nd</sup>       |
| 7b        | 3.64                   | 1 <sup>st</sup>       |

Table 3.6 HTHS viscosity of all the oils tested



3.9.3 FRICTION RESULTS FOR HIGH TEMPERATURE TESTS

The following set of experiments is focused on the friction peaks data as they are acquired for high temperature testing. The following test is considered to give evidence to the high friction peaks noticed when the oil viscosity changes due to temperature. The following set of oils has been tested at maximum load (3371 N/m):

| Blend Code             | 002A/02 | 005A/02 | 003B  |
|------------------------|---------|---------|-------|
| Grade                  | 10W-40  | 0W-20   | 0W-30 |
| HTHS (mPa s)           | 4.05    | 2.14    | 3.30  |
| V <sub>100</sub> (cSt) | 14.97   | 6.04    | 12.16 |
| V <sub>40</sub> (cSt)  | 97.8    | 31      | 68.93 |
| VI                     | 160     | 146     | 182   |

Table 3.7 Tested oil properties

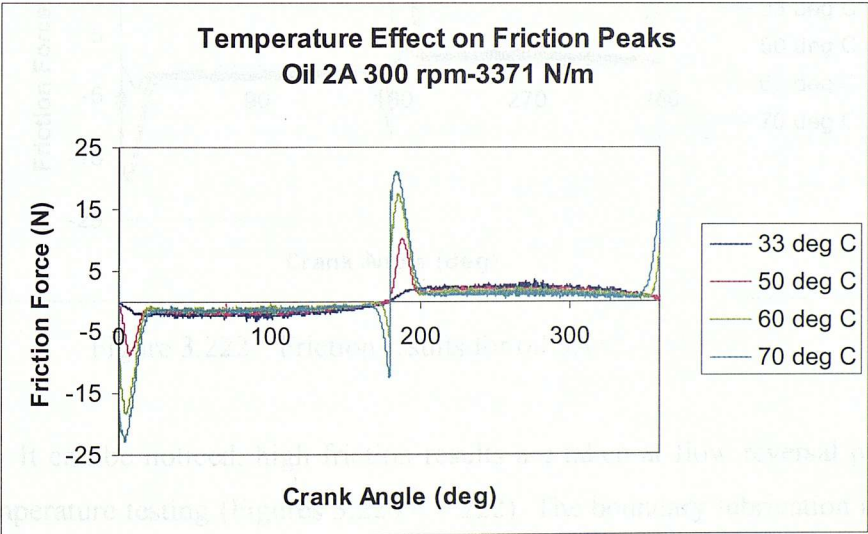


Figure 3.220 Friction results for oil 2A

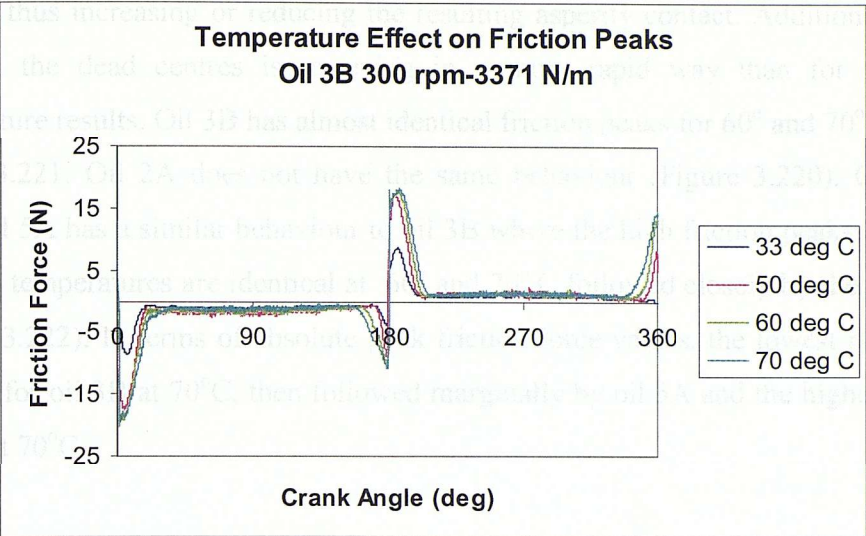


Figure 3.221 Friction results for oil 3B

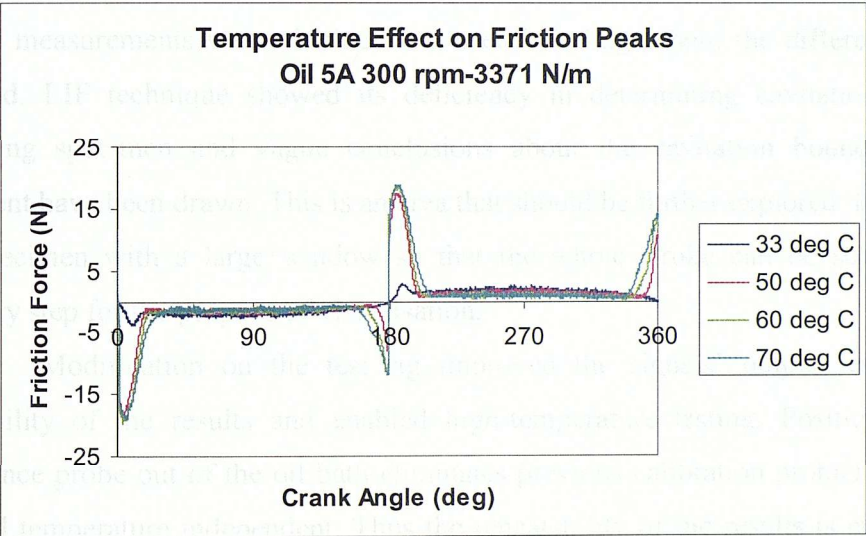


Figure 3.222 Friction results for oil 5A

It can be noticed, high friction results are taken at flow reversal points for the high temperature testing (Figures 3.220 – 3.222). The boundary lubrication at high loads and temperature is more intense. Friction force increases at high temperature when the interface between the liner surface and the piston ring is under boundary lubrication conditions. The formulation of the lubricant is an important factor because at higher temperatures the additives of the lubricant react in a different way with the surfaces in



contact, thus increasing or reducing the resulting asperity contact. Additionally, friction force at the dead centres is reversing in a more rapid way than for the ambient temperature results. Oil 3B has almost identical friction peaks for 60° and 70° C as seen in Figure 3.221. Oil 2A does not have the same behaviour (Figure 3.220). On the other hand, oil 5A has a similar behaviour to oil 3B where the high friction peaks observed for the high temperatures are identical at 60° and 70° C followed closely by the 50°C results (Figure 3.222). In terms of absolute peak friction force values, the lowest peak pressure appears for oil 3B at 70°C, then followed marginally by oil 5A and the highest one is for oil 2A at 70°C.

### 3.10 CONCLUSIONS

An extensive lubricant parametric study has been conducted. OFT, friction and LIF measurements were obtained and useful findings about the different oils were presented. LIF technique showed its deficiency in determining cavitation under the piston-ring specimen and vague conclusions about the cavitation boundary and its movement have been drawn. This is an area that should be further explored. Introducing a liner specimen with a large window so that the whole stroke can be scanned is the necessary step for the purpose of visualisation.

Modification on the test rig improved the signals' output, enhanced the repeatability of the results and enabled high-temperature testing. Positioning of the capacitance probe out of the oil bath eliminates previous calibration problems and is oil type and temperature independent. Thus the repeatability of the results is enhanced and eventually more oil parameters can be studied in a much safer way. Even if simplified, the results are repeatable and according to theory.

Higher viscosity and viscosity index oils are related to thicker OFT values. At high temperatures, OFT has a certain value which decreases slightly as temperature rises even more. At high loads the temperature effect is not evident.

HTHS has an effect on friction force. Furthermore, friction peaks at flow reversal move closer to the dead centres as load increases and away from the dead centres

as speed increases. This observation is a useful finding for the visualisation results that follow in the next chapter.

The numerous figures show the trend noticed during the parametric study. The viscosity effects were evident during testing and there are also distinguishable results for speed, load and high temperature testing. It also showed the compatibility of the friction force reversal at the dead centres according to theory for the high temperature results. Capacitance OFT measurements proved to be a robust technique which enabled testing at higher temperatures and therefore useful results relative to that specific parametric study to be drawn.

## CHAPTER 4

### TEST RIG

#### OIL FILM VISUALISATION AND PRESSURE MEASUREMENTS

#### INTRODUCTION

The test rig developed and described by Arcoumanis et al (1995) has been modified to allow the lubricant flow to be visualised. The onset and development of cavitation in the lubrication film present throughout the stroke between the piston ring and the cylinder liner in the test rig was visualized by a charge-coupled device (CCD) camera and monitored by a miniature pressure sensor. Various transient cavitation patterns were identified during the stroke, ranging from developing fern-shaped cavities to string cavities extending gradually up to the trailing edge of the piston ring. Both the onset and the development of these cavities seem to be affected by piston speed and load; at higher speeds cavities appear later in the stroke and are larger in size, while at higher loads they appear earlier, are more numerous and, thus, smaller in size.

A parametric study of the effect of speed, load, temperature and piston ring curvature on oil film pressure is also conducted.

#### 4.1 VISUALISATION EXPERIMENTS

Researchers from Toyota (Inagaki et al, 1995) have developed a fluorescent diagnostic system that can measure the oil film distribution through two-dimensional images. They had visualised the lubricant behaviour under different engine conditions and pictures were taken through a fully transparent liner. As a result, the lubricant film distribution at the thrust-side piston skirt zone and at the piston rings could be obtained.

Another group of researchers, Kim et al (1995) carried out pressure measurement experiments and visualisation of oil film simultaneously in an idealised single ring test rig. In this test rig, the piston ring is simulated by a straight glass bar test piece and the cylinder by a reciprocating flat plate. The researchers observed negative

pressures of the full film region before oil film rupture and formation of cavities. The shape and dynamic behaviour of oil film rupture are greatly influenced by revolution speed. They also found that the first oil film rupture occurs in the diverging part of the full film region and results in the generation of internal cavities. With the generation of internal cavities, tension is released but negative pressure is still maintained in internal cavities.

MIT researchers (Thirouard, 2001) have conducted experiments in both single cylinder diesel and spark ignition engines. In both cases, sections of the liner were replaced by synthetic fused silica and sapphire window sections respectively and thus the oil flow could be visualised. A two dimensional visualisation system using laser induced fluorescence was developed to provide detailed images of the oil distribution. Observations about the oil flow, transport and behaviour through the piston ring pack were made. Oil was observed flowing through the top ring groove in all engine conditions and the top ring up-scraping was found to be the main oil supply mechanism to the crown land at high engine speed and low load. Oil motion on the second land was observed in both circumferential and axial directions. Top ring down scraping appeared to be a major source of oil accumulation on the piston second land and part of the oil accumulated on the second land is transported to the crown land as liquid oil through the ring groove or as mist through the top ring gap.

The Laser Induced Fluorescence experiments that have been carried out with the single-ring test rig within the engine group at Imperial College, showed that cavitation occurs at relatively high speeds. The implementation of a new liner specimen, introducing a glass area inside an aluminium frame was considered a necessary action for the visualisation experiments. Eventually, pictures can be taken through the glass window at different crank angles which revealed that cavitation is a process happening at even low speeds.

#### 4.1.1 GLASS LINER

The glass liner specimen can be seen in Figure 4.1. The aluminium frame is in Figure 4.2. Plastic shims were placed underneath the glass liner as is shown in Figure 4.3,

where one can see how the glass liner was assembled. This is to ensure that the glass surface which is going to be in contact with the ring specimen remains parallel to the surfaces 1, 2 shown in figure 4.3.

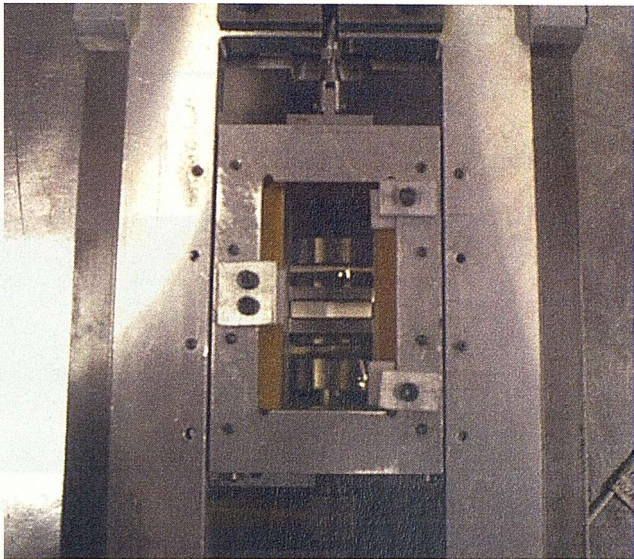


Figure 4.1      The glass liner in the single ring test rig

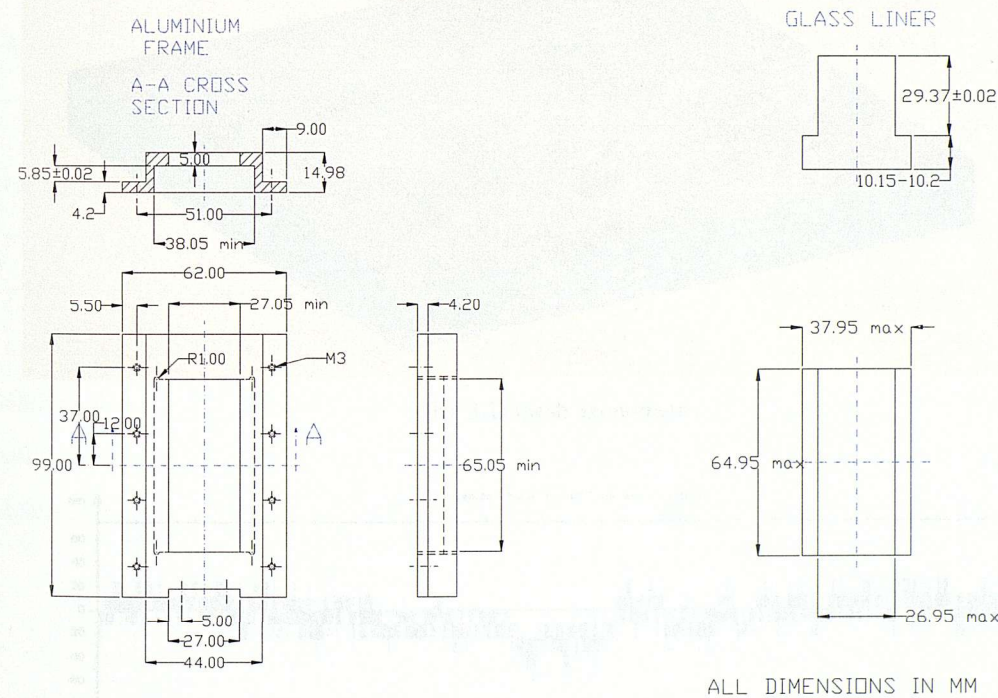
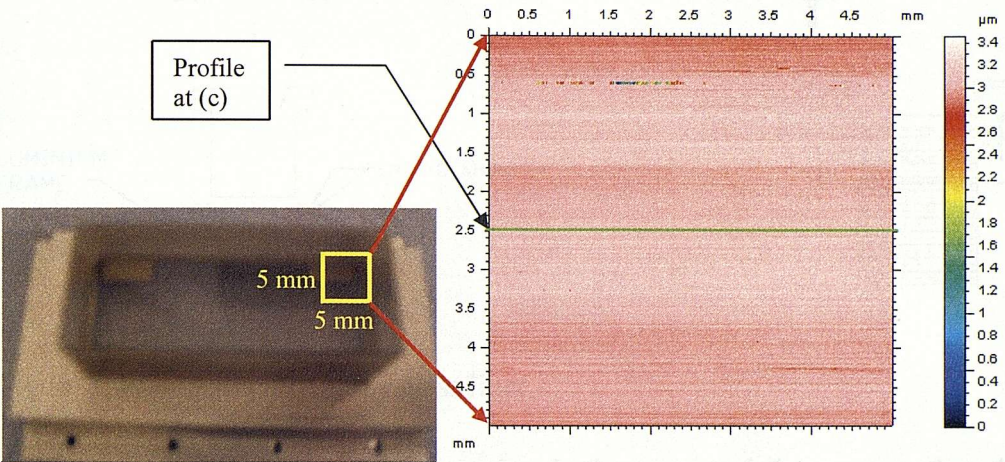


Figure 4.2      The aluminium frame holder and the glass liner

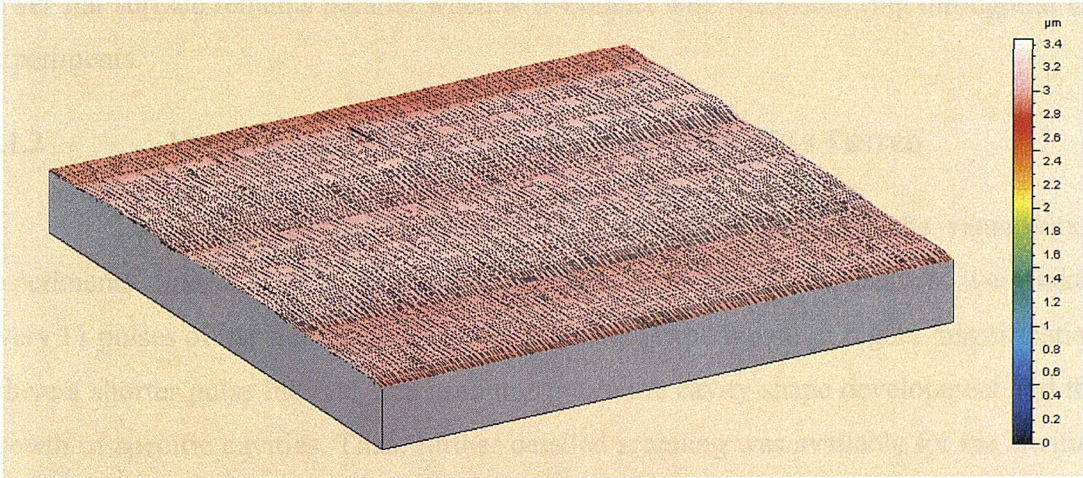
Figure 4.3      2-D and 3-D surface coordinate mesh for the glass liner specimen



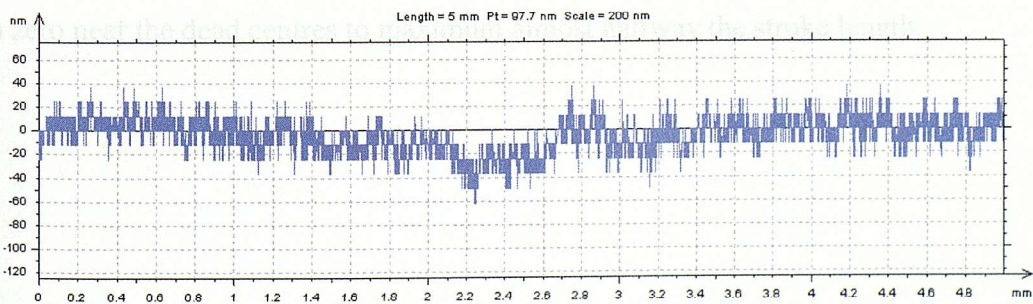
Figure 4.3 shows the roughness test results taken from a 5 mm × 5 mm area of the glass liner surface that is in contact with the ring specimen. 250 profiles were taken and the results are shown in Figure 4.3.



(a) 2-D surface roughness test of the highlighted glass liner section



(b) 3-D mesh axonometric



(c) profile taken from (a) at 2.5 mm

Figure 4.3 2-D and 3-D surface roughness tests for the glass liner specimen



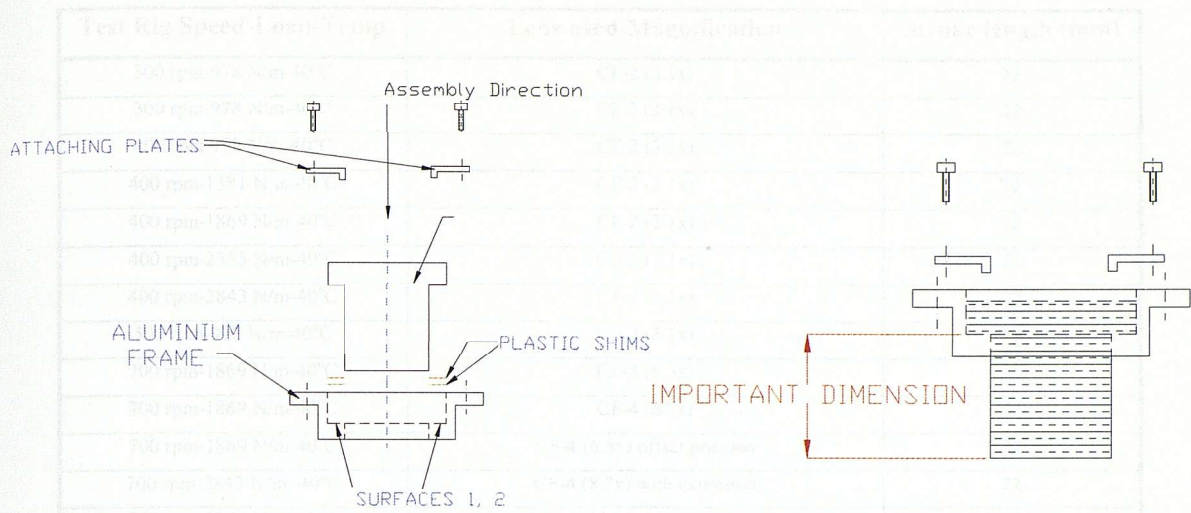


Figure 4.4 Assembly of the glass liner

In Figure 4.4 the important dimension is the one needed to ensure that the lower flat surface remains parallel when is in contact with the piston ring throughout the experiments.

4.1.2 VISUALISATION EXPERIMENTS CONDITIONS - OILS TESTED

The following matrix shows the test conditions used for the visualisation experiments. Speed, load, stroke and temperature were varied and pictures were taken every 11 pulses of the shaft encoder, except for some experiments at higher magnification where a shorter pulse interval was used to capture the cavity shape development and the growth of specific cavities. Thus, further detailed scanning was available for the cavities shapes that were initially identified with the low magnification lens to make better observations about their initiation, shape and transformation as the flow speed grows from zero near the dead centres to maximum almost halfway the stroke length.



Figure 5 Schematic diagram of the test rig

| Test Rig Speed-Load-Temp | Lens used-Magnification                    | Stroke length (mm) |
|--------------------------|--------------------------------------------|--------------------|
| 300 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 22                 |
| 300 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 22                 |
| 400 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 22                 |
| 400 rpm-1381 N/m-40°C    | CF-2 (3.1x)                                | 22                 |
| 400 rpm-1869 N/m-40°C    | CF-2 (3.1x)                                | 22                 |
| 400 rpm-2355 N/m-40°C    | CF-2 (3.1x)                                | 22                 |
| 400 rpm-2843 N/m-40°C    | CF-2 (3.1x)                                | 22                 |
| 500 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 22                 |
| 700 rpm-1869 N/m-40°C    | CF-3 (5.3x)                                | 22                 |
| 700 rpm-1869 N/m-40°C    | CF-4 (8.7x)                                | 22                 |
| 700 rpm-1869 N/m-40°C    | CF-4 (6.8x) offset position                | 22                 |
| 700 rpm-2843 N/m -40°C   | CF-4 (8.7x) with extension                 | 22                 |
| 700 rpm-2843 N/m-40°C    | CF-4 (8.7x) with extension offset position | 22                 |
| 100 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |
| 200 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |
| 300 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |
| 400 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |
| 400 rpm-978 N/m-65°C     | CF-2 (3.1x)                                | 50                 |
| 400 rpm-1381 N/m-40°C    | CF-2 (3.1x)                                | 50                 |
| 400 rpm-1869 N/m-40°C    | CF-2 (3.1x)                                | 50                 |
| 400 rpm-2355 N/m-40°C    | CF-2 (3.1x)                                | 50                 |
| 400 rpm-2843 N/m-40°C    | CF-2 (3.1x)                                | 50                 |
| 500 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |
| 600 rpm-978 N/m-40°C     | CF-2 (3.1x)                                | 50                 |

Table 4.1 Visualisation Experiments conditions

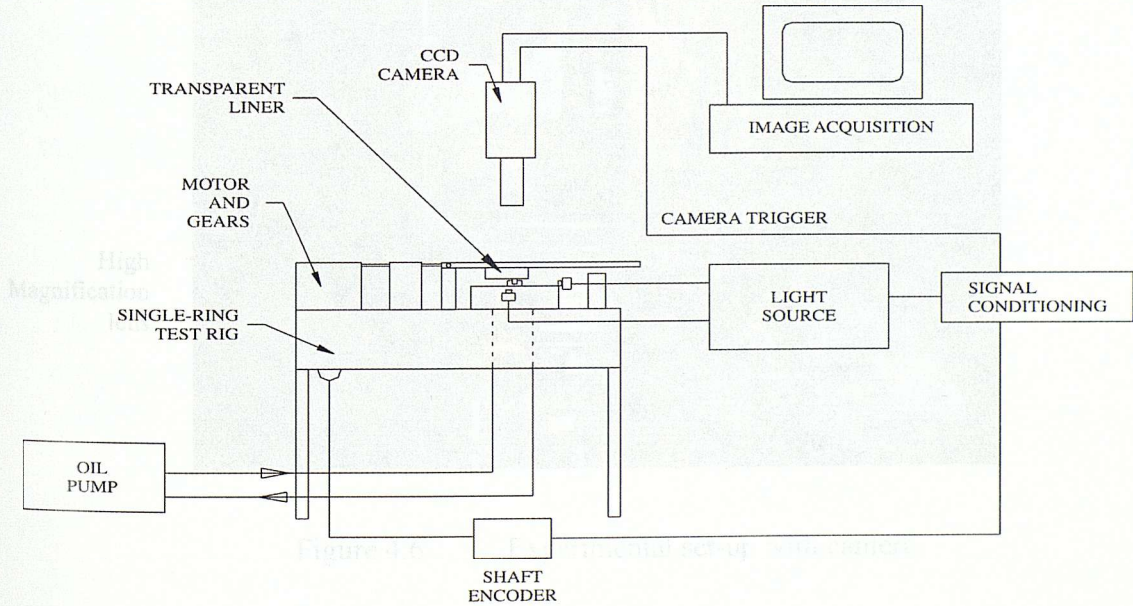


Figure 4.5 Schematic of the set-up



A high speed CCD camera was used to take pictures of the oil film during the whole stroke. Images were taken every  $2^\circ$  crank angle, corresponding to a range of 0.0152 – 0.8725 mm as a function of speed. The hardware trigger used for the camera was a National Instruments SC-2345 signal conditioning unit with a National Instruments 16-bit 6035E PCI data acquisition card. This new acquisition system with a faster A/D converter board was considered necessary for triggering the camera at high speeds. The shaft encoder signal is directly connected to the electric motor, which in turn provides a single signal for the TDC. This is used to provide the trigger pulse for the CCD camera and the flash light lamp needed to provide adequate light for the imaging in every cycle. 2000 pulses/revolution were used for data sampling as in the older configuration for the parametric series of experiments on the single ring test rig. All the encoder signals are inputs to the signal conditioning unit and Labview software with its G programming language is used for developing computer programs. The National Instruments NI-DAQ 6.9.3 was used for the development of the data acquisition program. The oil used was Castrol 1A.

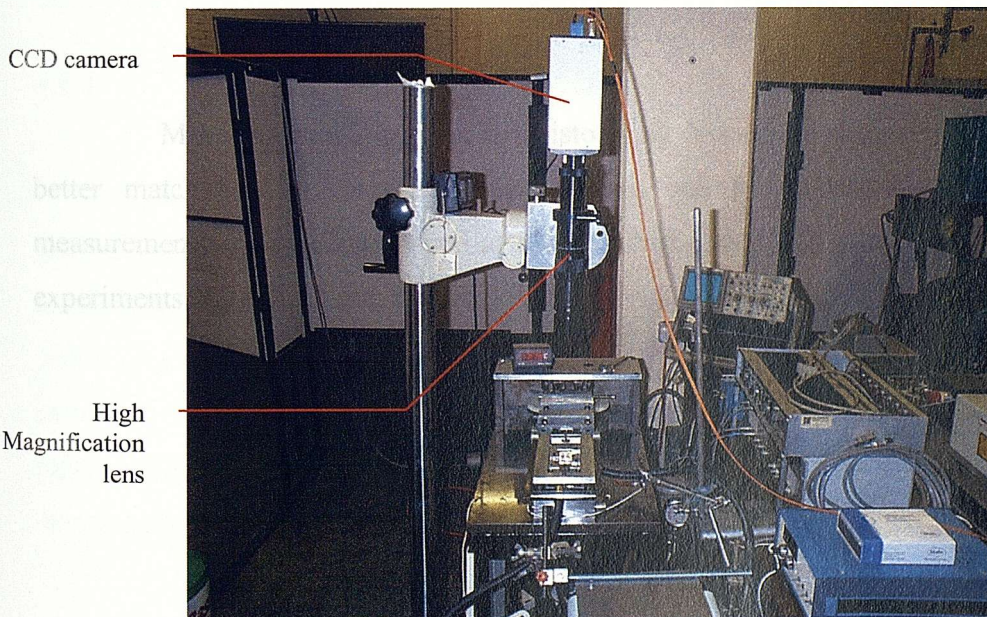


Figure 4.6 Experimental set-up with camera



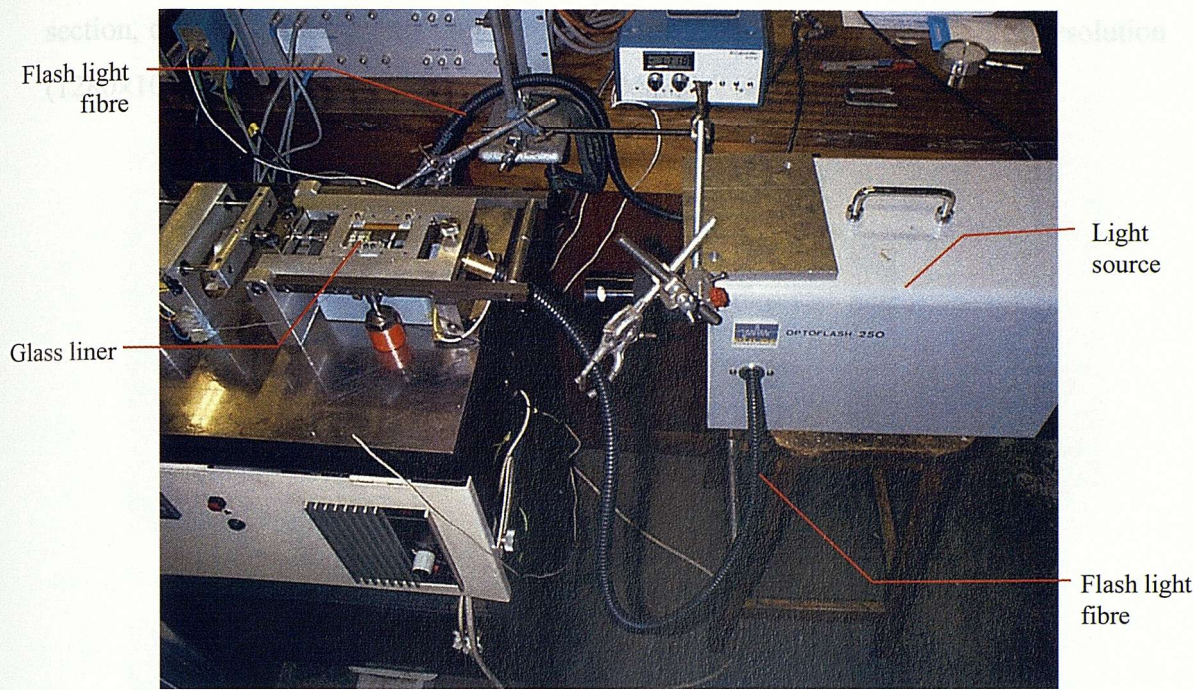


Figure 4.7 The test rig with the glass liner and flash lamp

4.1.3 TEST CONDITION FOR OIL FILM VISUALISATION: 300 RPM – 22 MM STROKE

More information about the piston-ring lubrication could be extracted and a better match for the pressure measurements can be presented. All the pressure measurements were accomplished at 22 mm stroke and so the respective visualisation experiments were done with the single ring test rig set at the same stroke.

|                       |         |
|-----------------------|---------|
| Blend Code            | 001A/05 |
| Grade                 | 5W40    |
| HTHS at 150°C (mPa s) | 3.6     |
| V100 (cSt)            | 13      |
| V40 (cSt)             | 80.5    |
| VI                    | 167     |

Table 4.2 Oil characteristics

Figure 4.8 shows the glass liner that had been used for the visualisation experiments as is fitted at the experimental set-up; the quartz liner has a 27 mm × 65 mm



section, contained within an aluminium frame. Photos were taken with a high resolution (1280×1024) 12-bit peltier cooled PCO CCD camera equipped with a zoom lens.

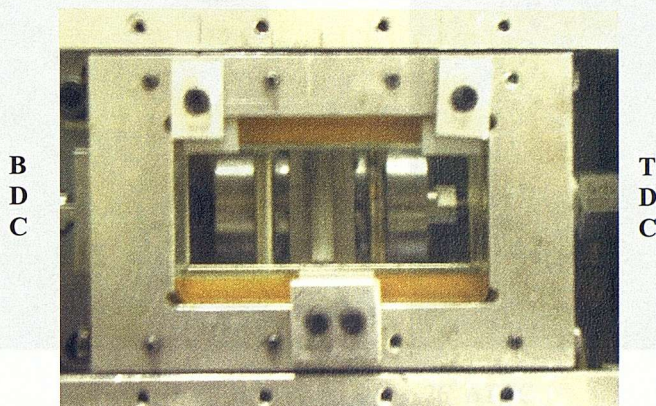


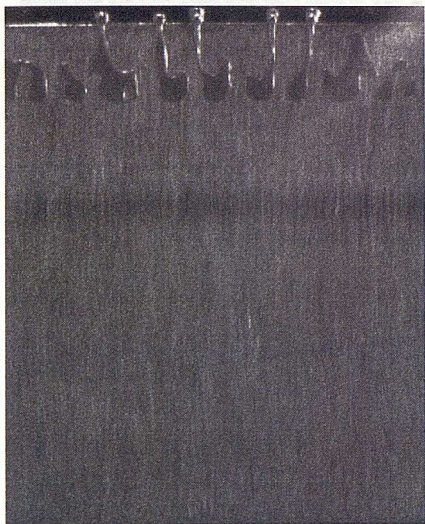
Figure 4.8 Close-up of the glass liner set-up

A flash lamp with optical fibres was used to illuminate the glass liner and piston-ring. Two fibres were used so that the light could be emitted in two directions providing the best possible illumination for the pictures.

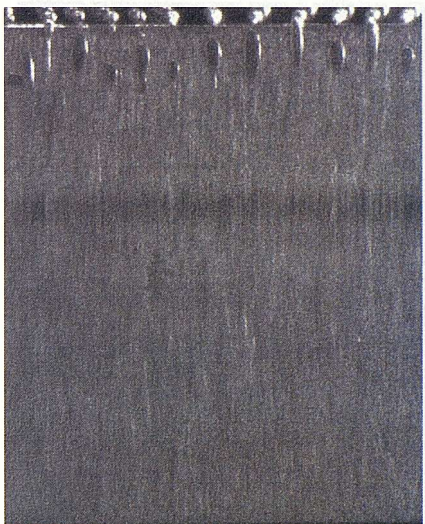
#### 4.1.4 OIL FLOW PICTURES AND CAVITY DEVELOPMENT AT 300 RPM

The following pictures show the flow development at 300 rpm. The testing showed that as the flow reverses internal cavities appear on the surface of the piston ring. Later in the stroke they develop into string cavities that reach the trailing edge of the piston ring. The cavities coalesce before they are formed into strings. These results were compared to the ones taken at different speeds and stroke length. There was good repeatability in terms of the cavity shapes that were observed. Cavities are formed at the diverging wedge of the piston ring specimen.





