Investigation of cavitation development in the lubricant film of piston-ring assemblies

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Abstract: The piston-ring assembly is one of the most loaded internal combustion engine sub-systems. Inadequate film characteristics between the ring and the liner may result due to a number of reasons. The relative motion between these components replenishes the lubricant, but it also generates cavitation conditions at the outlet of the ring-liner conjunction. The onset as well as the development of cavities is heavily dependent of the relative velocity. At higher velocity the cavitation region develops later and lasts longer. Therefore, to predict the cavitation behaviour, the mechanisms of oil film formation and pressure distribution should also be considered. Cavitation conditions are expected to occur when the local lubricant pressure drops below the vapour saturation pressure. Ideally such experiment is performed in a combustion engine under firing conditions. However, the main disadvantage of such an approach is the complexity required to decouple individual contribution of separate physical phenomena. The current research proposes an experimental technique where the cavitation is monitored in a test rig which simulates the lubrication conditions in a piston-ring assembly. Simultaneous measurements of the oil film pressure and oil film thickness are compared with fast speed camera recordings.

1. Introduction
The interface between piston-rings and cylinder liner represents one of the most hostile environments in the internal combustion (IC) engine. Due to the negative influence of the mechanical friction and wear over the engine efficiency, durability and noise, the interest of automotive industry in engine friction research has been continuously growing. Therefore, optimal rings-liner lubrication can be obtained considering the influence of several factors. The most important ones are the energy lost (due to friction), oil consumption, wear and exhaust emissions. Thin lubricating films encourage a high ring-liner friction force which decreases engine efficiency, increases fuel consumption and encourages surface interaction leading to wear. Alternatively, thick films may lead to greater oil consumption, which is a direct contributor to hydrocarbon and particulate emissions. In a modern IC engine, the engine friction could be responsible for up to 10-20% of the fuel used [1-4]. Of this, the contribution of the piston-liner can be up to 60-70% [5], with most of the energy lost in the rings-liner interface [6].

The physical phenomenon of cavitation can take place in fluids if the local pressure drops below the atmospheric pressure. It is believed that in most engineering applications cavitation has a negative impact on performance [7]. In a lubricated conjunction, cavitation occurs in the diverging part of the contact and it is responsible for a partial or complete collapse of the lubricated film. Consequently, it reduces the load-caring capacity of the interface and affects the lubricant film thickness, friction force...
and lubricant flow rate [8-11]. Therefore, for the case of the ring-liner interface it is desirable to diminish the effect of cavitation as much as possible.

To control the physical mechanisms leading to cavitation inception, it is important to firstly classify and understand them. A large wealth of literature is reported for the theoretical modelling and experimental verification of cavitation in journal bearing [8, 9, 12, 13]. In contrast, little attention has been paid specifically to the nature of the cavitation in the piston-rings conjunction [14-16]. Although the basic principle of cavitation formation is very similar for both cases, there are several important differences. The large majority of journal bearings (used in engineering applications) have a continuous rotating motion. This encourages the film formation, diminishing friction losses and protecting the two sides of the lubricated conjunction from direct interaction and consequent wear. During engine operation, the piston reciprocates from one dead centre to the other. Consequently, the entrainment velocity of the oil in the lubricated conjunction fluctuates in a broad range. It is zero at the dead centres and it reaches maximum between the two positions. The load applied on the ring-liner conjunction is directly proportional to the pressure behind the ring. This is usually a fraction of the pressure in the combustion chamber, and therefore, it varies extensively during the engine cycle, reaching maximum near top dead centre. Additionally, during the reciprocating motion of the piston, a very thin film of oil is left behind the rings. Near top dead centre, the film is subjected to a very high temperature cycle and it is exposed to the combustion gasses. Therefore, it is expected that localised oil degradation takes place. During the following cycle, the degraded oil mixes with fresh oil and this can have dramatic influence over the lubrication regime. Thus, the ring-liner conjunction operates in inherently transient lubricating conditions, and therefore, the importance of understanding the mechanisms that governs oil film formation/cavitation is further emphasized.

