In this series of experimental data, the oil film can be visualised, as seen on the surface of the piston-ring. The darker area on the surface of the piston ring is a polished region (Figure 5.33), that has been derived from the interaction between the ring and the liner as the piston reciprocates. This area is getting bigger at certain areas of the piston ring. The reason behind this is that the tangential tension of the piston ring is getting bigger towards the edge of the ring next to the ring gap and the oil film is being squeezed. Eventually, the ring and liner surface are prone to more friction close to the gaps.

b) Second compression ring

The second compression piston ring (flat ring) is being visualised, and an oil film region can clearly be seen, leaving a minimum area uncovered, at the upper edge of the second piston ring specimen. Oil droplets are also visualised.
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Figure 5.36  615.24° CA exhaust stroke, camera at 9.17 cm from top edge

c) Scraper ring

Figure 5.37  621.90° CA exhaust stroke camera at 9.17 cm from top edge

Oil droplets appear on the third piston ring specimen. Their appearance is subject to uncertainties.
In the above image, oil film is clearly seen on the surface of the third piston-ring specimen. Fully flooded conditions are observed for the second compression ring and the scraper ring.

Image 5.39 shows string cavities formed with the piston moving downstroke at the top compression ring.

Figure 5.40 493.20° CA expansion stroke, camera at 9.17 cm from top edge
Figure 5.41 475.20° CA "expansion" stroke camera at 9.17 cm from top edge

Figure 5.42 475.20° CA "expansion" stroke camera at 9.17 cm from top edge

Figure 5.43 540° CA exhaust stroke, camera at 9.17 cm from top edge
Figure 5.44  540° CA exhaust stroke camera at 9.17 cm from top edge

Figure 5.45  540° CA exhaust stroke camera at 9.17 cm from top edge

Figure 5.46  540° CA exhaust stroke, camera at 9.17 cm from top edge
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Figure 5.47  540° CA exhaust stroke camera at 9.17 cm from top edge

Figure 5.48  540° CA exhaust stroke camera at 9.17 cm from top edge

Images 5.41 – 5.48 provide evidence of the existence of oil droplets and oil mist. The droplets are similar to the ones noticed in Figures 5.36 and 5.37. The evidence of oil mist, though, (Figure 5.41) is unclear.

5.3.2.3 ENGINE ANTI-THRUST SIDE - CF-3 UPPER WINDOW

In Figure 5.49 the piston ring gap can be directly compared to the ring side clearance on top of the second compression piston ring. The images were taken from the upper window.
Figure 5.49  303.66° CA compression stroke, camera at 4.67 cm from top edge

Figure 5.50  254.88° CA compression stroke camera at 8.67 cm from top edge

The above image shows the developed oil film on the surface and droplets underneath the piston ring, at the lower piston ring side clearance.

Figure 5.51  255.24° CA compression stroke, camera at 8.67 cm from top edge
Figure 5.51 shows oil film breakage as there is a big “gap” – cavity on the surface of the ring, resembling an irregular shape of the string cavities.

Figure 5.52  255.24° CA compression stroke, camera at 8.67 cm from top edge

Figure 5.53  255.24° CA compression stroke, camera at 8.67 cm from top edge

Figure 5.54  255.24° CA compression stroke, camera at 8.67 cm from top edge
Again, Figures 5.55 and 5.56 provide the shape of more irregular shaped string cavities.

The strings do not appear on this image, as they have been expanded and appear as a larger cavity which spreads to a comparatively greater length on the piston-ring surface.
As the tension close to the ring gap is higher, the oil film is squeezed further towards the inner quartz window surface (Figure 5.57).

Figure 5.57  254.88° CA compression stroke camera at 8.67 cm from top edge

Figure 5.58  218.34° CA compression stroke, camera at 11.67 cm from top edge

Figure 5.59  218.34° CA compression stroke, camera at 8.67 cm from top edge
In this paragraph oil droplets at the ring side clearance were visualised (Figures 5.52 and 5.53). The effect of ring tension on the oil film became also evident, causing along with ring twist irregular shaped cavities, emerging from string diminishing (Figures 5.55 – 5.59).

### 5.3.2.4 ENGINE FRONT – SIDE CF-4 LOWER WINDOW

This set of images is using the highest magnification of the available CCD camera lenses. Figures 5.60 and 5.61 show the piston skirt and oil mist generated by the oil control ring, as the piston moves towards BDC. In both of the following images, a "lubricant cloud" is evident. Lubricant is transported forming oil mist.

![Figure 5.60](image1.png) 18° CA induction stroke, camera at 9.07 cm from top edge

![Figure 5.61](image2.png) (a) 48.60° CA induction stroke, camera at 9.07 cm from top edge
(b) 142.20° CA induction stroke, camera at 16.07 cm from top edge
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Figure 5.62  122.94° CA induction stroke, camera at 16.07 cm from top edge

Bubbles appear at the piston-ring bottom side clearance (Figure 5.62)

Figure 5.63  124.20° CA induction stroke, camera at 16.07 cm from top edge

Figure 5.64  124.20° CA induction stroke, camera at 16.07 cm from top edge
String cavities on the surface of the piston ring combined with oil droplets on the top ring clearance (Figure 5.65 and magnified section)

Figure 5.66  124.20° CA induction stroke, camera at 16.07 cm from top edge
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Figure 5.67 124.20° CA induction stroke, camera at 16.07 cm from top edge

Figure 5.68 124.20° CA induction stroke, camera at 16.07 cm from top edge

Figure 5.69 124.38° CA induction stroke, camera at 16.07 cm from top edge
The images above show droplets gathered at the top and bottom of the piston ring side clearance (Figures 5.62 – 5.69). There is also a quite unclear appearance of a string cavity on the surface of top compression ring at engine front side (Figures 5.67 and 5.68). These string cavities appear to be more even than the cavities noticed in paragraph 5.3.2.2 (engine anti-thrust side). Since the CF-4 images are not clear enough, it is unsafe to draw any further conclusions.

The identification of oil droplets in general can be directly compared to the research that was presented by Toyota (Inagaki et al, 1995). They have captured droplets in their images, as well as oil spouts moving towards the combustion chamber as the piston is moving to BDC. This was identified as one way to transport oil towards TDC. In paragraph 5.6, the oil mist generation is explained with images that capture droplets emerging from the ring groove. In Figures 5.41, 5.60 and 5.61 oil mist is observed. It is possible when the gas is travelling with a high enough velocity that droplets of oil may be sucked from the surface of the oil film as it passes. Droplet entrainment from the oil film on the cylinder wall to the ring gaps, where the gas velocity is high (Tian et al, 1998) should also be considered. A greater amount of oil mist generation is likely to occur at the piston ring gap due to the faster flowing blow-by gas at this point.

5.4 CAVITATION IN ENGINE COMPRESSION RING - INCEPTION

The research approach to understanding cavitation that was clearly identified in images 5.28 – 5.32 has followed the outcome of the visualisation research study of the single-ring test rig. The idealised single-ring test rig is being used as a large scale model in contrast to the real sized engine test rig. In the case of the single-ring test rig, the ring was covered in the majority of the stroke with string cavities. During flow reversal, the cavitation is initiated by fern-shaped cavities which grow further, to form up bigger ferns and fissure type cavities. Later on in the stroke they develop to strings on the surface of the piston-ring and they grow so that they will be reaching its trailing edge.

In the case of the engine, however, the stages of cavitation inception and initial development are not clear because the size of visualisation window has hindered imaging of the oil flow at the reversal points close to the dead centres. On one hand, the
quartz sectioned liner provided cavitation images similar to the ones recorded for the single-ring test rig, on another, the overall size and their span compared to the stroke length is not adequate to "scan" the whole of the engine stroke length. In Chapter 4, in the visualisation parametric study for the single-ring test rig, it was evident that as speed increases, the fern cavities appear later on in the stroke and in the case of the load increase, the opposite happens. For the motoring tests, it was not possible to have a good picture of the squeeze film effect and how does cavitation initiation change in the case of speed and load increase. Although speed was maintained in quite low levels its inception was not captured. For higher speed imaging, where the cavities are formed later on in the stroke, still no clear signs were identified. It should be pointed out, though, that for the case of the anti-thrust and front side windows, for the same speed and load, different cavitation initiation points in the stroke should be considered, as the load on the ring changes along the circumference due to piston tilt.

The following schematic shows the most probable cavitation development in the engine during the stroke according to the images taken. The stages derived, are postulated from engine imaging and the simulation test rig results visualisation data (Chapter 4).
Possible Cavitation Stages

(A) Fern-Shaped Cavities

Possible Cavitation Stages

(B) Irregular Fern Growth

Visualised Cavitation Stage

(C) String Cavities

Figure 5.70 Cavitation stages on the Lister-Petter engine
In the above schematic, there are three cavitation stages. (A) and (B) are possible cavitation stages that lead to the latest cavitation development (C) which is visualised in the experiments (string cavities). The schematic also combines the probable oil film pressure for the cavities in every stage respectively. As in the case of the single ring test rig, the cavitation inception is being accompanied by negative or sub-atmospheric pressures. In the converging edge of the piston-ring, the pressure is the combustion chamber pressure and at the diverging wedge the pressure reading would be that of the second land. Later on in the stroke (Figure 5.70 (B) – irregular fern growth) the ferns grow, and they are combined with sub-atmospheric oil film pressure at the diverging wedge of the piston-ring. Again, the pressure reading at the diverging wedge ends to the pressure measurement of the second land. The string cavities (Figure 5.70 (C) can reach the trailing edge of the piston ring or they can be cut midway. The visualised cavity shapes are not uniform. The pressure reading at the diverging wedge is that of the second land. Figure 5.70 refers to the compression stroke.

The ideal situation of the single ring test rig is confronted with the real size engine test rig experiments, where more parameters are affecting the piston ring pack during reciprocation.