24° ATDC



26° ATDC



28° ATDC

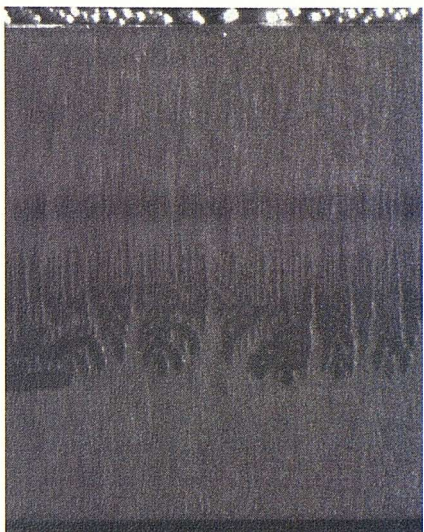


30° ATDC

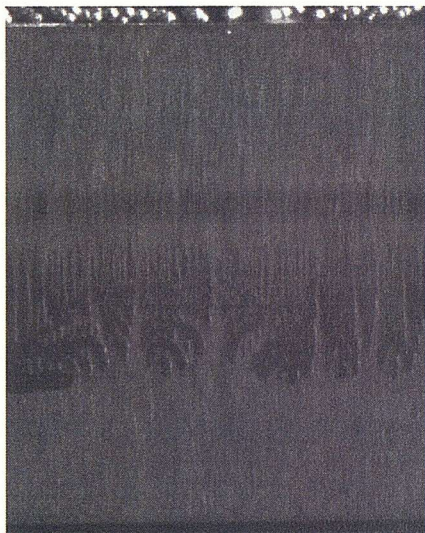




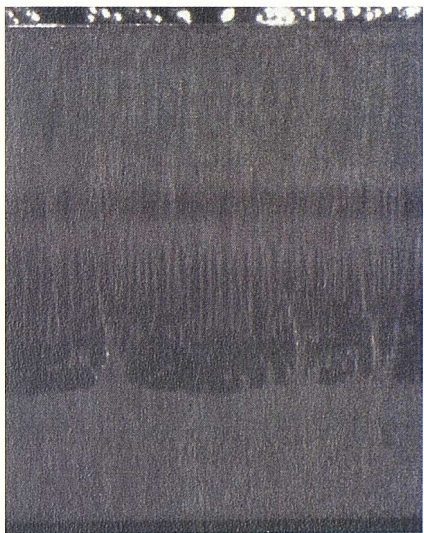
32° ATDC



34° ATDC

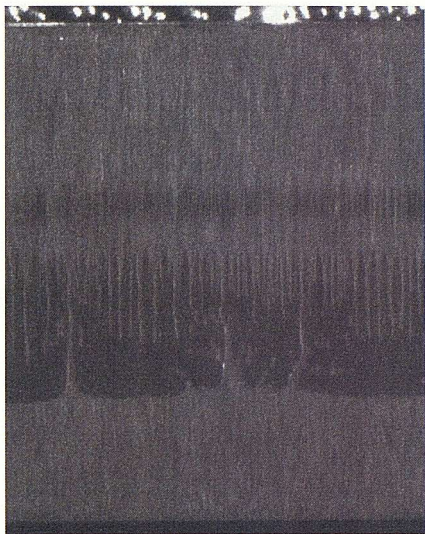


36° ATDC

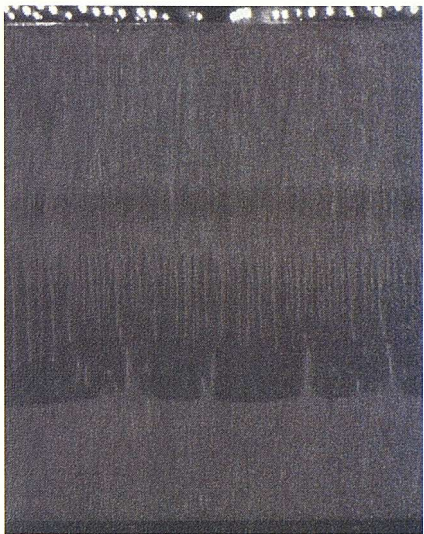


38° ATDC





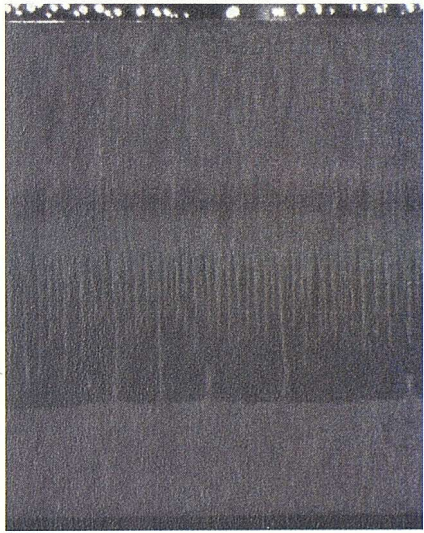
40° ATDC



42° ATDC

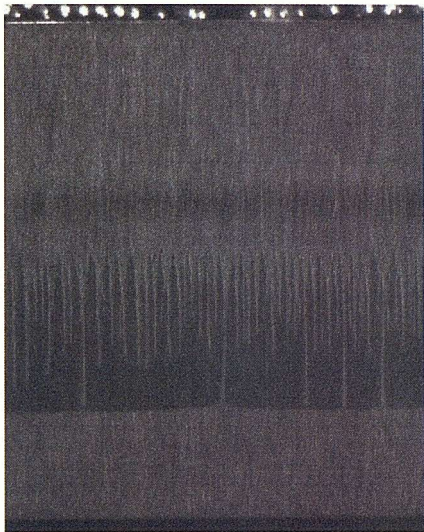


44° ATDC



46° ATDC

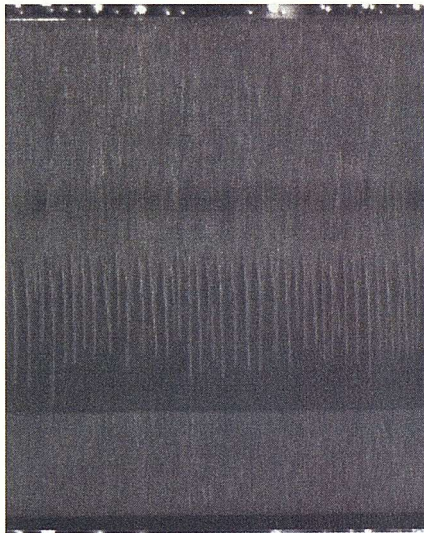




48° ATDC



50° ATDC

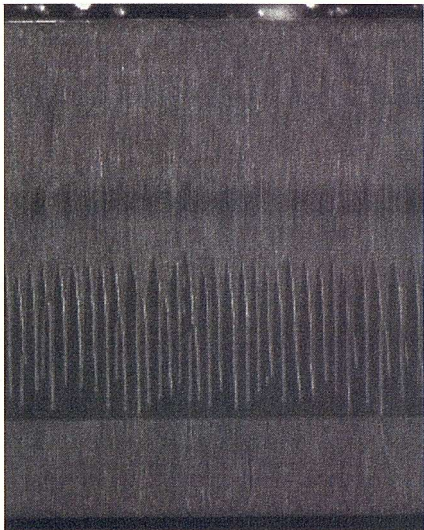


52° ATDC

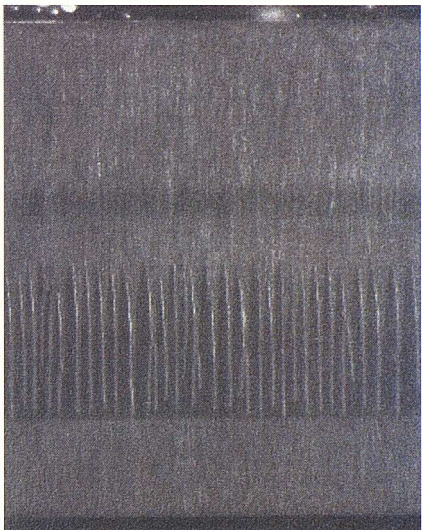


54° ATDC





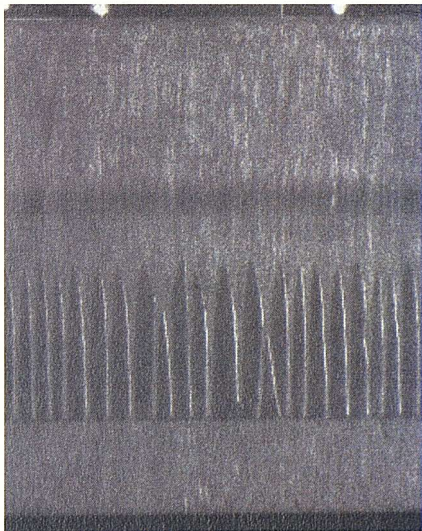
64° ATDC



66° ATDC



90° ATDC



92° ATDC



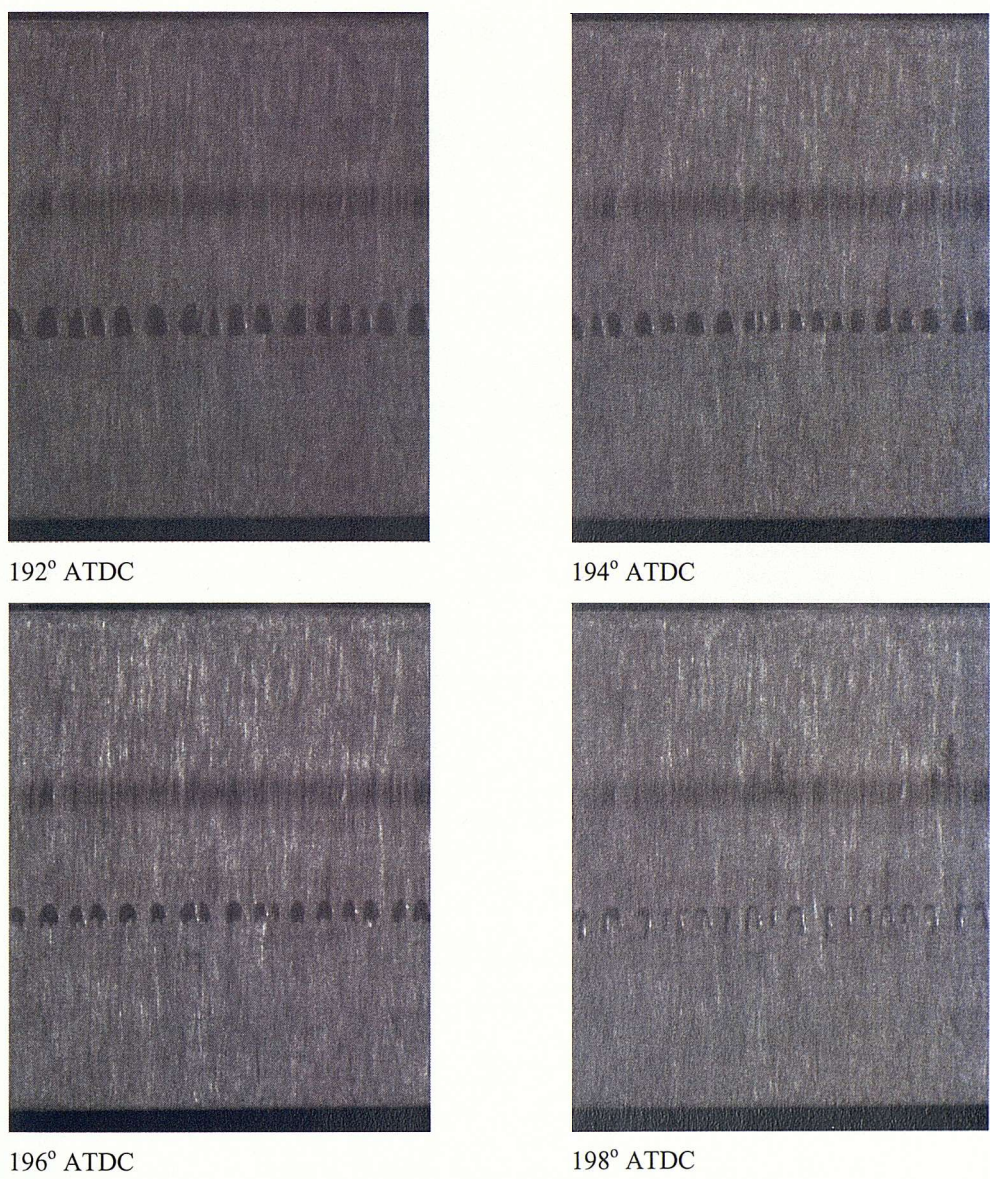


Figure 4.9 Oil film rupture pictures at different crank angles

This series of experiments, revealed the initiation and development of the cavitation throughout the stroke. More details about their stage and development are going to be discussed later on in this chapter, where they are correlated with the pressure measurements. Previous experiments which identified the cavitation under the ring at mid-stroke (Pyke, 2000) were unable to reveal the cavities that start appearing a few degrees after the dead centre. With the new set of experiments, the development



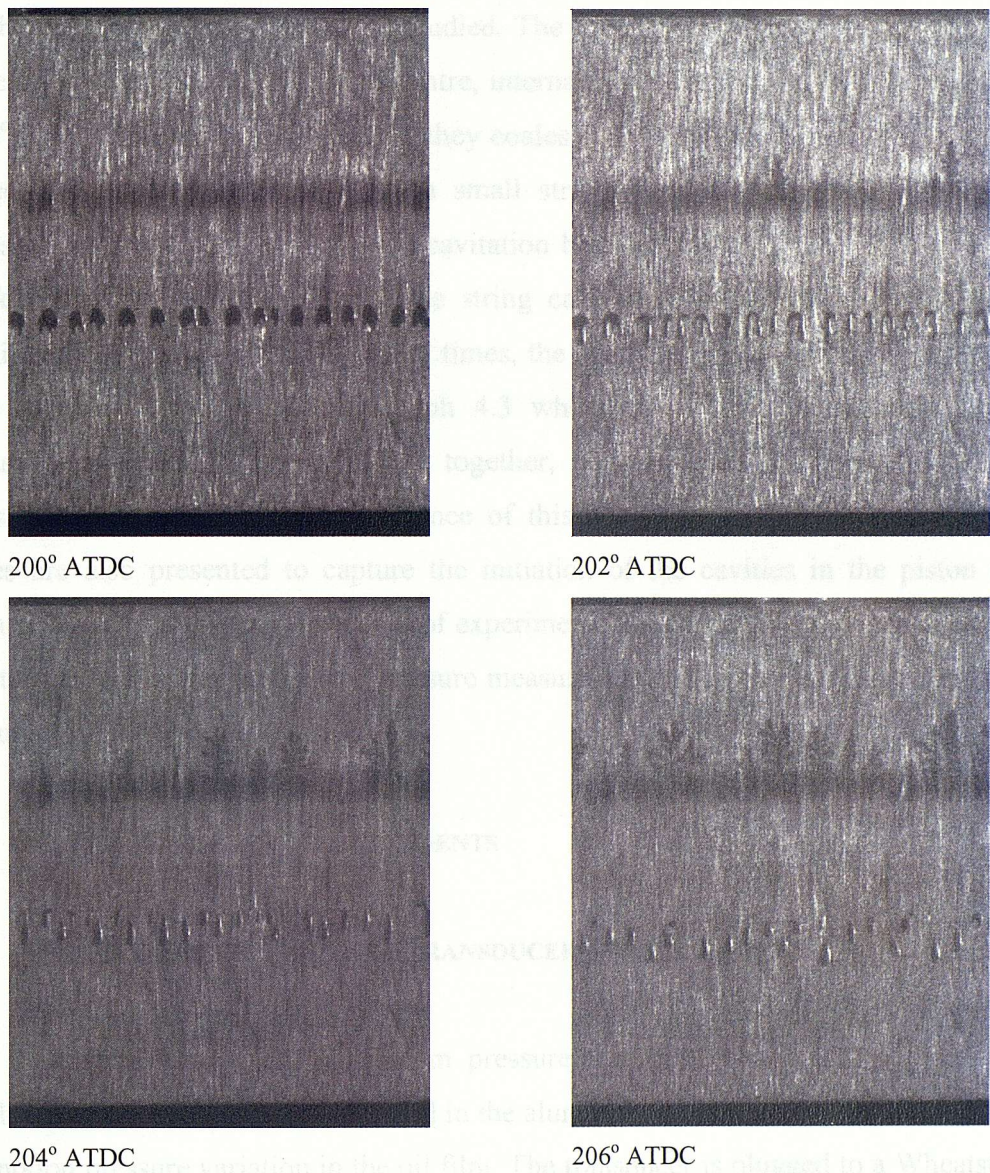


Figure 4.9 Oil film rupture pictures at different crank angles

This series of experiments, revealed the initiation and development of the cavitation throughout the stroke. More details about their shape and development are going to be discussed later on in this chapter, where they are combined with the pressure measurements. Previous experiments which identified the cavitation under the ring at mid-stroke (Pyke, 2000) were unable to reveal the cavities that start appearing a few degrees after the dead centre. With the visualisation experiments, the development



throughout the stroke could be further studied. The above pictures reveal that as the flow reverses, a few degrees after the dead centre, internal cavities appear in the shape of ferns (26-34° CA). These ferns grow further, they coalesce and cavities shaped as fissures are pictured afterwards (36-42° CA). Then small strings appear from inside the fissure cavities which grow until they reach a cavitation boundary (48-92° CA). Finally as the flow reverses later on in the stroke the string cavities diminish and the cavities are becoming bubbles (202-206° CA). At all times, the cavity shapes appear at the diverging wedge of the piston ring. In paragraph 4.3 where the pressure measurements and visualisation experiments are combined together, a clearer picture is being given that reveals the physics behind the appearance of this phenomenon. Higher magnification pictures are also presented to capture the initiation of the cavities in the piston ring specimen surface. Following this series of experiments, the higher magnification ones are presented, as well as the respective pressure measurements to get a clearer picture of the lubrication in the idealised test rig.

## **4.2 OIL FILM PRESSURE MEASUREMENTS**

### **4.2.1 MINIATURE PRESSURE TRANSDUCER AND LIF SENSOR**

A miniature silicon diaphragm pressure transducer manufactured by Kulite (model XCQ-080-500 Psia) was installed in the aluminium liner and was used to measure the temporal pressure variation in the oil film. The transducer is plugged to a Wheatstone bridge conditioner and an amplifier. According to the calibration certificate from Kulite Semiconductors Products Inc, the sensitivity of the sensor is 0.333 mV/PSI A. The compensated temperature range is 25°C to 80°C, output impedance 1370 Ohms and input impedance 1690 Ohms. For the test conditions the rated pressure is 500 PSI A and maximum pressure 1000 PSI A, tested at 10 VDC excitation. The maximum excitation is 15 VDC and the operational mode is absolute. The instrumentation is in conformance with MIL-STD-45662A.

The transducer was mounted in a new liner specimen, positioned at the middle, next to the optical fibre that is used for the LIF experiments in the idealised test rig that is

used to simulate the lubrication between the piston-ring and the cylinder liner. The configuration chosen was to mount the transducer next to an LIF sensor, slightly recessed from the liner's bottom surface (Fig. 4.13). The 2 mm diameter transducer is fitted in the moving liner specimen (Figure 4.10) and the oil goes through the 0.1 mm wide slot in Figure 4.13b to provide the pressure reading. There is a void between the edge of the pressure transducer and the surface of the liner in contact with the piston-ring. It was not mounted flush with the surface of the liner. Had the configuration been like that, the transducer area that would provide averaged oil film pressure data was going to be quite big relative to the piston ring width. In that case the output pressure signal was not going to represent the instant pressures under the piston ring and therefore a good resolution of the pressure measurements would have been lost.

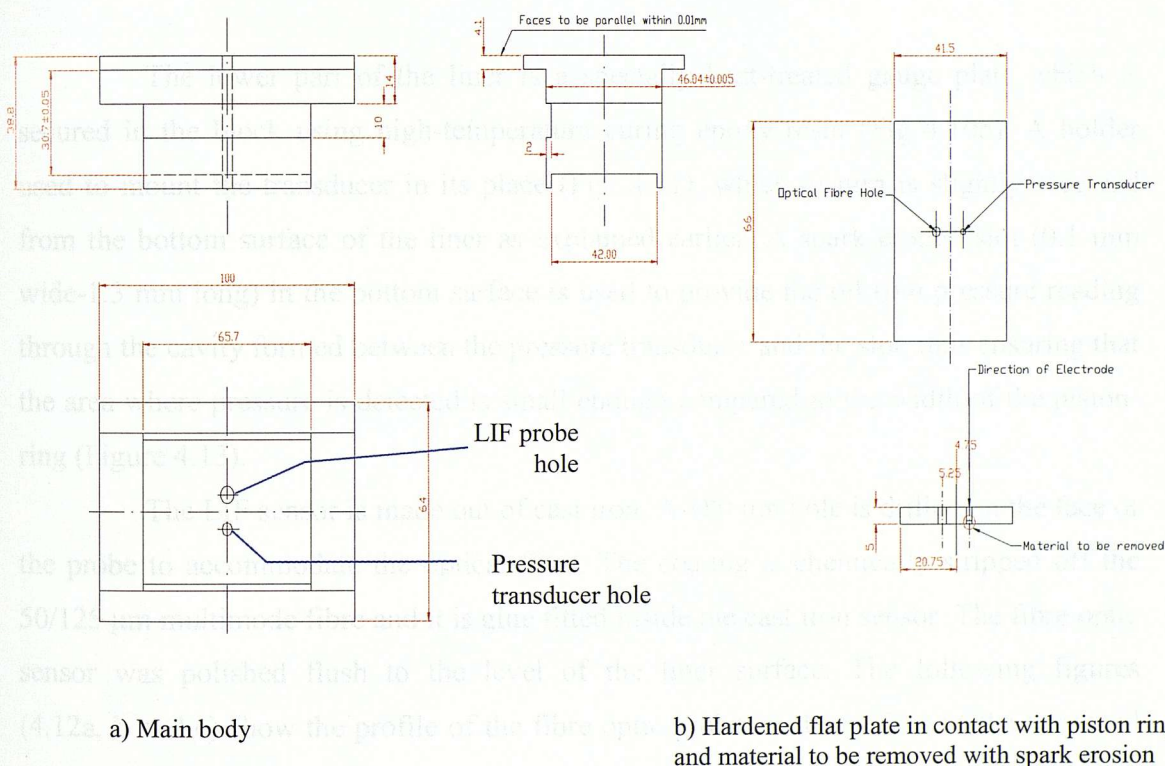


Figure 4.10 Liner consists of two pieces

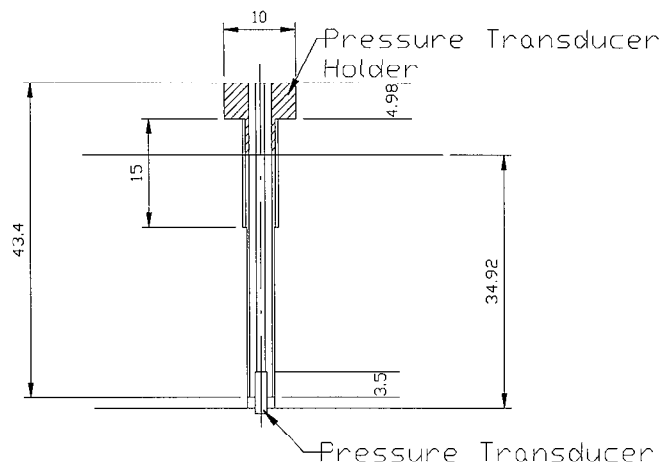
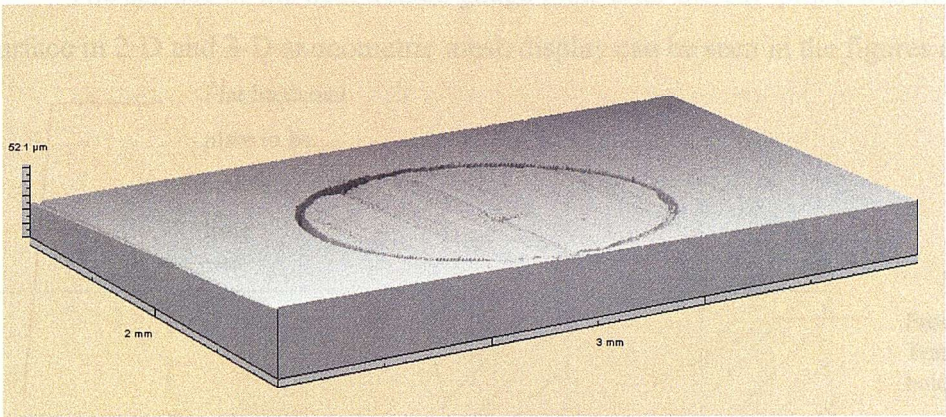


Figure 4.11 The Pressure Transducer holder on the liner main body

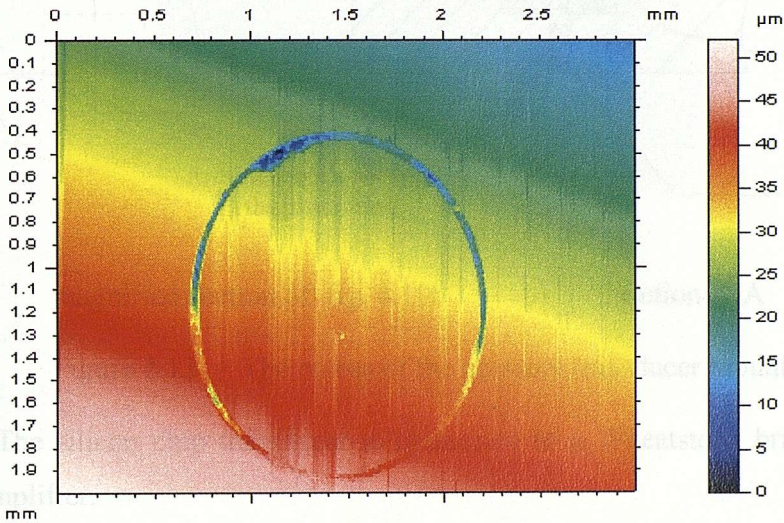
The lower part of the liner is a specially heat-treated gauge plate which is secured in the block using high-temperature curing epoxy resin (Fig 4.10b). A holder used to mount the transducer in its place (Fig. 4.11), which in turn is slightly recessed from the bottom surface of the liner as explained earlier. A spark eroded slot (0.1 mm wide-1.3 mm long) in the bottom surface is used to provide the oil film pressure reading through the cavity formed between the pressure transducer and the slot, thus ensuring that the area where pressure is detected is small enough compared to the width of the piston-ring (Figure 4.13).

The LIF sensor is made out of cast iron. A 180  $\mu\text{m}$  hole is drilled on the face of the probe to accommodate the optical fibre. The coating is chemically stripped off the 50/125  $\mu\text{m}$  multimode fibre and it is glue fitted inside the cast iron sensor. The fibre optic sensor was polished flush to the level of the liner surface. The following figures (4.12a, b and c) show the profile of the fibre optic probe on the surface of the hardened plate. As shown in the following figure, it is flush with the liner surface and only around the probe there is a slight recess that shows the outer line of the steel probe that includes the fibre. The tolerances of the probe fitting are -0.03 mm. This profile measurement shows how much is the lubrication on the surface of the liner affected by the “intrusion” of the fibre optic probe.

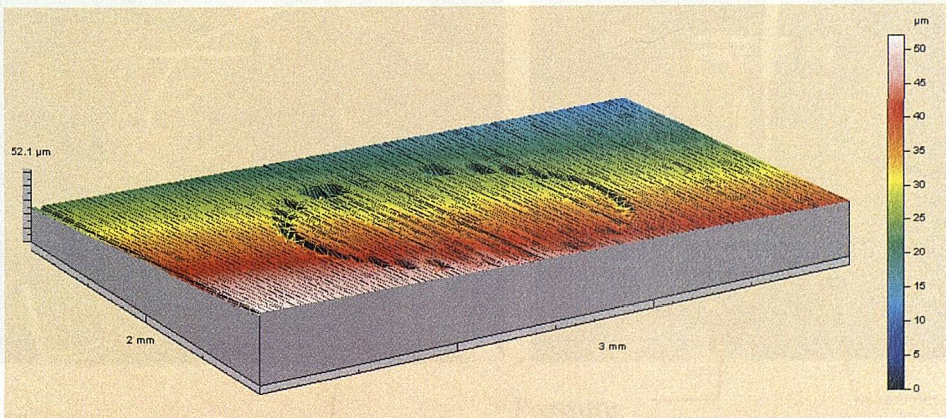




a) 3-D greyscale



b) 2-D mesh



c) 3-D axonometric

Figure 4.12 3-D surface maps and 2-D mesh of the probe fitting on the liner surface



4.2.2 The surface of the heat treated gauge plate after the LIF probe was mounted on the surface in 2-D and 3-D axonometric mesh display can be seen in the figures above.

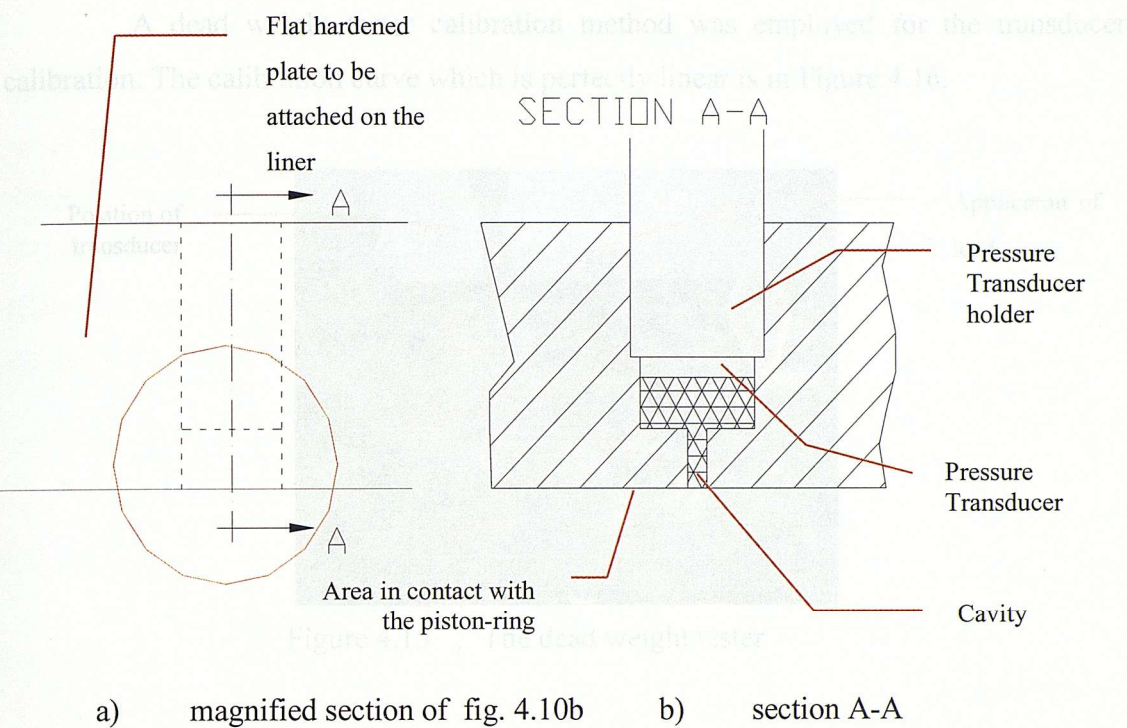


Figure 4.13 The recess of the pressure transducer mounting

The silicon chip transducer is connected to a Wheatstone bridge conditioner unit and amplifier.

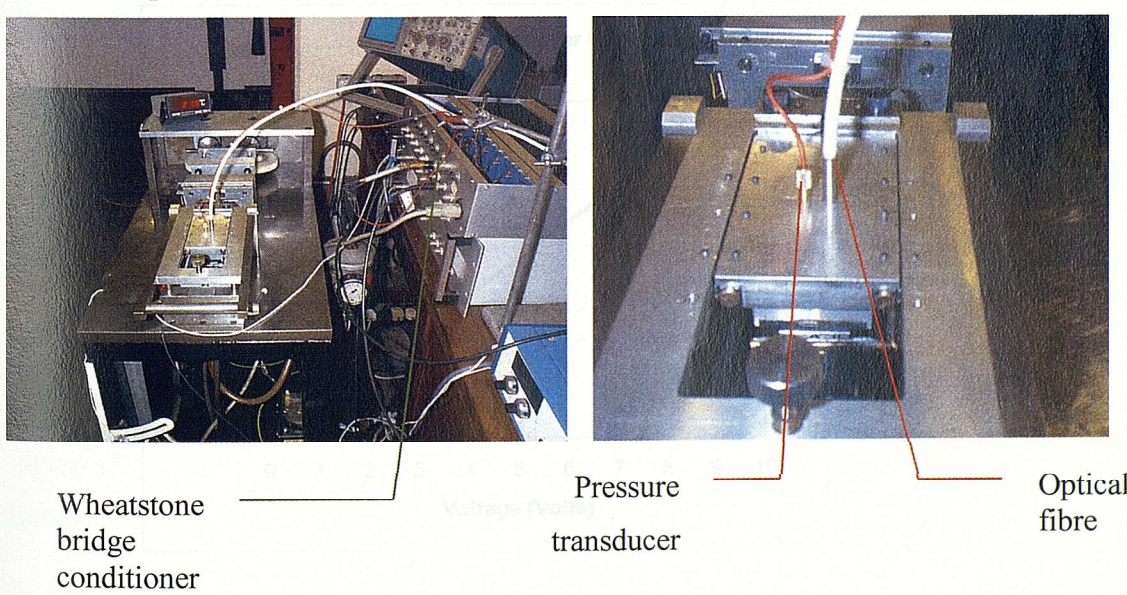


Figure 4.14 The pressure transducer fitted in the new liner



4.2.2 PRESSURE TRANSDUCER CALIBRATION

A dead weight tester calibration method was employed for the transducer calibration. The calibration curve which is perfectly linear is in Figure 4.16.

set at 50 mm. The pressure under the piston may at midstroke is in Figure 4.17. This pressure was taken at the midstroke position. The pressure was measured using a pressure transducer (Figure 4.18) was installed in the test rig. The pressure transducer was connected to a multimeter (electrical compound) to measure the pressure. The pressure transducer was connected to the oil film pressure to the test rig.

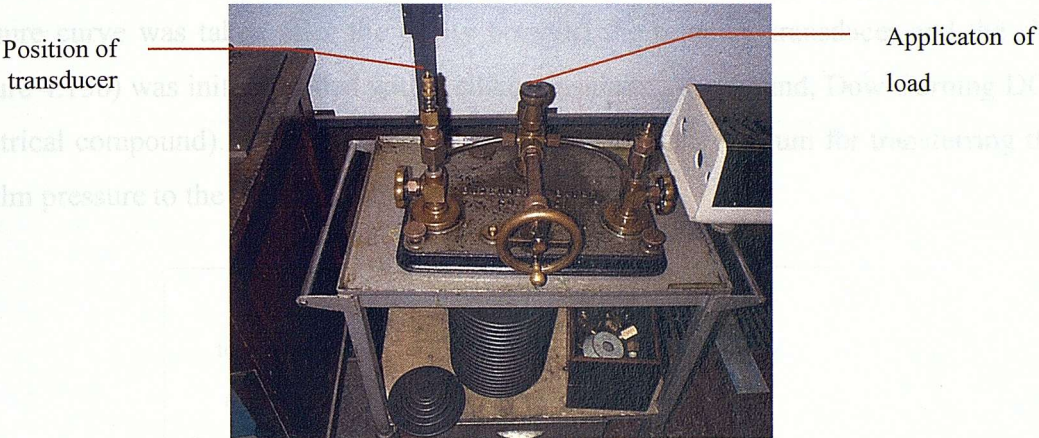


Figure 4.15 The dead weight tester

The dead weight tester used for the calibration is in Figure 4.15. Different loads were implemented and each time a reading from a FLUKE 87 multimeter was taken. Up to 500 psi load was applied to the transducer.

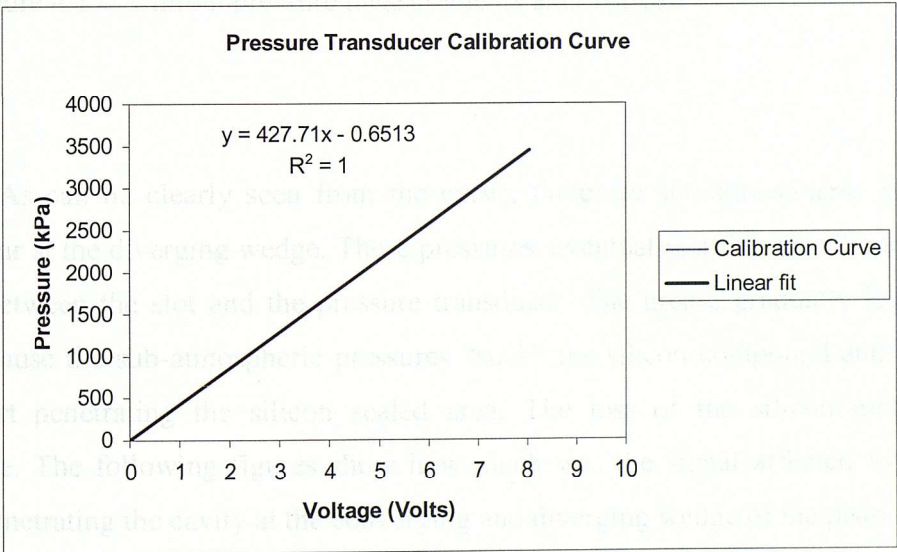


Figure 4.16 Pressure transducer calibration curve

4.2.3 PRESSURE TRANSDUCER REPEATABILITY

The initial experiments were accomplished with the pressure transducer set to measure the pressure under the piston-ring specimen at mid-stroke. The stroke length was set at 50 mm. The pressure under the piston ring at midstroke is in Figure 4.17. This pressure curve was taken after the cavity between the pressure transducer and the slot (Figure 4.13b) was initially filled with a silicon insulating compound, Dow Corning DC4 (electrical compound). The silicon compound was used as a medium for transferring the oil film pressure to the transducer through the existing slot.

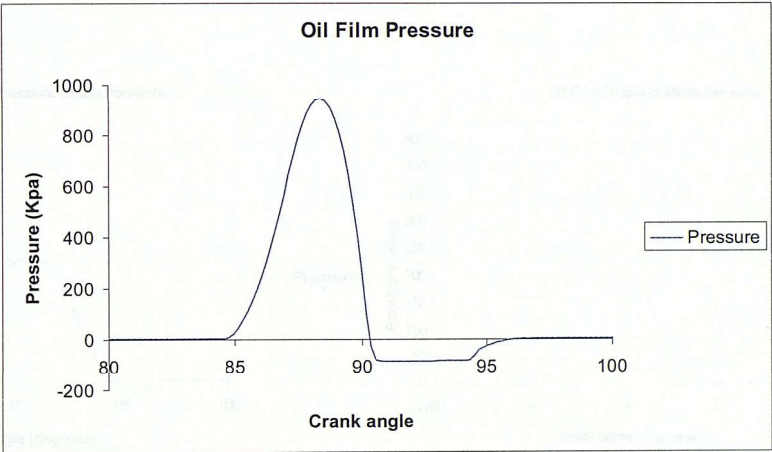


Figure 4.17 Initial pressure measurements at midstroke

As can be clearly seen from the curve, there are sub-atmospheric pressures which occur at the diverging wedge. These pressures, eventually, affect the silicon grease that lies between the slot and the pressure transducer. The grease gradually leaves the cavity because the sub-atmospheric pressures “suck” the silicon compound and pockets of air start penetrating the silicon sealed area. The loss of the silicon medium is undesirable. The following figures show how much was the signal affected by the air bubbles penetrating the cavity at the converging and diverging wedge of the piston ring.



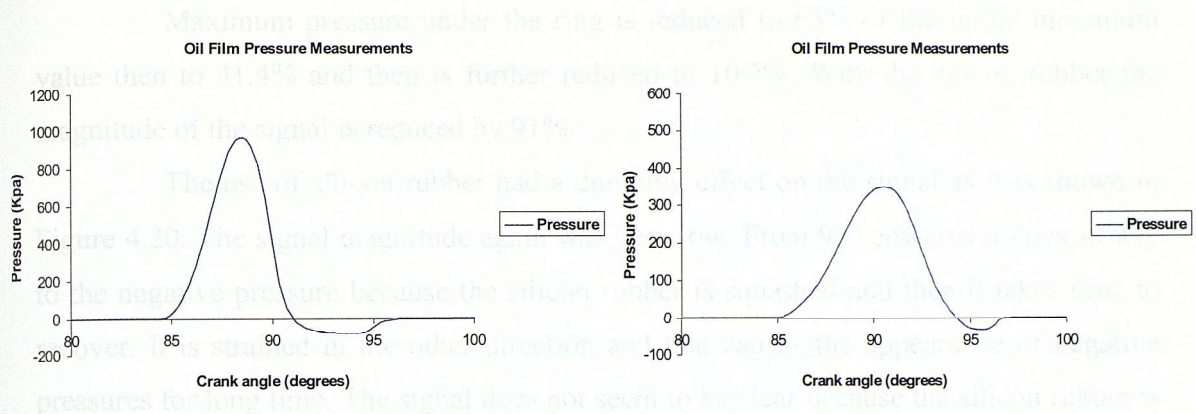


Figure 4.18 The measurement magnitude is reduced by the presence of air bubbles

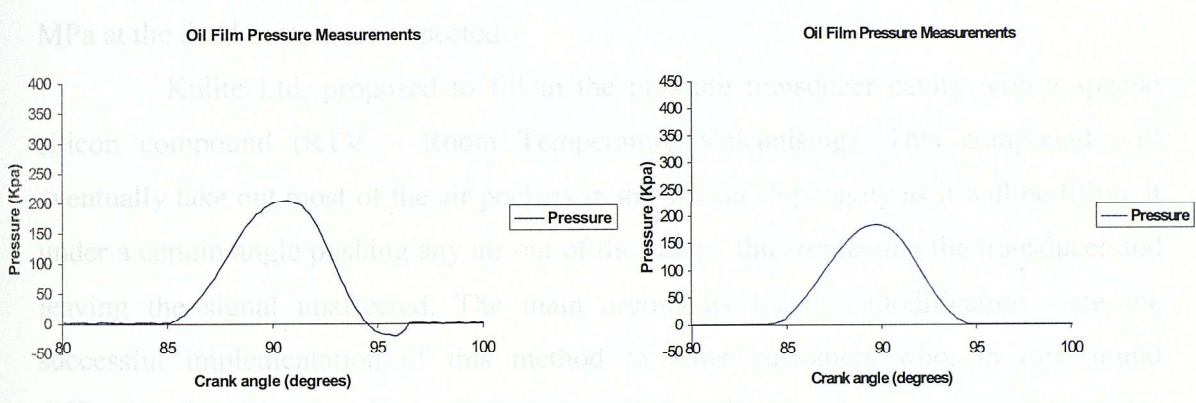


Figure 4.19 Further reduction of the measurement magnitude

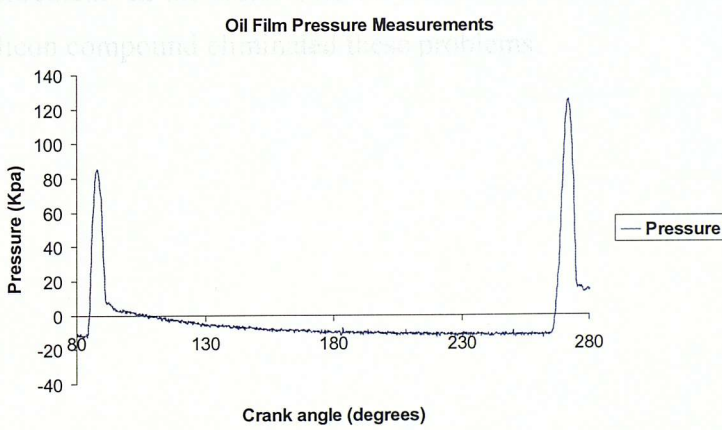


Figure 4.20 The pressure signal with the use of silicon rubber



Maximum pressure under the ring is reduced to 63% of the initial maximum value then to 41.4% and then is further reduced to 10.7%. With the silicon rubber the magnitude of the signal is reduced by 91%.

The use of silicon rubber had a dumping effect on the signal as it is shown in Figure 4.20. The signal magnitude again was very low. From 93° onwards it goes slowly to the negative pressure because the silicon rubber is squashed and then it takes time to recover. It is strained in the other direction and that causes the appearance of negative pressures for long time. The signal does not seem to be clear because the silicon rubber is oscillating as well when it is not in the higher pressure areas of the stroke.

According to Ostovar (1996) who modelled the lubrication in the single-ring test rig, pressures of the magnitude of 900 kPa (no extra load) at mid-stroke and up to 10 MPa at the dead centres are expected.

Kulite Ltd. proposed to fill in the pressure transducer cavity with a special silicon compound (RTV – Room Temperature Vulcanising). This compound will eventually take out most of the air pockets in the silicon chip cavity as it will be filling it under a certain angle pushing any air out of the cavity, thus protecting the transducer and leaving the signal unaffected. The main arguments for this modification were the successful implementation of this method to other customers who, in turn found difficulties in measuring air pressures in wind tunnels. In many cases, small particles were accelerated by the air flow and by hitting the silicon chip, they affected the real air pressure measurement. In the worst case they were damaging the transducer. Filling the cavity with silicon compound eliminated these problems.