Lubricant films cannot withstand large negative pressures, which lead to cavitation formation. Therefore, most converging-diverging conjunctions could exhibit a discontinuous liquid film [11, 13, 14]. Pockets (or cavities) of gas may interrupt the film, producing film rupture (or cavitation). Dowson et al. [17] propose a simple classification of the cavitation phenomena based on the main mechanism that governs it. They consider that there are two types of cavitation: “vaporous cavitation” and “gaseous cavitation”. The first type occurs when the lubricant pressure is reduced to its vapour pressure, at which point evaporation or boiling will result. The second is encountered when the lubricant pressure falls below the saturation pressure and dissolved gases are emitted from the solution. A pressure reduction below ambient conditions may either encourage suspended bubbles of gas to grow or draw gas into the lubricating film from an external source such as atmosphere. This form of gaseous cavitation is called ventilation [17].

2. Test rig and experimental setup

2.1. Experimental test rig

The current research considers an idealised test rig which simulates the lubricating conditions between the piston rings and the cylinder liner in a reciprocating engine. The advantage of this approach is that the tribological conditions can be isolated from the plethora of physical phenomena occurring in the piston-liner assembly under firing conditions (such as ring and piston dynamics, thermal and elastic deformations of the rings, circumferential ring variations, lubricant degradation and blow by). Additionally, a test rig approach allows an easy access to the ring-liner interface, and therefore, a better understanding of the lubricant characteristics.

Figure 1 shows a schematic representation of the test rig. A flat horizontal plate (3) reciprocates above a stationary ring specimen (2). The liner is made of heat treated gauge steel (representative for the cylinder liner) and secured in an aluminium block (4). To measure the pressure between the liner and the ring specimen, the method proposed by Dellis and Arcoumanis [15] was adapted for the current conditions. Controlled load (27) is applied on the ring-liner contact by a loading arm. Low friction between the loading arm and the liner specimen is ensured by a set of high precision low
friction roller bearings (9). The reciprocating motion of the liner is obtained using a variable speed DC electric motor (17), which was mechanically isolated from the rig to minimize any vibrations. The motor is coupled to the “crank guide linear bearing” (12), through a crank mechanism (13). This arrangement converts the rotation of the crank mechanism into the reciprocating motion of the liner.

Isolation of the shaft is achieved by a flexible neoprene coupling (16). In order to maximize the speed stability of the system and reduce vibrations, provision is made to increase and adjust the system’s inertia using a pair of contra-rotating weights (14) driven from the shaft and coupled together by a pair of low noise nylon helical gears (15). To improve the measurement accuracy, the electric motor’s angular velocity is monitored by a shaft encoder (18) and the corresponding linear sliding motion of the liner is computed considering the mechanism geometry. The shaft encoder, with a resolution of 2000 pulses per revolution from the process unit (19), allows measurements every 0.18 degrees. The liner holder is connected to the drive mechanism by a removable connecting joint (11). Access to the ring specimen is allowed by removing the joint and rocking the liner around pivot (10). The ring specimen is fixed in the ring holder which is further placed in the ring assembly (1), which is attached to the base. The ring holder sits on a knife edge which allows it to tilt in the transverse direction ensuring good conformity between the ring and the liner.

![Schematic diagram of single ring test rig](image)

The lubricant is stored in a tank (25). An electric pump (20) circulates the oil firstly through a Parker oil filter (23), secondly through a heat exchanger (24) and then feeds it to the ring/liner interface by eight jets located on both sides of the ring, ensuring a fully flooded area. The oil is collected in a bath which is then drained back to the tank (25). The flow is controlled by a pressure regulator (21). A K-type thermocouple monitors the temperature oil sprayed to the ring-liner interface. The oil temperature can be increased from room temperature to 80°C via a controlled box (18). During operation the oil temperature has a variation of ±0.5°C.
To investigate the effect of contact load on the specimen ring, the applied weights (27) can be changed. Arcoumanis et al. [16] showed that the load on the ring can be expressed as a function of speed as:

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

where \(g\) is the acceleration due to gravity, \(m_h\) is the mass applied on the hanger, \(R_{crank}\) is the crank radius, \(\omega(\theta)\) is the crank angular velocity, \(b\) is the ring width and \(B\) is the ring length.