Segments of the surface profile of the liner with the quartz windows fitted are given in Figures 5.71 – 5.78. The Talysurf surface profilometry provided these results that show which is the difference between the two surfaces in terms of surface roughness and how big is the gap in the longitudinal direction between the liner surface and the quartz window, after motoring the engine and having acquired the imaging data.

a) Anti-thrust side – limits between liner and windows

![Modified Profile](image-url)
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Figure 5.72  Upper side of lower window and liner limit

Figure 5.73  Lower side of upper window

Figure 5.74  Lower side of lower window (BDC)
(b) Front side – Limits between windows and liner surface

Figure 5.75 Upper side of lower window

Figure 5.76 Lower side of upper window

Figure 5.77 Upper side of upper window (TDC)
There are relatively big gaps in every window and liner limit. The range of the gap is between 40 and 650 microns. Those big gaps are affecting the lubrication as the piston reciprocates. The link between the surfaces is not uniform. When the engine block was taken out after the experiments, the windows surface was no longer as clear as it had been during fitting. The interaction between the piston – rings and the windows, made the surface look blurred. It should be noted, that the Talysurf was unable to provide a profile in the circumference of the liner due to geometrical restrictions. So, no surface roughness profiles were taken along the liner’s diameter.

5.4.1 TEST RIG – LISTER ENGINE VISUALISATION EXPERIMENTS COMPARISON

In this paragraph an attempt is made to give a comparison of the imaging taken from both experimental test rigs, based on the testing parameters and the image properties.

5.4.1.1 CAVITATION FACTOR

In Figure 5.79 the fully developed string cavities at the single-ring test rig, using the optical liner at 300 rpm, 971 N/m load and at 90° (mid-stroke) is shown. This image captures the whole piston – ring specimen which has a width of 5 mm. The sizes of
the fully developed string cavities compared to the known size of the piston ring specimen are compared in Figure 5.79:

![Image](image_url)

Figure 5.79 Single ring test rig visualisation at 300 rpm midstroke

<table>
<thead>
<tr>
<th>Test Conditions</th>
<th>Single Ring Test Rig</th>
<th>Lister – Petter Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed (rpm)</td>
<td>300</td>
<td>398</td>
</tr>
<tr>
<td>Load (N/m)</td>
<td>971</td>
<td>motoring</td>
</tr>
<tr>
<td>Stroke Length (mm)</td>
<td>50</td>
<td>110</td>
</tr>
<tr>
<td>Linear Velocity (m/sec)</td>
<td>1.728</td>
<td>2.016</td>
</tr>
<tr>
<td>Distance from (mm)</td>
<td>TDC: 25</td>
<td>top edge of top land at TDC: 111.99</td>
</tr>
</tbody>
</table>

Table 5.3 Test-rig and engine test conditions comparison

The following image (Figure 5.80) is taken with a CF-2 magnification lens with the PCO digital camera at the Lister-Petter test rig, at the anti-thrust side of the engine. The image shows the whole width of the piston ring groove and first compression barrel-faced piston ring, as well as the width of the strings that clearly appear again on the engine visualisation photos. From the image a coefficient can be derived for comparison of the cavities appearing on the surface of the piston – rings of the two test rigs. The single-ring test rig is considered a large scale model for studying the lubrication aspects of the real size piston-cylinder engine assembly, which is the Lister – Petter PHW-1
engine. The comparison between the two experimental test rigs necessitates a “cavitation factor” coefficient to be determined. This coefficient is the ratio of the width or the length of the string cavities divided to the piston ring width.

\[
\text{Cavitation factor}_1 = \frac{\text{width of string}}{\text{width of piston ring}} \quad (5.1)
\]

Following the measurement scale of the image, the string distances for every case (test rig – Figure 5.79 and engine – Figure 5.80) are presented (Table 5.4), so that averaged string distances measured from the images can be taken for every case.

<table>
<thead>
<tr>
<th>Measured string distances (mm)</th>
<th>Single-Ring Test Rig</th>
<th>Lister-Petter Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.292</td>
<td>0.531</td>
<td></td>
</tr>
<tr>
<td>0.167</td>
<td>0.368</td>
<td></td>
</tr>
<tr>
<td>0.250</td>
<td>0.344</td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td>0.478</td>
<td></td>
</tr>
<tr>
<td>0.333</td>
<td>0.514</td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td>0.589</td>
<td></td>
</tr>
<tr>
<td>0.167</td>
<td>0.414</td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.250</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.250</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.333</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.275</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.333</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.367</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.267</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.208</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.208</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.292</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Average width</strong></td>
<td>0.268</td>
<td>0.463</td>
</tr>
<tr>
<td><strong>σ</strong></td>
<td>0.056</td>
<td>0.090</td>
</tr>
</tbody>
</table>

Table 5.4  Measured string distances for test rig and engine
For the Lister – Petter engine, the cavitation coefficient derived for the average string distance as calculated in Table 5.4 is:

\[ \text{Cavitation factor}_1 = \text{CF}_{\text{engine}} = 0.194 \]  

(5.2)

whereas for the single ring test rig is \( \text{CF}_{\text{test rig}} = 0.054 \) (5.3)

\( \text{CF}_{\text{engine}} \) is 72.16% greater than \( \text{CF}_{\text{test rig}} \).

Another comparison of the cavitation factors derived from imaging is going to be made, this time looking at the length of the string cavities compared to the width of the piston ring. For the single ring test rig, there are differences between upstroke and downstroke as far as the length of the strings is concerned. The downstroke case is examined:

\[ \text{Cavitation factor}_2 = \text{CF}_2 = \frac{\text{length of string}}{\text{width of piston ring}} \]  

(5.4)

In the following table (Table 5.5), the lengths of the strings for every case (engine and test rig) are measured and the cavitation factor for each case is calculated.
Table 5.5  Measured string lengths for test rig and engine

<table>
<thead>
<tr>
<th>Measured string length (mm)</th>
<th>Single-Ring Test Rig</th>
<th>Lister-Petter Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2.563</td>
<td>1.941</td>
</tr>
<tr>
<td></td>
<td>2.142</td>
<td>2.109</td>
</tr>
<tr>
<td></td>
<td>2.530</td>
<td>2.305</td>
</tr>
<tr>
<td></td>
<td>2.470</td>
<td>2.243</td>
</tr>
<tr>
<td></td>
<td>1.939</td>
<td>2.154</td>
</tr>
<tr>
<td></td>
<td>2.589</td>
<td>2.106</td>
</tr>
<tr>
<td></td>
<td>2.395</td>
<td>2.099</td>
</tr>
<tr>
<td></td>
<td>2.753</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.656</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.656</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.736</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.677</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.757</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.723</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.664</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.437</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.226</td>
<td></td>
</tr>
<tr>
<td>Average width</td>
<td>2.536</td>
<td>2.137</td>
</tr>
<tr>
<td>σ</td>
<td>0.233</td>
<td>0.116</td>
</tr>
</tbody>
</table>

The cavitation factors respectively are:

\[ CF_{\text{engine}} = 0.898 \]  \hspace{1cm} (5.5)

\[ CF_{\text{test rig}} = 0.507 \]  \hspace{1cm} (5.6)

The lengths of the string cavities are covering the majority of the piston ring surface at the engine. In the case of the single ring test rig, they are covering approximately 50% of the piston ring length and have a very uniform shape, due to the limited factors affecting the lubrication of the single ring test rig. \( CF_{\text{engine}} \) is 43.5% greater than \( CF_{\text{test rig}} \).

According to Yu et al (1999) the more flat the ring is, the better is the hydrodynamic lubrication. The flat ring has a tendency to weaken the hydrodynamic lubrication at the middle of the stroke and it generates a smaller cavitation zone than the barrel-faced one. A curved profile does enhance the hydrodynamic lift, but at the same time it extends the cavitation region, which does not contribute to the load carrying
capacity of the piston-ring. Therefore, a flat face ring has a stronger load capacity than the one with a curved profile.

Images taken from the second compression ring (Figures 5.36a and b) as well as images from the 3rd ring (Figures 5.37a and b and 5.38) show a fully developed oil film on the surface of the rings, which is in accordance to what is mentioned by Yu et al (1999).

Starvation and cavitation make the effective load region of the ring to reduce significantly. Normally the effective load region is only 15-40% of the nominal region of the ring (Yu et al, 1999) which following the results of equation 5.5 for the top compression ring, is in good accordance to the lowest percentage.

5.4.2 Residence Time

For the structures described above, it is worth calculating the residence time of the string cavities for both cases that experimental data were acquired.

For the testing conditions described in Table 5.3, the residence time of the string cavities formation can be easily derived for the test rig. The residence time is calculated as the time from the crank angle that they appear reaching the trailing edge of the piston ring specimen, up to the respective crank angle that they start diminishing on the same stroke.

For the single ring test rig, the string cavities reach the trailing edge of the piston ring specimen at 88° CA approximately and they start diminishing at 192° CA. The residence time of the cavities, calculated for the sinusoidal liner movement is:

\[ t_{\text{res test rig}} = 58\ \text{msec} \]

For the engine, the pulses that were used for the image acquisition triggered by the software gave away the crank angle of the piston position as it moves from BDC to TDC. So, these structures were noticed at 215.28°CA and at 318.24°CA. The residence time of the string cavities, calculated for the piston movement is:

\[ t_{\text{res engine}} = 43\ \text{msec} \]

The above consideration for the engine, though, is putting the residence time calculated under risk. The estimation of the time span in the engine stroke that the string
cavities appear through the visualisation images is not accurate, but it takes into account the approach limited by the overall viewing area of the window. As far as this is concerned, the cavity shapes appear at both the edges of the windows, from the lower part of the lower window up to the upper part of the upper window. It is therefore important to stress that the accuracy of the calculation should be further enhanced with the aid of a larger window area, which will enable the coverage of the whole stroke length and not part of it, as it applies to the existing liner modifications.

5.5 CF-3 Magnification Images – Oil Mist

In this set of images, at the anti-thrust side of the Lister engine, more evidence to the appearance of the oil mist became clear. These images show the top compression ring. In figure 5.81 there is no sign of oil droplets at the induction stroke. After one revolution, oil droplets emerge from the ring side clearances (Figures 5.82, 5.83 and 5.84).