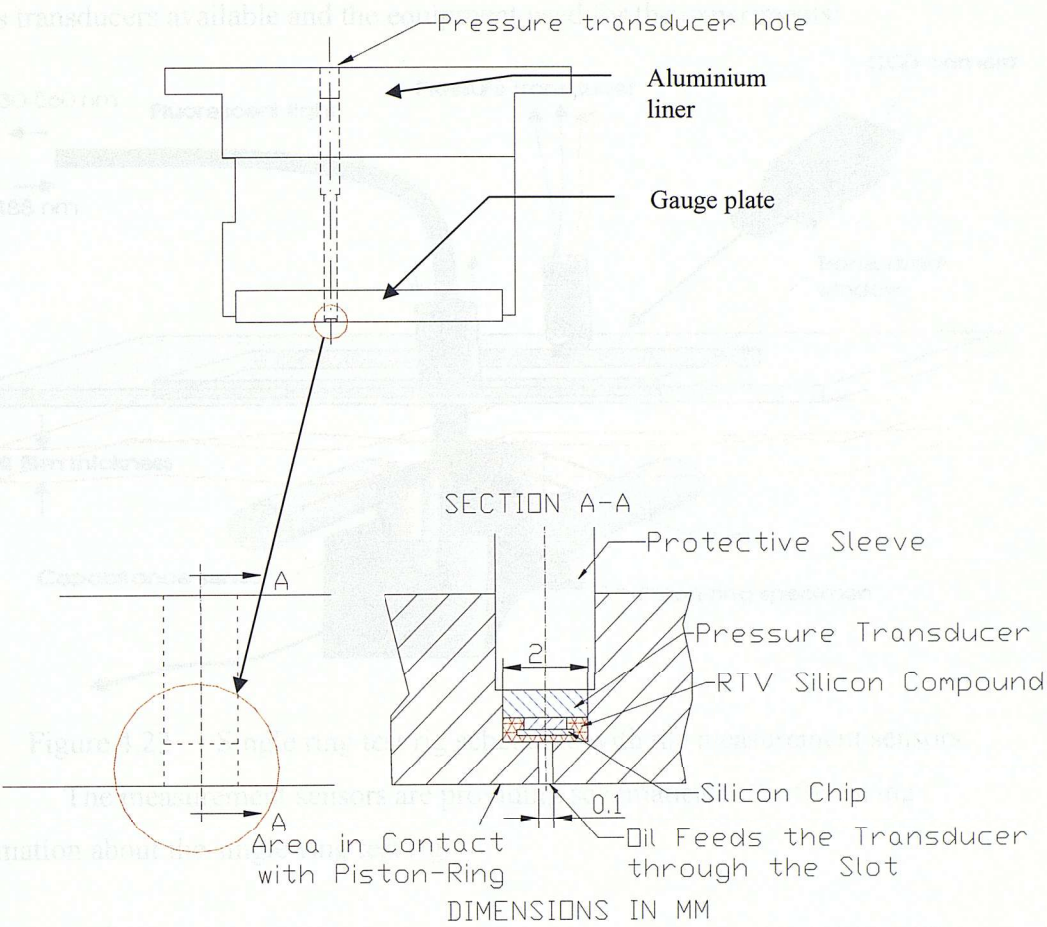


Figure 4.21 Mounting of pressure transducer in the aluminium liner

After the successful modifications took place, the single ring test rig can provide information about all aspects of cavitation phenomena. The following schematic shows transducers available and the equipment used for the experiments:

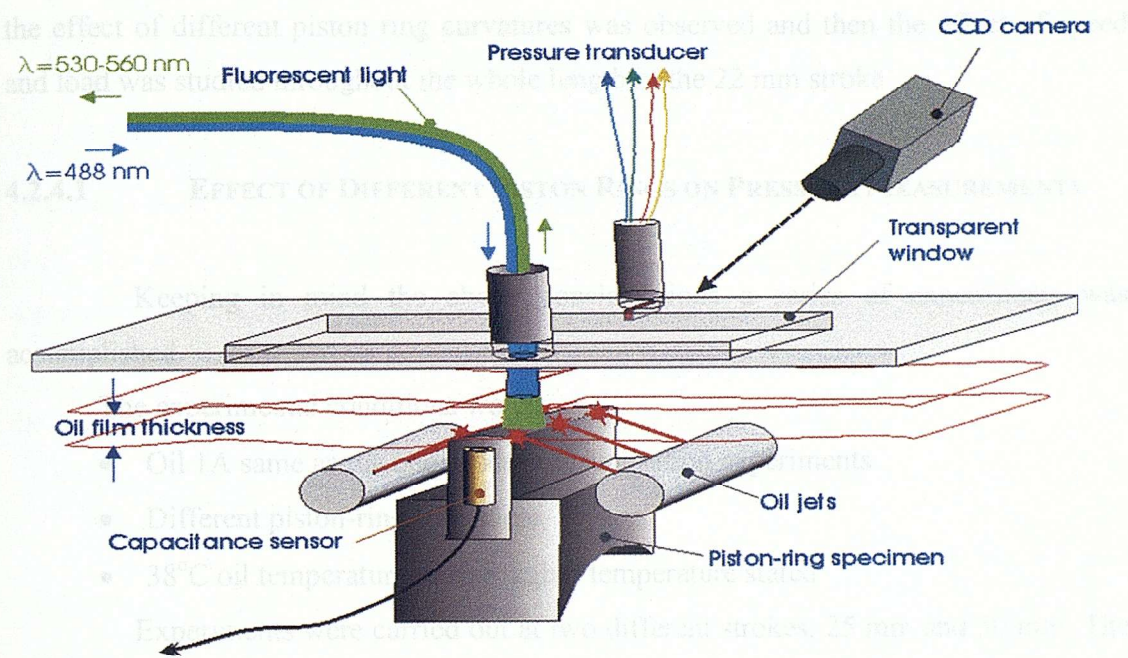


Figure 4.22 Single ring test rig schematic with the measurement sensors

The measurement sensors are providing schematically the following information about the single-ring test rig:

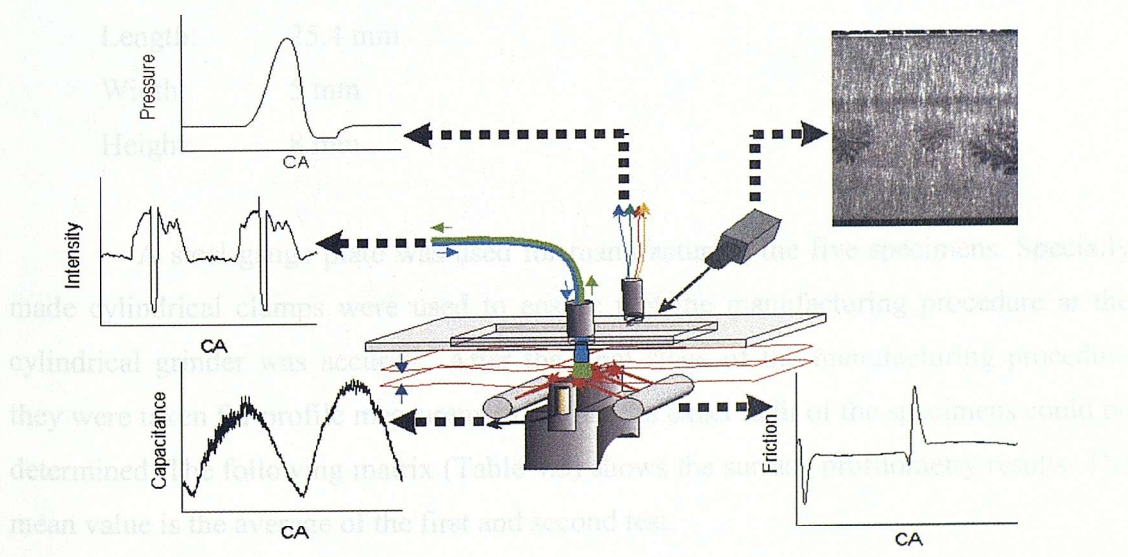


Figure 4.23 The installed transducers' output

4.2.4 OIL FILM PRESSURE MEASUREMENTS PART I

The pressure measurement experiments were accomplished in two series. First the effect of different piston ring curvatures was observed and then the effect of speed and load was studied throughout the whole length of the 22 mm stroke.

4.2.4.1 EFFECT OF DIFFERENT PISTON RINGS ON PRESSURE MEASUREMENTS

Keeping in mind the above considerations a series of experiments was accomplished.

The experimental conditions were:

- Oil 1A same as the one used for visualisation experiments
- Different piston-ring specimens
- 38°C oil temperature unless higher temperature stated

Experiments were carried out at two different strokes, 25 mm and 50 mm. The pressure transducer is set at mid-stroke where in both cases the highest liner speeds are reached.

Five new piston rings were made in a cylindrical grinding machine. The dimensions of all the piston rings are:

|         |         |
|---------|---------|
| Length: | 25.4 mm |
| Width:  | 5 mm    |
| Height: | 8 mm    |

A steel gauge plate was used for manufacturing the five specimens. Specially made cylindrical clamps were used to ensure that the manufacturing procedure at the cylindrical grinder was accurate. After the final stage of the manufacturing procedure they were taken for profile measurement so that the exact radii of the specimens could be determined. The following matrix (Table 4.3) shows the surface profilometry results. The mean value is the average of the first and second test.



| Ring Specimen        | Radius (mm)-First Test | Radius (mm)-Second Test | Radius (mm)-Mean Value |
|----------------------|------------------------|-------------------------|------------------------|
| Small curvature      | 40.0387                | 39.7754                 | 39.907                 |
| Medium curvature (1) | 73.6739                | 73.8940                 | 73.784                 |
| Medium curvature (2) | 79.5018                | 76.8305                 | 78.166                 |
| High curvature       | 106.2673               | 106.9802                | 106.623                |
| Used Piston-Ring     | 81.8560                | 77.1177                 | 79.487                 |

Table 4.3 Piston-ring curvatures

The different piston-ring specimens were used to study the effect of different curvatures on the pressure profile. In figures 4.24 and 4.25 we see the pressure variation at 25 mm and 50 mm stroke respectively for 5 different piston-ring specimens. The repeatability of the signal proved to be very good and in every experimental case 50 cycles were averaged so that the pressure curves could be derived. Figure 4.25 c shows the error bars between consecutive measurements from the acquired cycles.

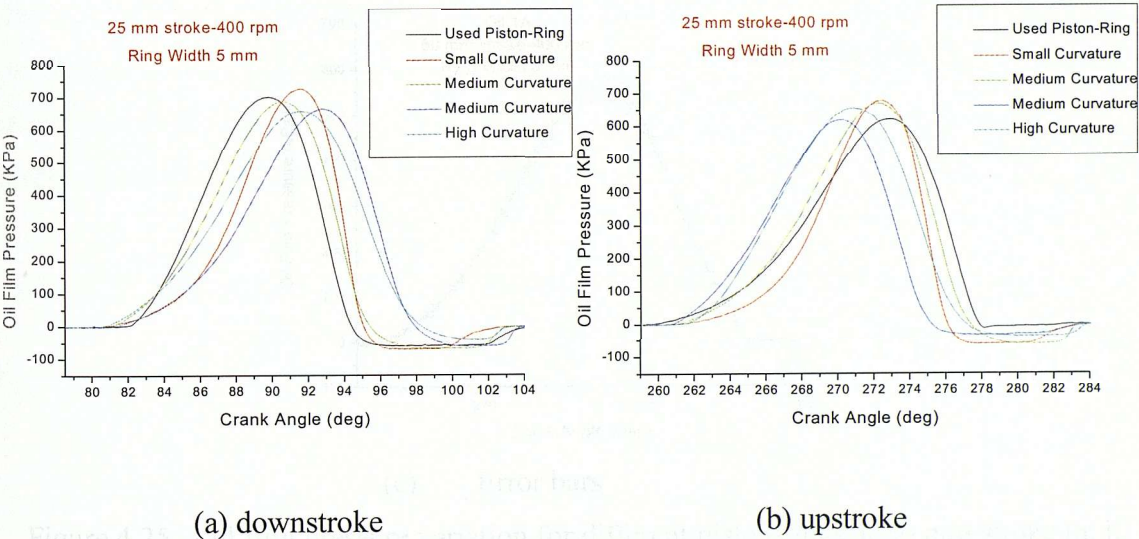


Figure 4.24 Oil film pressure variation for different piston-rings at 25 mm stroke

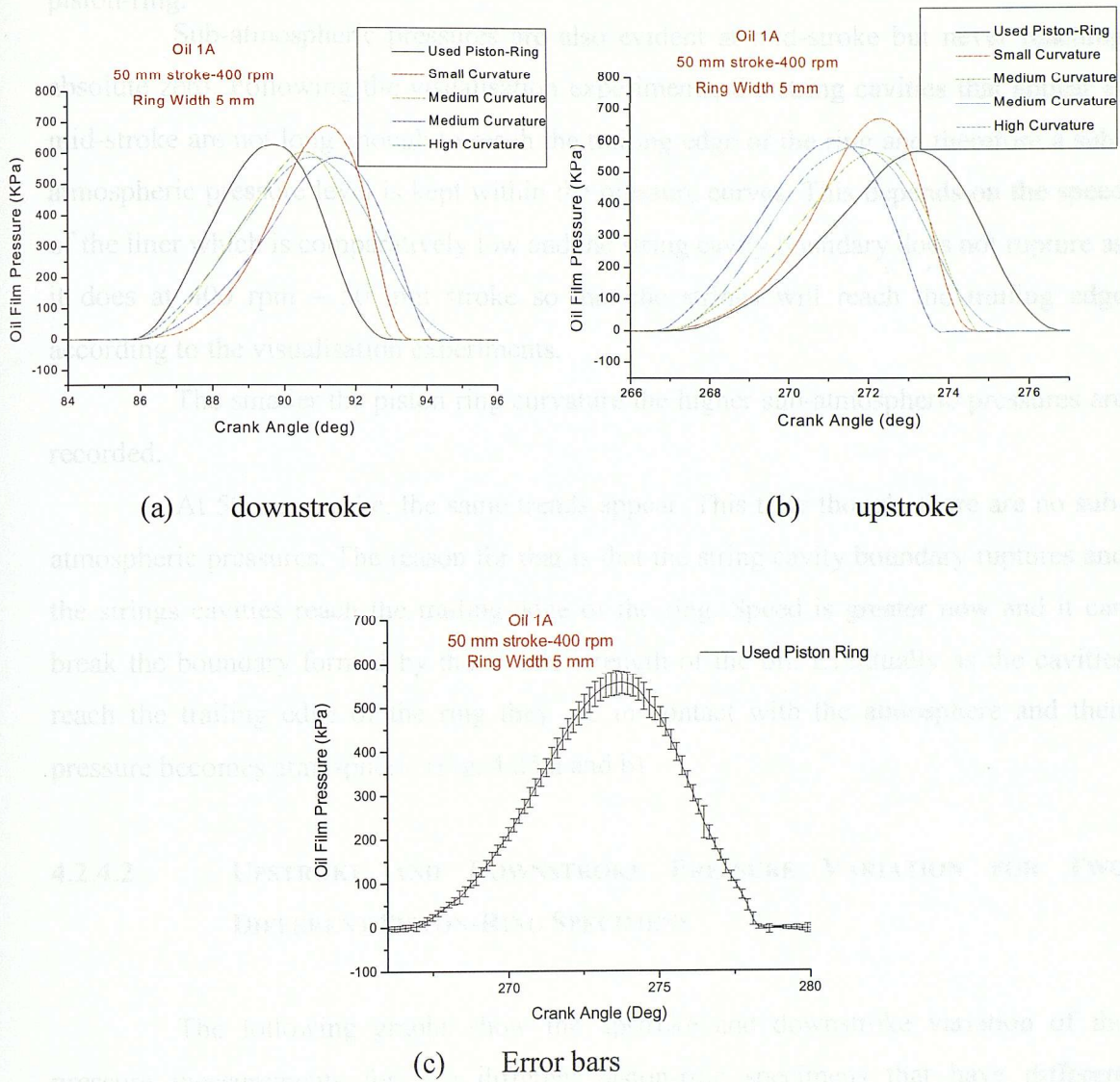


Figure 4.25 Oil film pressure variation for different piston-rings at 50 mm stroke (a, b) - Used piston ring error bars upstroke, based on 20 cycles (c).

The curves have a slightly different central point, because for removing and reinstalling the different piston ring, the liner set-up changes by a small margin, due to re-winding of the threaded link between the liner and the reciprocating mechanism. The threads in both sides of the link (right and left hand) could not ensure the exact position of the measuring point relative to the previous setting.

The crank angle in X axis at all cases is in accordance to the width of the piston-ring.

Sub-atmospheric pressures are also evident at mid-stroke but never reaching absolute zero. Following the visualisation experiments, the string cavities that appear at mid-stroke are not long enough to reach the trailing edge of the ring and therefore a sub-atmospheric pressure level is kept within the pressure curves. This depends on the speed of the liner which is comparatively low and the string cavity boundary does not rupture as it does at 400 rpm – 50 mm stroke so that the strings will reach the trailing edge according to the visualisation experiments.

The smaller the piston ring curvature the higher sub-atmospheric pressures are recorded.

At 50 mm stroke, the same trends appear. This time though, there are no sub-atmospheric pressures. The reason for that is that the string cavity boundary ruptures and the strings cavities reach the trailing edge of the ring. Speed is greater now and it can break the boundary formed by the tensile strength of the oil. Eventually as the cavities reach the trailing edge of the ring they are in contact with the atmosphere and their pressure becomes atmospheric (Fig. 4.25 a and b).

#### **4.2.4.2 UPSTROKE AND DOWNSTROKE PRESSURE VARIATION FOR TWO DIFFERENT PISTON-RING SPECIMENS**

The following graphs show the upstroke and downstroke variation of the pressure measurements for two different piston-ring specimens that have different curvatures. In the next paragraph an upstroke-downstroke comparison is made for the capacitance results and pressure measurements of different piston ring specimens.



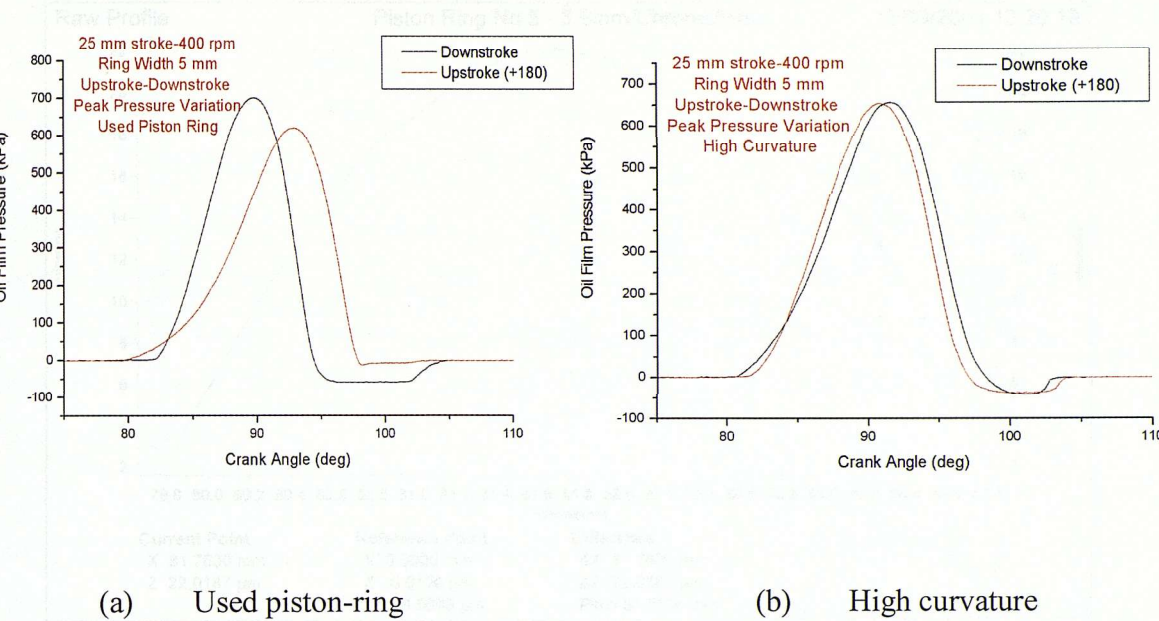


Figure 4.26 Upstroke and downstroke variation of pressure for the used (79.48 mm) and the high curvature piston ring (106.623 mm).

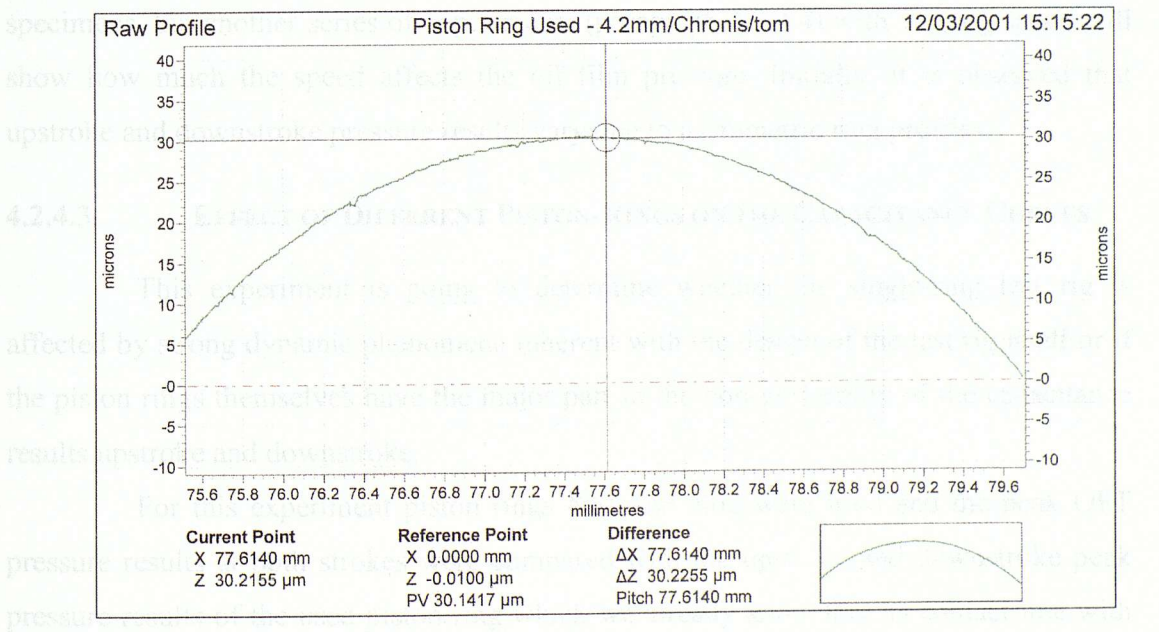


Figure 4.27 Profile of the used piston ring specimen



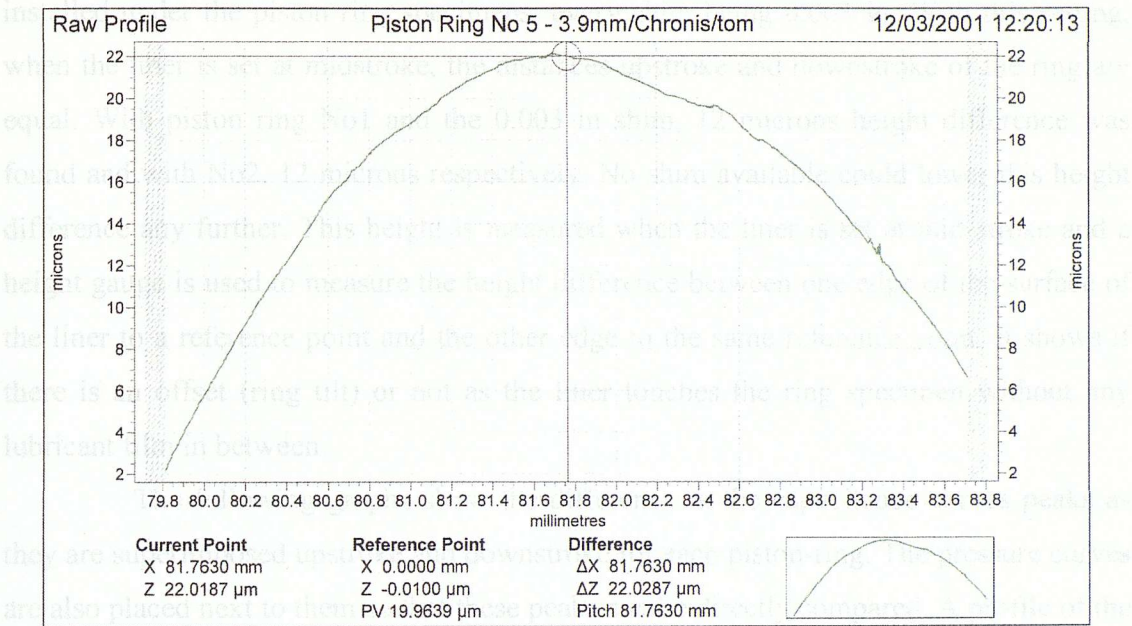


Figure 4.28      Profile of the high curvature piston ring specimen

The peak pressures in Figure 4.26 a and b are comparable for the two ring specimens, but another series of experiments (paragraph 4.2.4.4) with varying speed will show how much the speed affects the oil film pressure. Initially, it is observed that upstroke and downstroke pressure results vary due to asymmetric ring profiles.

4.2.4.3      EFFECT OF DIFFERENT PISTON-RINGS ON THE CAPACITANCE CURVES

This experiment is going to determine whether the single-ring test rig is affected by strong dynamic phenomena inherent with the design of the test rig itself or if the piston rings themselves have the major part in the non-uniformity of the capacitance results upstroke and downstroke.

For this experiment piston rings No1 and No2 were used and the peak OFT pressure results at both strokes were compared with the upstroke and downstroke peak pressure results of the used piston ring which we already know that its contact line with the liner specimen is non symmetric. Their surface curvature is shown at Table 4.3.

These piston rings are:

Used piston-ring, Piston-Ring No1, Piston-Ring No2 and for every experiment we made sure that they are as flat as possible in reference to the liner surface. Plastic shims were

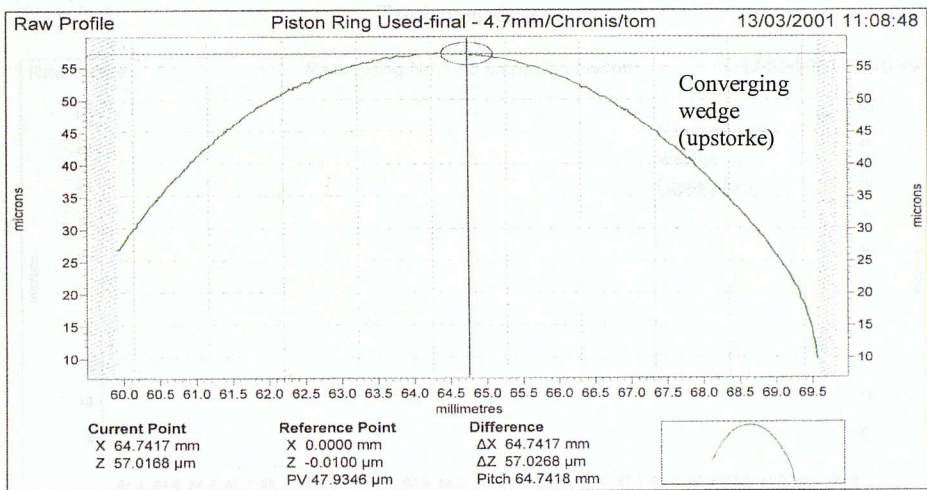
installed under the piston ring specimens, every shim being 0.003 in. With this setting, when the liner is set at midstroke, the distances upstroke and downstroke of the ring are equal. With piston ring No1 and the 0.003 in shim, 12 microns height difference was found and with No2, 12 microns respectively. No shim available could lower this height difference any further. This height is measured when the liner is set at mid-stroke and a height gauge is used to measure the height difference between one edge of the surface of the liner to a reference point and the other edge to the same reference point. It shows if there is an offset (ring tilt) or not as the liner touches the ring specimen without any lubricant film in between.

The following graphs show the difference in the capacitance curves peaks as they are superimposed upstroke and downstroke for each piston-ring. The pressure curves are also placed next to them so that these peaks can be directly compared. A profile of the piston rings is represented as well, so that great differences on the curvatures of the piston-rings can be observed between the converging and diverging wedge.

The differences in the peak pressures between upstroke and downstroke for the same piston ring specimen, as in Figure 4.26a, is not only attributed to the dynamic phenomena concerning the oscillatory mechanism of the idealised single-ring test rig or the fact that the ring is not perfectly in line with the liner surface (i.e. shims will have to be used under the piston-ring specimen). The Talysurf profilometry (Figures 4.29a, 4.30a and 4.31a) showed that the high curvature piston ring (symmetric) had exactly similar results upstroke and downstroke in contrast to the used piston ring. This urges us to assume that the variation noticed in the results for different piston ring specimens upstroke and downstroke is also due to profile asymmetry of the piston ring.

Used Piston-Ring

Taylor Hobson



(a) profile of the used piston-ring

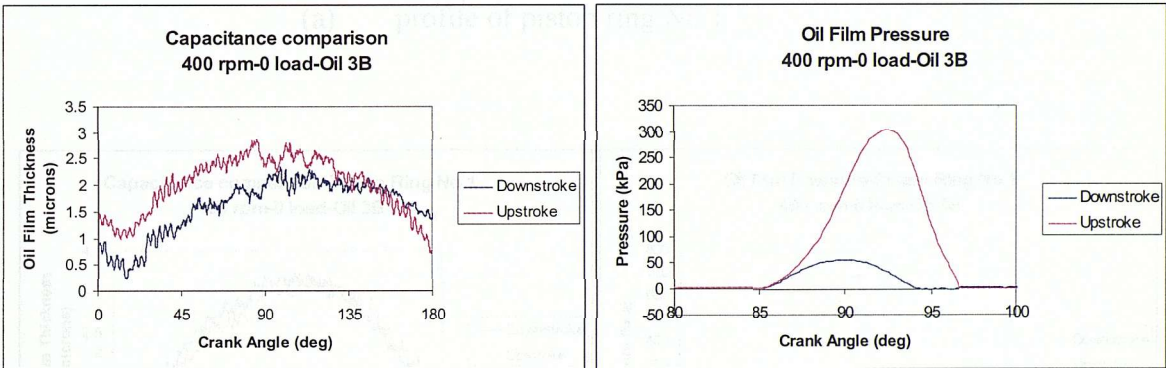
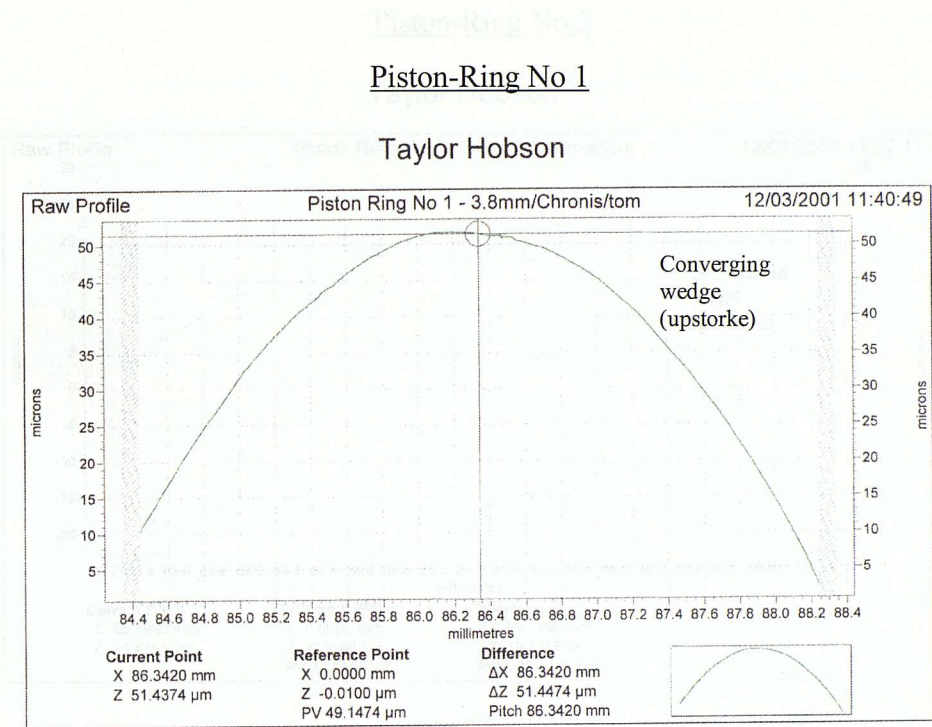


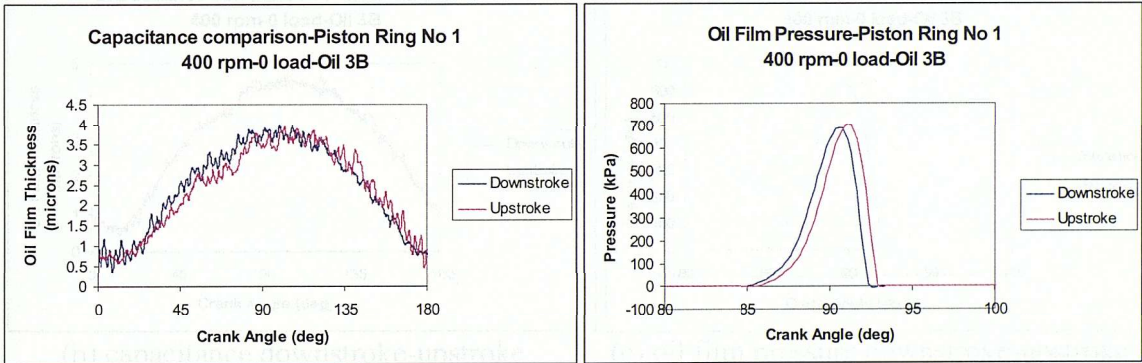
Figure 4.29 Used Piston-Ring Capacitance and oil film pressure results for oil 3B at 400 rpm, 971 N/m.

For the used piston ring, the inclination of the converging wedge is  $\phi_{\text{conv-upstroke}} = 44.71^\circ$  whereas the respective angle for the diverging wedge is  $\phi_{\text{div-upstroke}} = 27.47^\circ$ . The converging wedge is shown in Figure 4.29 a for the upstroke case test.





(a) profile of piston ring No 1



(b) capacitance downstroke-upstroke      (c) oil film pressure downstroke-upstroke

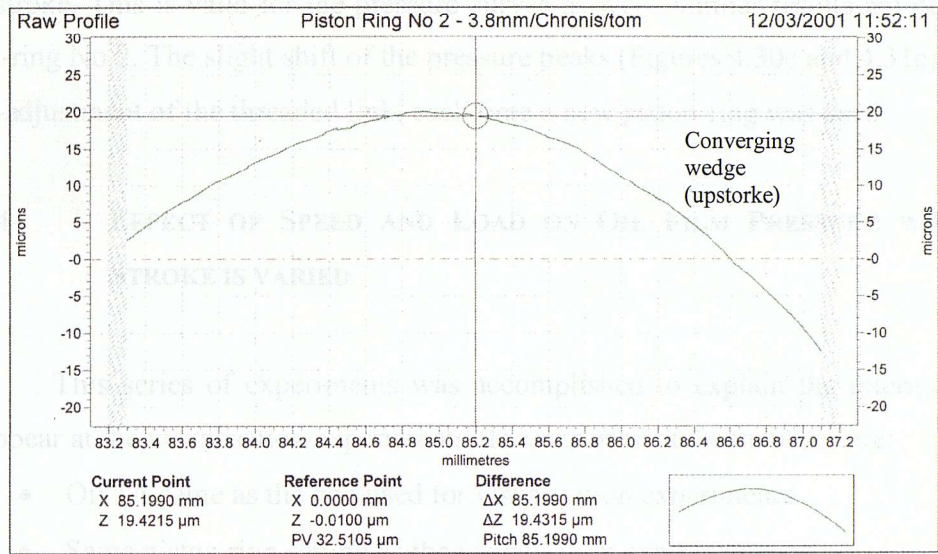
Figure 4.30    Piston-Ring No 1 Capacitance and oil film pressure results for oil 3B at 400 rpm, 971 N/m.

For piston ring No 1, the inclination measurements are  $\varphi_{\text{conv-upstroke}} = 33.14^\circ$  and  $\varphi_{\text{divg-upstroke}} = 33.50^\circ$ . Figure 4.30 a, shows the converging wedge for the upstroke tests.

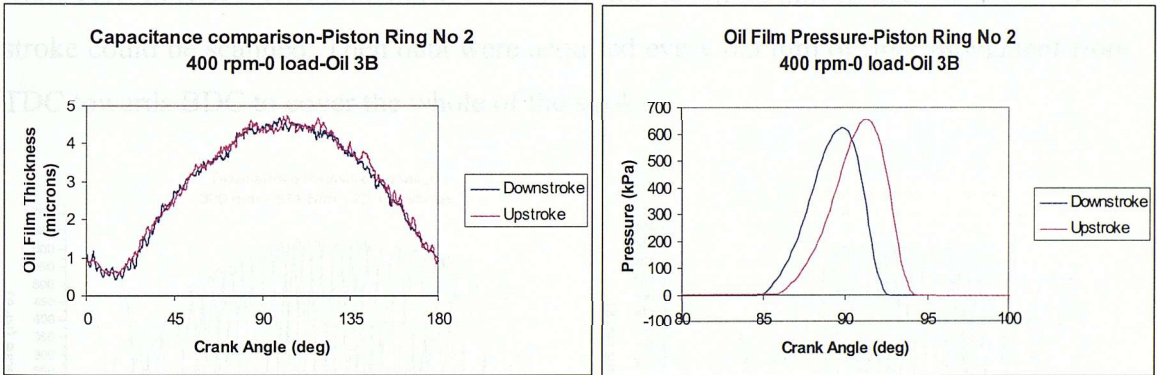


Piston-Ring No 2

Taylor Hobson



(a) profile of piston ring No 2



(b) capacitance downstroke-upstroke

(c) oil film pressure downstroke-upstroke

Figure 4.31 Piston-Ring No 2 Capacitance and oil film pressure results for oil 3B at 400 rpm, 971 N/m.

For piston ring No2,  $\phi_{\text{conv-upstroke}} = 22.29^\circ$  and  $\phi_{\text{divg-upstroke}} = 18.77^\circ$ . Figure 4.31 a, shows the converging wedge for the upstroke.

Figure 4.29 shows that the used piston-ring with the non uniform wear (contact line is closer to one end than the other) has an effect on the upstroke and downstroke

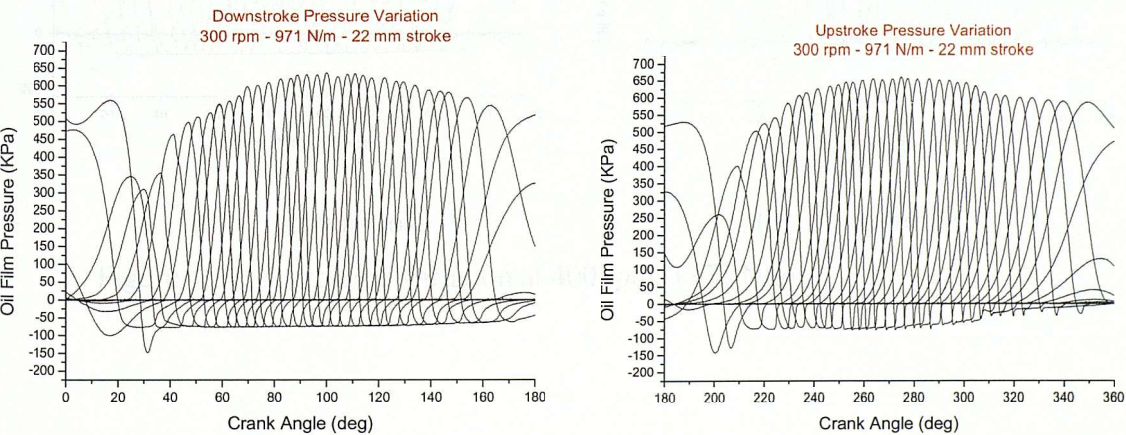
pressures resulting in respective differences on the capacitance signal. In Figure 4.30, piston ring No 1 is giving almost identical results when compared upstroke and downstroke. This is valid for the pressure curves as well. Similar results are derived for piston-ring No 2. The slight shift of the pressure peaks (Figures 4.30c and 4.31c) is due to the re-adjustment of the threaded link, each time a new piston-ring was used.

4.2.4.4 EFFECT OF SPEED AND LOAD ON OIL FILM PRESSURE WHILE THE STROKE IS VARIED

This series of experiments was accomplished to explain the internal cavities that appear at the visualisation experiments. The experimental conditions are:

- Oil 1A same as the one used for visualisation experiments
- Same piston ring setting as the visualisation experiments
- 38°C oil temperature

The effect of load is studied and partly that of speed while the stroke is varied from TDC to BDC. This time the stroke length was set to 22 mm so that the whole of the stroke could be scanned. Then data were acquired every 0.5 mm of liner movement from TDC towards BDC to cover the whole of the stroke.



(a) downstroke (b) upstroke

Figure 4.32 Pressure variation at 300 rpm, 971 N/m



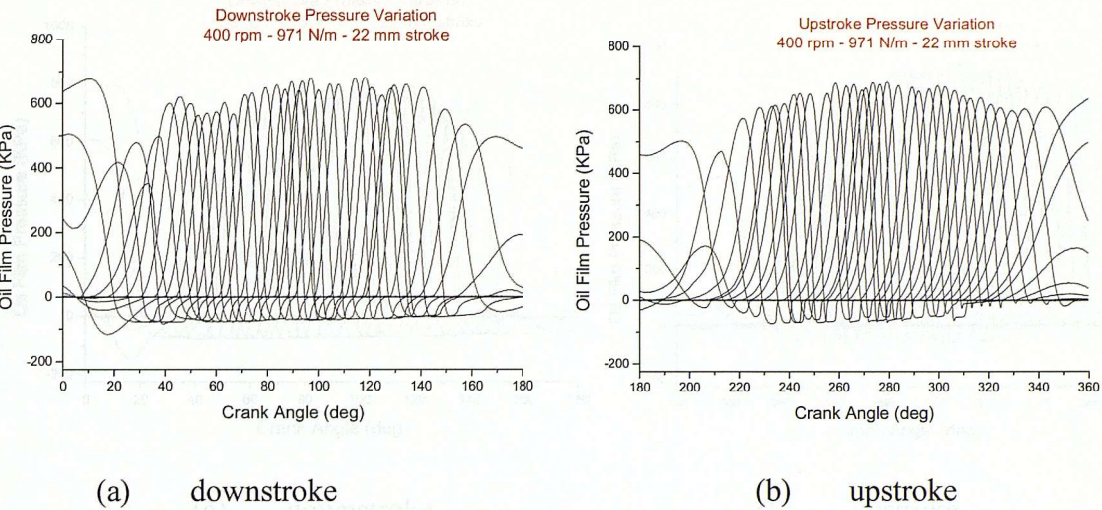


Figure 4.33 Pressure variation at 400 rpm, 971 N/m

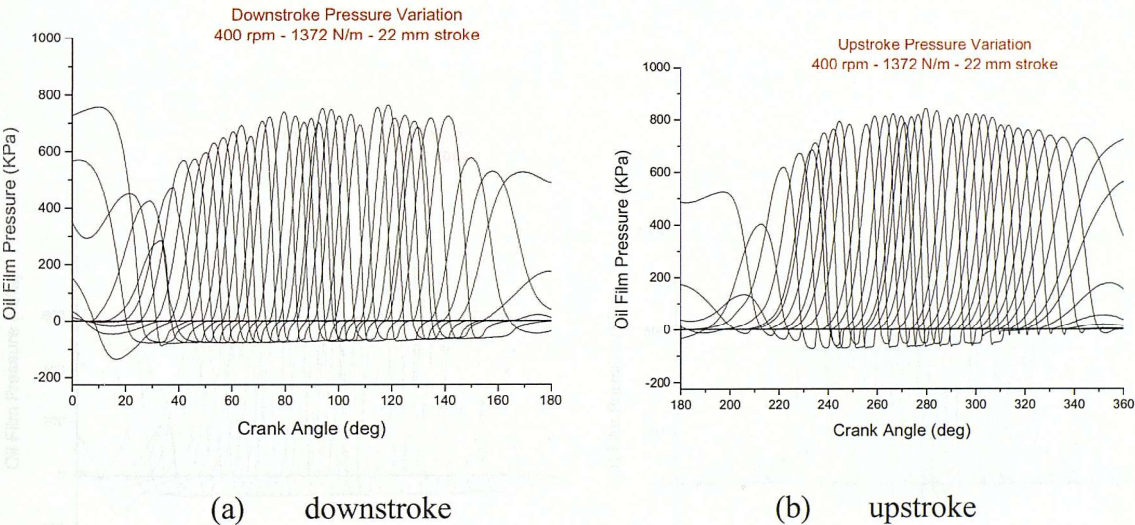


Figure 4.34 Pressure variation at 400 rpm, 1372 N/m

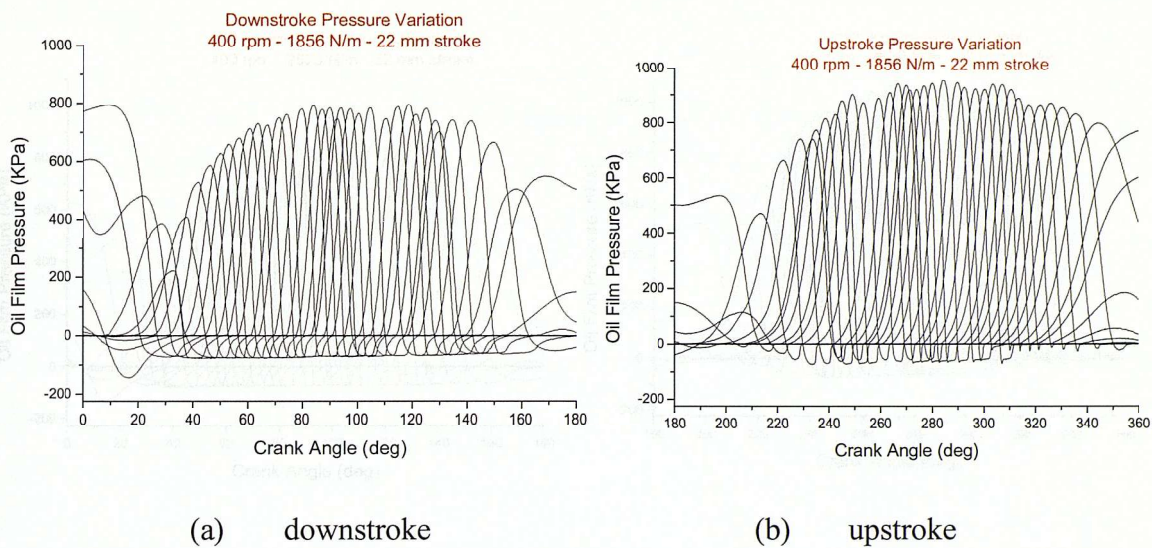


Figure 4.37 Pressure variation at 400 rpm, 2323 N/m

Figure 4.35 Pressure variation at 400 rpm, 1856 N/m

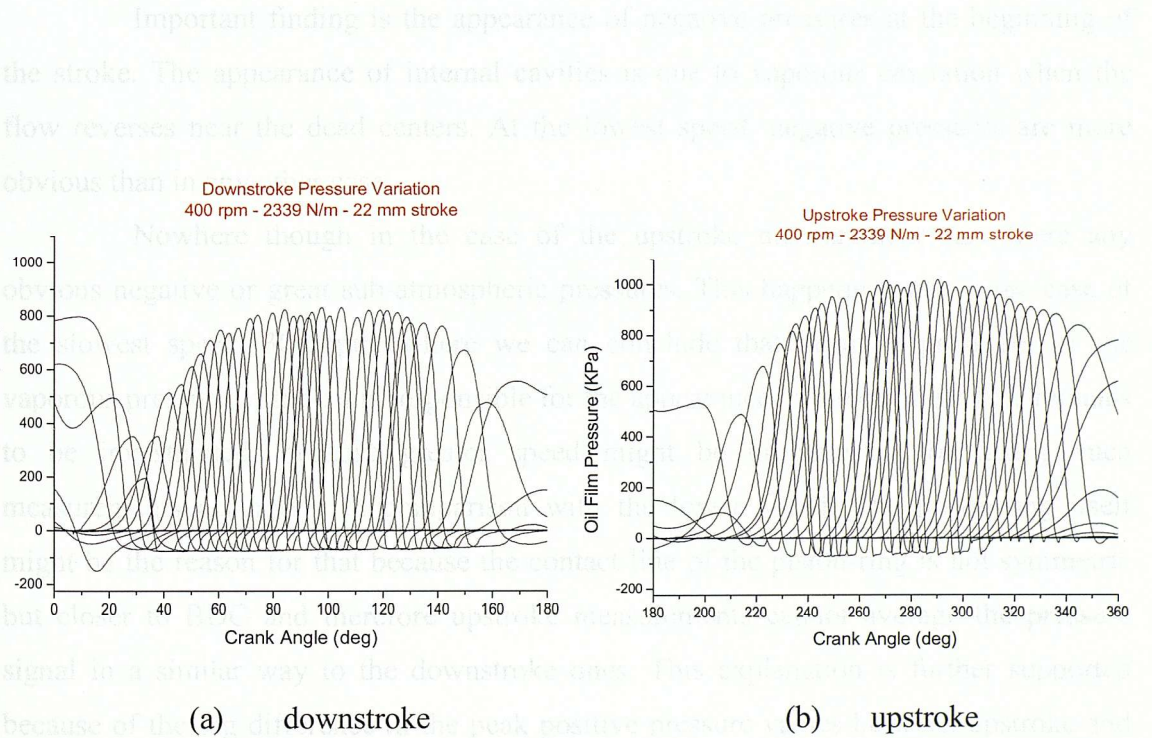


Figure 4.36 Pressure variation at 400 rpm, 2339 N/m

Figure 4.35 Pressure variation at 400 rpm, 1856 N/m



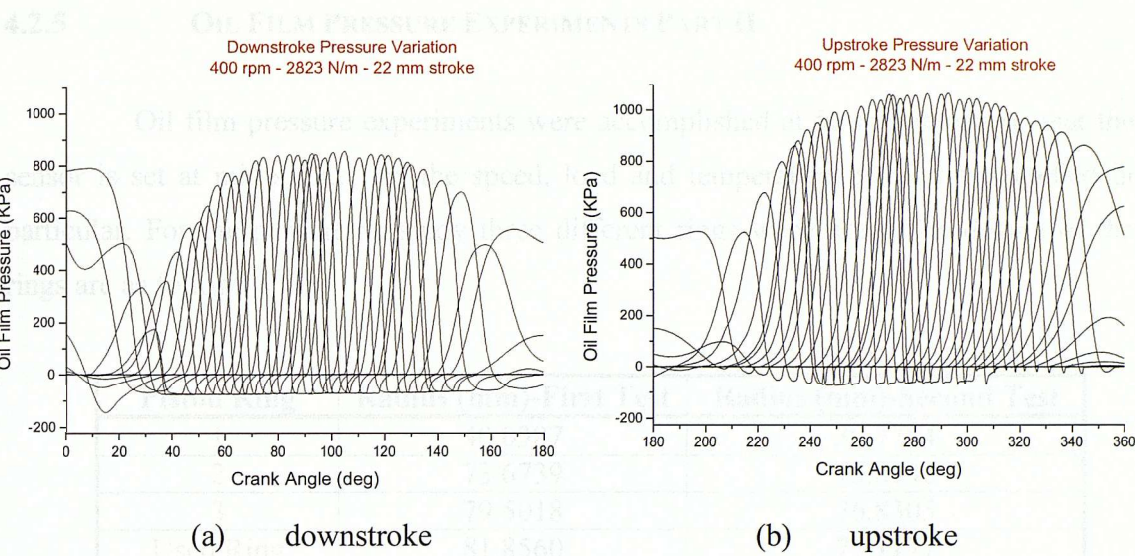


Figure 4.37 Pressure variation at 400 rpm, 2823 N/m

Important finding is the appearance of negative pressures at the beginning of the stroke. The appearance of internal cavities is due to vaporous cavitation when the flow reverses near the dead centers. At the lowest speed, negative pressures are more obvious than in any other case.

Nowhere though in the case of the upstroke measurements are there any obvious negative or great sub-atmospheric pressures. This happened only in the case of the slowest speed, 300 rpm, where we can conclude that negative pressures or the vaporous pressure of the oil is responsible for the appearance of the cavities. This remains to be investigated, because greater speed might be problematic for taking such measurements accurately in comparison with the lower speed. The piston-ring itself might be the reason for that because the contact line of the piston-ring is not symmetric but closer to BDC and therefore upstroke measurements cannot average the pressure signal in a similar way to the downstroke ones. This explanation is further supported because of the big difference in the peak positive pressure values between upstroke and downstroke.

The negative pressure in Figures 4.32, 4.33, 4.34, 4.35 and 4.36 is moving earlier in the stroke. As we saw in the visualisation and friction experiments as load increases the flow reverses earlier in the stroke. The fact that we have pressures below zero (i.e. negative) shows that we are moving towards the tensile strength of the liquid.

4.2.5 OIL FILM PRESSURE EXPERIMENTS PART II

Oil film pressure experiments were accomplished at 50 mm stroke so that the sensor is set at mid-stroke and the speed, load and temperature effect were studied in particular. For the parametric study three different rings were used. The details of the rings are as follows:

| Piston Ring | Radius (mm)-First Test | Radius (mm)-Second Test |
|-------------|------------------------|-------------------------|
| 1           | 40.0387                | 39.7754                 |
| 2           | 73.6739                | 73.8940                 |
| 3           | 79.5018                | 76.8305                 |
| Used Ring   | 81.8560                | 77.1177                 |

Table 4.4      Piston-ring curvatures used for pressure experiments part II

4.2.5.1 PISTON RING NO 1

Tests were carried out at the following conditions:

| Speed @ 971 N/m | Load @ 400 RPM | Temp @ 400 RPM |
|-----------------|----------------|----------------|
| 100 rpm         | 971 N/m        | 65 °C          |
| 200 rpm         | 1372 N/m       |                |
| 300 rpm         | 1856 N/m       |                |
| 400 rpm         | 2339 N/m       |                |
| 500 rpm         | 2823 N/m       |                |
| 600 rpm         |                |                |

Table 4.5      Test Matrix for Piston Ring No 1



4.2.5.1.1 SPEED EFFECT

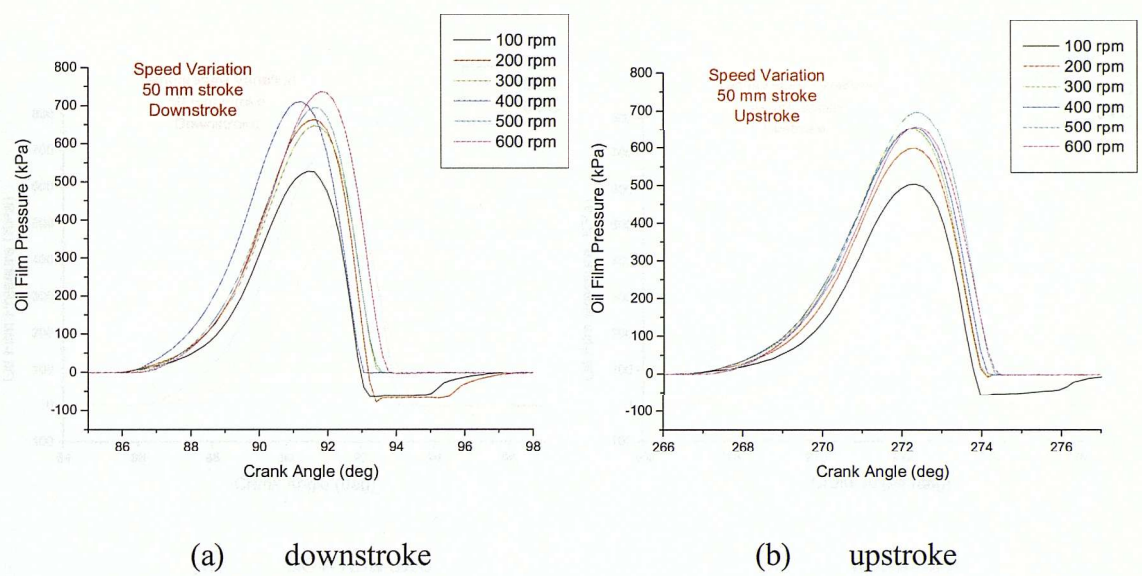


Figure 4.38 Pressure measurements: Speed effect on piston ring No1, Oil 1A, 971 N/m

4.2.5.1.2 LOAD EFFECT

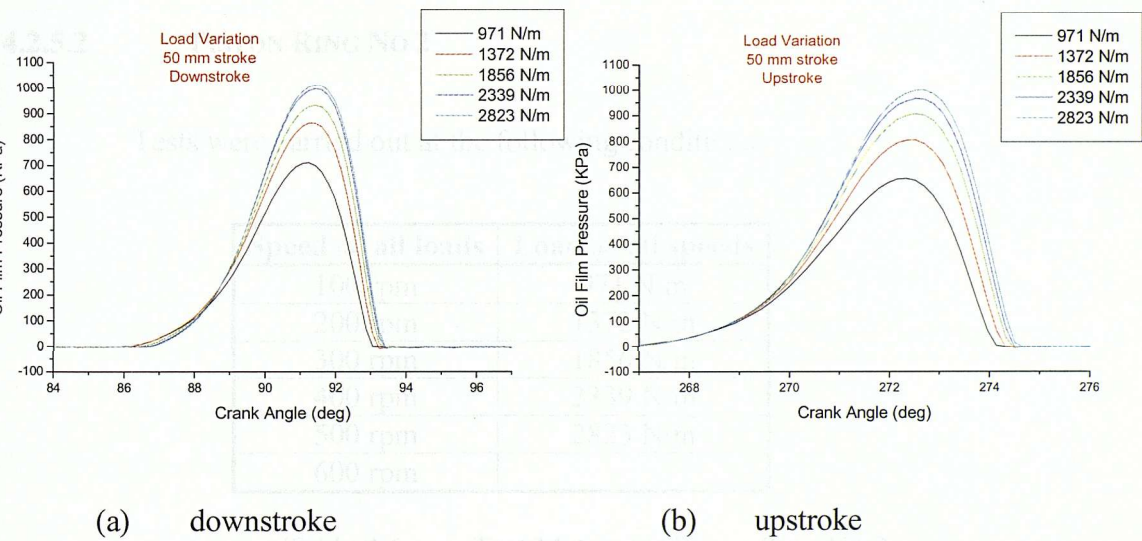


Figure 4.39 Pressure measurements: Load effect on piston ring No1, Oil 1A, 50 mm stroke, 400 rpm

4.2.5.1.3      TEMPERATURE EFFECT

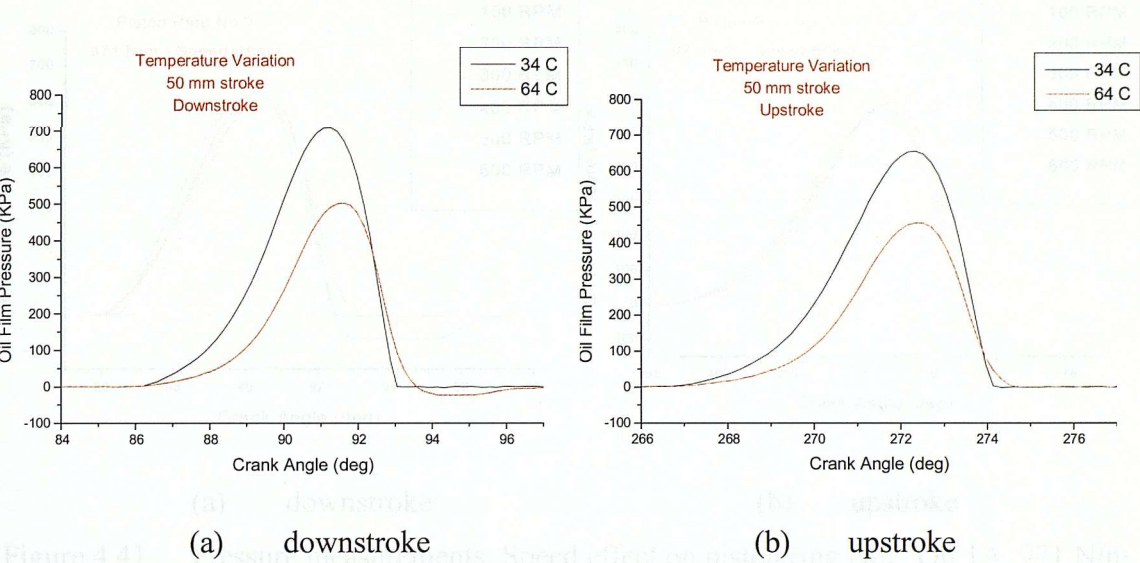


Figure 4.40      Pressure measurements: Temperature effect on piston ring No1, Oil 1A, 971 N/m, 50 mm stroke at 400 rpm

4.2.5.2      PISTON RING NO 2

Tests were carried out at the following conditions:

| Speed @ all loads | Load @ all speeds |
|-------------------|-------------------|
| 100 rpm           | 971 N/m           |
| 200 rpm           | 1372 N/m          |
| 300 rpm           | 1856 N/m          |
| 400 rpm           | 2339 N/m          |
| 500 rpm           | 2823 N/m          |
| 600 rpm           |                   |

Table 4.6      Test Matrix for Piston Ring No 2



4.2.5.2.1 SPEED EFFECT

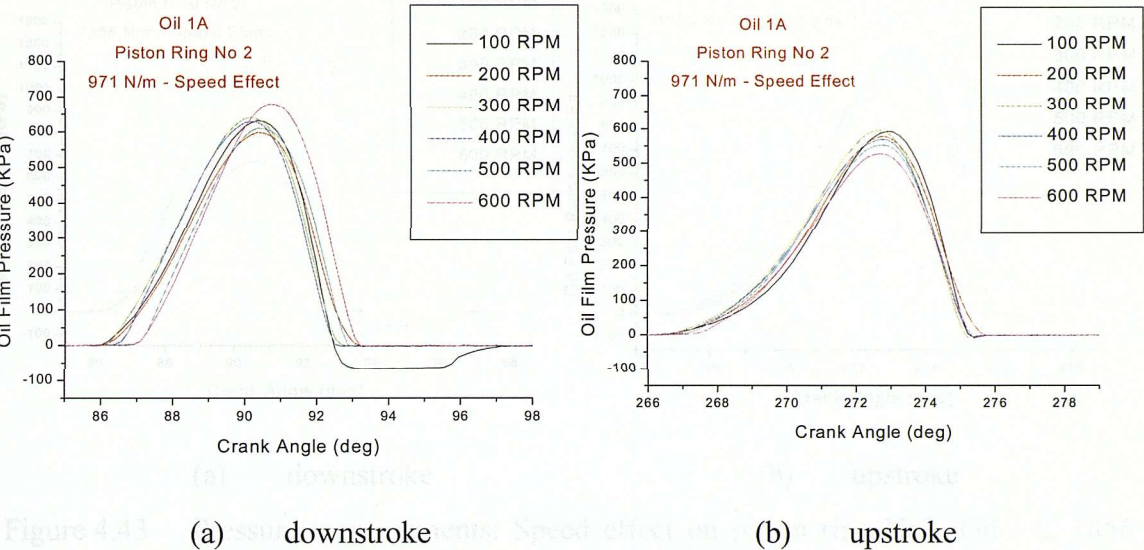


Figure 4.41 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 971 N/m

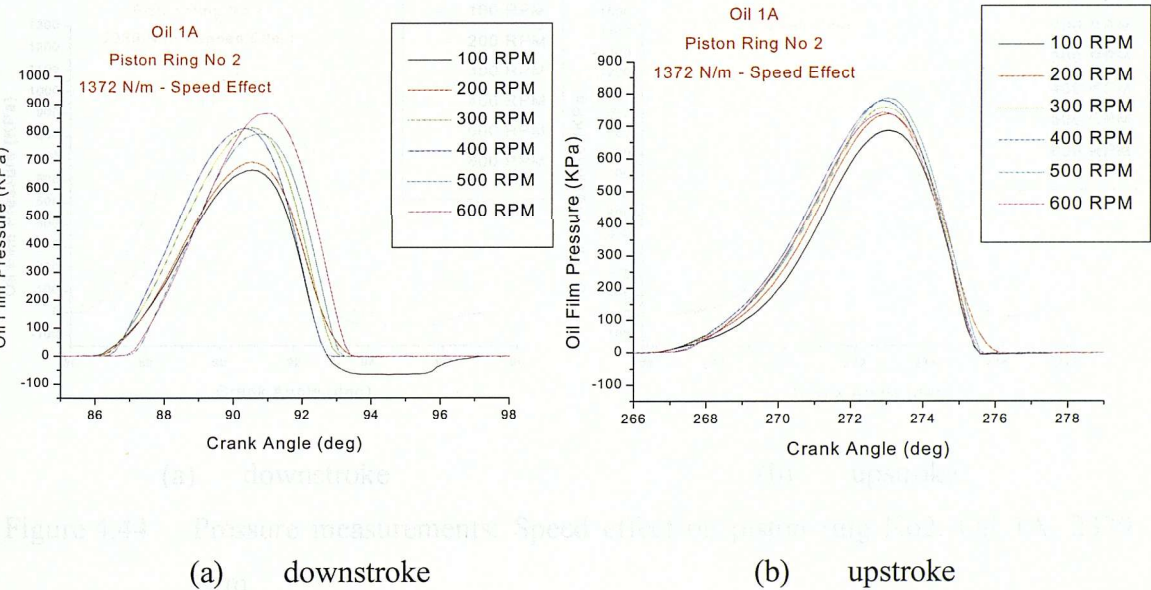


Figure 4.42 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 1372 N/m

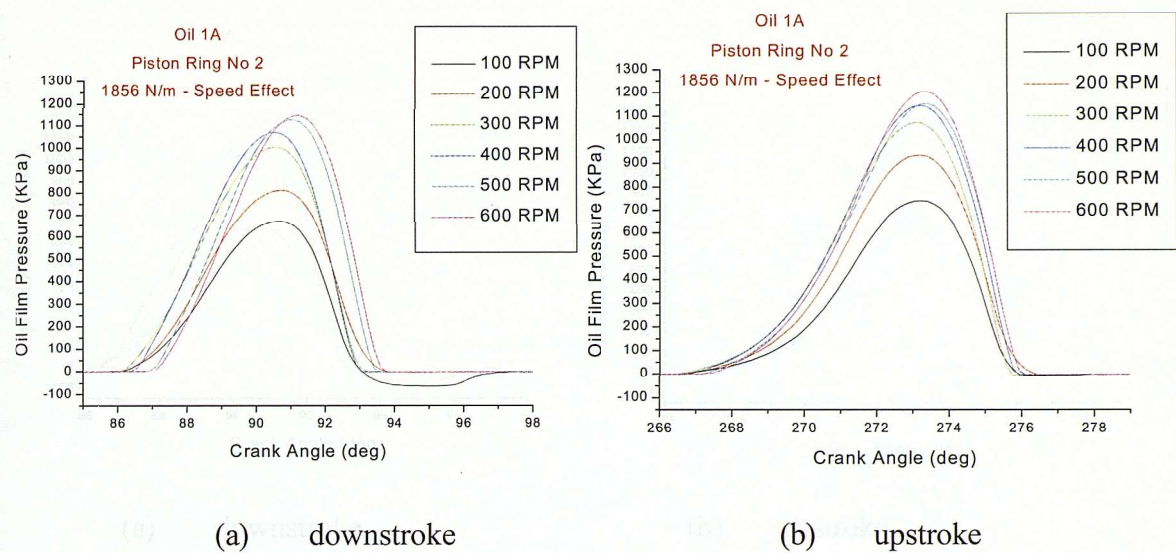


Figure 4.43 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 1856 N/m

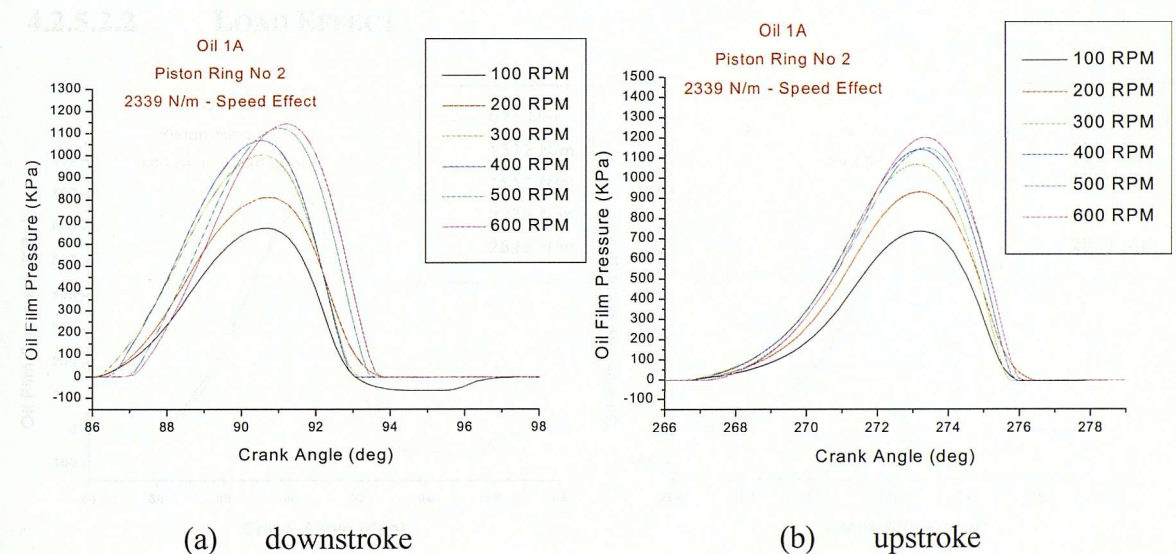


Figure 4.44 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 2339 N/m

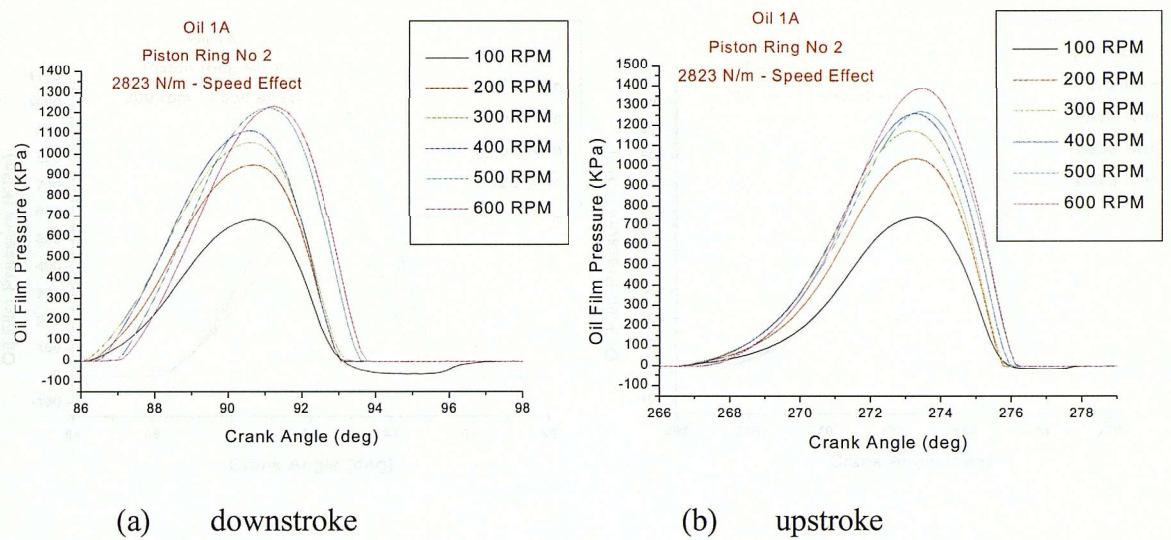


Figure 4.45 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 2823 N/m

4.2.5.2.2 LOAD EFFECT

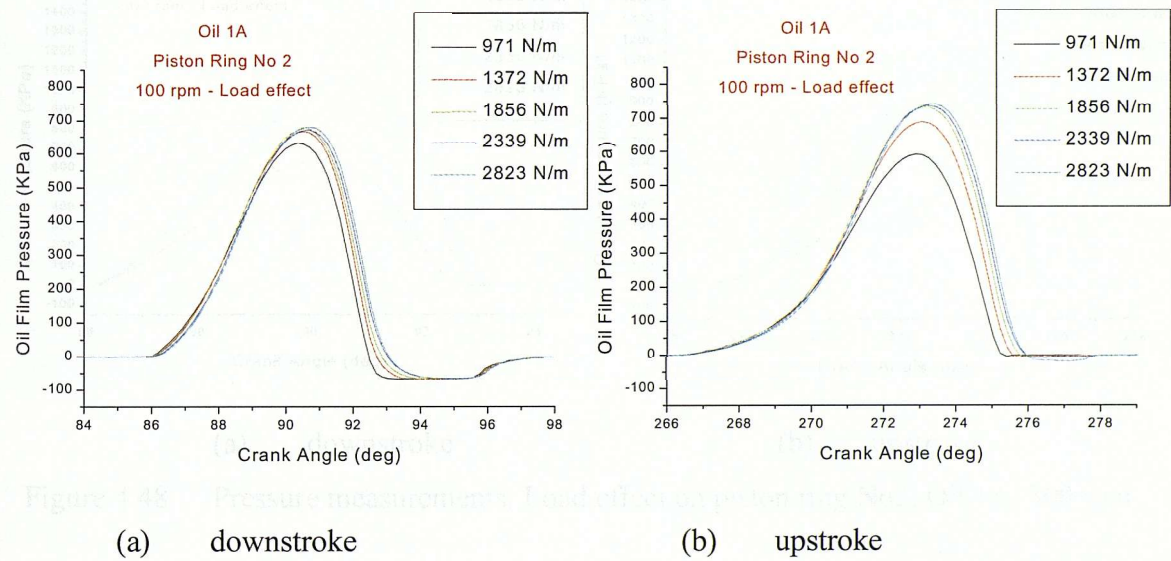


Figure 4.46 Pressure measurements: Load effect on piston ring No2, Oil 1A, 100 rpm



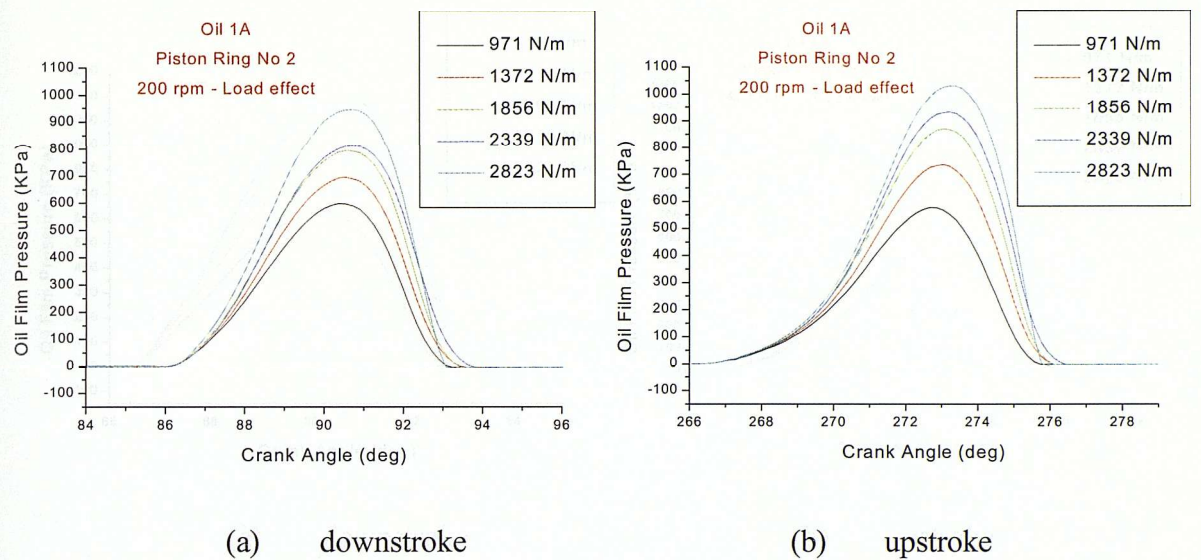


Figure 4.47 Pressure measurements: Load effect on piston ring No2, Oil 1A, 200 rpm

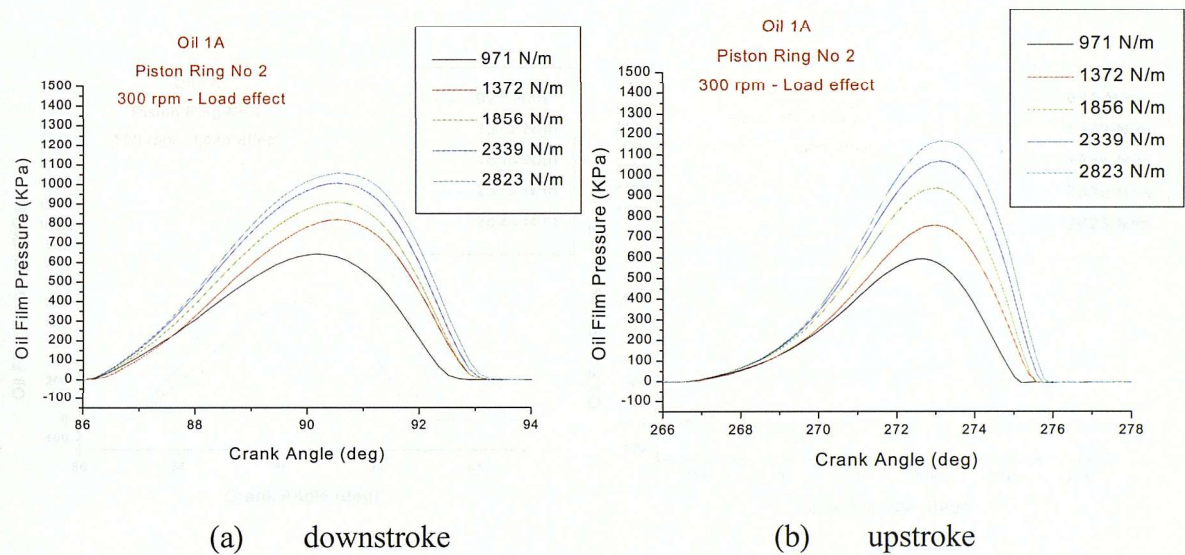


Figure 4.48 Pressure measurements: Load effect on piston ring No2, Oil 1A, 300 rpm



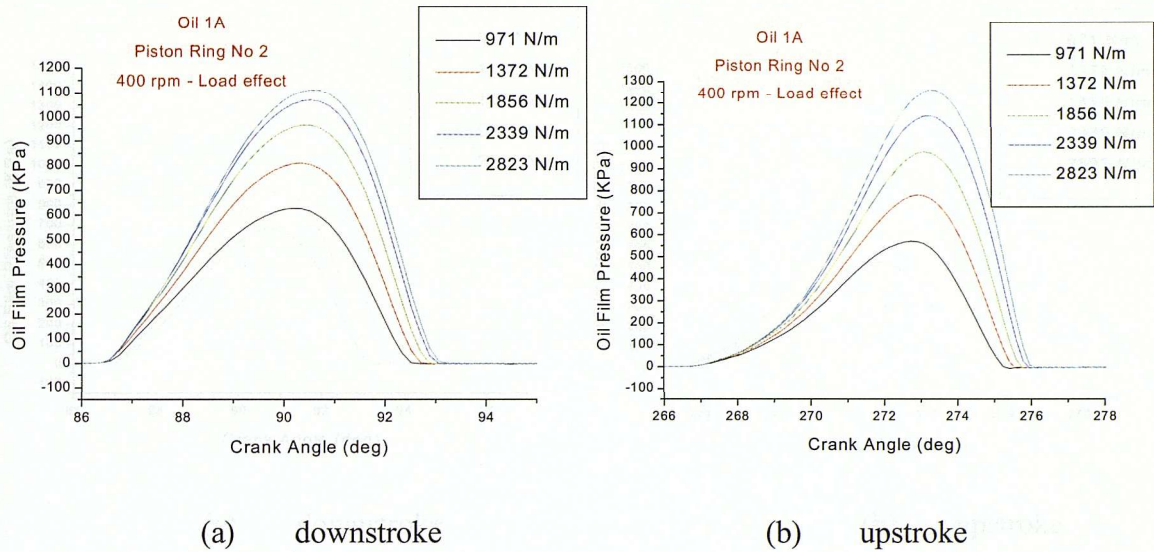


Figure 4.49 Pressure measurements: Load effect on piston ring No2, Oil 1A, 400 rpm