![Image](image_url)

Figure 5.81 44.28° CA, induction stroke 3.67 cm from top edge
Figure 5.82  403.92° CA, expansion stroke 3.67 cm from top edge

Figure 5.83  403.92° CA, expansion stroke 3.67 cm from top edge

Piston movement

Oil mist generation area

Figure 5.83  403.92° CA, expansion stroke 3.67 cm from top edge

(a)
The oil mist that is captured in the images in front of the piston rings is generated by the oil droplets at the bottom ring side clearance. During the expansion stroke, the oil droplets which are located under the piston ring at the ring side clearance and at the piston ring groove are emerging, creating a “cloud” of droplets which is moving towards the combustion chamber. The droplets are squeezed out of the lower side clearance with the aid of the high pressure that prevails on the combustion chamber during the expansion stroke. The oil mist can initiate a bigger oil transport as it might entrain more oil during its upstroke movement and is already discussed in paragraph 5.4.3.4. The oil spouts identified by Inagaki et al (1995) at higher engine speeds and for lower magnification imaging should be derived by the droplets as shown in the images above. Higher speeds should turn the entrainment to an intensified process that is covering a larger area in the stroke.

5.6 CONCLUDING REMARKS

The visualisation images show the string cavities with different magnification lenses. Oil mist can also be noticed with the higher magnification lenses (CF-3) as well as oil droplets on top of the piston rings and at the ring side clearance (paragraph 5.5). Oil
droplets were also noticed when the piston was at BDC (Figures 5.43 – 5.48) and in Figures 5.36, 5.37 and 5.40 in front of the second compression ring, the scraper ring and the top piston land. The most probable explanation is that these droplets adhere to the inner surface of the window. The outcome of the ring - window interaction is the development of a rough surface. Talysurf results showed up to 8 μm groove depth after the visualisation testing. The inner surface of the window has long linear grooves/scratches that are formed across the circumference of the liner. With the piston reciprocating, oil can penetrate the groove and it produces droplets depending on the local depth of the groove. So, Figures 5.36, 5.37, 5.40, 5.43 - 5.48 present how the lubricant inside these cracks is shaped as droplets. Images taken from a cracked region of a visualisation window (long and deep groove) showed that larger oil droplets were formed inside these crevices.

Strings were observed only on the surface of the top compression ring. There are images, however, that show oil film developed on the surface of the top compression ring as well as the second compression and scraper ring (Figures 5.33, 5.34, 5.35, 5.36, 5.37, 5.38, 5.42, 5.51, 5.52, 5.53, 5.54, 5.55, 5.56, 5.57 and 5.58) but no string cavities visualised as such in Figures 5.28, 5.29, 5.30, 5.31, 5.32 and 5.55. Possible reasons for the appearance of the film on one hand and the strings on another (for the same piston-ring but in different stroke and distance from TDC), is ring fluttering which causes oil pumping into the ring groove (Figure 5.3) and thus affecting the oil availability at the converging wedge. So, as the piston ring changes position oil is squeezed or enters the side clearances.

Both Gamble (2002) and Thirouard (2001) mention the contribution of oil mist into oil transport through the ring pack. The experimental visualisation data showed that oil could be transported out of the ring gap as oil mist. Other identified mechanisms, according to Thirouard (2001) that drive oil film from the lands to the ring groove clearance were identified as follows:

1. ring squeezing
2. lateral motion of the piston relative to the rings

Inertia force generates a pressure gradient which, in turn, transports oil only in the OCR. The rates of oil transport resulting from the above mechanisms depend on:
a. ring and piston dynamics and consequently vary with engine speed and load
b. amount of oil available in the vicinity of the ring groove clearances

The oil droplets and their source of appearance was captured, though, and CF-3 magnification provided good imaging quality of how these droplets emerge through the ring side clearances.
CHAPTER 6
CONCLUSIONS AND FURTHER WORK

6.1 CONCLUSIONS

The oils required by on-going developments in engine technology, increasingly stringent emission regulations, the use of alternative fuels and the trend towards the introduction of "fuel efficient" oils of lower viscosity, will require new lubricants to be formulated. The new oils will be expected to operate at higher temperatures and over extended drain intervals. A thorough understanding of the complex phenomena associated with the lubrication of the piston assembly will be essential for the engine designers and lubrication engineers to address these issues. The measurement of in-cylinder lubricant film thickness in a firing engine can provide valuable information about lubrication conditions of the piston assembly, performance of a lubricant, its distribution on the piston-liner interface and its transport through the ring-pack.

The LIF method has proved to be a good technique for oil film thickness point measurements. Engine applications of the LIF, however, have uncertainties due to the signal calibration problem. There have been different approaches to the subject so far; static and dynamic calibrations are the two methods used, each of them having certain disadvantages. It is, however, important to measure the oil film as accurate as possible because its contribution to emissions has proved to be important, despite the fact that the cause-and-effect relationship remains somewhat unclear.

In this work, the simulation test-rig was used to study the piston-ring lubrication under idealized conditions, to improve the acquired signals and to implement new techniques so that the fundamental aspects of piston-ring lubrication and the effect of different oils could be further understood and interpreted.

Useful results were obtained from the single-ring test rig. These are summarized as follows:
Extensive testing showed good repeatability. The modification of the capacitance probe position outside of the oil bath proved to be successful in terms of signal quality and film thickness variation was found to be very close to the hydrodynamic lubrication theory curve.

The new position of the capacitance transducer eliminates previously encountered calibration problems and is oil temperature independent which is very important for the correct interpretation of high temperature results.

The LIF results validated the capacitance results and further showed cavitation to take place at mid-stroke.

Friction results were found to be repeatable although not very symmetric upstroke and downstroke due to the manufacturing of the test rig that causes the appearance of dynamic phenomena during its reciprocation. Another factor having significant effect on the asymmetric signals is the asymmetric piston ring specimen itself.

The friction results close to the dead centers where the peaks were recorded, are validated by the capacitance results and further supported by the imaging. Increasing load is providing good stability for the results.

Vibration elimination to avoid extra error sources on the signals was considered necessary and steps towards this direction were made. Bolting of the test rig bed directly to the floor, manufacturing of steel clamps for the electric motor and welding of steel brackets to enhance the test rig’s rigidity were successful modifications as the capacitance signal comparison to the previous standard showed. Further on, more results from lubricant parametric study were derived:

Viscosity of the oils tested remains an important factor. Although the oils tested in the first phase were not in the extremes in terms of viscosity, the more viscous oil developed as expected a thicker oil film than the other oils.

Oils of extreme viscosity value (very low - very high) were tested and a trend dependant on the lubricant viscosity index was noticed. The
MOFT of oils of the same grade at ambient temperature is viscosity index dependant.

- For specific oil testing at higher temperatures decreases the MOFT.
- Load tests at high temperatures showed that the temperature effect at high loads is not evident - the MOFT variation from ambient temperature to 70°C is marginal.
- The oil with the higher kinematic viscosity retains the maximum MOFT with temperature variation than the other oils tested.
- Friction parametric study showed that the friction peaks move closer to the dead centers as the load increases. This was a useful indication which later on was verified by the visualisation experiments.
- Higher HTHS value gives lower friction peaks close to the dead centers.
- Higher friction results are combined with high temperature testing (lower viscosity, higher friction) where the formulation of the lubricant is also an important factor as during boundary lubrication conditions the increased temperature has a pronounced effect on friction force measurements.

A grooved piston-ring specimen was developed in order to enable the study of lubricants’ fluorescence quantum yield at ambient and elevated temperature and to quantify the LIF signal acquired from the measuring probes. This study will enable the use of the simulation test rig as the standard which will provide the dynamic LIF calibration coefficients for engine use. The results showed that:

- The dynamic calibration LIF results had a very big difference in the thickness value than the capacitance results.
- Background noise was considered as the source of this great variation and it was further established by the non-linearity of the photomultiplier voltage over the measured distance from the piston ring, 5 to 40 μm.
If the OFT measurement is below 5 μm, which is the case for the MOFT according to the capacitance results, it is not accurate because the ratio between the voltage and the thickness is too small.

For films thicker than 10 μm, the static calibration curve can be used for the results.

The dynamic calibration coefficient (μm/V) is increasing respectively with temperature as the higher temperature tests have shown.

Furthermore, the power output of the different fibre probes differs by a very big percentage when they are compared to each other. This should be taken into account when any calibration attempt is being made using different fibre probes.

The initiation and development of cavitation throughout the stroke was identified with the introduction of the glass liner specimen and images acquisition with a high speed CCD camera and magnifying lenses. Testing revealed that:

- The cavities start their appearance very early in the stroke (a few degrees after TDC) and they continue to exist until very late in the same stroke.
- They take the shape of ferns, grown ferns, fissures, strings and bubbles.
- At all times the cavity shapes appear at the diverging wedge 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.
- Transition between boundary cavitating conditions was identified in both capacitance and pressure - visualisation results with the oil film pressure transiting from \( P_{\text{vapour}} \) to \( P_{\text{atm}} \).

These results were evident during the speed and load parametric study of the visualization experiment.
To further supplement the cavitation pictures, a miniature pressure transducer was installed in a new aluminium liner specimen. A piston-ring specimen parametric study showed that:

- The smallest curvature gave the highest oil film pressure results.
- The non-symmetric wear of the piston-ring specimen is responsible for the peak pressure and MOFT variation upstroke and downstroke.
- Pressure measurements throughout the stroke identified subambient pressures which appear at the beginning of the stroke.
- The appearance of ferns on the surface of the piston-ring is accompanied by negative or sub-atmospheric pressures and shows that pressure either reaches the tensile strength of the liquid or the liquid's vaporous pressure.

Parametric speed, load and temperature oil film pressure study on the single ring test rig resulted in the following:

- Higher oil film pressures are reached when higher load is applied on the piston ring specimen.
- For the speed tests the trend observed was not as clear as the load tests were but again higher speeds are combined with higher pressure.
- The shift of oil film pressure peaks for the same piston ring specimen when tested at different speed and load is due to the variation of the ring "effective" width.

On the engine the modification of the block enabled imaging experiments.

- The visualization of strings provided further cavitation evidence and verified the findings of the single-ring test rig. There are however differences to the lubricant flow on the test rig.
- There is evidence that the top compression ring operates under starved lubricating conditions.
- The motoring tests gave a close insight of the droplets and mist that is generated and transports the lubricant.
High magnification imaging allowed for identification of oil droplets that emerge from the ring side clearances during the expansion stroke. These droplets contribute to oil transport as it is already a recognized mechanism.

Concrete evidence was provided only for the string type of cavities and clearly further work is needed to capture the whole cavitation process throughout the stroke.

Other types of cavities already identified in the single-ring test rig are assumed to be the initial stages of cavitation on the top compression ring.