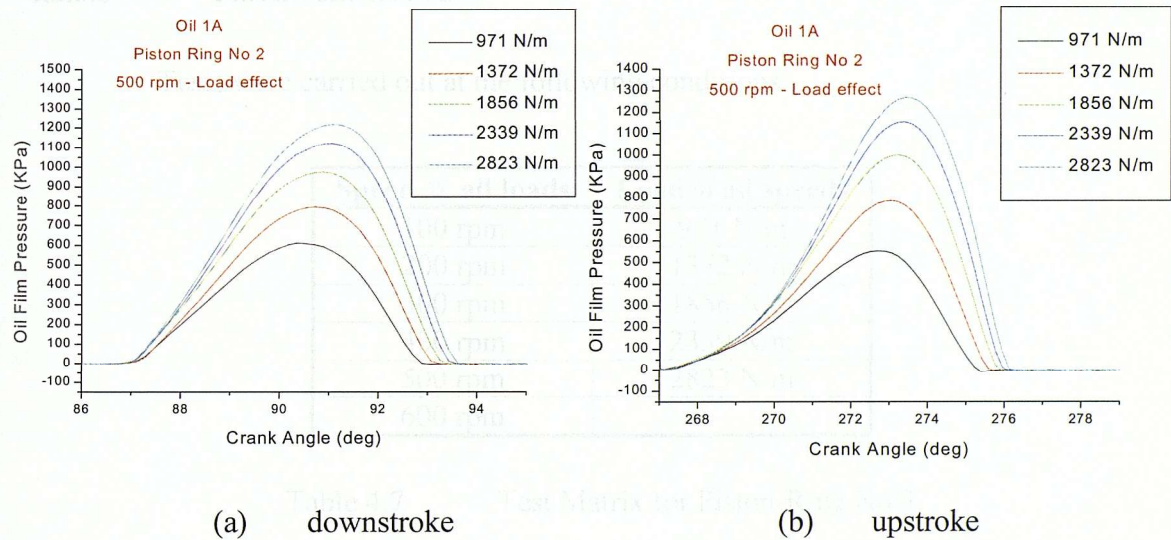


Figure 4.50 Pressure measurements: Load effect on piston ring No2, Oil 1A, 500 rpm

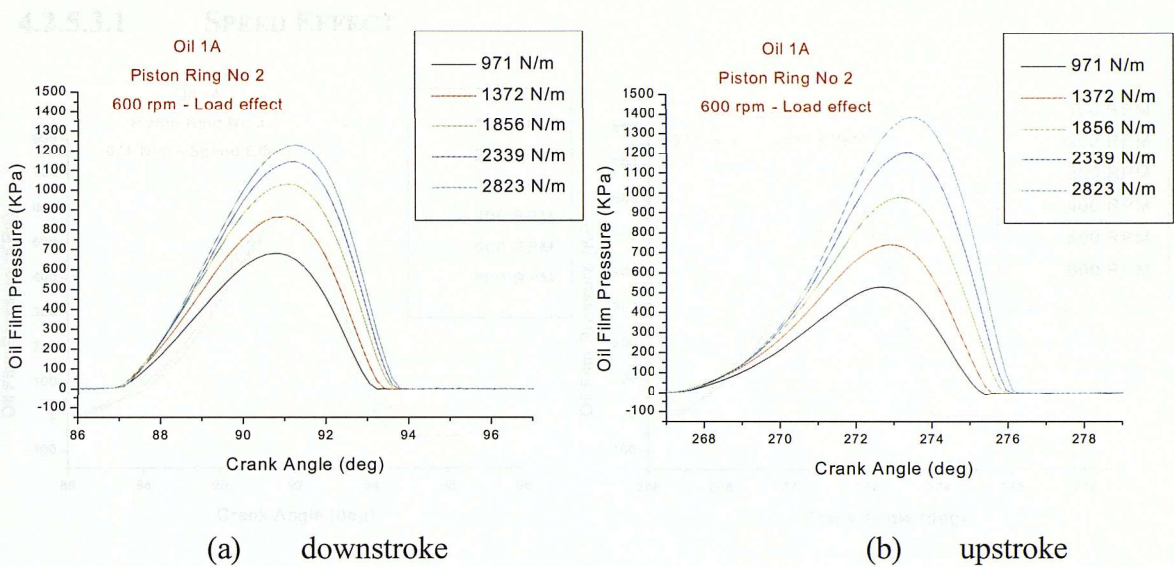


Figure 4.51 Pressure measurements: Load effect on piston ring No2, Oil 1A, 600 rpm

Figure 4.52 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 971 N/m

4.2.5.3 PISTON RING NO 3

Tests were carried out at the following conditions:

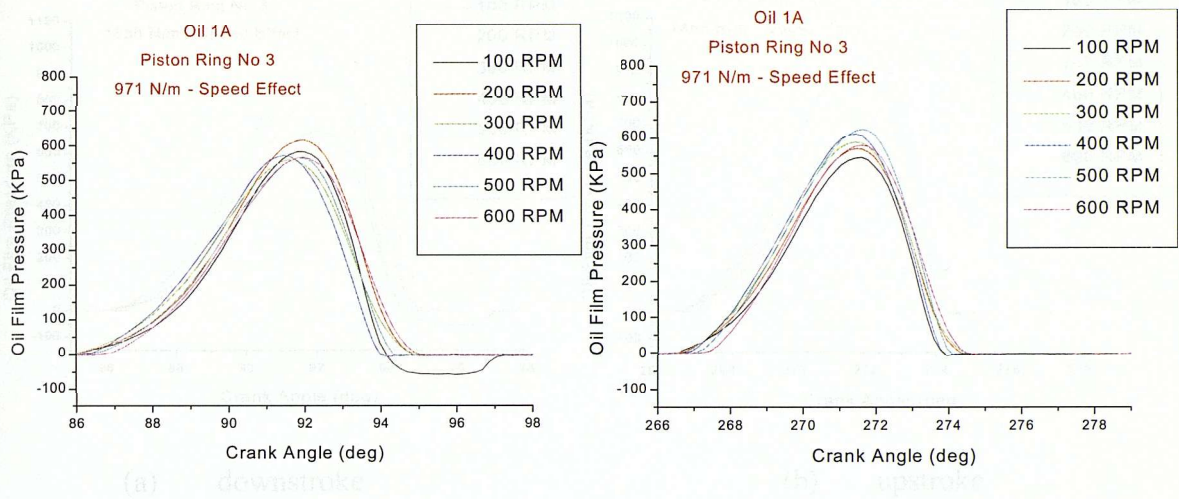
| Speed @ all loads | Load @all speeds |
|-------------------|------------------|
| 100 rpm           | 971 N/m          |
| 200 rpm           | 1372 N/m         |
| 300 rpm           | 1856 N/m         |
| 400 rpm           | 2339 N/m         |
| 500 rpm           | 2823 N/m         |
| 600 rpm           |                  |

Table 4.7 Test Matrix for Piston-Ring No 3



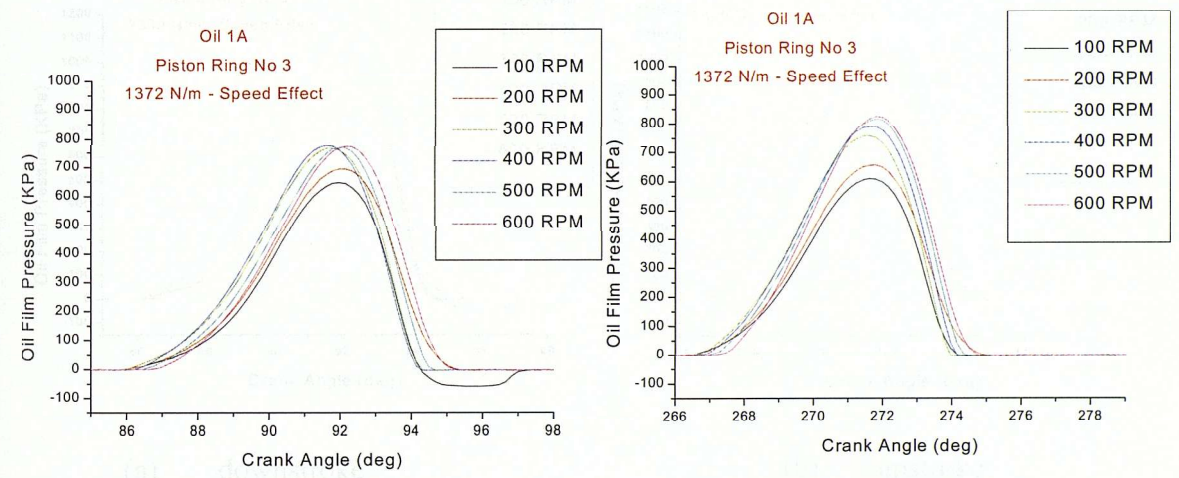
Figure 4.53 Pressure measurements: Speed effect on piston ring No2, Oil 1A, 971 N/m

4.2.5.3.1 SPEED EFFECT



(a) downstroke (b) upstroke

Figure 4.52 Pressure measurements: Speed effect on piston ring No3, Oil 1A, 971 N/m



(a) downstroke (b) upstroke

Figure 4.53 Pressure measurements: Speed effect on piston ring No3, Oil 1A, 1372 N/m



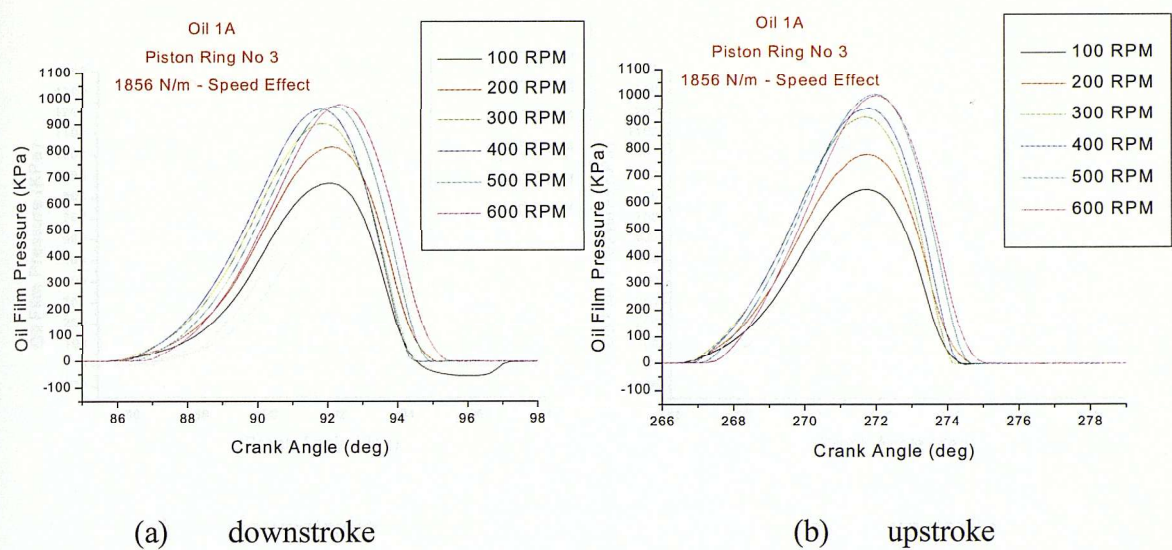


Figure 4.54 Pressure measurements: Speed effect on piston ring No3, Oil 1A, 1856 N/m

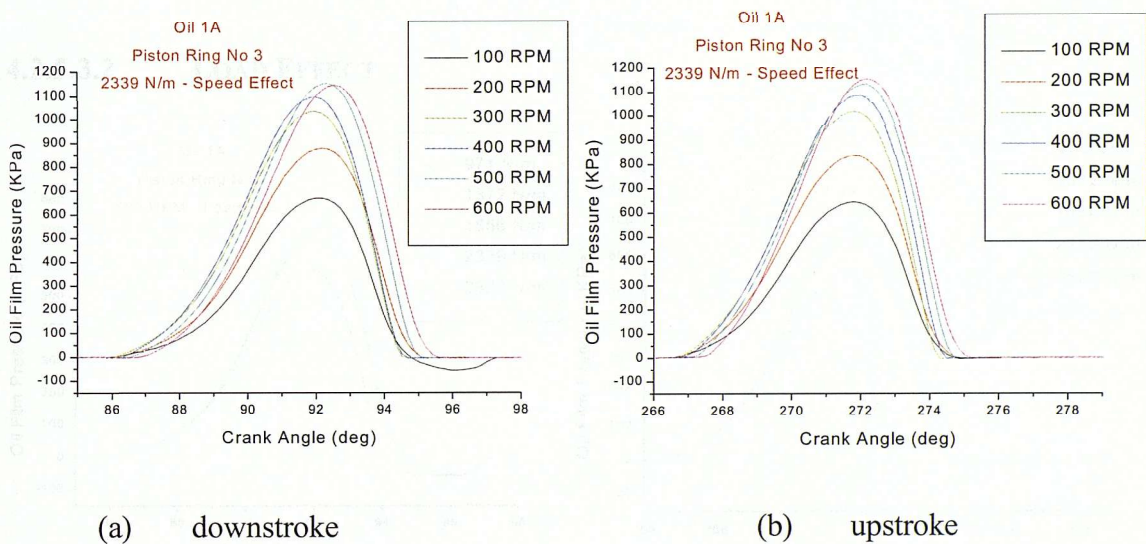


Figure 4.55 Pressure measurements: Speed effect on piston ring No3, Oil 1A, 2339 N/m



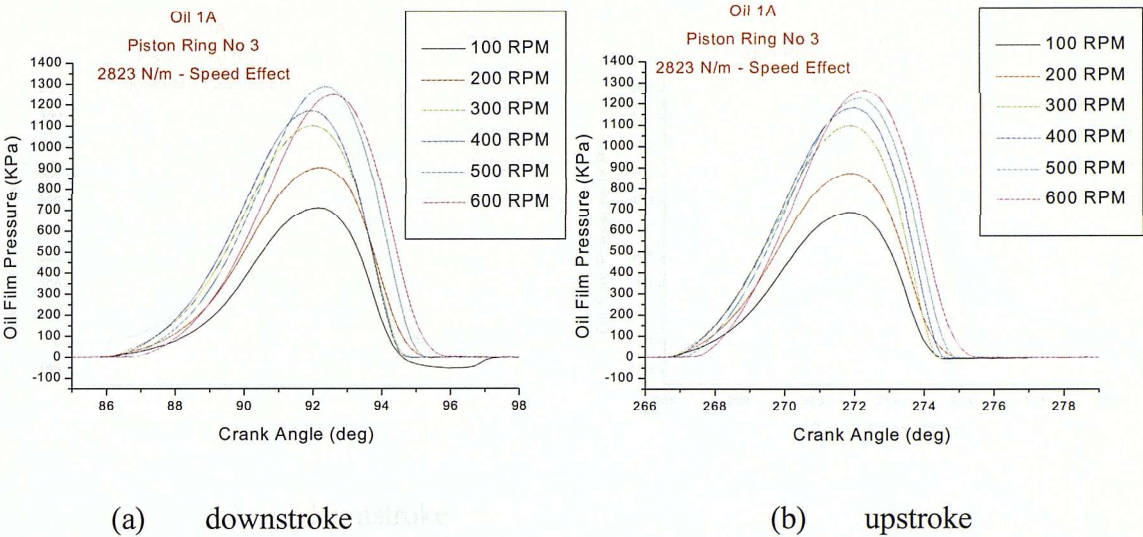


Figure 4.56 Pressure measurements: Speed effect on piston ring No3, Oil 1A, 2823 N/m

4.2.5.3.2 LOAD EFFECT

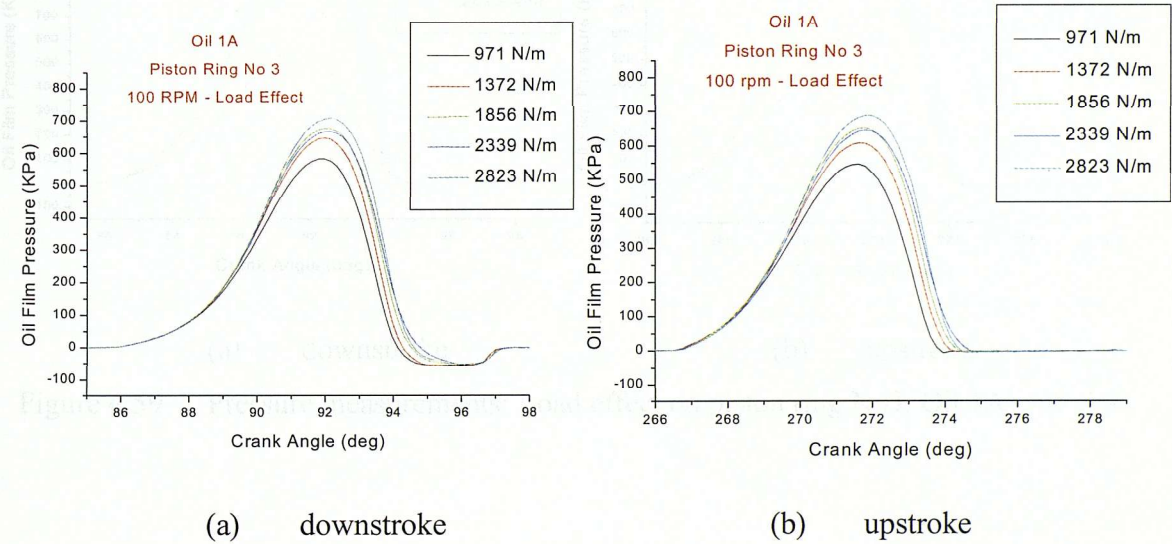


Figure 4.57 Pressure measurements: Load effect on piston ring No3, Oil 1A, 100 rpm

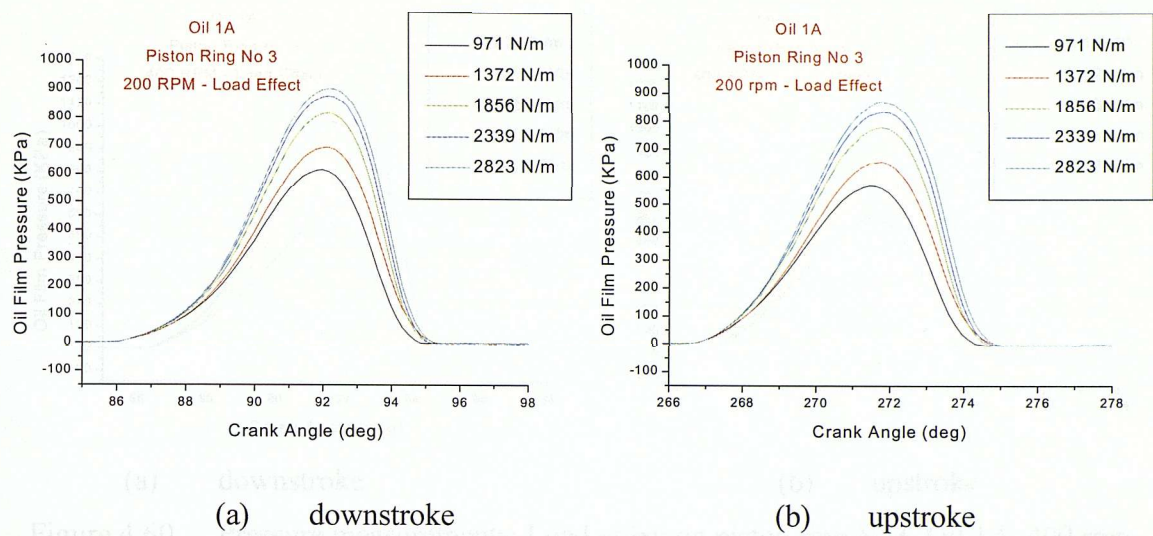


Figure 4.58 Pressure measurements: Load effect on piston ring No3, Oil 1A, 200 rpm

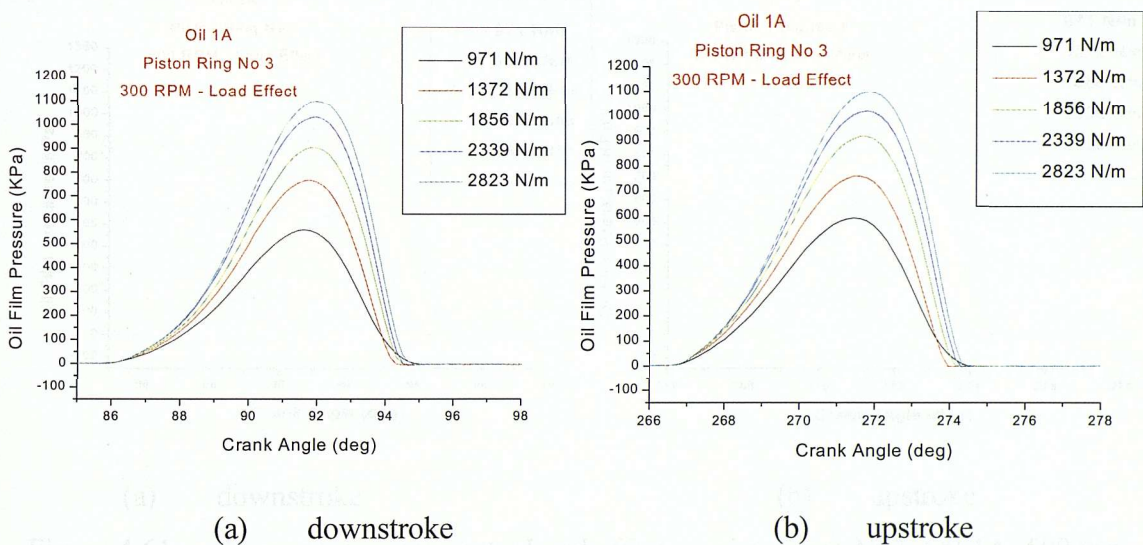


Figure 4.59 Pressure measurements: Load effect on piston ring No3, Oil 1A, 300 rpm

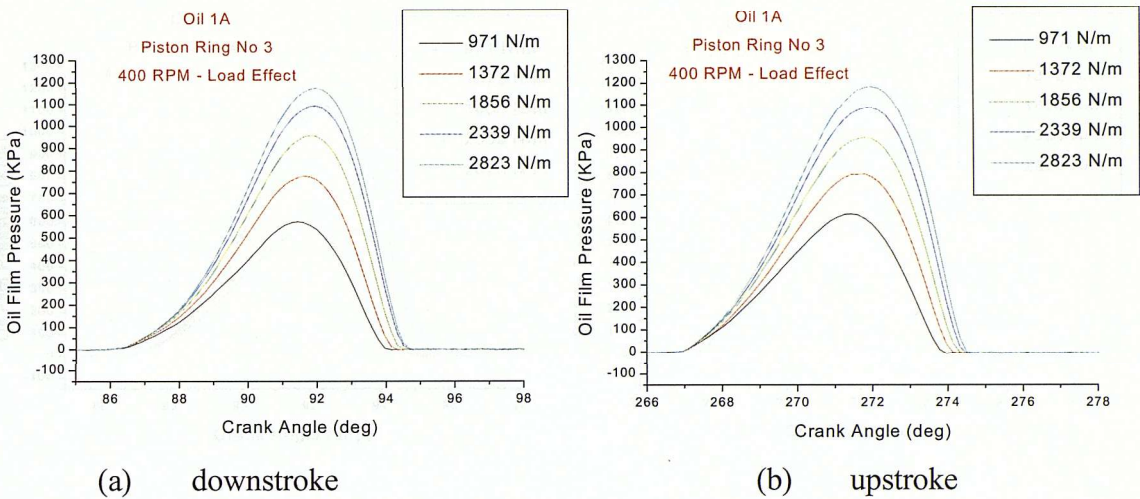


Figure 4.60 Pressure measurements: Load effect on piston ring No3, Oil 1A, 400 rpm

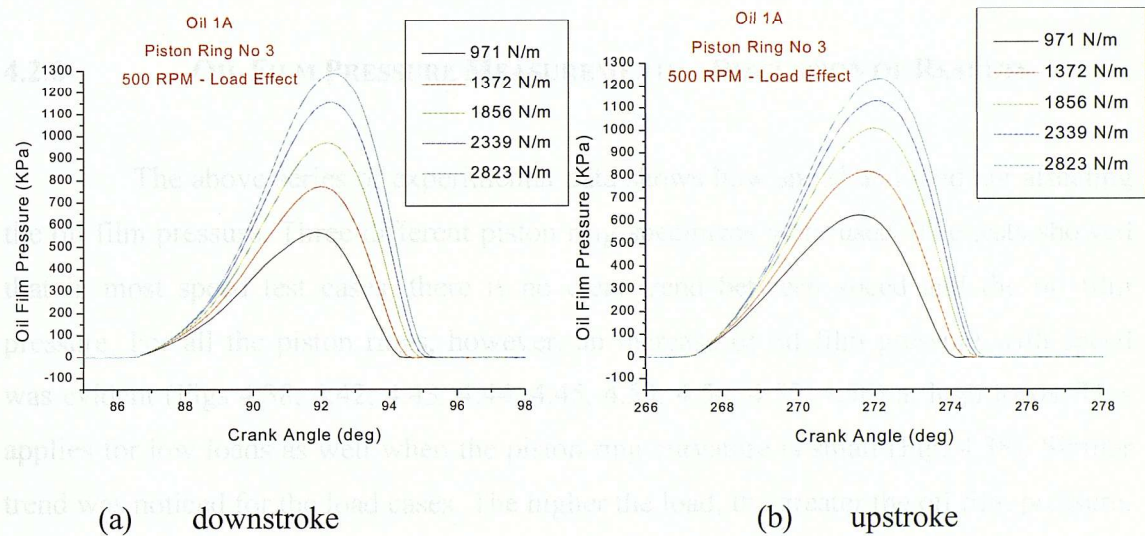


Figure 4.61 Pressure measurements: Load effect on piston ring No3, Oil 1A, 500 rpm



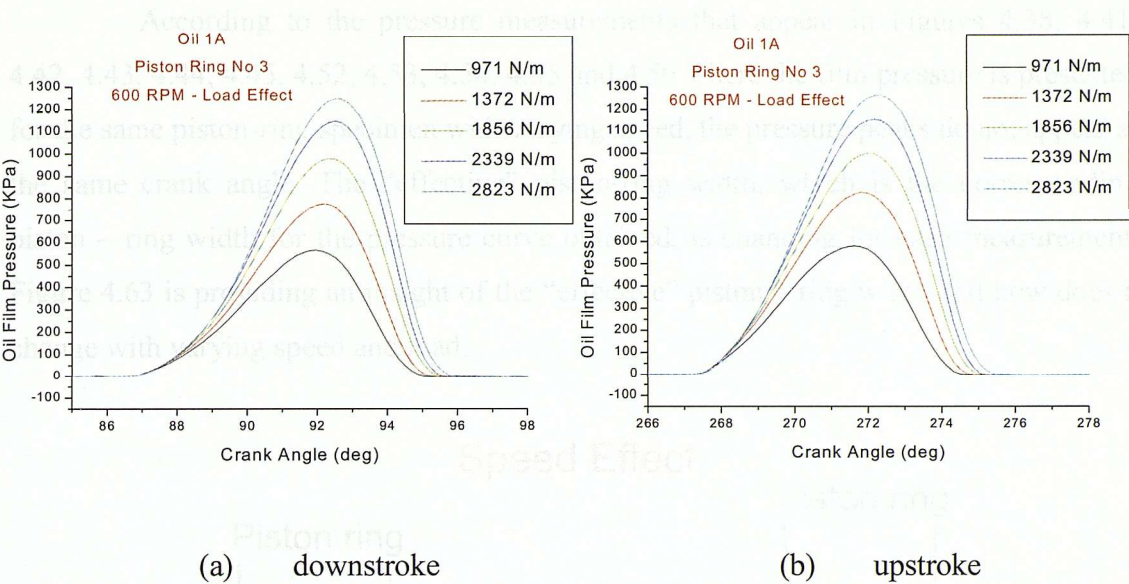


Figure 4.62 Pressure measurements: Load effect on piston ring No3, Oil 1A, 600 rpm

4.2.6 OIL FILM PRESSURE MEASUREMENTS – DISCUSSION OF RESULTS

The above series of experimental data shows how speed and load are affecting the oil film pressure. Three different piston ring specimens were used. The tests showed that in most speed test cases, there is no clear trend between speed and the oil film pressure. For all the piston rings, however, an increase of oil film pressure with speed was evident (Figs 4.38, 4.42, 4.43, 4.44, 4.45, 4.53, 4.54, 4.55, 4.56) at high loads. This applies for low loads as well when the piston ring curvature is small (Fig. 4.38). Similar trend was noticed for the load cases. The higher the load, the greater the oil film pressure. The results were repeatable for every load test case (Figs 4.39, 4.46, 4.47, 4.48, 4.49, 4.50, 4.51, 4.57, 4.58, 4.59, 4.60, 4.61 and 4.62). Temperature tests showed that at higher temperature the oil film pressure decreases (Fig. 4.40).

The speed results showed that as we are moving towards higher loads, there is a quite clear trend showing higher reciprocating speeds accompanied by higher oil film pressure measurements. This is not in accordance to Ostovar (1996) model predictions as for higher speeds he presents the lower pressure results. In terms of absolute measurement, the magnitude of the midstroke pressure measurement (downstroke) has a 14% variation, with the model providing the higher pressure of the two.



According to the pressure measurements that appear in Figures 4.38, 4.41, 4.42, 4.43, 4.44, 4.45, 4.52, 4.53, 4.54, 4.55 and 4.56 where the film pressure is presented for the same piston-ring specimen with varying speed, the pressure peaks do not appear at the same crank angle. The “effective” piston-ring width, which is the corresponding piston – ring width for the pressure curve obtained, is changing for each measurement. Figure 4.63 is providing an insight of the “effective” piston – ring width and how does it change with varying speed and load.

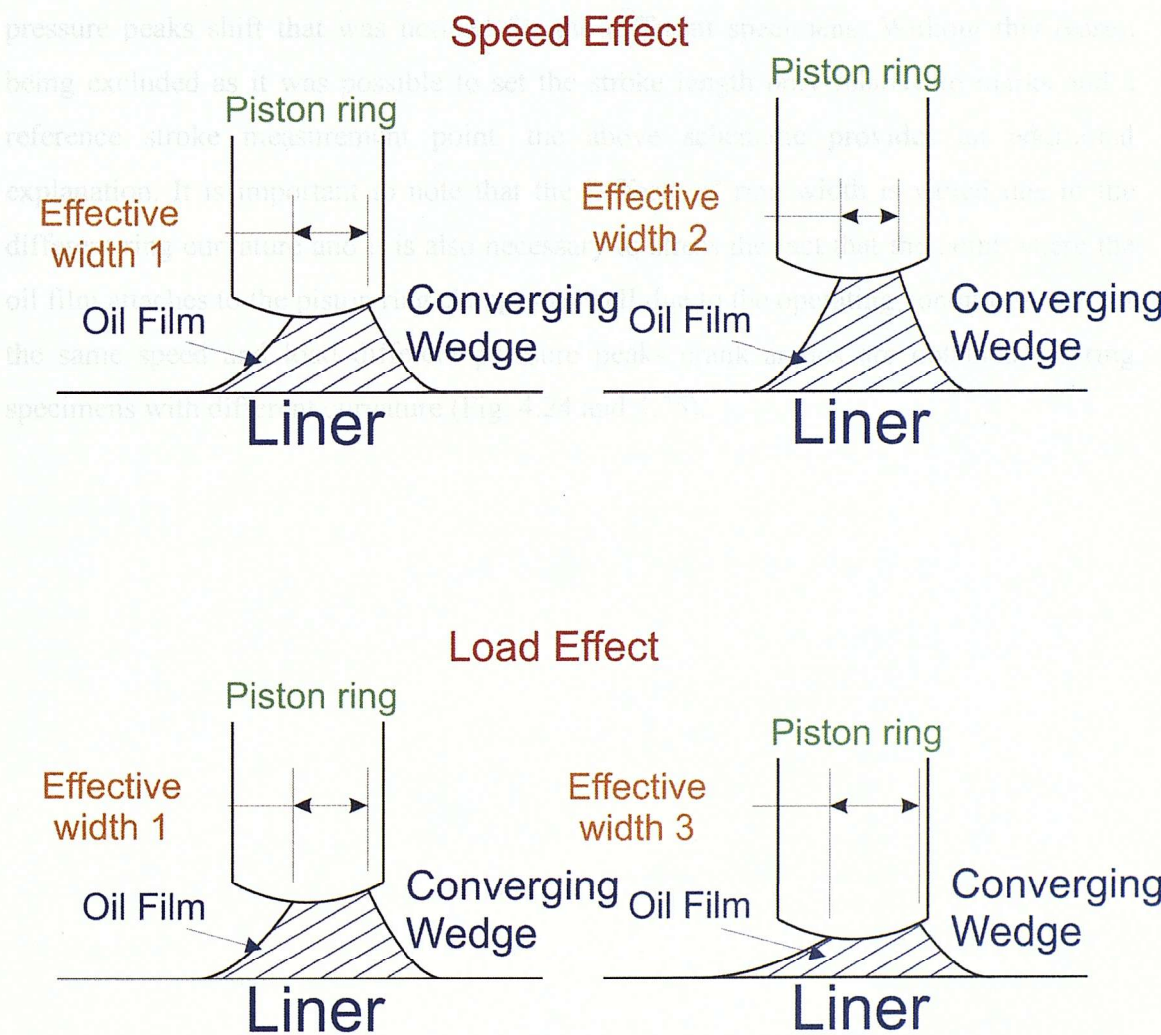


Figure 4.63     Speed and load effect on the “effective” piston – ring width

What should be noticed in the case of the parametric oil film pressure measurements for each piston – ring specimen, is that for the load variation tests the pressure peaks appear more uniform. The “effective” ring width is altered as it gets wider by a small margin while load increases (Fig. 4.63), whereas the pressure peaks do not have a big variation for each testing condition (the curve peaks are slightly shifted between them; Figures 4.39, 4.46, 4.47, 4.48, 4.49, 4.50, 4.51, 4.57, 4.58, 4.59, 4.60, 4.61 and 4.62). Initially, in the pressure testing results for different piston – rings, the slight variation of the mid-stroke setting of the liner relative to the ring was blamed for the pressure peaks shift that was noticed for the different specimens. Without this reason being excluded as it was possible to set the stroke length only relative to marks and a reference stroke measurement point, the above schematic provides an additional explanation. It is important to note that the “effective” ring width is varied due to the different ring curvature and it is also necessary to stress the fact that the point where the oil film attaches to the piston ring changes as well due to the operating conditions. So, for the same speed and load different pressure peaks crank angles are obtained for ring specimens with different curvature (Fig. 4.24 and 4.25).



4.3 HIGHER MAGNIFICATION PICTURES COMBINED WITH PRESSURE RESULTS

Figure 4.64 shows a low magnification picture of the piston-ring area where images of the film were obtained with the CCD camera. The different coloured squares show the covered area in each series of photographs with higher magnification lenses.

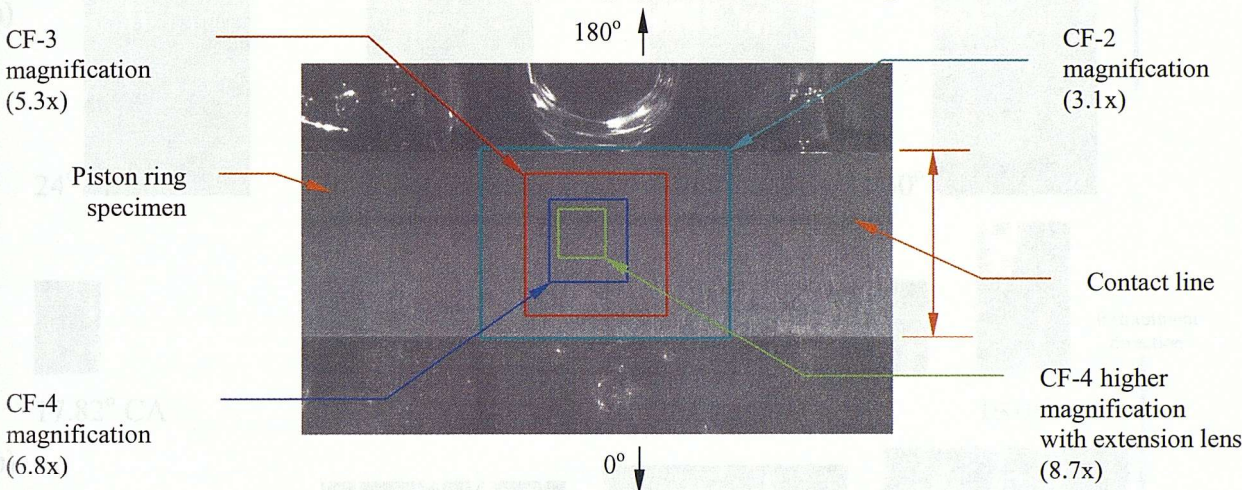


Figure 4.64 Details at low magnification of the piston-ring

As flow reverses cavities start forming in the surface of the piston ring. In Figure 4.65 the highlighted areas indicate the shift of flow reversal later in the stroke due to increasing speed.

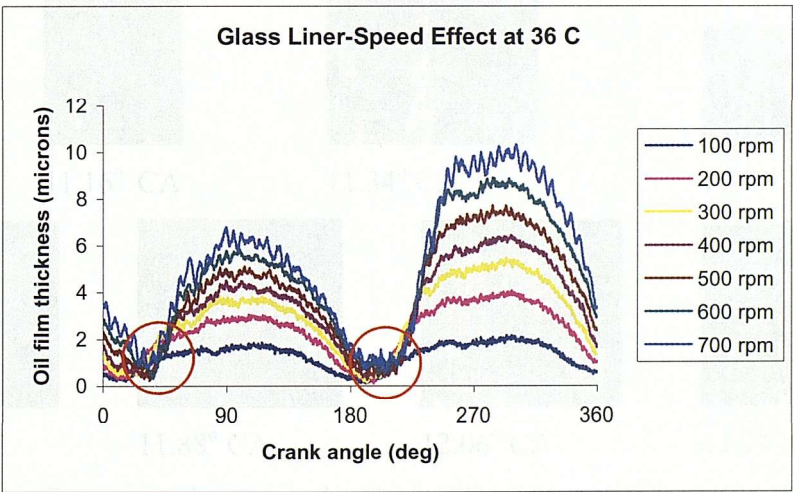


Figure 4.65 Minima MOFT curves shift to the right with speed at the highlighted areas (measured with the capacitance transducer in 100 r/min increments)

Higher magnification photos enabled better identification of internal cavities at crank angles near TDC where the flow reverses which is illustrated in the photos below:



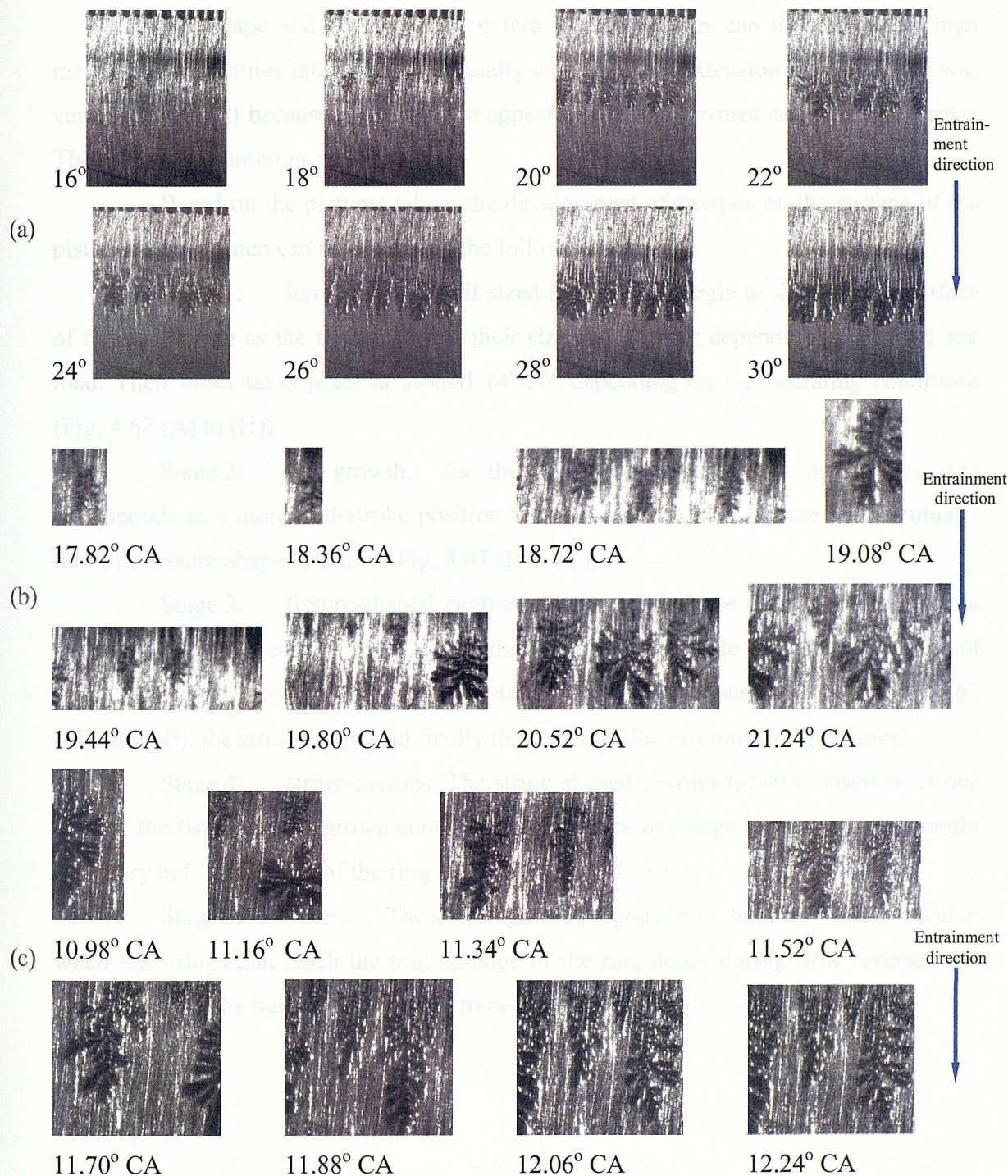


Figure 4.66 Fern growth captured using different magnification systems. (a) CF-3 magnification ( $\times 5.3$ ), 700 r/min, 22mm stroke, 1869 N/m. (b) CF-4 magnification ( $\times 6.8$ ), 700 r/min, 22mm stroke, 1869 N/m. (c) CF-4 magnification with extension lens ( $\times 8.7$ ), 700 r/min, 22mm stroke, 2843 N/m. In (b) and (c) images were taken at smaller crank angle intervals to focus on the fern-shapes.



The shape and development of fern shaped cavities can be seen in the high magnification pictures taken above especially with the CF-4 extension lens. The load was varied (increased) because it enables the appearance of the cavities earlier in the stroke. They are more numerous as well.

Based on the pictures taken, the development of cavities on the surface of the piston-ring specimen can be divided in the following stages:

Stage 1: fern growth. Small-sized fern cavities begin to shape on the surface of the piston-ring as the flow reverses; their size and number depends on the speed and load. Their onset takes place at around  $14^{\circ}$ - $30^{\circ}$  depending on the operating conditions (Fig. 4.67 (A) to (H)).

Stage 2: fern growth. As the liner reciprocates and the piston ring corresponds to a more mid-stroke position the fern cavities grow in size and eventually turn into fissure-shaped cavities (Fig. 4.67 (I) to (K)).

Stage 3: fissure-shaped cavities. Their initial shape (Fig. 4.67 (L)) is thin towards the trailing edge of the ring and thicker at the other side where the first sign of string shaped cavities is evident as they start to grow. The fissures coalesce (Fig. 4.67 (M) and (N)), the strings grow and finally they are transformed into string cavities.

Stage 4: string cavities. The string shaped cavities initially observed at one edge of the fissures, have grown enough to reach the trailing edge of the ring or a straight boundary before the edge of the ring (Fig. 4.67 (O) and (P)).

Stage 5: bubbles. These strings disintegrate into bubbles which develop when the strings that reach the trailing edge of the ring decay during flow reversal; the pressure inside the bubbles is expected to be atmospheric.

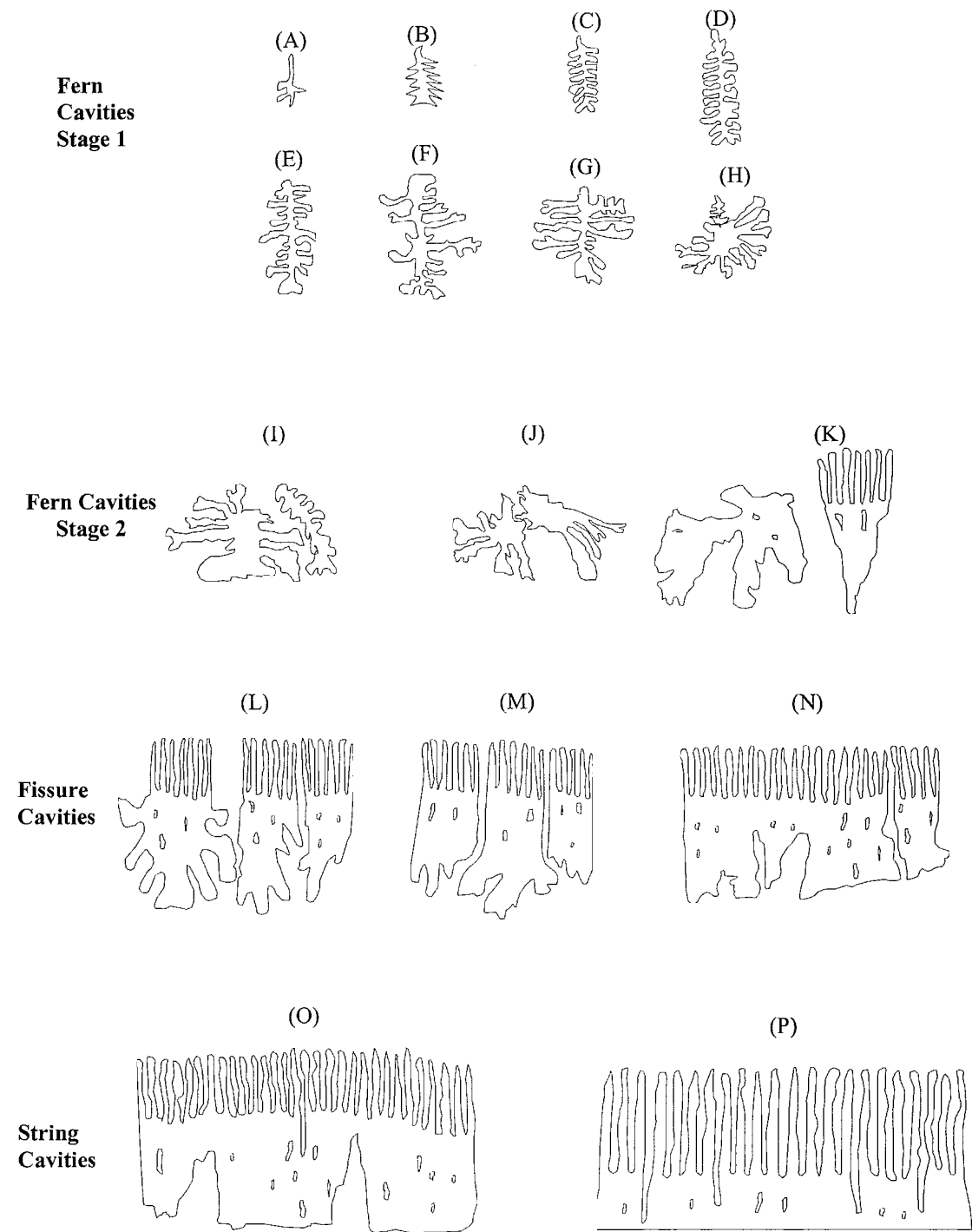


Figure 4.67 Development of the cavities as they appear on the surface of the piston-ring

The following image (Figure 4.68) shows the pressure distribution and the respective cavity shapes.

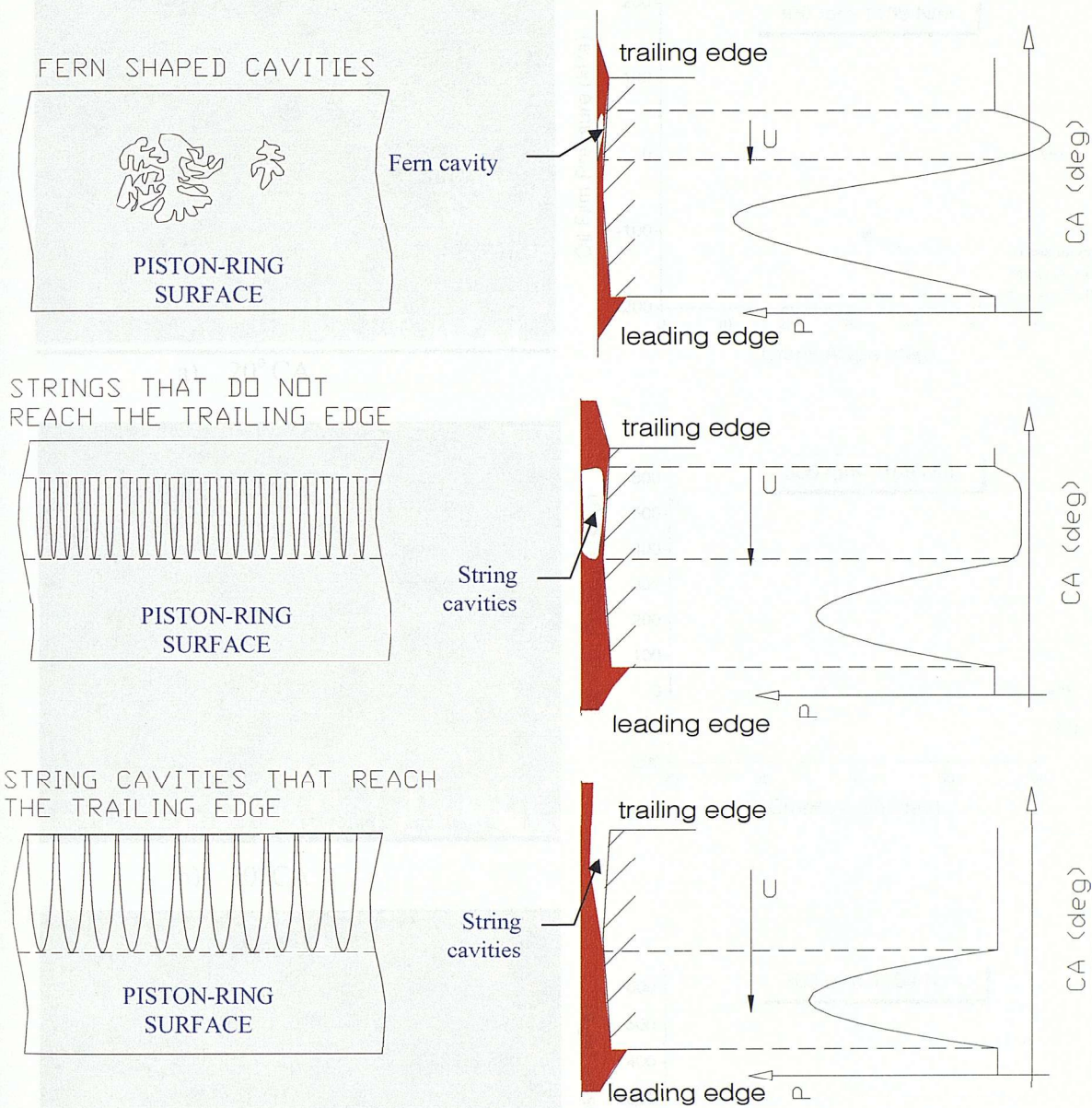
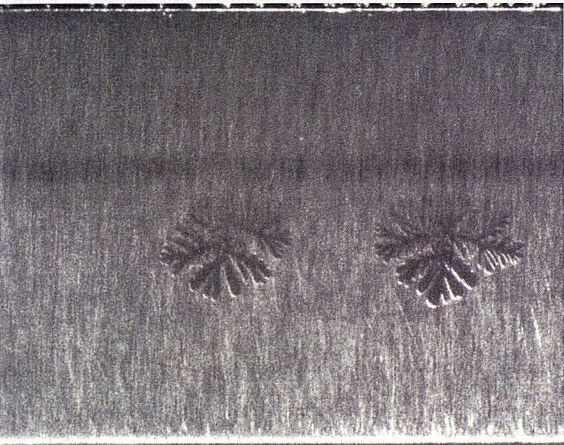


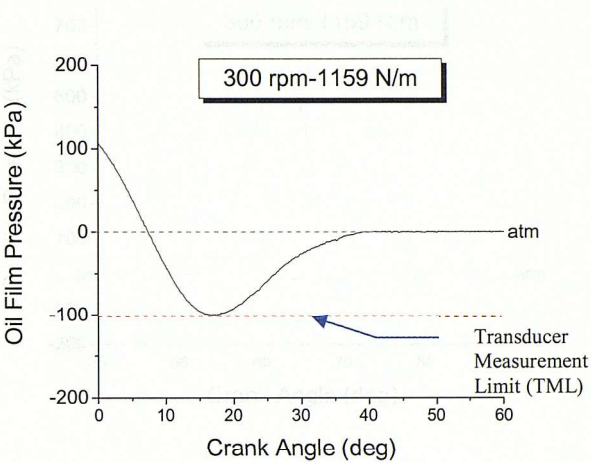
Figure 4.68 Cavity types and corresponding pressure profiles



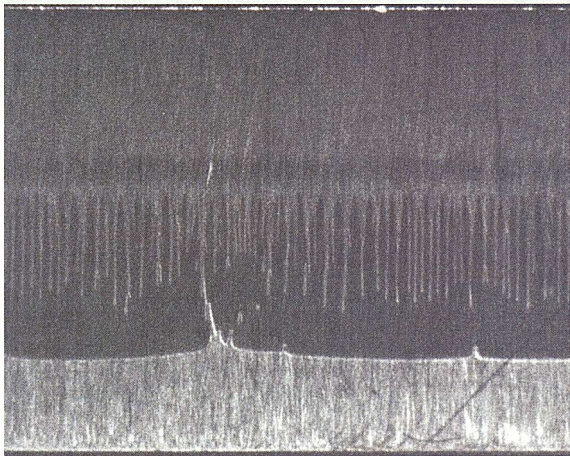
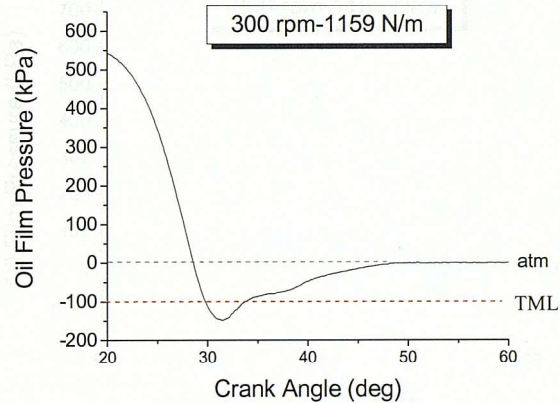
4.3.1 DEVELOPMENT OF CAVITIES DURING THE STROKE



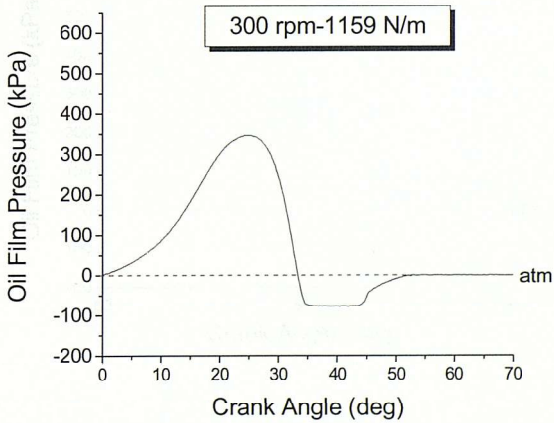
a) 20° CA



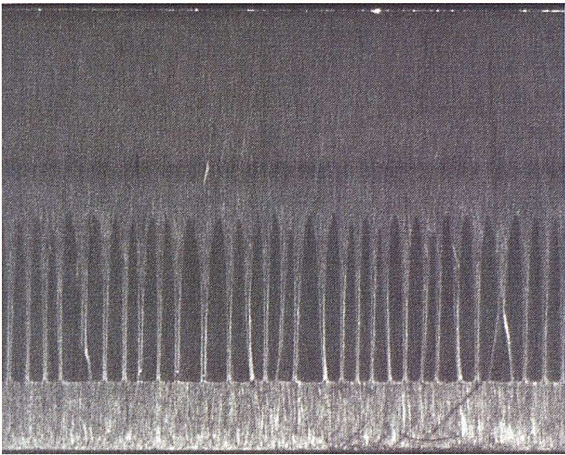
b) 30° CA



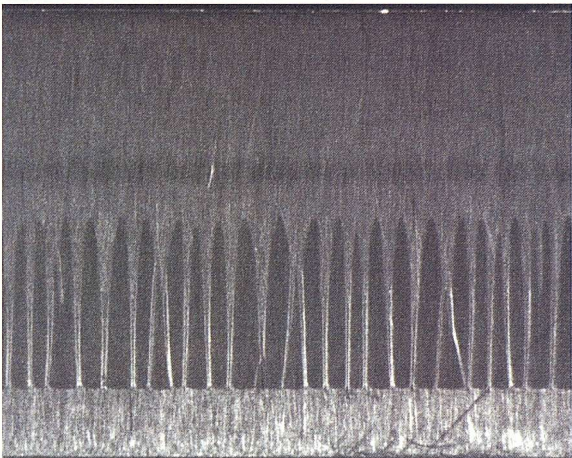
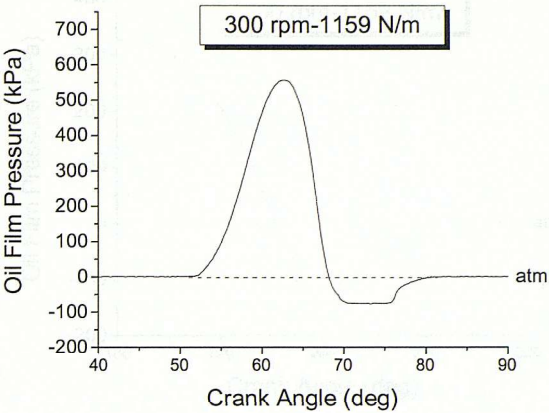
c) 35° CA



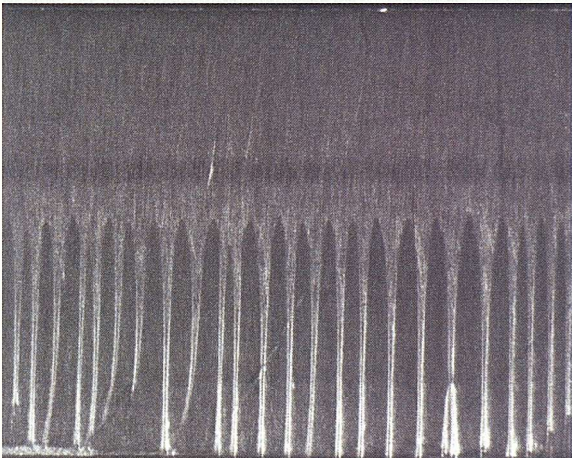
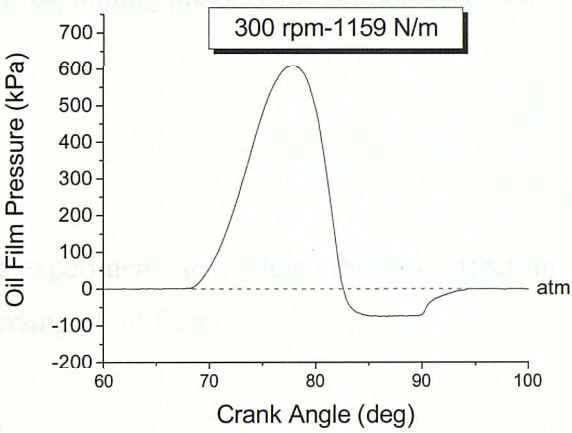




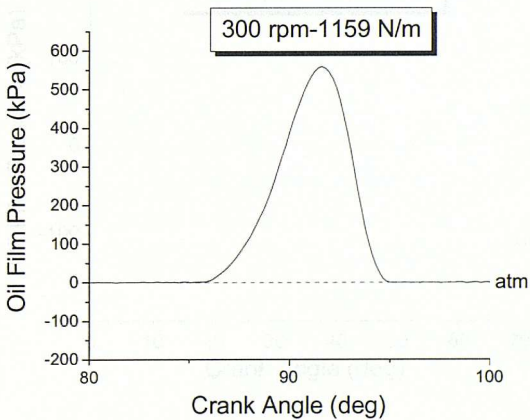
d) 70° CA



e) 81° CA



f) 90° CA





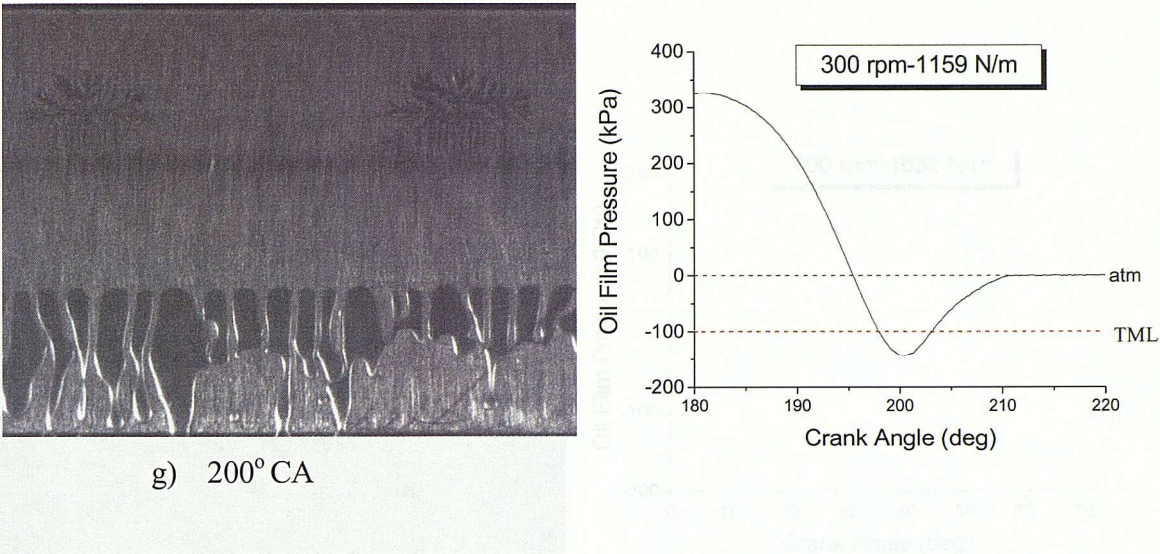
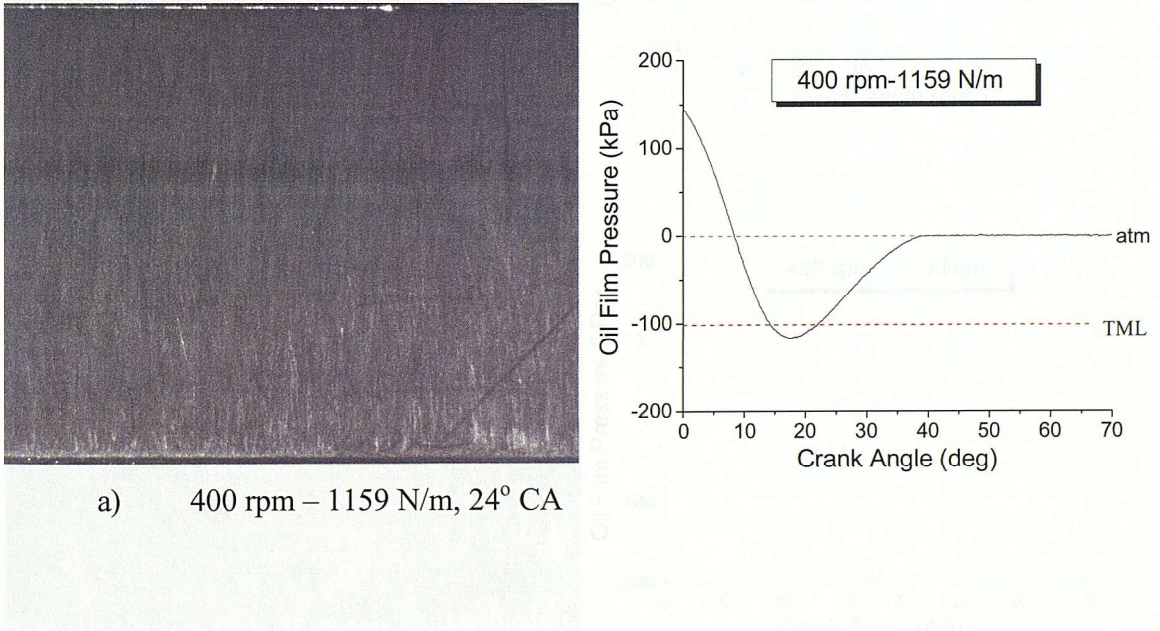


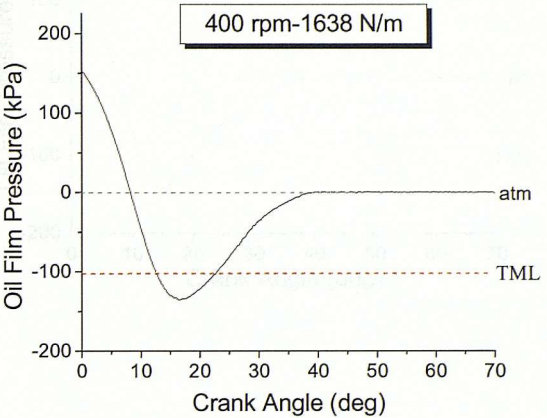
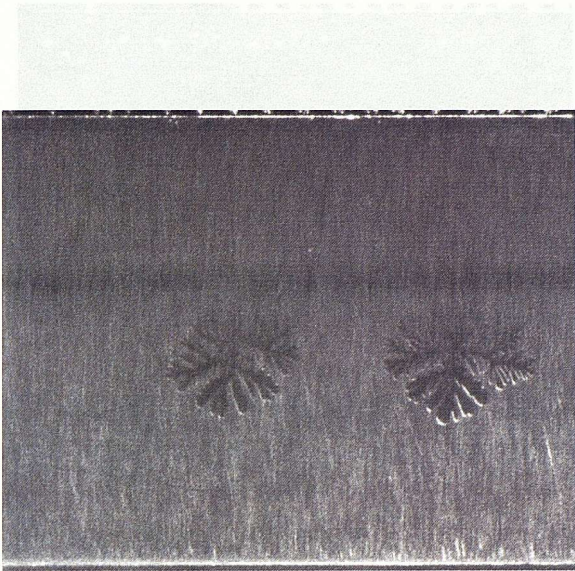
Figure 4.69 Development of cavities and corresponding pressure measurements at 300 rpm, no external load

4.3.2 LOAD TESTS

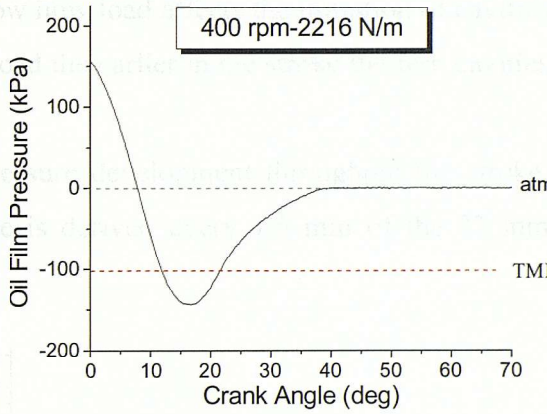
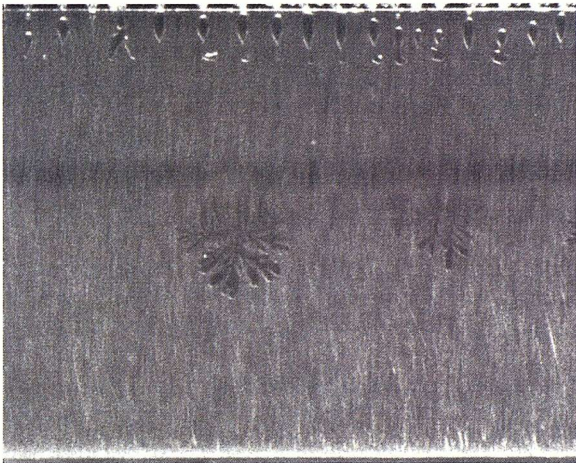
Different loads were used for these experiments to evaluate the load effect on the appearance of fern cavities during the reversing of oil flow.



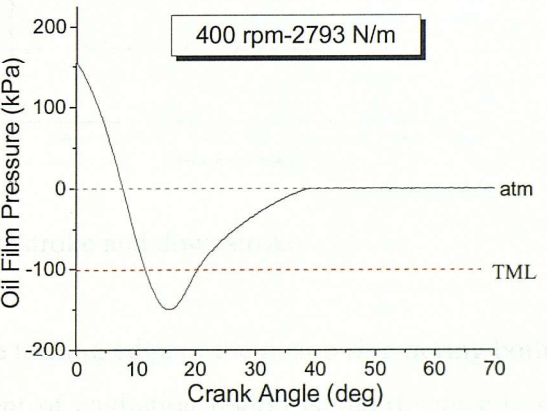
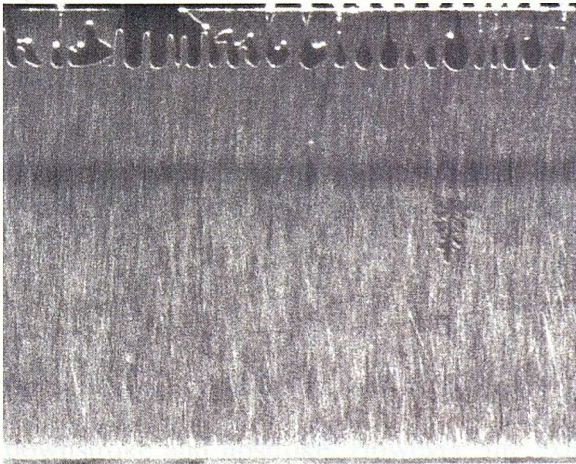




b) 400 rpm-1638 N/m, 20° CA



c) 400 rpm-2216 N/m, 16° CA



d) 400 rpm -2793 N/m, 16° CA



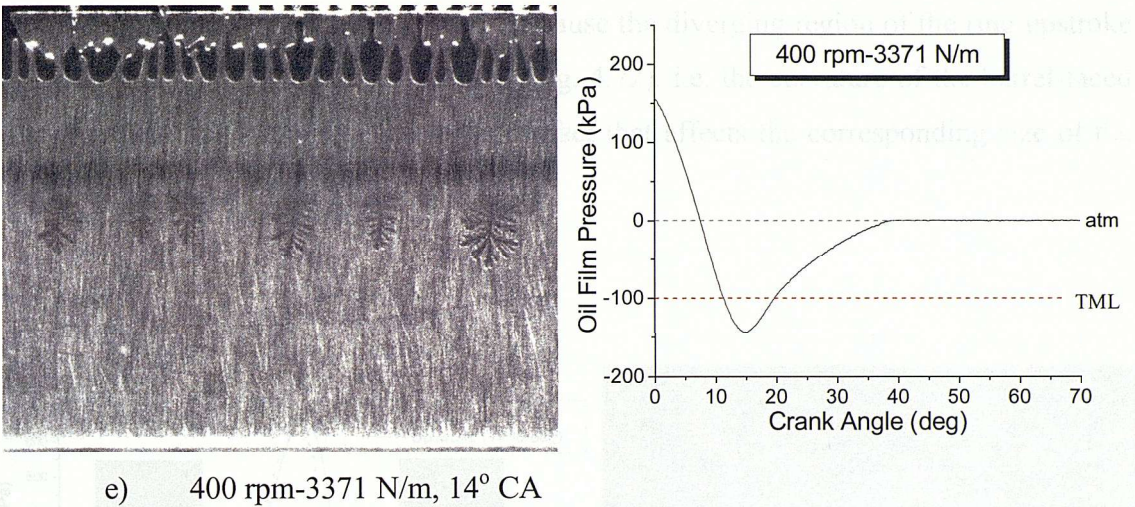


Figure 4.70 Effect of load on reversing oil flow and formation of cavitation patterns

The images in Figure 4.70a to e show how load affects the initiation of cavities at the beginning of the stroke; the higher the load the earlier in the stroke the fern cavities appear.

The following graphs show the pressure development throughout the stroke, downstroke (a) and upstroke (b). Each curve is derived every 1.6 mm of the 22 mm stroke:

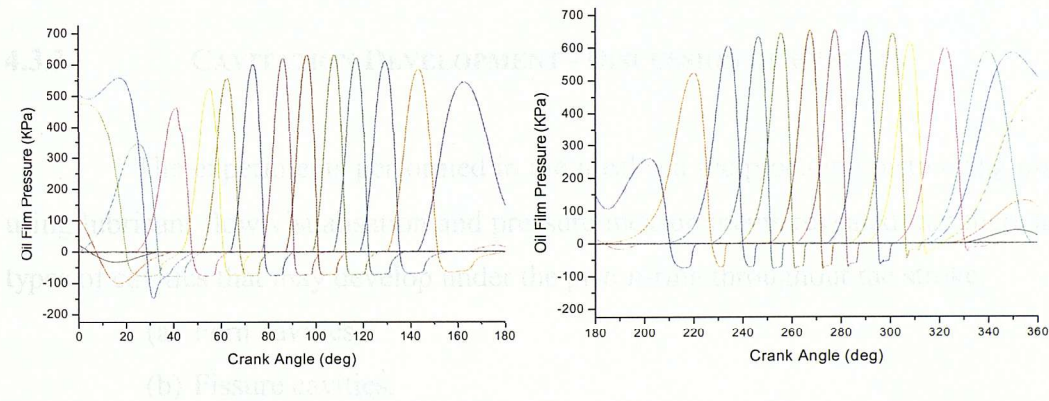


Figure 4.71 Pressure development upstroke and downstroke

Oil reaches vaporous pressure at the trailing edge of the piston ring during both downstroke and upstroke. However, the extent of cavitation region is significantly less



upstroke than in the downstroke results because the diverging region of the ring upstroke is much smaller than in the downstroke (Fig. 4.72), i.e. the curvature of the barrel-faced piston-ring is not symmetric but has an offset that affects the corresponding size of the cavitation region.

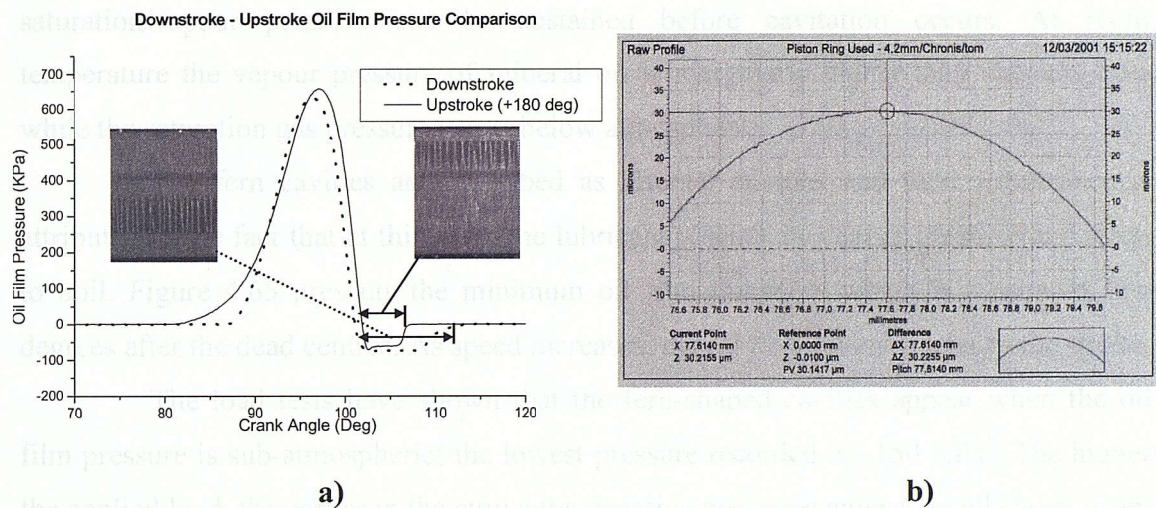


Figure 4.72    a) Vaporous pressure difference between downstroke and upstroke strings  
b) Ring profile as measured by surface profilometry

4.3.3            CAVITATION DEVELOPMENT - DISCUSSION OF RESULTS

The experiments performed in the idealised reciprocating piston-ring assembly using lubricant flow visualisation and pressure measurements revealed that there are four types of cavities that may develop under the piston-ring throughout the stroke:

- (a) Fern cavities,
- (b) Fissure cavities,
- (c) String cavities,
- (d) Bubbles.

The fern cavities are internal cavities and they are formed as the lubricant reaches vapour pressure or negative absolute pressure. It seems that negative absolute

pressures involving tension may arise just before oil film rupture occurs. This tension is released with the generation of internal cavities but the negative absolute pressure is still maintained causing a reduction of the minimum film thickness and thus extension of the squeeze film to a larger fraction of the stroke.

Various studies (Temperley and Chambers, 1946; Vincent, 1943) of the tensile strength of liquids have shown that for short periods of time pressures far below the gas saturation/vapour pressure can be sustained before cavitation occurs. At room temperature the vapour pressure of mineral oil is marginally higher than absolute zero while the saturation gas pressure is just below atmospheric.

The fern cavities are described as internal cavities and their appearance is attributed to the fact that at this stage the lubricant reaches its vapour pressure and starts to boil. Figure 4.65 presents the minimum oil film thickness which is reached a few degrees after the dead centres. As speed increases, the oil flow reverses later in the stroke.

The load tests have shown that the fern-shaped cavities appear when the oil film pressure is sub-atmospheric; the lowest pressure recorded is  $-150 \text{ KPa}^\dagger$ . The higher the applied load, the earlier in the stroke the negative pressures appear. In all cases, when the flow is reversing near the dead centres, fern-shaped cavities appear at the diverging part of the piston-ring assembly.

The pictures taken at 300 rpm (Figure 4.69) show the development of the cavities from fern-shaped to strings up to the BDC reversal point, where the same phenomenon is repeating itself upstroke. The fern cavities are very small in the beginning and as they start growing they take the shape shown in Figure 4.69 c ( $35^\circ \text{ CA}$ , fissures). The strings start growing from inside the fern cavities before they reach a boundary limited by the tensile strength of the liquid. This is happening in every case but, depending on reciprocating speed, the string cavities that have already developed may coalesce with external cavities formed at atmospheric pressure and sucked from the diverging part of the piston-ring. Eventually the string cavities reach the trailing edge of

---

<sup>†</sup> Since the pressure sensor does not have the ability to measure pressures below absolute zero, any negative results below  $-100 \text{ kPa}$  absolute are not considered as a realistic value. The resolution of the sensor (35 bar) is too high to provide accurate results in the range of 1.4% of the sensor range under vacuum. A lower resolution sensor would have measured the subambient pressures in that region in a much better way. It should also be stressed that the sensor is preloaded as its face is being squeezed just above the lubricant measuring slot. This offset is being recorded and subtracted from the results. The measurement limit of the pressure transducer is indicated in Figures 4.69 a), b), g) and 4.70 a), b), c), d), e) as a red dotted line. Further investigation is underway to resolve this discrepancy.



the piston-ring; the higher the rotating speed the earlier in the stroke coalescence takes place.

The crank angle when the fern cavities initially appear depends on speed of reciprocation and applied load; speed and load also have an effect on the number and size of cavities. The higher the speed, the later in the stroke the fern cavities appear and the bigger their size is. On the other hand, the higher the load the earlier in the stroke the cavities appear, the greater their number and the smaller their size. This is due to the squeezing action of the film that causes the appearance of negative pressures much earlier in the stroke than at lower loads.

Oil film pressure measurements have revealed that when the strings reach the trailing edge of the piston-ring, atmospheric pressure is recorded at the diverging part of the ring. However, at lower speeds (100 - 200 rpm) the film pressure in the diverging part of the ring is sub-atmospheric and equal to the vapour pressure of the lubricant.

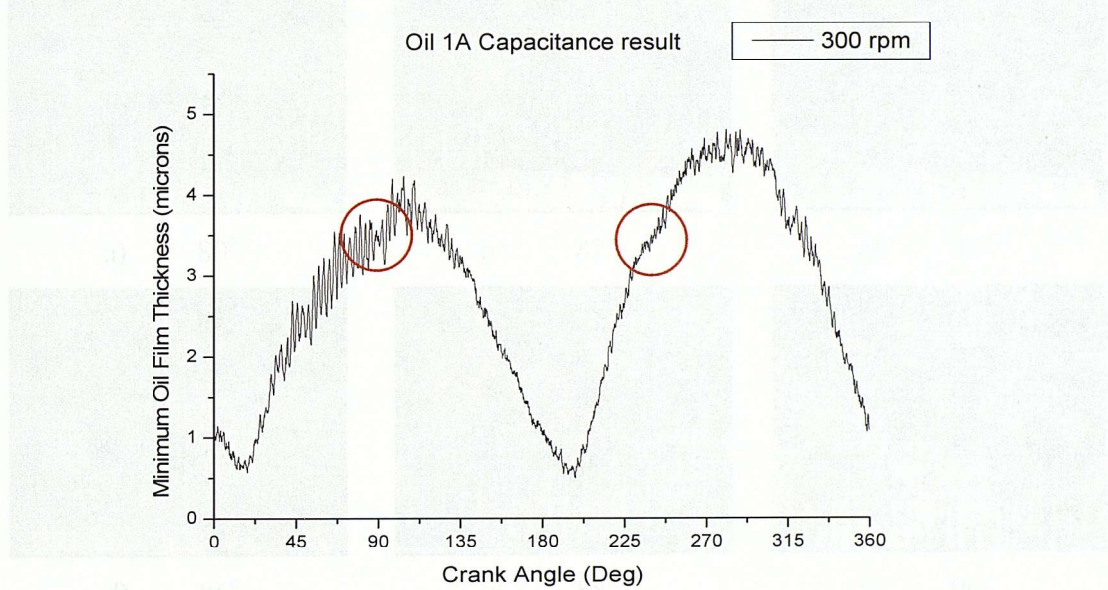


Figure 4.73 Effect of the cavitating conditions transitions in the capacitance signal

The shift in oil film thickness as it is observed in the highlighted area in Figure 4.73 on the capacitance signal, occurs in the region between 65-85° CA for the downstroke and 260-290° CA for the upstroke. From the photos taken for this crank angle region, the string shaped cavities start approaching the trailing edge of the ring;



eventually there is a sudden transition between two different boundary cavitating conditions. The first cavitating condition is when the pressure at the trailing edge of the piston-ring becomes equal to vapour pressure. The second is when the pressure at the trailing edge of the piston-ring becomes equal to atmospheric. The sub-ambient pressures help support the load (Jones et al, 1999) so when the transition to atmospheric pressure takes place only a slight decrease of the OFT is observed. This oil film behaviour is similar to the higher load effect where the film decreases.