The approach to analyze the engine imaging results was based on the calculation of the cavitation structures residence time and the introduction of cavitation factors, so that a direct comparison to the large scale model of the single – ring test rig can be made. The comparison showed that:

- The cavitation phenomena are more pronounced and to a larger extend than those on the test rig.
- 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 distinct and captured by the camera lens.
- The unavailability of the oil film pressure measurements for the engine does not support as strongly the findings from imaging. Implementing pressure transducers to measure the oil film pressure in the cylinder liner is the next step towards getting a complete picture of the cavitation inception and development throughout the engine stroke.
6.2 Further Work

The successful implementation of the miniature pressure transducer combined with the glass liner visualization results, offers ground for engine applications. Measuring stations of oil film pressure along with the LIF probes are considered necessary so that information about the lubrication in the engine can be acquired. The visualisation experiments themselves will provide further information about the oil film transport. The quartz windows application should be covering a longer part of the circumference on the surface of the liner as well as the piston stroke, so that more valuable information along and across the piston travel can be provided.

Another aspect of the LIF technique which merits consideration is the problem of fuel-oil interaction and the resulting dilution of the oil film. This clearly has an effect on the calibration of the LIF signal sampled from the engine. The LIF technique can be used simultaneously with real-time measurements of oil consumption to investigate the effect of oil transport through the ring pack on the rate of particulate emissions.

Engine application of the pressure measurements and visualisation should be combined with modeling work so that an analytical relationship expressing the oil displacements as a function of the piston speed and the oil volume can be derived. As a result, the evolution of the oil distribution along the axial direction of the piston lands could be computed through the engine cycle and therefore correlated with the oil supply to the ring/groove clearances. Respective correlations should be derived for the circumferential oil flow.

Eventually, use of pressure measurement transducers on the already machined slots on the surface of the liner is the next step towards getting a clearer image of the guiding forces behind what has been visualised already.

Such information will be of great use for the future development of oil transport simulation tools. To bring understanding of the oil transport process between the piston lands, through the grooves and gaps and their interactions, identification and characterization of the mechanisms responsible for the oil flows along the piston have to be made. The driving forces behind axial oil displacements and the circumferential flow
should be identified and verified with previous findings. Factors such as inertia-driven oil displacements, ring and piston dynamics, ring/groove relative angle, ring squeeze action, region by region analysis of the piston-ring pack, dragging effect of gases flowing through the ring groove clearances and around the piston circumference, ring and land geometry, ring fluttering, ring instability, flutter induced gas flow, ring pumping, clearance of the ring grooves, land oil distribution, lateral motion of the piston relative to the rings and from the engine side piston speed, engine load and intake manifold pressure should be checked individually and in combination so that the mechanism of the oil transport could be evaluated. Eventually, a major step towards the development of analytical tools for oil consumption reduction and potentially providing practical guidance for the piston and piston-ring designs would be achieved.

Further work involving the engine includes, along with the modifications already proposed, emissions instrumentation to establish the link between particulate emissions and oil film thickness. Then, the total picture between oil properties and formulation, oil transport phenomena and engine operating conditions will be much clearer. The DELIF (Dual Emission Laser Induced Fluorescence) is being considered as a very useful technique. DELIF experiments will preserve the fast time response of intensity based technique and at mean time the intensity dependence on the Laser induced energy is eliminated. The availability of high power ultraviolet LED to excite in the near UV opens up an easy route via the DWF (Dual Wavelength Fluorescence) approach to the separation of temperature and oil thickness data and their simultaneous measurement, avoiding interference from the natural fluorescence of the oil itself. Simultaneous OFT and temperature measurements can be carried out on the engine under firing conditions with the DWF. Novel DWF sensor scheme includes: the use of two distinct fluorescence dyes to study the characteristics of the engine in terms of both lubricant film thickness and engine temperature. Data will yield information on the oil film thickness from the fluorescence intensity and the oil temperature from the fluorescence decay time or intensity ratio.

An important use of a fast responding scheme such as fluorescence lifetime monitoring is in the creation of the temporal temperature profile history in a complete engine cycle.
The aims will be:

- to investigate how critical conditions for oil consumption (high speed / high load, high speed / low load) affect the oil transport in the piston – ring pack
- establish transit lubricant distribution during cold start for both gasoline and diesel engine
- experimental validation for engine friction modeling
- study fuel absorption on lubricant film with comparison between gasoline port injection and GDI engine
- calibration: Provide a robust and reliable calibration technique for oil testing.

The necessary steps towards this direction include the modification of the engine to allow for changing oil when the engine is running. Thus, fluorescent dye tracers can be added in the lubricant and lubricant transport will be monitored by fluorescent signals detected by the fibre optics.

Figure 6.1 Larger viewing area proposed for the new visualization engine block
Additionally, the larger viewing areas of a purpose built engine block for visualization only (Figure 6.1) will make possible the calculation of the residence time of the cavitation structures for the engine. The new viewing areas will enable capturing the piston during its reversal points at both BDC and TDC. Then, firing tests should be considered to get the full picture of cavitation.

Overall, although very interesting results and insight have been obtained on a number of aspects of piston-ring lubrication centred around cavitation, there is an increasing interest on the subject as means of reducing friction and improving the efficiency of both gasoline and diesel engines. It is widely considered that reduction of friction through design considerations and oil formulation represents one of the most tangible goals in the evolutionary development of reciprocating engines and it is hoped that the work presented in this thesis represents a small step in this direction.
REFERENCES


REFERENCES


REFERENCES


Williams, P.R. and Williams, R. L., “Measurement of the Cavitation Threshold of Oils under Dynamic Stressing by Pulses of Tension”,


## APPENDIX A

**LITERATURE REVIEW – SUMMARY**

### Table 1.1 OIL FILM THICKNESS MEASUREMENTS

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental Set-up</th>
<th>Technique</th>
<th>Major Findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Greene (1969)</td>
<td>Single-cylinder spark ignition engine with a transparent cylinder liner and a dry sump lubrication system. Motored tests performed.</td>
<td>• Ultraviolet light photography of engine oil mixed with a fluorescent dye. • Scattered light photography.</td>
<td>• The ultraviolet technique detected all the oil film between the piston and the liner but it couldn’t be determined which surface the lubricant is adhered to. • Scattered light technique could only detect the lubricant adhered to the surface of the cylinder liner. • The piston skirt was found to be flooded with oil. • Piston rings were separated from the liner by a continuous oil film. • The oil carried on the piston land was found to be mostly in contact with the piston.</td>
</tr>
<tr>
<td>Wing and Saunders (1972)</td>
<td>Piston assembly of a firing engine</td>
<td>• Inductance</td>
<td>First oil film thickness measurement</td>
</tr>
<tr>
<td>Hamilton and Moore (1974a)</td>
<td>Firing Diesel engine</td>
<td>• Three capacitance probes placed in the thrust side of the liner. Calibration using slip blocks</td>
<td>0.4 to 2.5 μm oil films were measured. Error could occur as a result of the variation of the dielectric constant.</td>
</tr>
<tr>
<td>Hamilton and Moore (1974b)</td>
<td>Single cylinder four stroke diesel engine and three capacitance gauges mounted in the</td>
<td>• Capacitance</td>
<td>Oil film thickness varied between 0.4 and 2.5 μm. • Variation of the oil film thickness throughout the strokes was 1 to 2 μm. • The diameter of the sensor should be increased to raise its</td>
</tr>
</tbody>
</table>
| **thrust side of the cylinder liner. The piston ring pack consisted of five rings.** | **electrical capacity and maintain its accuracy for oil films thicker than 10 μm.**  
- Ring tilt did not change the minimum oil film thickness.  
- Increase in the oil film thickness under the top ring was attributed to combustion gases passing under the ring and lowering the dielectric constant.  
- Cavitation could lower the dielectric constant and lead to overestimated oil film thicknesses. |
| --- | --- |
| **Parker et al (1975)** | **Firing engine**  
- Capacitance probes on top and third ring giving minimum oil film thickness throughout stroke.  
- Inductance probes in liner.  
- Similar oil film thicknesses were found under all three compression rings. Large variations were observed in the film thicknesses over time, thought to be caused by ring rotation. |
| **Ford & Foord (1978)** | **A 100W mercury lamp emitted light through UV filters and illuminated the lubricant film on a spinning shaft. The fluorescent light was collected by a blue and green sensitive photocathode. The predominant wavelength of the mercury lamp was 365 nm.**  
- Mercury lamp induced fluorescence  
- 325 nm laser-induced fluorescence  
- 441.6 nm laser induced fluorescence.  
- oil films between 0.1 and 1000 microns could be measured.  
- The fluctuation of the laser power could affect the experimental results.  
- The fluorescent signal decayed with time when the UV laser was used (photobleaching effect)  
Attenuation of the laser power did not eliminate the problem  
- Visible laser light did not cause the above mentioned effects in the lubricants tested  
- Fluorescent dyes were tested at five different concentrations between 0.05 and 0.5% by volume; it was found that lower concentrations should
The mercury lamp was then replaced first by a 2 mW He-Cd laser emitting UV light of 325 nm and then by a visible blue laser light of 441.6 nm and 18 mW.