Figure 4.74 shows the cavities formed at 50 mm stroke, 300 rpm, no external load where they reach the trailing edge of the piston-ring. Unfortunately, pressure measurements could not be obtained for this condition. The transition between the two cavitating conditions, is evident in the series of pictures shown in Figure 4.74. In pictures 4.74 b and c an external cavity can be seen being sucked from the diverging part of the ring. Eventually this helps the transition process between the two cavitating conditions.

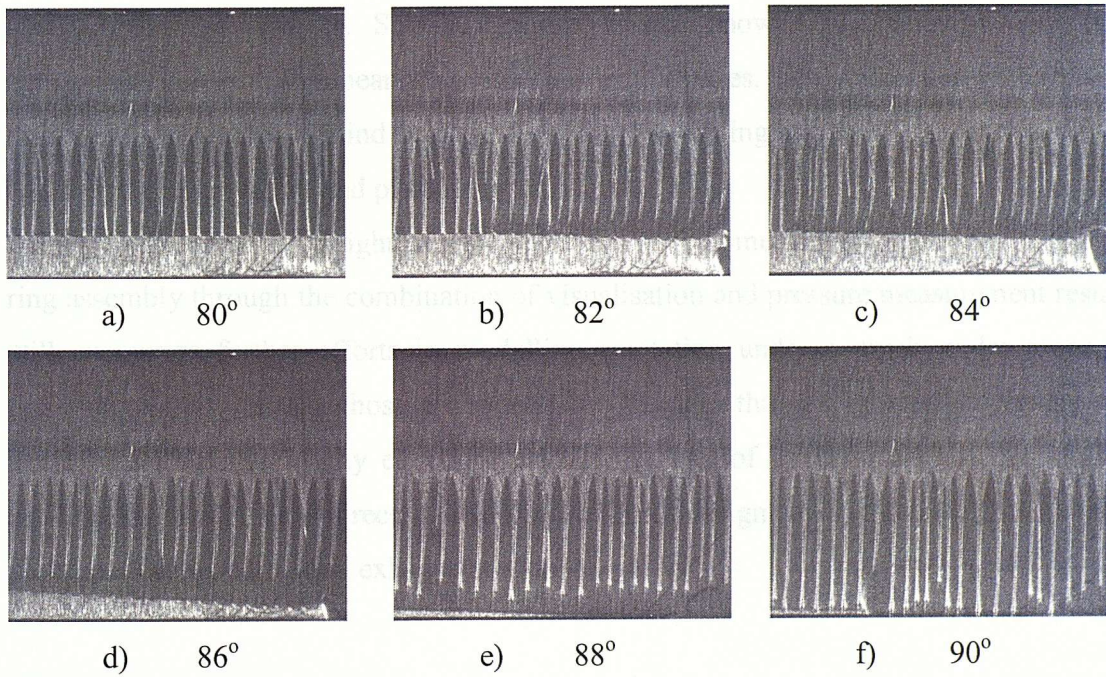


Figure 4.74 Visualisation of the transition between different cavitating conditions



## 4.4 CONCLUSIONS

An idealised test rig that simulates the lubrication condition between a piston-ring and the cylinder liner in reciprocating engines has been modified to allow oil film visualisation and pressure measurements. The lubricant film thickness has a major influence on the cavitation behaviour, which in turn, affects the shape of the hydrodynamic pressure profile in the entrainment direction and consequently the load carrying capacity of the ring-pack.

The use of a silicon chip ultra miniature transducer has provided both qualitative and quantitative pressure data, which proved to support the flow visualisation results, while its robustness and good repeatability offer promise for its application to the piston-ring pack of production engines. Parametric study showed the effect of speed, load, temperature and piston-ring curvature on the oil film pressure measurements results upstroke and downstroke. Stroke variation results showed that negative and sub-atmospheric pressures appear close to the dead centres. The “effective” ring width variation is the reason behind pressure peak shifting during parametric testing (different speed, load, temperature and piston-ring curvature).

The obtained insight into the onset and development of cavitation in the piston-ring assembly through the combination of visualisation and pressure measurement results will encourage further efforts in modelling cavitation under a much wider range of operating conditions than those examined here. It seems that a combined theoretical and experimental approach may eventually lead to a level of understanding of piston-ring lubrication which can directly influence engine design and, through this, engine performance, economy and exhaust emissions.

## CHAPTER 5

### ENGINE VISUALISATION EXPERIMENTS

#### INTRODUCTION

After the successful modifications that took place on the single-ring test rig, which included the pressure measurements and the visualisation experiments, a similar investigation has been attempted into the single-cylinder diesel engine. These modifications included fitting quartz windows to visualise the lubrication film characteristics between the piston-rings and the cylinder liner and the introduction of measurement stations at top dead centre, mid-stroke and bottom dead centre. Reviews of previous work on oil film visualisation and oil transport has been used to assist the interpretation of the visualisation results in the Lister-Petter engine. Similar type of cavities to the one observed in the test rig have been identified although the quality of the images is, as expected, inferior in the case of the engine.

#### 5.1 PREVIOUS WORK ON ENGINE OIL FILM VISUALISATION

Researchers from MIT (Thirouard, 2001) and Toyota (Inagaki et al, 1995) have previously published research centred at the visualisation of the oil film at the wall of the cylinder liner. For these experiments they used fully optical liners or liners with windows and pictures were taken under both firing and motoring conditions. Their investigations will be reviewed here in order to provide background information for the present experiments.

5.1.1 MIT RESEARCH ON OIL TRANSPORT (THIROUARD, 2001)

MIT researchers (Thirouard, 2001) used two engines to visualise the oil flow:

- a) a Kubota single cylinder engine
- b) a PSA single cylinder engine

a) Kubota engine (Thirouard 2001): A quartz (synthetic fused silica) window was installed in the removable liner of the Kubota engine. The window has been designed with a trapezoidal section in both vertical and horizontal direction to better support the combustion pressure. The window was located in the anti-thrust side (Figure 5.1).

The synthetic fused silica provides:

- Good optical quality (excellent transmittance of the visible wavelength)
- Good mechanical properties (very scratch resistant)
- Inexpensive raw material and low cost machining

The engine incorporated a relatively small window in a well-cooled cast iron cylinder liner. It was located below the mid-stroke position and therefore not exposed to the very high temperatures encountered during the combustion process.

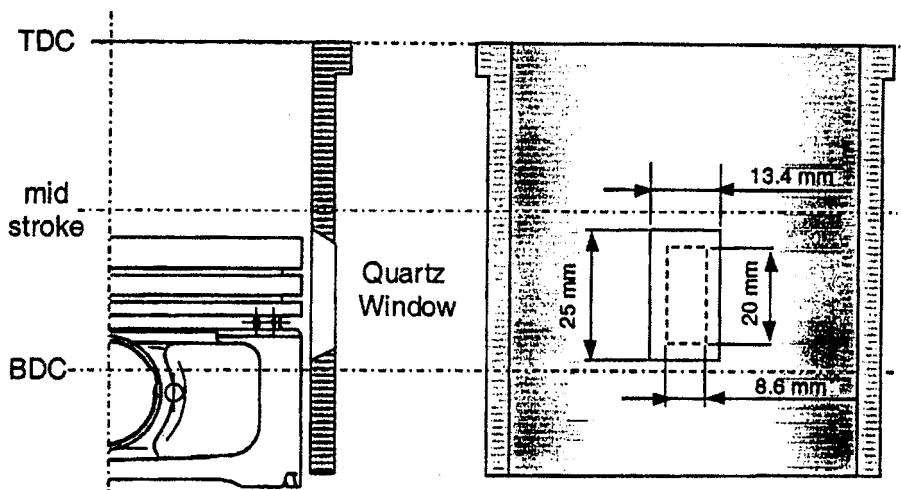
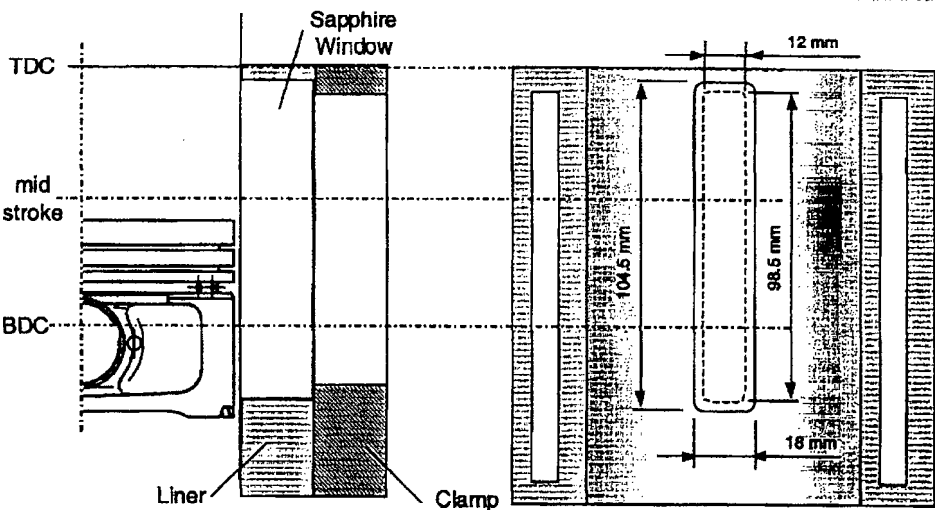


Figure 5.1 Kubota engine window

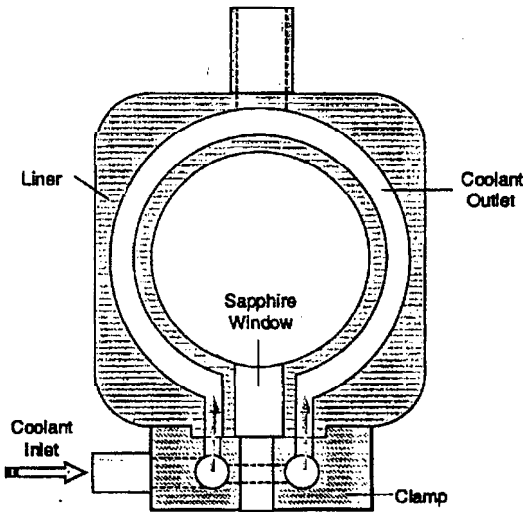
b) PSA engine (Thirouard, 2001): In order to verify experimentally more aspects of piston-ring oil transport, a spark ignition engine was equipped with extensive optical



access in the form of a long window of 104.5 mm×18 mm dimensions. The window was made of sapphire, which has superior mechanical properties and is therefore more likely to resist the stresses induced by the combustion chamber pressure. Since sapphire and cast iron have very similar thermal expansion coefficients, it is expected that with a sapphire window the gap between the window and the liner will not increase as temperature rises, thus minimising the risk of leakage. It should be mentioned that the window was glued to the liner with high temperature resistant and high thermal conductivity epoxy glue. Both windows were polished on the outside and honed on the inside.



(a) side view



(b) top view

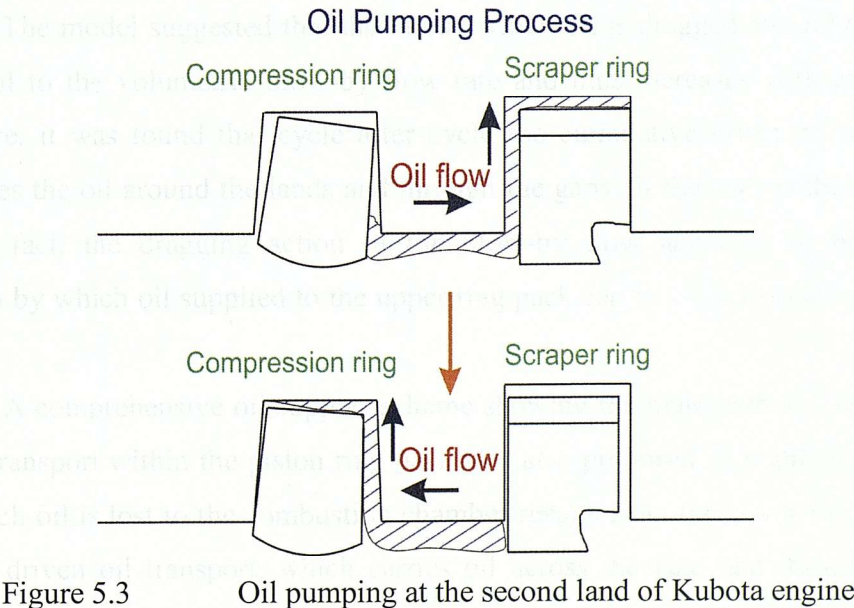
Figure 5.2 PSA spark ignition engine and top view

A visualisation system using multiple dye Laser Induced Fluorescence was implemented to observe oil displacement among various regions of the piston ring pack in both a diesel and a spark ignition engine. The two dimensional oil distribution measuring system was developed using a pulsed laser. The fluorescent dyes are sensitive to temperature and keeping in mind that the temperature in the ring pack varies from the piston skirt to the crown land, the sensitivity of the LIF signal to oil temperature should be minimised. Rhodamine 590 which is quite stable with temperatures up to 250°C was used. The second harmonic of a Nd-YAG laser was used to excite the doped oil (532 nm). The fluorescent signal was acquired with an intensified CCD camera. Direct reflection from the window was suppressed using a narrow band filter. High resolution images and video recordings of the oil distribution were acquired over a range of running conditions representative of passenger car engine operation.

On the piston lands, oil was observed to move in both the axial and circumferential directions. The axial oil displacement was found to be driven by the inertia forces resulting from the piston alternating motion. The circumferential flow appeared to result from the dragging action of the blow-by gases as the calculated oil velocity is about 2 mm/sec (second piston-land), which is the same order of magnitude as the observed velocity. The shear stress at the interface of gas and oil can drag the oil to move with the gas. Effect of the gas dragging eventually removes the oil out of the second land through the ring gaps. Through experimental observation, it was established that the inertia forces dictate the oil repartition along the axial direction of the piston lands. In particular, it forces oil to repeatedly accumulate against the rings and spread over the piston surface. The spreading of the oil puddles by the inertia force was modelled taking into account two dimensional aspect of the oil flow occurring at the front edge of the puddles. The purpose of the modelling was to predict the effect of the inertia force on the oil motion and compare to the volume of the oil puddle as measured during engine operation. Measurements were obtained as OFT, calibrated intensity of the dual emission wavelength technique. Signal was acquired by the intensified CCD camera for 2-D imaging of the oil distribution (integration of the fluorescence emission over the fluorescence process).

The predictions were in good agreement with the oil distribution profile measured during engine operation. Furthermore, the model suggests that for each engine speed and land width there is minimum oil volume required for the inertia force to establish a net oil transport across the land. The evolution of the oil distribution along the axial direction of the piston lands could now be computed through the engine cycle and therefore correlated with the oil supply to the ring/groove clearances. Such information will be of great use for the future development of oil transport simulation tools.

Two major mechanisms were found to be responsible for oil flows from the piston lands to the ring grooves: the ring pumping which was observed experimentally (Figure 5.3) and the lateral motion of the piston relative to the rings. The quantity of oil flowing into the grooves appeared to be mostly determined by the amount of oil available in front of the ring groove clearances at the time of the ring lift and/or the piston lateral motion. Accordingly, it can be concluded that the rate at which oil from the lands is supplied to the grooves of the top two rings is strongly dependant on the timing of the inertia driven oil displacement and the ring and piston dynamics. By changing the second ring and third land geometry, it was experimentally proven that the amount of oil that accumulates in front of the lower clearance of the second ring groove greatly affects the oil flow rate through the back of the scraper ring. In particular, the combined use of a Napier ring and a cut-off at the top of the third land was found to minimize the rate of oil transport from the third to the second land.



The oil flows from the ring grooves to the piston lands appeared to be governed by the ring squeezing action and the dragging effect of the gases flowing through the ring groove clearances, this latter mechanism being only significant when ring flutter occurs. The experimental observations of the land oil distribution during top ring fluttering clearly showed that gas flows resulting from ring instability could remove most of the oil accumulated in the top ring groove, preventing oil from flowing to the crown land. Furthermore, it was found from engine test results and analytical calculations that the actual gas flow resulting from ring fluttering has little effect on the land oil distribution. The experimental observation further suggested that the direction of the net oil flows through the back of the rings was determined by the location of the main oil supply to the ring groove. Typically, because more oil was supplied to the lower clearance of the ring grooves, oil was flowing from the lower to the upper land. The ring/groove relative angle, which will affect the squeezing process and the flutter-induced gas flow, appeared to affect the rate at which the oil was flowing through the ring grooves but not the direction of the net transport.

The combination of the oil flows across the piston lands and through the ring grooves appeared to be the main path of oil transport towards the crown land. Since oil displacements across the lands and thus oil supply to the lower clearances of the ring grooves depend on the intensity of the inertia force, it was concluded that the rate at which oil is transported to the combustion chamber increases with engine speed.

The model suggested that the rate at which oil is dragged around the lands is proportional to the volumetric blow-by flow rate and thus increases with engine load. Furthermore, it was found that cycle after cycle the cumulative effect of the blow-by gases carries the oil around the lands and through the gaps all the way to the upper skirt region. In fact, the dragging action of the blow-by flow appeared to be the only mechanism by which oil supplied to the upper ring pack can be recycled to the crankcase region.

A comprehensive oil support scheme showing the main path and mechanisms of the oil transport within the piston ring pack was also proposed. It highlighted that the rate at which oil is lost to the combustion chamber results from the competition between the inertia driven oil transport, which carries oil across the land and through the ring



grooves towards the top of the piston, and the gas-driven oil removal, which returns oil to the crankcase. Eventually, a region by region analysis of the piston ring pack indicated that there are three main locations where the rate of oil transport to the combustion chamber can be controlled.

1. The oil control ring which determines the oil supply to the third land and the part of the liner located between the scraper ring and the OCR.
2. The second ring and its groove where the rate of oil transport from the third to second land can be controlled by detailed geometrical features and/or second ring flutter.
3. The second land where the balance between the oil supply and the gas driven oil removal determines the total amount of oil accumulated on the second land and thus the rate of oil supply to the top ring groove.

5.1.2 TOYOTA RESEARCH (INAGAKI ET AL, 1995)

The Toyota researchers (Inagaki et al, 1995) used a single cylinder spark ignition engine and two types of cylinder liners were prepared for the engine. For the motoring test a whole transparent cylinder was used, made of optical glass (BK7) with a thickness of 10 mm. For the firing test a cast iron liner having a pair of tapered slit holes in the axial direction at the thrust and anti-thrust side. Special care was taken so that the steps at the edges of the windows in the bore face were finished to less than 5  $\mu\text{m}$  in height.

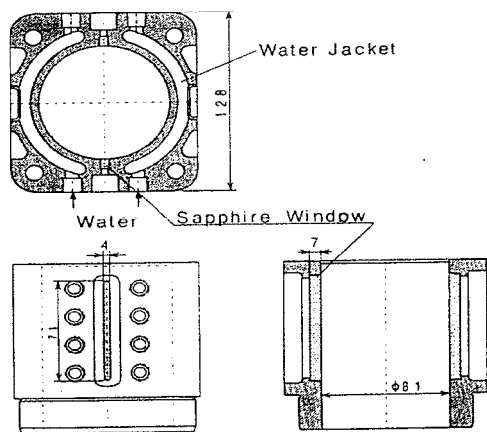


Figure 5.4 Toyota engine – window layout

Sapphire was chosen for the increased temperature gradient along the window (if firing tests were going to be accomplished) and for its superior mechanical properties. Because of the similar thermal expansion coefficient with the sapphire window the gap between the window and the liner will not increase as temperature rises. The window should be glued in the liner and ground afterwards, so that the resulting window and liner surface are perfectly flush.

The researchers used a Xe-flash lamp (blue filter), a filtered charge-coupled device video camera, a PC-based signal analyser and fluorescent dye in lubricating oil. The blue light with a wavelength of 300-500 nm flashed into oil between piston and transparent window with a period of less than 10  $\mu$ s. A fluorescent dye mixed in the lubricant is induced by the blue light and consequently a yellowish-green fluorescence with a peak wavelength of 510 nm is emitted with an intensity appropriate to the oil film thickness through two-dimensional images.

The calibration method measures the OFT distribution between a cylindrical-shaped surface and an optical flat glass. Under high speed (5000 rpm) oil spouts were observed from the first ring gap. The measured oil consumption rate agrees with the variation in the rate of spouted oil film thickness. Very clear images of the oil film thickness distribution on the piston skirt, piston lands and piston rings were obtained under both motoring and firing operating conditions.

### **5.1.3 LEEDS UNIVERSITY RESEARCH ON OIL FILM TRANSPORT (GAMBLE, 2002)**

Gamble (2002) is dealing with the modelling of the piston ring pack and he is also monitoring the lubricant condition. He is giving an extensive description of the oil transport mechanisms in the piston ring assembly so that the lubricant degradation can be studied more effectively. In more details, modelling is divided into two sections:

- a) Piston Ring Pack and b) Piston

#### **5.1.3.1 LUBRICANT CONDITION MONITORING (SAMPLING)**

Oil sampling was conducted at the cylinder wall in diesel engines in the past. Latterly the piston assembly was used for sampling. Samples from top ring show steady degradation throughout a run.

### 5.1.3.2 LUBRICANT DEGRADATION

Degradation will be controlled primarily by the conditions it encounters. Conditions include high temperatures and contact with the combustion gases. As the oil is being transported around the engine it encounters different operating conditions.

### 5.1.3.3 LUBRICANT TRANSPORT

Oil was observed passing through the top and second ring gaps, particularly during engine braking conditions. Hoult and Shaw (1994) observed oil only being transported into the combustion chamber during reverse blow by.

**Rings and cylinder wall:** Gas flow is determined through the gaps and inter-ring gas pressures. Small amount of gas flow passes through the ring-pack, adding to the ring sealing force against the cylinder, thus discouraging oil transport towards the combustion chamber.

Oil may be transported as mist which can travel around the piston in the gas flow. The secondary motion of the piston can produce a large variation in the gas flow through the ring gaps. Circumferential position of piston ring gaps is also significant.

**Oil Transport Mechanisms in the Piston Assembly:** It is necessary to determine how the oil moves around the piston assembly components and what the residence time is. If all the oil film ahead of the ring does not flow beneath the ring face as the piston moves along the cylinder, oil will begin to accumulate at the leading edge of the ring.

Major transport mechanisms taking place in the piston assembly are:

- gas driven flow on piston surfaces
- oil mist entrainment in gas flow around the piston assembly and those on the cylinder wall

- flow at ring cylinder interface
- accumulation in inter-ring volumes
- flow on the cylinder wall through the piston-ring gaps
- inertia driven flow

Shape of piston rings and configuration of piston ring pack have a significant impact on the amount and direction of oil transport. Condition of piston assembly is likely to have a major influence on the length of time that lubrication remains in this region and hence its degradation. Rapid degradation was identified in the experiments because no additives such as antioxidants were present; so additionally to temperature, the combustion process influences the degradation.

TDC oil was sampled through a hole of 1.2 mm diameter drilled radially inward from the back of top compression ring groove and gas chromatography was used to determine if there is any fuel in the oxidised oil. The ring pack samples contain significantly larger fuel content than the sump samples.

Reverse blow-by has been noted to transport oil from the piston second land to the combustion chamber. General trends observed were that oil consumption was found to increase with speed and to some extent load. It also increased with increasing fitted ring gap size. Oil consumption generally increased with speed and size of the top ring gap.

The level of fuel dilution has been shown to influence lubricant transport in the piston assembly which is important for oil consumption. MOFT was significantly reduced when the effect of fuel dilution on lubricant viscosity was considered.

#### 5.1.3.4 CONCLUSIONS

The major points that the research had concluded are:

- Piston assembly tribology is influenced by piston secondary motion through increase or decrease of ring pack gas flow and inter-ring pressures.
- Knowledge of film thickness is crucial. Lubricant transport on the piston was influenced by inertia and gas flow. MOFT is reduced by fuel dilution in lubricant, especially for top compression ring.



- Ring rotation should be taken into account.
- Significant lubricant transport is caused by gas flow and due to inertia.
- The assumed shape for oil accumulation ahead of the piston ring is an important consideration
- Change of piston rings profile has an impact on the amount of lubricant in the piston assembly and direction of oil transport.
- Oil mist production: Oil can be moved between piston assembly and cylinder wall and vice versa in this manner.

5.2 ENGINE SPECIFICATIONS-ENGINE TEST BED

The direct-injection, single-cylinder water-cooled Lister-Petter PHW1 diesel engine used for the LIF engine experiments in previous research projects carried out within the lubrication group (Duszynski, 1999; Pyke, 2000) was coupled to a Plint TE/46 engine test bed. Table 5.1 shows the Lister-Petter engine specifications and Figure 5.4 the dynamometer specifications.

|                         |                     |
|-------------------------|---------------------|
| Cylinder Bore           | 87.31/87.35         |
| Stroke                  | 110 mm              |
| Stroke/Bore ratio       | 1.26:1              |
| Swept volume            | 659 cm <sup>3</sup> |
| Compression ratio       | 16.5:1              |
| Connecting rod length   | 231.9 mm            |
| Conrod/crank ratio      | 4.22:1              |
| Oil capacity            | 2.8 l               |
| Max engine torque       | 32.5 Nm@1700 rpm    |
| Max Speed               | 2000 rpm            |
| Tangential Ring Tension |                     |
| Top ring                | 14.36-19.21 N       |
| 2 <sup>nd</sup>         | 14.54-19.43 N       |
| 3 <sup>rd</sup>         | 13.16-17.57 N       |
| 4 <sup>th</sup>         | 21.17-28.6 N        |

Table 5.1 Lister-Petter Engine Specifications



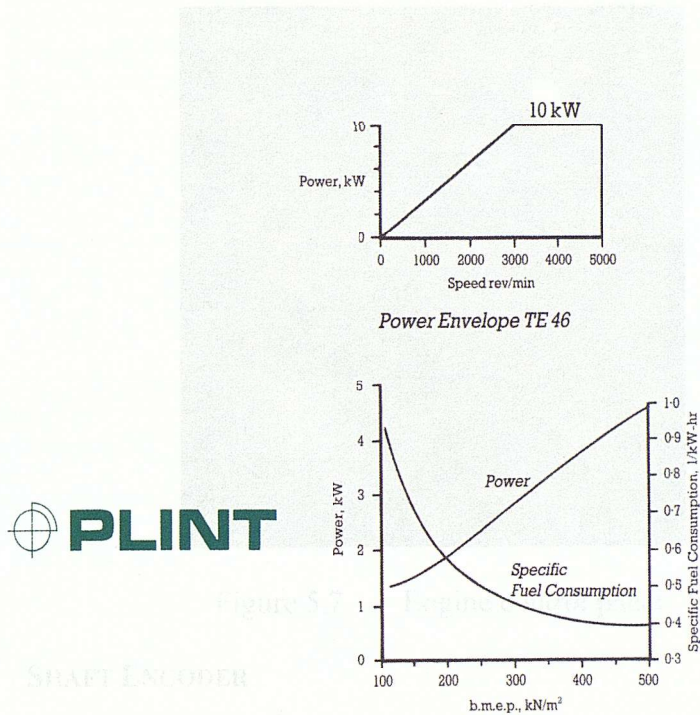


Figure 5.5 Plint Dyno specifications

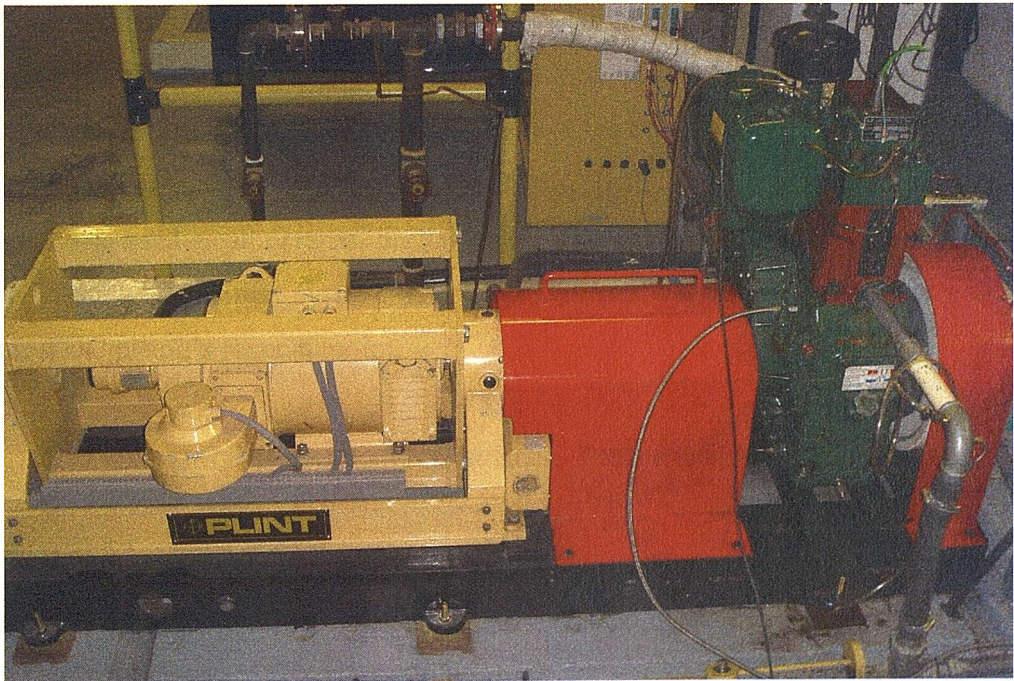


Figure 5.6 Plint dyno coupled to the Lister Petter engine





Figure 5.7 Engine control panel

### 5.2.1 SHAFT ENCODER

A shaft encoder, Muirhead Vactric Electric Model E25BC-4H-1000, was coupled directly to the engine's crankshaft and provided 3 TTL pulse trains of output signal. Two were in quadrature channels of 1000 and one channel was 1 ppr (pulse per revolution). In order to further enhance the resolution of the encoder, the two 1000 ppr channels were electronically gated (XOR) and, as a result, a choice of 1000 or 2000 ppr signals was available. That corresponded to 0.36 or 0.18 deg of crank angle per pulse. That signal was used as an external digital trigger providing the frequency of a triggered scan for the data acquisition system. The 1 ppr channel was used as a marker pulse to detect the TDC piston's reversal point.

### 5.2.2 EXISTING MEASUREMENT PROBES – PREVIOUS ENGINE BLOCK

The engine cylinder head was fitted with a Kistler 7061A quartz pressure transducer allowing the cylinder pressure to be recorded throughout the engine cycle by the data acquisition system. Thermocouple probes were used to measure the temperature in the oil sump, the coolant entrant and outlet, the air inlet and the exhaust gas temperature. The block and liner was fitted with optical fibre sensors, one at TDC thrust

side and three at TDC, mid-stroke and BDC. The old block was later replaced with a new one incorporating quartz windows as well.

5.2.3 CYLINDER HEAD PRESSURE TRANSDUCER CALIBRATION – PROXIMITY SENSOR

The Kistler 7061 pressure transducer mounted on the cylinder head of the Lister-Petter engine was taken out to be recalibrated but eventually failed. So, it was replaced by an AVL 144 (120P – 250 C) water cooled pressure transducer. The transducer will give the reference pulse (top dead centre-TDC at compression stroke) so that an accurate triggering can be set for the experiments that require knowledge of the piston position at all times.

For the calibration procedure a dead weight tester was used and a Type 5007 Kistler Charge Amplifier which is connected directly to an AVL transducer. The amplifier was set at 10 units sensitivity (10 mV/psi) and the gain was set at 80% when no pressure was applied at the dead weight tester. The calibration data were obtained with the amplifier set to “SHORT”, so that the voltage won’t be decaying while taking the readings from the dead weight tester. Decay happens due to the nature of the transducer. Since the first tests will be under motoring conditions, it was not considered necessary to go much further than the maximum motoring pressure stated from the manufacturer which is  $P_{MAX}=20.68$  bar. Maximum combustion pressure is rated at 93.08 bar.

The following calibration curve was obtained:

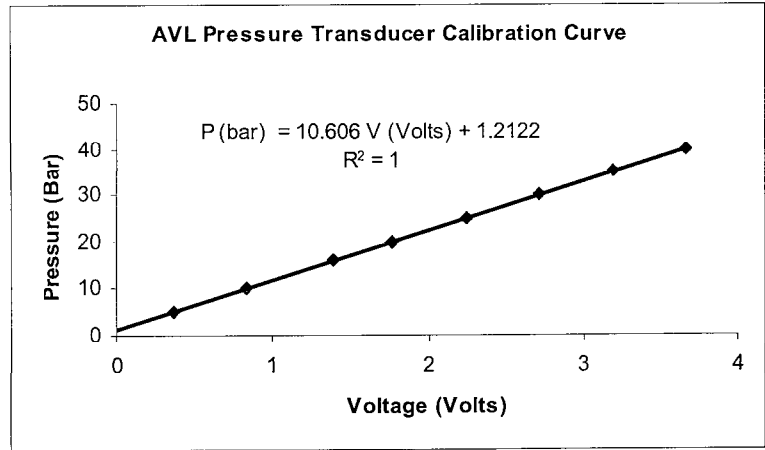


Figure 5.8 AVL pressure transducer calibration curve



The above curve gives a very good linear fit for the pressure transducer calibration data.

Another reference pulse is also needed along with the shaft encoder pulses (1 per revolution and 1000 or 2000 pulses per revolution) and this pulse is being given by a proximity gap sensor, manufactured by IMO (with M12 thread) which is fitted on top of the valve rocker adjuster and will detect the opening of the exhaust valve. This will provide the reference pulse needed so that the exact position of the piston can be found at all times while the stroke is identified as well. The output signals that will enable taking a trigger output at the desired crank angle are:

- 1 pulse per 4-stroke cycle, at TDC compression
- 1 pulse per 4-stroke cycle, at TDC exhaust
- 1 pulse for the marker output, the digital signal of the AVL pressure transducer

### 5.3 EXPERIMENTAL SET-UP

A new engine block was fitted to the Lister-Petter engine. The block was heavily modified, so that images from the lubricant as the engine is motored could be taken.

#### 5.3.1 MODIFIED ENGINE BLOCK AND LINER VIEWS

For the purpose of visualisation, pockets had to be machined in the sides of the engine block so that the viewing area is large enough to enable the imaging process. Figures 5.9 and 5.10 show the drawings of the machined pockets in the liner surface. The pockets were spark eroded according to Figure 5.10. The quartz windows were made to suit the spark eroded pockets. Above, between and below the window pockets, holes have been machined to accommodate the pressure transducer and the optical fibres.

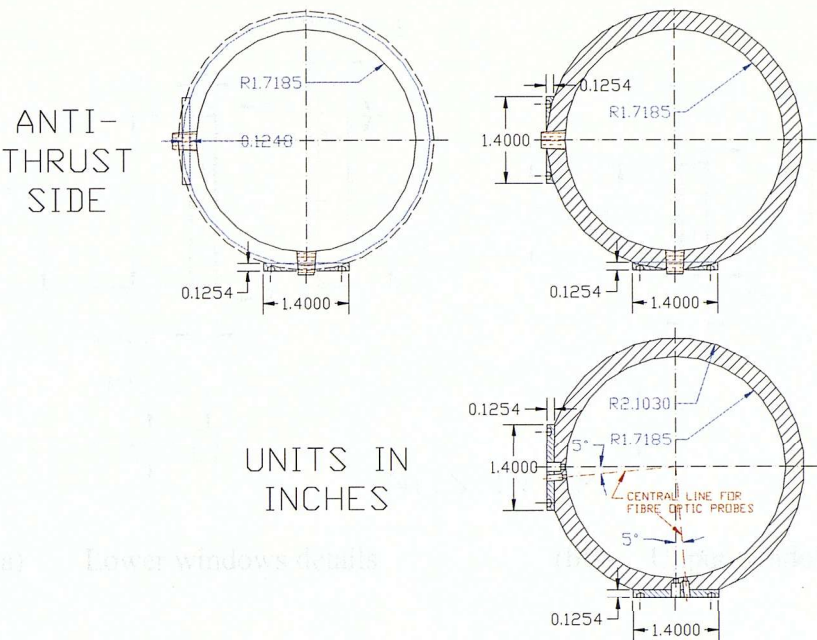


Figure 5.9 Quartz window locations on the Lister-Petter engine liner

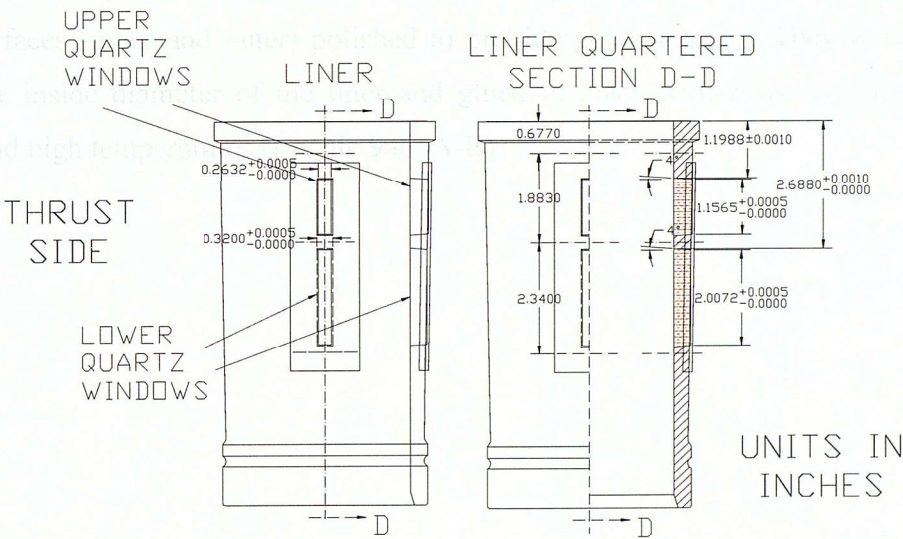


Figure 5.10 The liner with the quartz window pockets

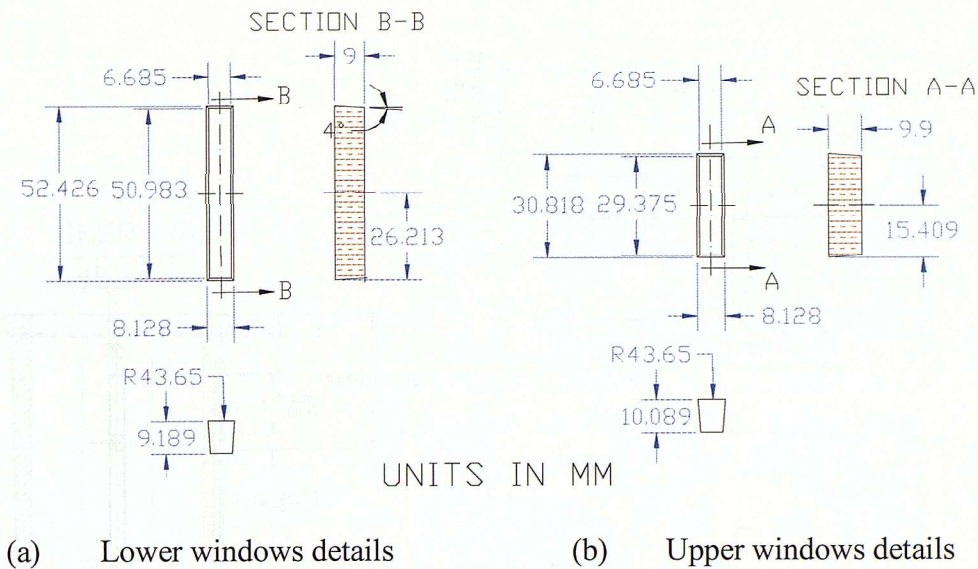


Figure 5.11 Lower and upper quartz window details

The quartz windows have the liner curvature at their inner surface and have both surfaces (inner and outer) polished to provide good imaging. They were installed from the inside diameter of the liner and glued in place with a special glue that can withstand high temperatures (Loctite 9464A-B).