<table>
<thead>
<tr>
<th>Study</th>
<th>Description</th>
<th>Methods</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wakuri et al (1979)</td>
<td>Single ring test ring, liner made out of glass</td>
<td>Interferometry</td>
<td>Good agreement between theory and experiment for both oil film thickness and location of cavitation boundary.</td>
</tr>
<tr>
<td>Ting (1980)</td>
<td>Single cylinder research engine with transparent cylinder liner</td>
<td>Laser induced fluorescence with a red fluorescent dye</td>
<td>The results were not calibrated but it was suggested that grooves of known depth cut into the piston rings could be used to calibrate the LIF signal. Piston ring clearance could affect oil consumption.</td>
</tr>
<tr>
<td>Shin et al (1983)</td>
<td>Piston assembly test rig</td>
<td>Capacitance sensors mounted on rings allowing minimum oil film thickness to be measured throughout the cycle</td>
<td>Oil film thickness was found to agree with theory.</td>
</tr>
<tr>
<td>Hoult and Rifai (1985)</td>
<td>Motoring tests. Engine used had a transparent cylinder</td>
<td>Laser induced fluorescence with dyes. Ring profile fitting was used for calibration. Probes were mounted at mid-stroke at two points in liner</td>
<td>Boundary conditions were different from the theory</td>
</tr>
<tr>
<td>Reference</td>
<td>Test Rig/Engine Description</td>
<td>Technical Details</td>
<td>Results/Findings</td>
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<tr>
<td>Sanda and Someya (1987)</td>
<td>Reciprocating test rig</td>
<td>• Two Eddy current. Minimum oil film thickness measured throughout cycle</td>
<td>• Relative displacement between reciprocating liner and stationary ring specimen was measured and the film thickness was calculated</td>
</tr>
</tbody>
</table>
| Hoult et al (1988)                | Test version of Cummins VT-903 engine fitted with a quartz glass window. He-Cd 14 mW laser at 441.6 nm | • Laser induced fluorescence  
• The results of a bench-top calibration were compared with ring face profiles discerned in the fluorescence signal acquired from the engine | • Good agreement between calibration methods-motored operation avoids the complications of temperature and lubricant degradation effects.  
• Film thicknesses of approximately 2 microns were measured at top dead centre.  
• The ring pack operated under starved lubrication conditions.  
• The quartz window fitted in the cylinder liner was found to be invasive. |
| Lux et al (1990)                  | Fired single cylinder diesel engine, with quartz window                                       | • Laser induced fluorescence. Tool marks on piston skirts used for calibration                                                                                                                                   | • Large differences were found between statically and dynamically obtained calibration coefficients  
• Oil mass balance performed but large differences between oil flows and actual oil consumption were observed. |
| Hoult et al (1991)                | Motored diesel engine (no temperature gradients across oil film)                               | • Laser induced fluorescence. Calibration was performed with ring profile fitting.                                                                                                                              | • Cavitation was not observed. Multigrade oils were observed to wet the rings less.                 |
| Richardson and Borman (1991)      | Cameron Plint wear tester and firing diesel engine for in cylinder measurements.              | • Laser induced fluorescence. Fibre optics probes mounted in cylinder liner at TDC, mid-                                                                                                                         | • Light of combustion might affect the signal  
• Thermally stable fluorescent dyes could be used  
• The TDC fibre broke, possibly due to thermal stresses in the liner when the engine was |
<table>
<thead>
<tr>
<th>Study</th>
<th>Methodology</th>
<th>Findings</th>
</tr>
</thead>
</table>
| Richardson and Borman (1992)  | Same as above                                                               | • Wear tester: fibre in liner to observe boundary conditions and ring to measure the oil film thickness  
• Diesel engine: four fibre optic probes.  
• Starvation was observed in the wear tester. Reynolds boundary condition was found to be best for lubrication prediction.  
• In the engine, the observations were as above for the wear tester. |
| Brown et al (1993)            | Single cylinder CAT 1Y73 engine with fused silica window. Two multigrade oils were used. | • Laser induced fluorescence. To calibrate the results a variable temperature calibration micrometer was used.  
• Fluorescence quenching increased at elevated temperatures.  
• Thermal expansion of the calibrated cell was observed.  
• Oil film pressure effects on fluorescent quantum yield were found negligible.  
• The oil film under the compression ring was not affected by engine speed.  
• At medium to high load, the lower viscosity oil provided, as expected, thinner oil films but at low engine speeds and low load the opposite was observed. |
| Grice and Sherrington         | Single cylinder Petter AV1L                                                  | • Capacitance  
• Low repeatability in motored tests. |
<table>
<thead>
<tr>
<th>Year</th>
<th>Configuration</th>
<th>Tests/Techniques</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1993</td>
<td>Firing diesel engine. Motored tests carried out as well.</td>
<td>Inductance tests, especially for the top compression ring. Cavitation was reported at the ring outlet. The inductance probe displayed problems with its signal to noise ratio and spatial resolution. The probes were insensitive to cavitation due to the large cycle to cycle variation. Ring profiles showed evidence of ring tilt. Superimposed ring profiles showed discrepancies in the outer ring profiles which were attributed to either cavitation or ring lateral movement.</td>
<td></td>
</tr>
<tr>
<td>1993</td>
<td>Single cylinder spark ignition engine with electronic fuel injection system.</td>
<td>Scanning laser induced fluorescence. Bench top calibration using a special gauge with nickel-plated steps.</td>
<td>The bleach effect was found to be negligible. During motoring tests the thinnest oil film was observed under the second compression ring. During firing tests, the lubricant film was thinnest under the top compression ring. The oil volume at the top and second land did not appear to be affected by the change in the profile of the ring face. The thickest oil films were observed under the oil control ring. The diameter of the laser beam was found too big to provide good spatial resolution of the scanning technique.</td>
</tr>
<tr>
<td>1993</td>
<td>Motored single cylinder engine</td>
<td>Laser induced fluorescence. Calibration by static test ring and then by ring profile</td>
<td>The fibres proved to have different calibration coefficients. Expected trends were found with speed and liner temperature.</td>
</tr>
</tbody>
</table>
Dynamic calibration is necessary and the reflectivity was important to take into account. Oil film thickness was found to be roughly proportional to the square root of the kinematic viscosity.

<table>
<thead>
<tr>
<th>Study</th>
<th>Engine Description</th>
<th>Methods</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Konomi et al (1993)</td>
<td>Toyota TRE3 research engine with a floating cylinder liner for friction measurements.</td>
<td>Scanning laser induced fluorescence, Floating liner for friction measurements</td>
<td>The measured LIF signal was found to have very good linearity in the oil film range between 2 and 100 μm. Both firing and motoring tests showed strong dependence of the friction force on the piston skirt profile. Curvature was found to be important to both oil film thickness and friction.</td>
</tr>
<tr>
<td>Dearlove and Cheng (1995)</td>
<td>Motored single cylinder. Piston guided by linear bearings. The piston had only one ring and a single window was used.</td>
<td>Laser induced fluorescence. Grooves were etched on ring for calibration purposes. Ring profile fitting.</td>
<td>Difficulties were encountered with conformity due to circumferential variation. Calibration was a problem. Friction appeared to be mixed.</td>
</tr>
<tr>
<td>Mattson (1995)</td>
<td>Heavy duty diesel engine.</td>
<td>Capacitance transducers mounted in the cylinder liner</td>
<td>The top ring oil film thickness increased at engine idle speed. Cavitation occurred under the top ring at higher engine speeds and loads. Piston tilt was detected at the beginning of the power stroke. Low repeatability was associated with the small number of engine cycles averaged. The TDC sensor showed that there was very little change in oil film thickness under the top compression rings. The transducer mounted at the lowest part of the cylinder liner showed thicker oil films.</td>
</tr>
</tbody>
</table>
Stiier and Ghandi (1997)  
**Single cylinder motored engine**
- Laser induced fluorescence. Capacitance was used for calibration.
- There was difficulty matching ring profiles measured by LIF, those measured by capacitance and those measured by Talysurf. Some probes produced better results than others.

**Single cylinder firing engine**
- Laser induced fluorescence
- The front and rear probes showed generally similar film depths for all the rings at all strokes.
- Thrust side showed differences between all results.
- The three piece oil rings showed less variation in oil film thickness resulting in lower oil consumption.
## Table 1.2 LUBRICATION AND EMISSIONS

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental Set-up</th>
<th>Major Findings Results and Remarks</th>
</tr>
</thead>
</table>
| Manni et al (1995) | VM Turbotronic 425 CHIE Diesel engine. AVL measuring devices for fuel and lubricants. Short term procedure to evaluate fuel consumption and emissions (run engine in urban and extra urban cycle). A long term procedure to evaluate the oil consumption (high power test). | - Multigrade 10W-40 was the only tested oil showing a significant variation of total particulate emission.  
- Fluid oils seemed to give a lower contribution of the oils to the soluble organic fraction.                                                                 |
- At low loads and idle the diesel engine produces many more particulates than either gasoline or CNG. The difference becomes smaller for medium loads.                                                      |
| Laurence et al (1996) | Single cylinder Hydra research engine. Constant volume sample dilution tunnel. Wet test meter. | - Total particulate emission rate increases with decreasing viscosity, because decreased viscosity leaves a thicker film on the liner during and after the combustion stroke, allowing more oil to participate in the combustion process.  
- Total particulate emission rate increases with an increased ring-gap width.  
- At hotter conditions, the extra oil is partially oxidized and emitted carbonaceous material.  
- Changes in lubricant parameters affect the fuel derived emissions as well, through the process of absorption and desorption.  
- Engine load has an effect on particulate emission rate and composition.                                                                 |
<table>
<thead>
<tr>
<th>Study</th>
<th>Equipment/Methodologies</th>
<th>Findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manni et al (1997)</td>
<td>Ricardo Hydra research diesel engine. Dilution tunnel for particulate sampling. Heating flame ionization detector for HC emissions. Non dispersive infra red analyzer for CO. Chemilluminescence analyser for NOx detection.</td>
<td>Tests at high constant load condition with oils of different rheology show that low viscosity oils give less fuel consumption and higher air fuel ratio (AFR) than the high viscosity ones. Decrease of CO, particulate emissions and smoke is achieved when low viscosity oils are used. Some synthetic basestocks give a low contribution to particulate emissions. The effect of traditional additives seems to be less important than the effect of basestocks. The use of full synthetic lubricant formulated with selected basestocks gives better emission performances at a high load condition than a mineral oil as regards CO, particulate and smoke.</td>
</tr>
<tr>
<td>Bennett et al (1997)</td>
<td>Two Volvo 850 vehicles (2.0 and 2.5 litre engines). Horiba emission analysers. Three different basestocks lubricants (mineral oil, poly-alpha olefin hydrocracked)</td>
<td>Emissions of total hydrocarbons, non-methane hydrocarbons, non-methane organic gases and speciated hydrocarbons were independent of lubricant composition. CO emissions were independent of lubricant composition. Significant reductions in NOx emissions were observed for both the poly-alpha olefin and hydrocracked based lubricants. Lubricant composition appeared to influence the catalyst conversion efficiency for NOx emissions.</td>
</tr>
<tr>
<td>Abdul-Khalek et al (1998)</td>
<td>Direct injection diesel engine. Mini dilution system for particulate measurement. Scanning mobility particle sizer (SMPS). Condensation particle counter (CPC). Electrical aerosol analyser.</td>
<td>The lowest number concentrations were observed at 75% load at both rated speed and maximum torque speed. The engine and dilution system show long stabilization times for measuring number concentrations. Measured size distributions fit well to bimodal, lognormal distributions.</td>
</tr>
<tr>
<td>Study</td>
<td>Engine Type</td>
<td>Methodology</td>
</tr>
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</tr>
<tr>
<td>Parks II et al (1998)</td>
<td>Kohler 6CMHS single cylinder engine with glass window. LIF spectroscopy.</td>
<td>- Fuel concentration in the cylinder wall oil film reaches maximum levels during cold start and non-stoichiometric operation and significant fuel concentrations exist during stoichiometric operation of a warm engine. - Fuel is present in the oil film throughout the engine cycle but film thickness varies during the engine cycle.</td>
</tr>
<tr>
<td>Schneider et al (1998)</td>
<td>Petrol engine. Radioactive tracer method (isotope dilution method) to measure the contribution of engine oils to deposits on exhaust system components and particulates in the exhaust gas of a gasoline engine.</td>
<td>- The average contribution of oil derived carbon to deposits in the exhaust manifold is 2±1% and is independent of engine speed. - For deposits in the exhaust parts the oil derived contribution is 6±2% and is also independent of speed. - Samples of exhaust filtrate have a 2% to 30% carbon contribution from engine oil, which increases in proportion to engine speed. - The contribution of oil derived carbon to deposits on oxygen sensors is 3±2% and does not have a major dependence on engine speed.</td>
</tr>
<tr>
<td>Graskow et al (1999)</td>
<td>Mitsubishi GDI engine. Dual stage variable residence time mini-dilution system. Particulate measurement instrumentation. Condensation nucleus counter (CNC). Scanning mobility particle sizer (SMPS).</td>
<td>- For operating conditions ranging from 32-90 km/hr the number weighted size distributions were well reported by broad monomodal lognormal distribution functions. - Most of the particles emitted under these operating conditions were in the accumulation mode. - For the 13 km/hr operating condition, the size distribution was bimodal.</td>
</tr>
<tr>
<td>Reference</td>
<td>Methodology</td>
<td>Findings</td>
</tr>
<tr>
<td>-----------</td>
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</tr>
</tbody>
</table>
| Andrews and Ahamed (1999) | Ford SI Zetec multi port fuel injection, 3 way catalyst and EGR. Smokemeter. | - Contribution of SI engine to total particulate emissions is significant.  
- Analysis of the particulates showed that the bulk of the mass was ash and the second largest fraction was unburnt lubricating oil.  
- The catalyst has to be at a minimum of 400°C to achieve effective particulate oil oxidation. |
| Khalek et al (1999) | Medium duty, turbocarged aftercooled, direct injection diesel. Microdilution system that allows systematic variation of dilution and sampling conditions. Scanning mobility particle sizer (SMPS). | - Most of the size distributions showed a bimodal structure, a nuclei mode with a geometric number median diameter.  
- The influence of dilution variables such as residence time, dilution temperature and dilution ratio on the total number emission of nanoparticles ($D_p<50$ nm) is dramatic.  
- The formation and measurement of increased diesel particle number concentrations are favoured by the following test conditions: lower dilution temperature, lower dilution ratio, longer residence time, higher relative humidity and higher fuel sulphur content. |
| Graskow et al (1999) | 2.3L 1993 GM QUAD-4 and 4.6L 1994 Ford V8. Single stage ejector diluter system. Condensation nucleus counter. Scanning mobility particle sizers. Number concentrations and size distributions were measured upstream the catalytic converter. | - The baseline size distributions for the two additives were quite different from each engine and from the non-additized fuel.  
- The size distributions measured from the Ford engine were lognormal and showed a significant increase in the number concentration of very small particles. |
### Table 1.3  OIL TRANSPORT IN PISTON RING ASSEMBLIES

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental Set-up</th>
<th>Technique</th>
<th>Major Findings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wong and Hoult (1991)</td>
<td>Two identical small production IDI diesel engines Kubota EA 300N.</td>
<td>LIF diagnostics for instantaneous film thickness data (He-Cd laser 441.6 nm) Single and multigrade lubricants in motoring and firing conditions Five different piston ring configurations Tritium as radioactive tracer for oil consumption</td>
<td>No noticeable variation in film profiles Radial tension has little effect on the piston lands. Three piece oil control rings allow less oil to reach the upper lands Two distinct oil recalculation loops appear to exist between the piston and the sump Crown land appears to run dry and does not contribute to oil consumption Oil consumption rates shows no correlation with oil transport rates</td>
</tr>
<tr>
<td>Nakashima, Ishihara, Urano (1995)</td>
<td>Four cylinder engine. 5 mm thick quartz sleeve inserted. Two observation windows were drilled in the cylinder prior to insertion of the quartz liner.</td>
<td>Video camera. Investigation of oil flow by visualisation and by measuring oil.</td>
<td>Main stream of oil flow to combustion chamber depends on ring gap positions Oil flow is affected by vacuum condition Oil flow was reduced by reducing end clearance or by using the top ring with a triangle step joint.</td>
</tr>
<tr>
<td>Mihara and Four</td>
<td>Four</td>
<td></td>
<td>Influence of top ring sliding</td>
</tr>
</tbody>
</table>
Inoue (1995) engines with different bore/stroke ratio. Top rings are full keystone. surface on oil consumption can be explained by the ratio of barrel face drop to effective sliding width. • Exhaust brake operation increases oil consumption due to the oil scraping effect. • Proper adjustment of top ring keystone angle as well as an asymmetric barrel face can reduce oil consumption.

<p>| Inagaki, Saito, Murakami and Konomi (1995) | Single cylinder spark ignition engine. Two compression rings and three piece oil control ring. Whole transparent cylinder for motoring tests and cast iron cylinder with slit windows for firing tests. | Two dimensional measuring system • Filtered Xe-Flash lamp (blue filter) • Filtered CCD camera • PC based signal analyser • Fluorescent dye in lubricating oil | The system showed clearly the distribution of lubricant film thickness on the piston skirt, piston rings and piston lands under motoring and firing conditions • The increasing rate of the mean oil film thickness agrees with that of the oil consumption rate. • Oil spouts were observed going through the top ring gap. Spouted oil film thickness is closely related with the oil consumption rate. • System enables the valuation and visualisation of the flow up oil which is the cause of the oil consumption. |
| Tian, Rabute, Wong and Heywood (1997) | • Ring dynamics and gas flow model for 4 cyl passenger car diesel engine. Barrel-faced top-ring with an upper inside chamfer | | The profile of the worn upper wedge of the top ring was found critical for oil consumption. • With the understanding of the top ring/groove wear mechanism and the worn top ring up-scraping mechanism, the ring profile, groove profile and ring/groove coating can be optimised to minimise oil loss from top ring upscraping. |
| Takiguchi, Nakayama, Furuhama, Yoshida | Single | • LIF, He-Cd laser 442 nm | Large amount of oil is supplied from the piston skirt to the oil ring at certain strokes on the thrust and anti-thrust sides |</p>
<table>
<thead>
<tr>
<th>Year</th>
<th>Study</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>1998</td>
<td>Thirouard, Tian, Hart</td>
<td>Single cylinder diesel engine equipped with quartz glass windows in the liner. The oils were doped with fluorescent dye. 2-D LIF system using Nd-YAG laser at 528 nm and a CCD camera. Sampling was done with a bandpass filter at 580 nm. Qualitative analysis of oil distribution in the ring pack was carried out. Accumulation of the lubricant above the top compression ring occurred only for engine speeds above 1600 rpm and was a function of piston tilt and ring twist. With the top ring gap fixed by a locating pin away from the observation window, oil accumulation on the crown land was not observed at all throughout the range of engine speeds. Locating the ring-gap pin close to the window resulted in increased amount of lubricant detected in the crown land area. Scraping the residual oil film up the cylinder liner by the upper side of the top compression ring was found to be the main oil transport mechanism into the piston crown land. Oil flow through the top ring groove was also observed.</td>
</tr>
<tr>
<td>1999</td>
<td>Audette and Wong</td>
<td>Heavy duty Cummins diesel engine modelled. Model that uses oil film thickness and allows for the change of oil composition and change in OFT due to vaporisation. Total vaporisation accounts for 10-17% of the total oil consumption. Vaporisation is strongly dependent on liner temperature and liner oil composition. Oil vaporisation has little dependence on engine speed.</td>
</tr>
<tr>
<td>1999</td>
<td>Ishihara, Nakashima, Urano</td>
<td>Water cooled 4 cycle, 4 cyl. Video camera to visualise motoring tests. Weighting of Oil injected from the large end of the connecting rod flowed from the lower piston ring gap through</td>
</tr>
<tr>
<td>Author(s)</td>
<td>Engine Type</td>
<td>Cylinder</td>
</tr>
<tr>
<td>-----------</td>
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<tr>
<td>Murata (1999)</td>
<td>Gasoline 1.6 l engine. Transparent glass cylinder</td>
<td>amount of oil after engine operation</td>
</tr>
<tr>
<td>Yu, Sawicki, Dunaevsky (1999)</td>
<td>Analytical solution of piston ring pack lubrication for 2 cyl. Truck air compressor</td>
<td>Full mass conservation boundary conditions defined by JFO theory</td>
</tr>
<tr>
<td>Takiguchi, Sasaki, Takahashi, Ishibashi, Furuhama, Kai, Sato (2000)</td>
<td>Six cylinder DI diesel engine</td>
<td>Capacitance sensors in the ring sliding surfaces</td>
</tr>
<tr>
<td>Seki, Nakayama, Yamada, Yoshida, Takiguchi (2000)</td>
<td>Optical measuring system in a pre-combustion chamber</td>
<td>LIF point measurements. 4 probes installed at 19°, 54°, 77° and 150°  - SAE 30 oil with</td>
</tr>
<tr>
<td>Engine Type</td>
<td>Coumarin 6</td>
<td></td>
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<tr>
<td>-------------</td>
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</tr>
<tr>
<td>5 kW, water-cooled single cylinder small diesel engine</td>
<td>Ring follow up motion to the cylinder liner</td>
<td></td>
</tr>
<tr>
<td>Bore/Stroke: 72 mm/72 mm</td>
<td>- OFT of compression ring becomes thinner at the liner upper portion due to oil starvation. OFT around TDC is determined by the compression ring and liner sliding surface roughness</td>
<td></td>
</tr>
<tr>
<td>Displacement: 293 cm³</td>
<td>- At TDC, the OFT increases as the tangential tension is reduced and decreases when the width is reduced</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- The amount of lubricating oil around each ring increases greatly as the engine load is reduced and the oil starvation observed at the compression ring around TDC is relieved.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>- The OFT of scraper ring shows similar tendency to that of the oil control ring.</td>
<td></td>
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</tbody>
</table>

  - Ring pack lubrication model to study the effect of piston secondary motion on the performance of a diesel engine piston ring pack in terms of gas flow and inter ring gas pressures
  - Piston secondary motion and the ring gap positions were shown to have an effect on the ring pack operating conditions, such as oil film thickness, friction, wear, oil transport and degradation.
  - Quasi static configuration considered (rings do not rotate). If this is taken into account a more complex dynamic behaviour is expected.