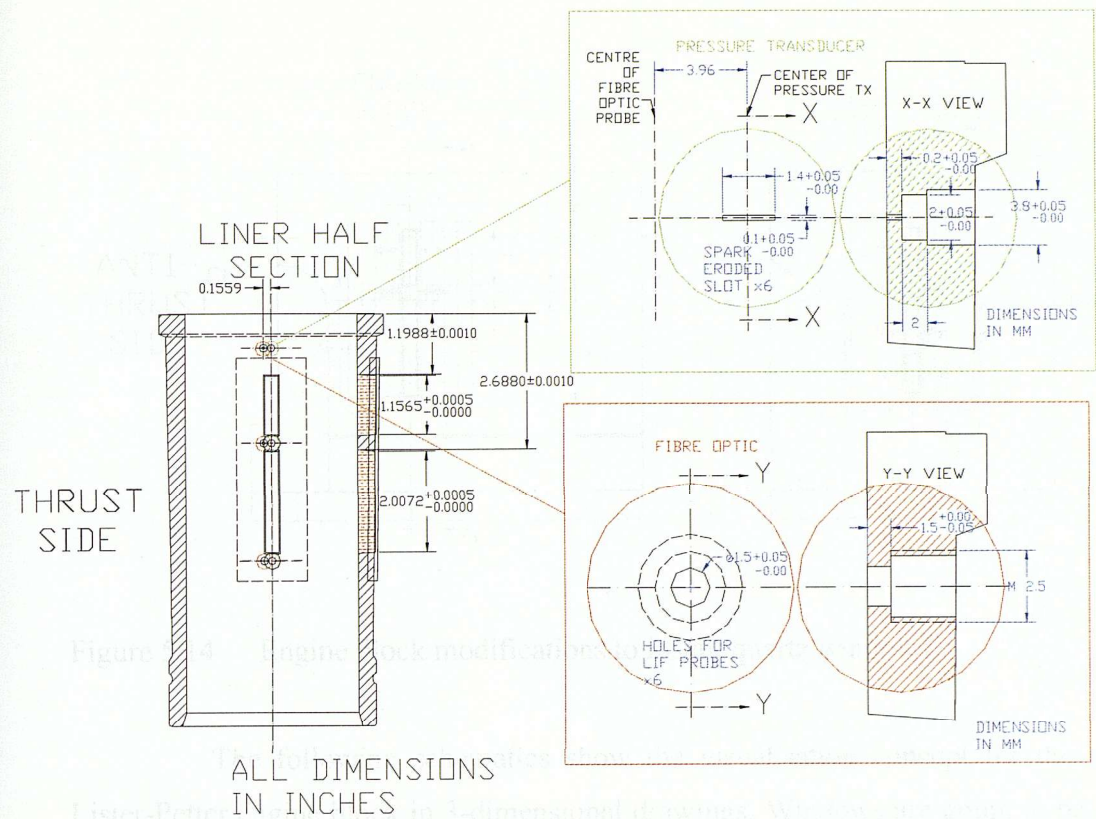


Figure 5.12 Details of the pressure transducer fittings and fibre holes

The 3-D drawing of the liner and block can be seen in Figure 5.15. Front side and anti-thrust sides of the block are used for the purposes of imaging and visualisation. The re-machined engine block drawings can be seen in Figures 5.13 and 5.14.

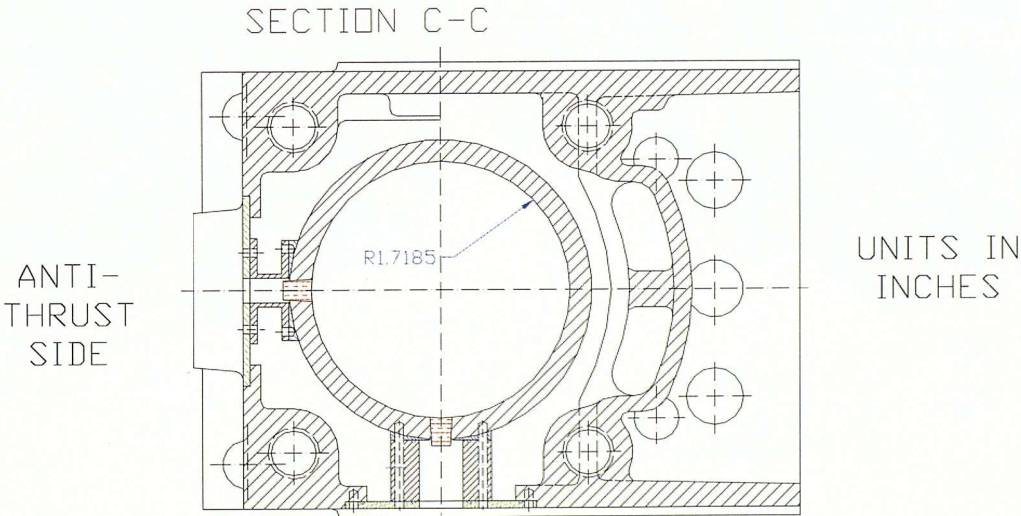


Figure 5.13 Cut Engine block and cylinder liner with quartz windows



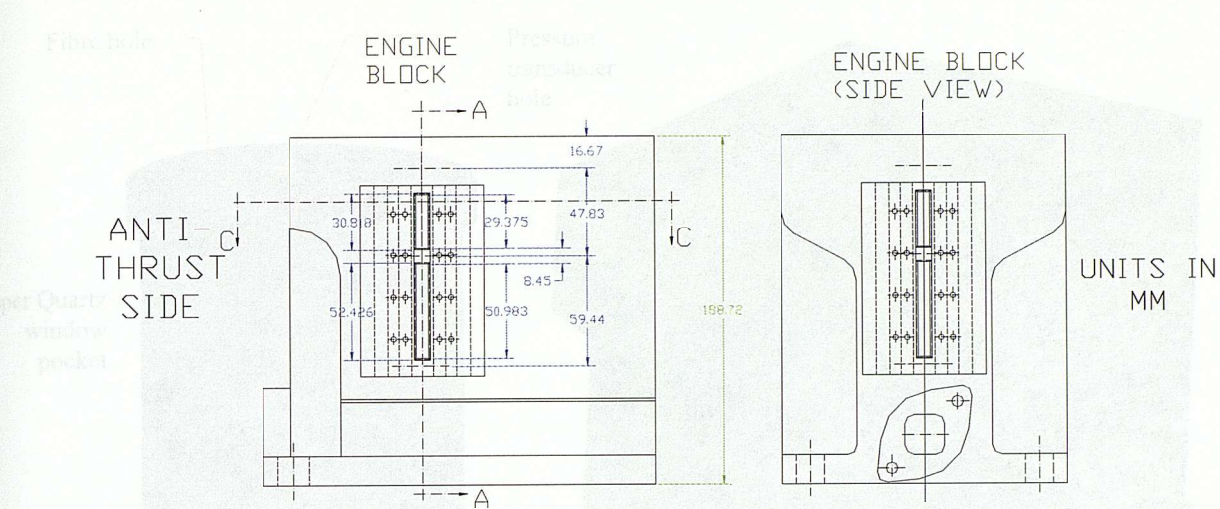


Figure 5.14 Engine block modifications to install quartz windows

The following schematics show the visualisation concept for the modified Lister-Petter engine block in 3-dimensional drawings. Windows are going to be installed in two sides of the liner assembly front side and anti-thrust side. The piston, block with window pockets, liner pressure transducer holes and optical fibre holes can clearly be seen. Photos of the block follow (Figure 5.16) as well as magnification of the areas to be visualised in the experiments at both sides.

Figure 5.15 3-D drawings of the modified Lister-Petter engine block



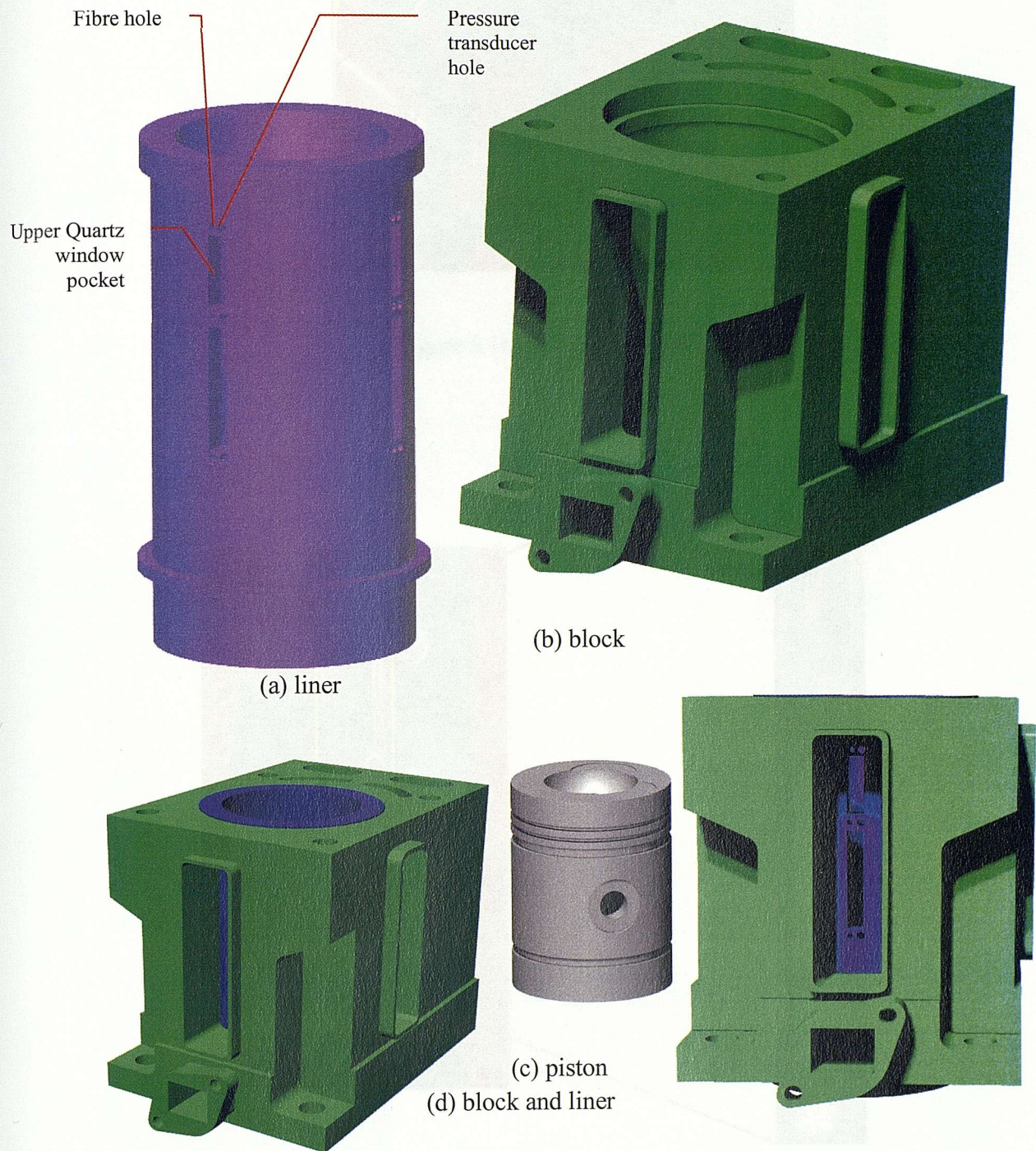


Figure 5.17 Upper and lower window, anti-thrust side window visualisation area

Figure 5.15 3-D drawings of the modified Lister-Petter engine block



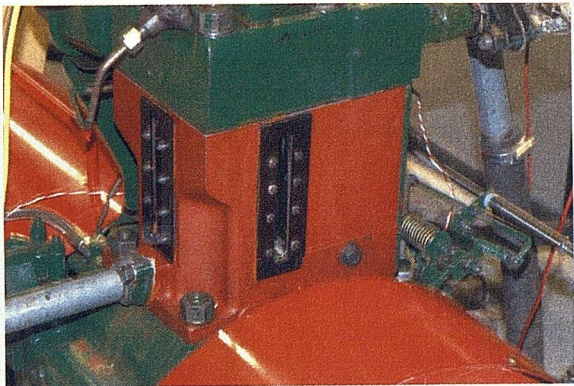


Figure 5.16 View of the block

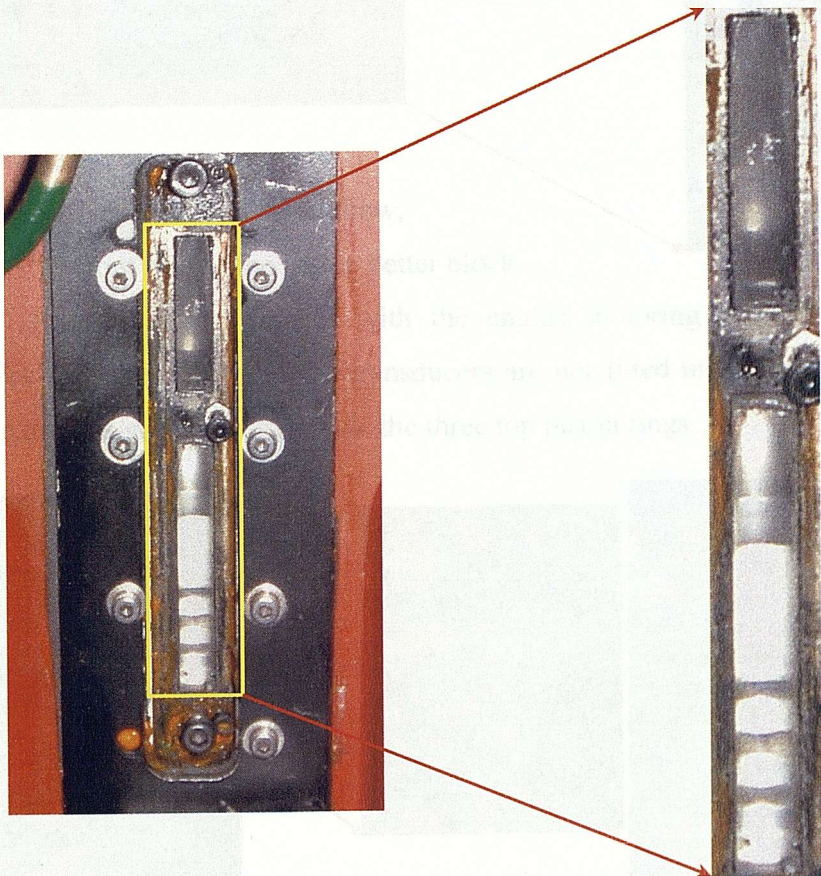


Figure 5.17 Upper and lower window, anti-thrust side window visualisation area



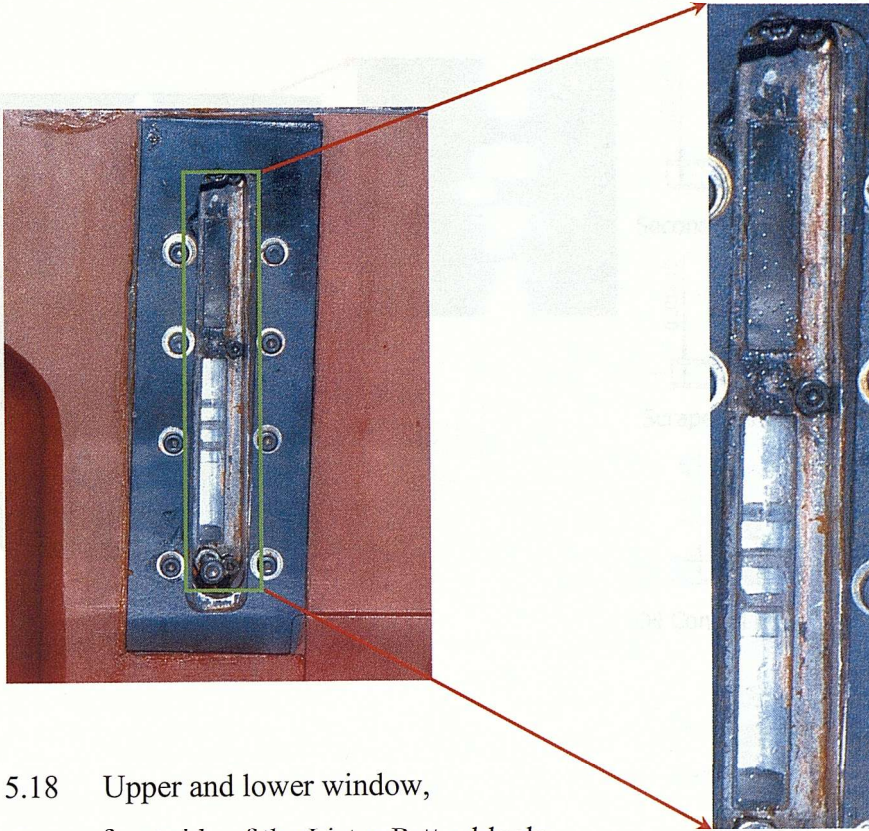


Figure 5.18 Upper and lower window,  
front side of the Lister-Petter block

Prior to taking images with the engine motoring only, the optical fibre transducers as well as the pressure transducers are not fitted in the block. The areas of interest in the piston ring pack include the three top piston rings.

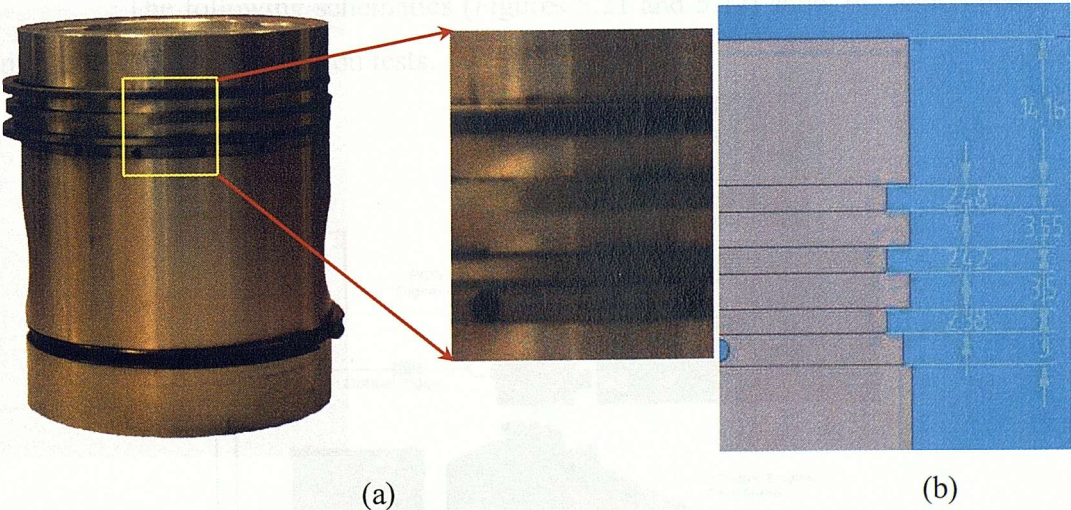


Figure 5.19 Visualisation of piston ring pack: (a) Piston anti-thrust side  
(b) Piston land and groove dimensions (mm)



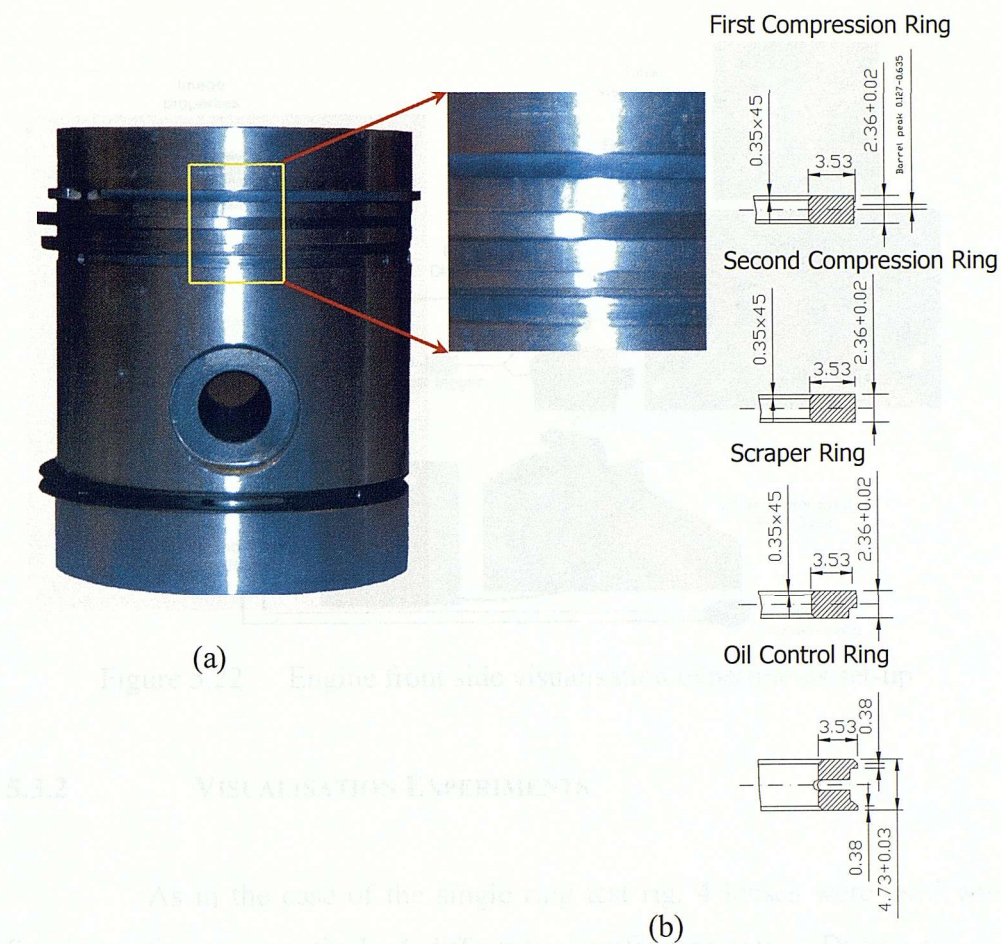


Figure 5.20 Visualisation of piston ring pack: (a) Piston front side (b) Piston ring-pack details

The following schematics (Figures 5.21 and 5.22) show the experimental set-up for the engine visualisation tests.

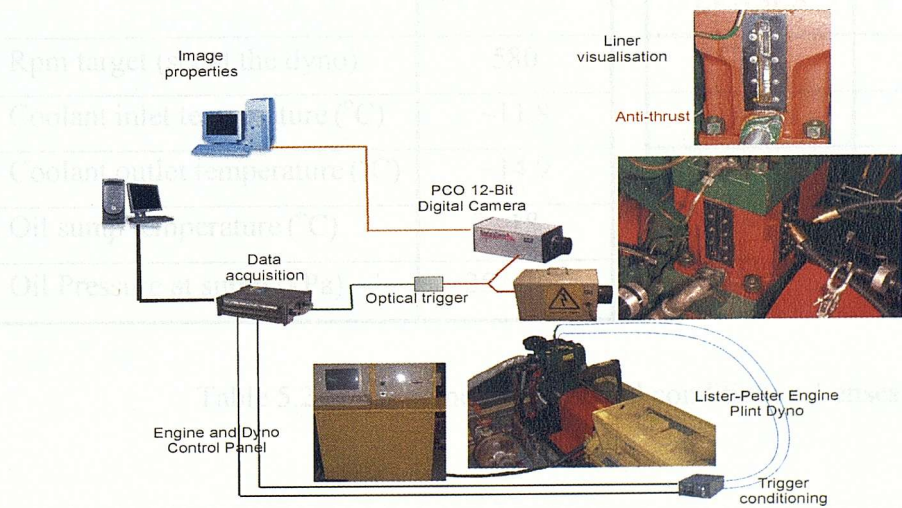


Figure 5.21 Engine anti-thrust side visualisation set-up

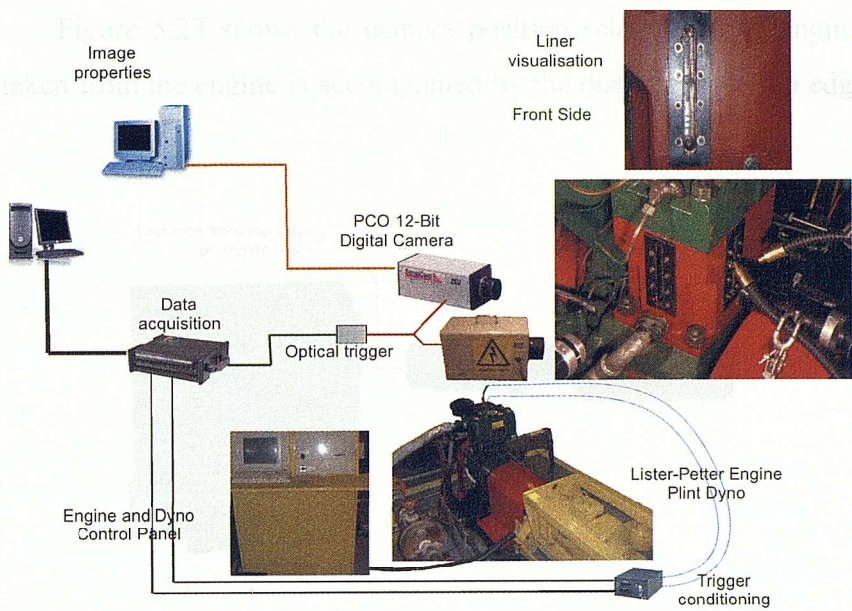


Figure 5.22 Engine front side visualisation experiments set-up

5.3.2 VISUALISATION EXPERIMENTS

As in the case of the single ring test rig, 4 lenses were used with the PCO Sensicam giving respectively 4 different magnification ratios. During the experiments, the engine was running under the following conditions:

|                                 |         |            |               |
|---------------------------------|---------|------------|---------------|
| Rpm                             | 346-388 | CCD LENSES | MAGNIFICATION |
| Rpm target (set at the dyno)    | 580     | CF-1       | 1.4           |
| Coolant inlet temperature (°C)  | ~11.8   | CF-2       | 3.1           |
| Coolant outlet temperature (°C) | ~14.9   | CF-3       | 5.3           |
| Oil sump temperature (°C)       | ~18     | CF-4       | 6.8           |
| Oil Pressure at sump (kPa)      | 252-264 |            |               |

Table 5.2 Engine experimental conditions – Lenses used



Figure 5.23 shows the camera position relative to the engine block. Every image taken from the engine is accompanied by the distance from top edge of the engine block:

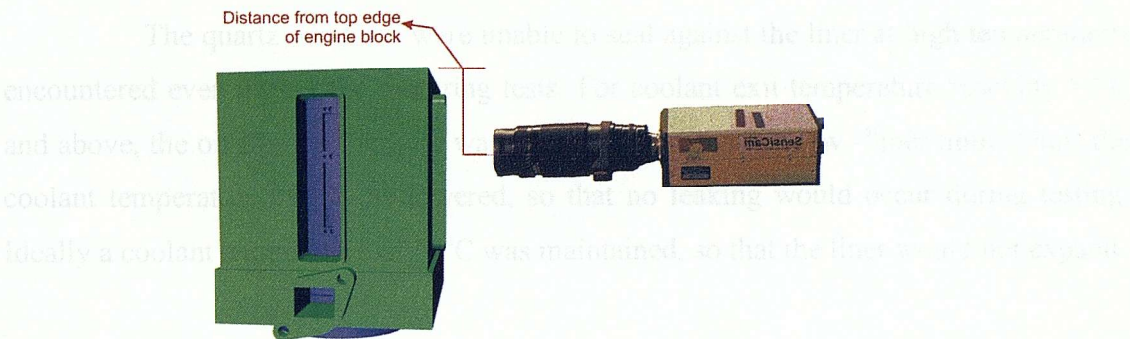


Figure 5.23 Camera distance from top edge of Lister-Petter engine block

The following set of visualisation images shows the piston-rings (top compression, second compression and scraper ring) as well as the window itself without the piston behind it, with oil droplets appearing on the photos.

During the Lister-Petter engine visualisation experiments, better images were taken from the lower windows, at both anti-thrust side and front side. The LIF measurement probes were not fitted prior to the acquisition of the signal from the pressure transducer. This Kulite unit, (XCQ-080 500 psi (absolute)) failed as the very small wiring of the Wheatstone bridge amplifier was snapped due to fitting and removal of the pressure measuring unit from the measuring stations. As far as the measuring stations are concerned, from the six pressure monitoring points the four were saved; the necessary machining on the outer surface of the engine liner distorted the bore. Thus, re-boring the liner (0.020 in) and oversize piston fitting was inevitable but eventually two of the spark eroded slots for the pressure measuring stations were lost (at mid-stroke, thrust and anti-thrust side). In Figure 5.72 the surface profile of the plug fitted at the anti-thrust side of the mid-stroke pressure measuring hole can be seen.

Poor visibility through the windows was encountered with the fibre optics light source (spark light) but still the CF-4 imaging of the strings proved to be poor. Blurred images, especially at high magnification, were due to the difference of the



focusing point distance. Even the minimal change would alter the quality of the image, as the focus was adjusted with the engine stopped and at the central vertical line of the image. Engine vibration during motoring and the curvature of the piston and rings that are going to be visualised are sources of focusing misalignment.

The quartz windows were unable to seal against the liner at high temperatures encountered even during the motoring tests. For coolant exit temperature reaching 50°C and above, the oil film on the liner was coming out of the window - liner limit. Thus, the coolant temperature had to be lowered, so that no leaking would occur during testing. Ideally a coolant temperature of 15°C was maintained, so that the liner would not expand.

5.3.2.1 ENGINE ANTI-THRUST SIDE - CF-1 LOWER WINDOW

The CF-1 set of images provide an overview of the piston ring pack.

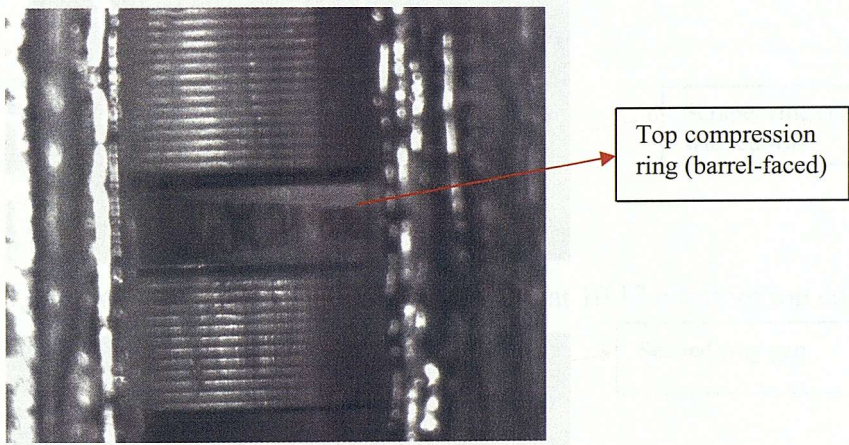


Figure 5.24 580.32° CA exhaust stroke camera at 10.17 cm from top edge



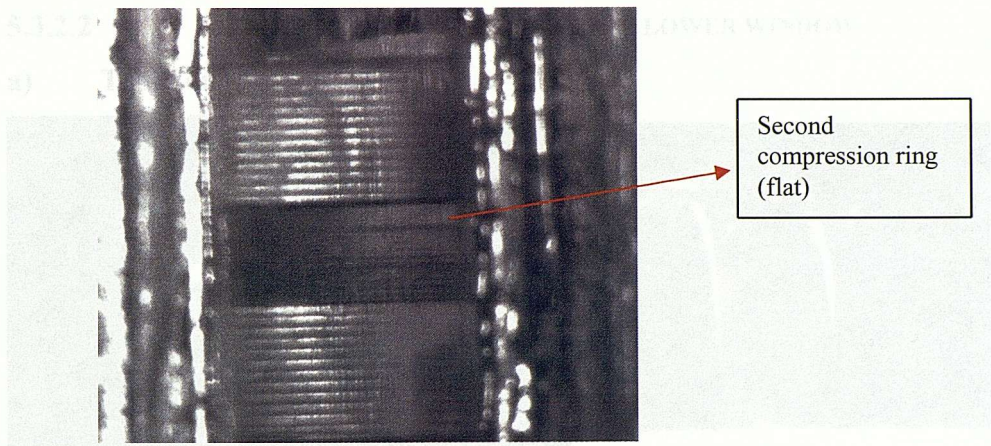


Figure 5.25 609.48° CA exhaust stroke, camera at 10.17 cm from top edge

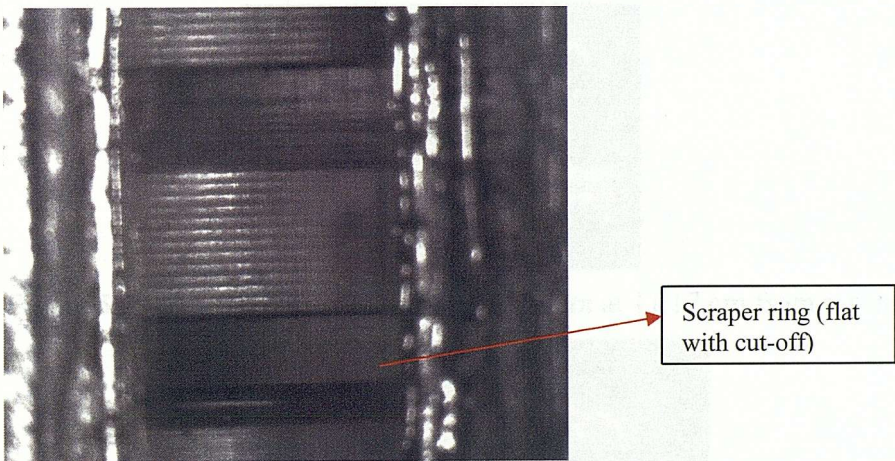


Figure 5.26 621.90° CA exhaust stroke, camera at 10.17 cm from top edge

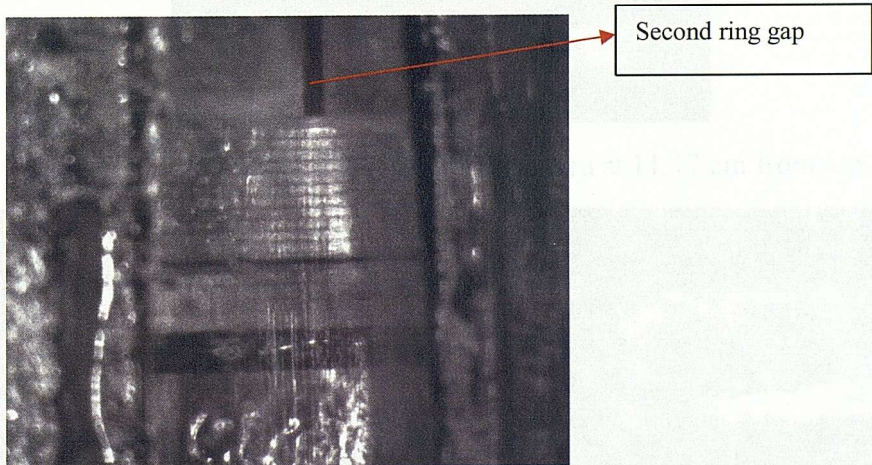


Figure 5.27 622.62° CA exhaust stroke, camera at 10.17 cm from top edge



5.3.2.2ENGINE ANTI-THRUST SIDE - CF-2 LOWER WINDOW

a)Top Compression Ring

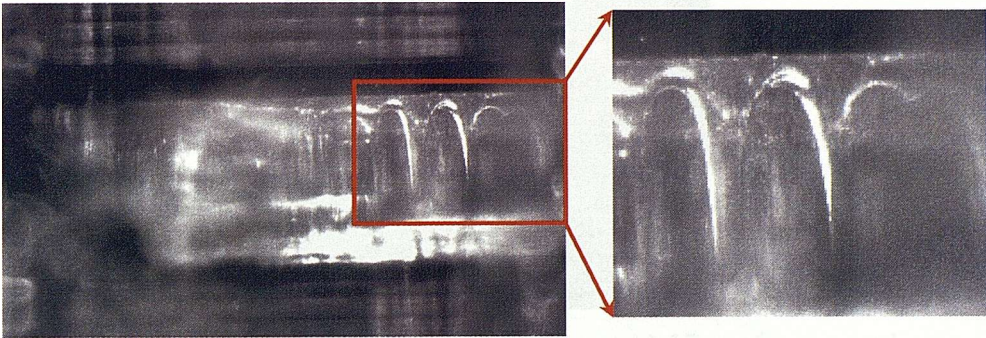


Figure 5.28576° CA exhaust stroke camera at 11.17 cm from top edge

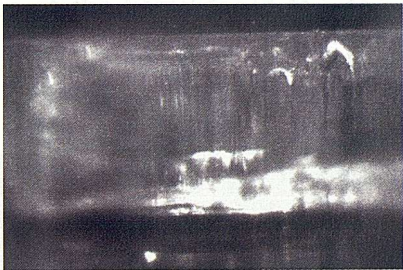


Figure 5.29576° CA exhaust stroke camera at 11.17 cm from top edge

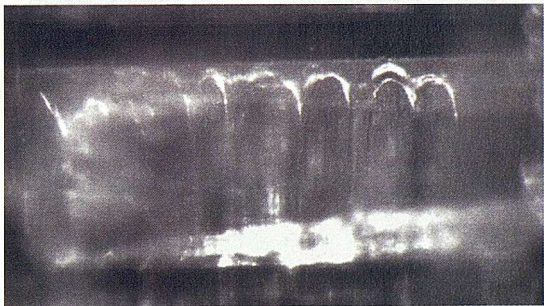


Figure 5.30576° CA exhaust stroke camera at 11.17 cm from top edge

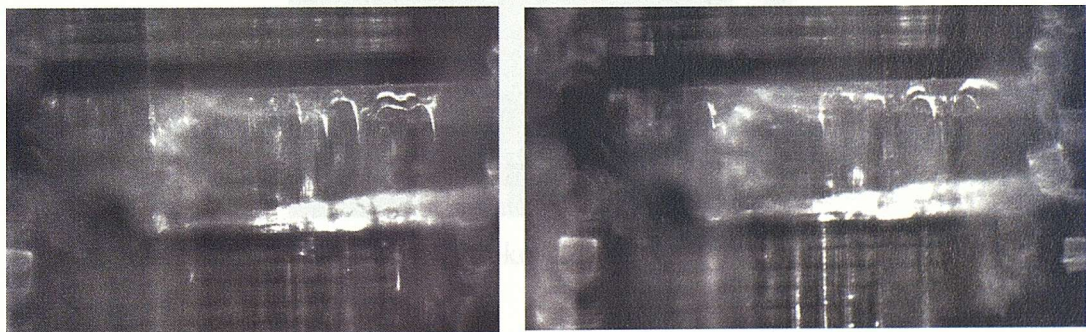


Figure 5.31576° CA exhaust stroke camera at 11.17 cm from top edge



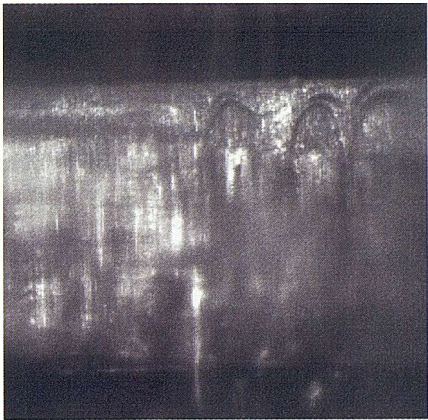


Figure 5.32 576° CA exhaust stroke camera at 11.17 cm from top edge

This set of pictures (Figures 5.28 – 5.32) gives evidence of the string cavities that appear on the surface of the barrel-faced top compression ring. In the images to follow the oil film is visualised on the piston-ring surface.

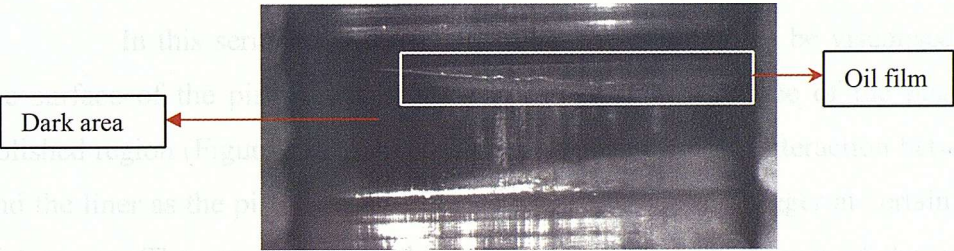


Figure 5.33 608.40° CA exhaust stroke camera at 9.17 cm from top edge

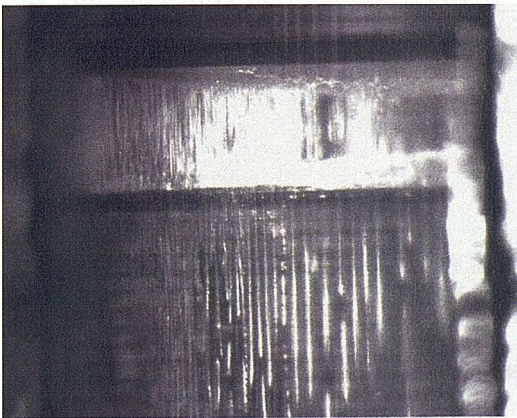


Figure 5.34 608.40° CA exhaust stroke camera at 9.17 cm from top edge