  - Theoretical studies were conducted concerning piston-ring dynamics. AVL GLIDE software, able to predict piston dynamics, piston-ring dynamics and oil consumption
  - Assymetrical gas load due to secondary piston movement causing ring fluttering
  - The higher inter-ring pressures in second piston land sharply increase the downward scraping effect. At the exhaust stroke the top ring received additional gas pressure on its lower ring side.
  - At top side, sealing was found to be responsible for a radial ring collapse.
<table>
<thead>
<tr>
<th>Tian and Wong (2000)</th>
<th>• Mixed lubrication 2-D model with consideration of shear thinning effects of multigrade oils. Oil squeezing and asperity contact were both considered for the interaction between the flanks of the twin land OCR and the ring groove.</th>
<th>• Around thrust and anti-thrust sides the difference between the minimum OFT of two lands can be as high as several microns due to piston dynamic tilt. • Piston dynamic tilt is able to create oil accumulation in different regions and results in oil transport in the upper piston regions.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rabuté and Tian (2001)</td>
<td>• Computer model to identify main mechanisms contributing to increased gas leakage and gas flow related oil transport</td>
<td>• High engine speed creates more ring instability in both axial and radial directions • Ring flutter and radial collapse generate more direct and stronger gas flow all around the circumference of the piston • At high engine load conditions, high bore expansion in engines with aluminium blocks gives significant increase to ring and mating parts, the onset of microwelding and scuffing can be delayed or completely avoided.</td>
</tr>
<tr>
<td>Gamble, Priest and Taylor (2003)</td>
<td>• Existing computer model extended to include additional oil transport mechanisms at the ring/cylinder interface and in the piston assembly</td>
<td>• Amount and direction of oil transport is dependant upon the condition of the piston ring assembly components (wear), top piston ring gap size and quantity of oil present on the piston lands • Further evaluation of the impact of oil accumulation to oil transport depends on shape and volume of oil accumulating ahead of the piston rings on the cylinder wall • Further work on modelling includes the inertia driven flow on the piston surfaces and oil mist generation</td>
</tr>
</tbody>
</table>
### Table 1.4 CAVITATION IN LUBRICANTS

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental Set-up Measurement Technique</th>
<th>Major Findings Results and Remarks</th>
</tr>
</thead>
</table>
| **Parkins, May-Miller (1984)** | Squeeze film between two flat horizontal surfaces was photographed and viewed through its lower surface. Two pressure transducers were mounted flush with the liner surface. Pressure and displacement were displayed on oscilloscope. | • Different cavitation regimes were identified.  
• Cavitation bubbles appeared which are fed from within the film  
• Film pressure increases more rapidly than it decreases  
• Transducer recorded bubble cavitating pressure equal to atmospheric. Sub-zero pressures were recorded when film thickness was at its minimum. |
| **Floberg (1965)** | Theoretical and experimental investigation of the geometry of a rotating cylinder lightly loaded against a stationary plane | • Excellent correlation between analytical and experimental results.  
• Depending on adhesion and cohesion forces of the oil, it will divide in streamlets whose behaviour can easily be experimentally determined.  
• Determination of number of striations or streamers in a width unit.  
• Subatmospheric pressure, as with a separation boundary condition could be tensile. The shape of the pressure curve depends on the boundary condition.  
• The influence of surface tension can be neglected even at very low loads. |
| Olsson (1965) | Rotating journal bearing | • Two types of cavitation appeared in the 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, thus having free connection with the atmosphere.
- Bubble cavitation, which occurs when the pressure reaches the absolute vacuum. Bubbles are gradually transformed into fingers.
• The cavitation has been visually studied in the experimental rig. The results showed good agreement with Jakobsson and Floberg theory.
• The thin streams of oil are assumed to be at ambient pressure and to have a mean flow velocity equal to the mean of the surface speeds. |
| Coyne and Elrod (1970) Part I and II | Theoretical model which assumes the presence of a continuous cavity-fluid interface beneath which all the fluid flows to form a uniform layer of certain thickness. | • Improved boundary conditions in the case of film rupture.
• Comparisons with separation data of other researchers gave credit to the theoretical predictions.
• In part II a slider-bearing application is confirmed by experiment. |
| Taylor (1973) | Theoretical results for the cylinder-plane and journal bearing configuration. Three outlet boundary conditions have been postulated for the lubricant-cavity interface:
- Swift Stieber
- Separation
- Floberg | • Cylinder-plane: Separation and Swift-Stieber gave very different results
• Journal bearing: Separation does not occur at eccentricity ratios below about 0.3 hence, a stable cavitation pattern should not emerge for such conditions. |
<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Description</th>
<th>Key Points</th>
</tr>
</thead>
</table>
| Brown and Hamilton (1975) | Examination of complete hydrodynamic behaviour of a piston ring in a motored rig with small piezo-electric transducer and a capacitance film thickness gauge fitted in the cylinder liner of a Petter AV1 diesel engine | • First measurement of oil film pressure under a piston ring  
• If the oil jets were stopped, the pressure only covered part of the ring indicating the occurrence of starvation.  
• Single and double tapered rings were used: the latter clearly showed cavitation as well as negative pressures  
• The pressures measured under the ring pack indicated normal hydrodynamic lubrication. |
| Savage (1976)             | Mathematical model for describing steady perturbations to a steady and uniform flow of a viscous fluid | • 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 |
| Brown and Hamilton (1978) | Piezoelectric pressure gauges with a range of $-2$ to $+2$ MN/m$^2$. A pair of capacitance gauges were mounted either side of the pressure gauge in the liner. | • 10 degrees before top dead center, the pressure profile corresponded to the Reynolds boundary condition.  
• 10 degrees after top dead centre, large positive pressures were measured at the inlet. At the outlet, negative pressures as low as $-0.7$ MN/m$^2$ were measured.  
• Cavitation was reproducible with an outlet pressure of $-0.1$ MN/m$^2$, corresponding to the oil vapour pressure. |
| Dowson, Taylor (1979)     | Theoretical study on cavitation in bearings                                  | • Two basic forms of cavitation in lubricant films are presented:  
  - Gaseous in which gas cavities arise in a lubricant due to ventilation from the surrounding atmosphere whenever subambient pressures occur.  
  - Vaporous when the lubricant may boil at ambient temperature if the pressure falls to its vapour pressure. |
<table>
<thead>
<tr>
<th>Source</th>
<th>Description</th>
</tr>
</thead>
</table>
| Moore and Hamilton (1980) | Small piezo electric transducers and capacitance gauges mounted in the liner, 9.5° below TDC. | • Problems occurred when the cylinder pressure was greater than the hydrodynamic pressure under the ring.  
• Shock wave hid the signal (when exhaust valve closed during the induction stroke) |
| Rastogi and Gupta (1992) | Theoretical analysis was carried out to describe the growth and collapse of cavities in a shear thinning fluid which was cyclically deformed in a coaxial disk geometry. Equation for film thickness were used and Newtonians and non-Newtonian fluids were assumed to have the same viscosity. | • Reverse squeezing and converging diverging wedge flow are responsible for cavitation in dynamically loaded journal bearings. The former is responsible for much of the damage.  
• Shear thinning influences cavitation behaviour by modifying the film thickness.  
• The cavitation zone is significantly larger in shear thinning than for Newtonian lubricants because of the lower tensile strength of the shear thinning fluids. |
| Deshimaru (1994) | Transparent acrylic aperture in tube for visual inspection of cavitation. Four paraffin oils with different viscosity levels were used. Heated hydraulic fluid under circulation condition. Upstream and downstream pressures were adjusted by relief valves. | • Cavitation tended to occur more readily with rising oil temperature and rising upstream pressure.  
• The size of air bubbles in the cavitation tended to refine with rising oil temperature. |
<table>
<thead>
<tr>
<th>Source</th>
<th>Methodology</th>
<th>Findings</th>
</tr>
</thead>
</table>
| Yang and Keith (1995) | Computer model which combines a compressible fluid model, a pressure viscosity relation and elastic surface deformation with cavitation. It determines full film, cavitation and reformation regions. | - Cavitation does not occur immediately after the piston ring passes through TDC. The film shape changes to a flatter shape due to elastic deformation and high trailing edge pressure forces the lubricant to replenish.  
- With EHL effects taken into account, the peak pressure in the full film region is about 14% lower.  
- Pressure reformation has a very strong effect on the peak pressure in full film region and cavitation location. |
| Kim, Azetsu, Yamauchi, Someya (1995) | Simulation rig: piston ring is a straight glass bar test piece and cylinder a reciprocating flat plate. The state of oil film is observed and photographed through the glass test piece. Also pressure pick up systems for oil film pressure measurements | - After dead center, negative pressure involving tension arises before oil film rupture occurs. The magnitude of negative pressures increases with the increase of crank angle.  
- In the diverging part of the full film region the first oil film rupture occurs and results in the generation of internal cavities. With their generation, tension is released but negative pressure is still maintained in internal cavities.  
- A new series of internal cavities is generated by suction from the rear edge of the test piece and together with the internal cavities they coalesce. Then, the negative pressure in the internal cavities returns to atmospheric pressure.  
- The shape and dynamic behaviour of oil film after oil film rupture are greatly influenced by revolution speed. |
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<tr>
<th>Author(s)</th>
<th>Title</th>
<th>Key Points</th>
</tr>
</thead>
</table>
- The results are sensitive to the choice of boundary conditions.  
- Significant negative hydrodynamic pressures suggest calculations using Coyne and Elrod separation. |
| Dowson, Taylor, Priest (2000) | Computation model for top piston ring of a Caterpillar 1Y73 single cylinder diesel engine. Focus on how cavitation is best modelled. Alternative cavitation models include:  
a) full Sommerfeld  
b) Reynolds, cavitation and reformation  
c) Flow separation  
d) Modified Reynolds separation | - Boundary conditions are sensitive to the assumed nature of the cavitation in the diverging lubricant film.  
- Alternative cavitation models gave significant differences in hydrodynamic pressure profiles, lubricant film boundaries, film thickness, oil flow and friction  
- Combination of theoretical and experimental approach is essential.  
- Reynolds cavitation and fluid film reformation may be applicable at high loads and Coyne and Elrod fluid film separation at low loads. |
<table>
<thead>
<tr>
<th>Bolander, Steenwyk, Sadeghi and Gerber (2005)</th>
</tr>
</thead>
<tbody>
<tr>
<td>• Good correlation between measured and calculated data. • Results encompass the entire range of lubrication regimes experienced by the piston-ring from boundary to full-film hydrodynamic lubrication and the transition between the regimes is presented in detail. • Shift of OFT at dead centres is due to squeeze-film effect.</td>
</tr>
</tbody>
</table>
Table 1.5  CALIBRATION ATTEMPTS OF THE LIF SIGNAL IN ENGINE APPLICATIONS

<table>
<thead>
<tr>
<th>Reference</th>
<th>Experimental Set-up</th>
<th>Major Findings Results and Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smart and Ford (1974)</td>
<td>Light by a mercury lamp illuminated lubricant film on a spinning shaft</td>
<td>• For the calibration of the signal 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.</td>
</tr>
</tbody>
</table>
| Ting (1980)                | Single cylinder research engine with transparent sleeve. A red fluorescent dye was used for the LIF experiments. | • The results were not calibrated but it was suggested that grooves of known depth cut into the piston rings could be used to calibrate the LIF signal.  
  • Piston ring clearance could affect oil consumption. |
| Hoult, Lux, Wong, Billian (1988) | Test version of Cummins VT-903 engine fitted with a quartz glass window. He-Cd 14 mW laser at 441.6 nm | • First attempt of in-situ calibration to the measurement of oil film thickness in the piston-ring zone.  
  • Two independent methods were employed for the calibration of the LIF signal:  
  1. **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.  
  2. **Dynamic calibration** 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 is compared to the ring measured beforehand. Piston-ring profiles were measured before installation with Talysurf.  
  • The researchers found that there is an agreement between in-situ dynamic and the bench calibration coefficients.  
  • The wear pattern on the cylinder liner demonstrates that the window inserted into the liner does affect the lubrication of the cylinder and piston.  
  • Afterwards researchers said that it is difficult or impossible to perform the ring profile fitting calibration because of oil starvation and ring twist. |
| Lux (1990)                 |                                                                                      | • Calibration was performed with marks on piston.  
  Identifying marks on piston: |

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| **APPENDIX A** | Fired single cylinder diesel engine, with quartz window | 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 μm.

- Large differences were found between statically and dynamically obtained calibration coefficients
- Oil mass balance performed but large differences between oil flows and actual oil consumption were observed. |

| Wong and Hoult (1991) | Two identical small production IDI diesel engines, five different ring configurations. | 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 μm and 19.6 μm. |

| Richardson and Borman (1991) | Cameron Plint wear tester and firing diesel engine for in cylinder measurements. | The researchers used small steps for calibration. The method of ring profile fitting has to deal with problems arising from oil starvation and ring twist.

**Options for calibration:**

1) Small step on the ring so that lubrication would not be affected as in the case of a groove around the ring.
2) Capacitance gauge, but its size is much larger than the fibre optics and are thus more intrusive and tend to average... |
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<tbody>
<tr>
<td>• Grooves in the piston skirt and top compression ring with depths that are readily machined or spark eroded (i.e. &gt;25 μm). The filling behaviour of the grooves was stochastic in nature and that generally there was insufficient oil in the contact to support complete filling. 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</td>
<td></td>
</tr>
<tr>
<td>• Readings over a larger area.</td>
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<tr>
<td>• <strong>Dynamic Calibration Method:</strong> Calibration steps were ground into the third piston-ring. One step was put on each side of the ring face. The steps were ground to depths of 15 and 20 μm. These were the smaller sized steps that the machinists could grind into the ring face. The third ring was pinned to ensure that the steps passed over the measurement region.</td>
<td></td>
</tr>
<tr>
<td>• <strong>Static calibration:</strong> Bench top fluorescence calibration cell which had the facility to continuously vary the film thickness. The researchers concluded that:</td>
<td></td>
</tr>
<tr>
<td>• Light of combustion might affect the signal</td>
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</tr>
<tr>
<td>• Thermally stable fluorescent dyes could be used</td>
<td></td>
</tr>
<tr>
<td>• The TDC fibre broke, possibly due to thermal stresses in the liner when the engine was fired</td>
<td></td>
</tr>
<tr>
<td>• Photobleaching could be neglected for engine tests</td>
<td></td>
</tr>
<tr>
<td>• The effect of combustion light was minimized by the height of the piston’s crown land.</td>
<td></td>
</tr>
<tr>
<td>• Starved lubrication conditions were found.</td>
<td></td>
</tr>
<tr>
<td>• The in-situ calibration did not work as expected; the calibration grooves of 15 and 20 μm in the third piston ring proved too deep.</td>
<td></td>
</tr>
<tr>
<td>• The oil film thickness decreased with engine speed.</td>
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</tbody>
</table>
| Phen, Richardson, Borman (1993) | 1Y73 engine with fused silica window. Two multigrade oils were used. | intensities and oil film thickness and the temperature dependence of the fluorescing quenching effect, a bench-top calibration procedure was chosen.  
- Fluorescence quenching increased at elevated temperatures.  
- Thermal expansion of the calibrated cell was observed.  
- Oil film pressure effects on fluorescent quantum yield were found negligible.  
- The oil film under the compression ring was not affected by engine speed.  
- At medium to high load, the lower viscosity oil provided, as expected, thinner oil films but at low engine speeds and low load the opposite was observed. |
| --- | --- | --- |
| Motored single cylinder engine | Ring profile fitting was used for calibration but it this method is subject to serious error due to variation of reflectance over the ring surface. The temperature of the liner is important because the oil viscosity is dependant on the liner and ring surface temperatures. Static calibration was used as well. The reflexion characteristics of the ring and piston were important in such a calibration. The static test rig showed that the calibration was linear.  
- Static and dynamic calibration: This calibration was not correct for some fibres and this was determined by comparing the actual measured profiles of the rings versus what was measured by the fibres as the ring passed by. The profiles did not correspond. 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.  
- The fibres proved to have different calibration coefficients.  
- Expected trends were found with speed and liner temperature.  
- Dynamic calibration is necessary and the reflectivity was important to take into |
<table>
<thead>
<tr>
<th>Source</th>
<th>Description</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sanda and Konomi (1993)</td>
<td>Toyota TRE3 research engine with a floating cylinder liner for friction measurements.</td>
<td>• Oil film thickness was found to be roughly proportional to the square root of the kinematic viscosity.</td>
</tr>
</tbody>
</table>
|                                |                                                                            | • Scanning LIF. Transparent cylinder and normal cylinder with quartz window.  
• Calibration: Special gauge with nickel-plated steps of 2.2 μm in the range of 2-10 μm.  
• 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.  
• The measured LIF signal was found to have very good linearity in the oil film range between 2 and 100 μm.  
• Both firing and motoring tests showed strong dependence of the friction force on the piston skirt profile.  
• Curvature was found to be important to both oil film thickness and friction.  |
| Dearlove and Cheng (1995)      | Motored single cylinder. Piston guided by linear bearings. The piston had only one ring and a single window was used. | • Difficulties were encountered with conformity due to circumferential variation.  
• Calibration was a problem.  
• Friction appeared to be mixed.  |
| Stiyer and Ghandi (1997)       | Single cylinder motored engine. Capacitance technique was used for the OFT measurements. | • Ring fitting method problems: 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. 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 fiber. Lastly the effect of ring twist will also alter the actual ring profile during engine operation.
- Some probes produced better results than others.

<table>
<thead>
<tr>
<th>Froelund, Schram, Noordzij, Tian, Wong (1997)</th>
<th>4 cylinder spark ignition engine, LIF focusing probe at 83% down from TDC</th>
</tr>
</thead>
</table>

- 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 to the same height on the LIF signal, the right calibration factor has been found, assuming that the toolmarks are fully flooded with oil.

<table>
<thead>
<tr>
<th>Arcoumanis, Duszynski, Lindenkamp, Preston (1998)</th>
<th>Lister-Petter single cylinder diesel engine. Firing engine LIF experiments fibres at TDC, mid-stroke and BDC.</th>
</tr>
</thead>
</table>

- Static calibration: A micrometer of enhanced resolution was used with an optical fibre, which was mounted flush with the anvil of the micrometer. This calibration device enabled a continuous variation of the oil film thickness.
- In order to compensate for the temperature effects in the calibration another bench-top calibration set-up was built. The surface of a fused silica block was etched and a series of grooves 5 mm wide, 5 mm apart and incremental in depth from 2 μm to 10 μm in 2μm steps were used for the calibration purposes. This custom built calibration standard enabled the study of temperature effects on fluorescence quantum yield by performing calibration of the oils at elevated temperatures.

<table>
<thead>
<tr>
<th>Nakayama, Morio, Takeshi and Okamoto (2003)</th>
<th>Static calibration apparatus is consisted of two thickness gauges. Their thickness could be changed from 0 to 100 μm. LIF sensor has same specification as the one fitted on the bearing for the film</th>
</tr>
</thead>
</table>

Calibration:
- Dynamic and static calibration values were consistent
- OFT from 0 to 100 μm could be measured with accuracy with both calibration methods

LIF results:
- Precision measurements of OFT for the engine bearing
- OFT differs due to shaft misalignment
- Measured results by LIF agree well with the calculation results by EHL analysis
measurement. A shaft with two attached thickness gauges that leave between them a step of known depth (filled with oil) was used for dynamic calibration.