2.2. Measurement techniques

The oil film thickness is measured using a Laser Induced Fluorescence (LIF) technique. The basic principle of this technique uses the wavelength shift of a laser beam passing through a film of lubricant to predict the amount of oil between the two faces of the conjunction. Figure 2 shows a schematic representation of the basic concept of the laser induced fluorescence technique. A multimode fibre was used to transmit laser light through the liner to the ring specimen. The oil between the ring specimen and the liner wall was illuminated with “blue light” laser (\(\lambda=488\) nm). The illuminated oil then naturally fluoresced in the green spectrum region (around \(\lambda=500\) nm). The fluorescence light was transmitted out the liner with the same multi-mode fibre.

![Figure 2. Basic setup for fibre optic in the liner to measure oil film thickness by LIF method](image)

Figure 3 shows the practical arrangement used in the experiment. The illuminating laser beam is provided by an Argon-ion air-cooled laser with an operating wavelength of 488 nm and the output power of 30 mW. A photomultiplier tube is used to detect the fluorescence light intensity. To correlate the fluorescence light intensity and the oil film thickness, the method described by Arcoumanis et al. [16] was carefully adapted. The laser light travels through an optical fibre to the liner side of the conjunction and it is emitted from the end of the fibre to illuminates the oil film. The refracted light is routed out of the contact interface through the same optical fibre along with the illuminating laser light. The dichroic mirror (see figure 3) is transparent to the green fluorescent light, but reflects the blue laser light. The green light passes through two further filters, one orange glass filter and one interference filter, which remove any noise. Finally the light intensity is converted into low current output by the photo-multiplier. A Digital Amplifier (DA) type socket amplifier converts the current into amplified voltage signal.
The voltage is converted into film thickness using an in-situ calibration technique. Figure 4 shows the experimental test rig during the calibration and figure 5 shows the calibration curve. The liner is positioned at mid stroke and a high precision dial gauge is placed on the liner side of the ring-liner conjunction. The liner is lifted by the calibration screw in 2 µm increments.

The calibration curve shows the average values of the film thickness as well as the error bars of maximum and minimum values. Due to technical limitation of the dial gauge calibration technique, the minimum film thickness measured during the calibration is 2 µm. Since the oil film thickness during normal operation conditions of test rig drops below this value, the calibration curve was extrapolated to include thinner films. It can be noted that the calibration line does not pass through zero. It is believed that this is due to the background radiation [15] and noise in which alter the signal in the photomultiplier. Another possible reason [18] is the incomplete filtration of the laser light resulting in DC offset.

To measure the oil film pressure the technique described by Dellis and Arcoumanis [15] and improved by Arcoumanis et al. [19] was carefully adapted. The pressure is measured by a miniature silicon diaphragm pressure transducer manufactured by Entran, model EPIH-412-S440-20B. The pressure transducer is connected to a Wheatstone bridge conditioner and its signal is amplified by a FLYDE amplifier. The value measured by the sensor represents an average over the measurement window. To further improve the measurement resolution, the pressure sensor and the oil film were linked through a narrow slit of 0.1mm. To avoid any empty space and to provide an uninterrupted conjunction “roof”, for the oil film, a small amount of silicon rubber was added to the surface of the
pressure sensor before inserting it into the liner. This rubber filled all the additional empty spaces. After it cured, any excess rubber present on the liner surface was carefully removed.

The data acquisition system used for the experiment consists of a National Instruments 16-bit 6035E PCI data acquisition card and National Instruments SC-2345 signal conditioning unit (5 in figure 1). The 6035E device features 16 channels with a 16-bits resolution. This card has a timing system, of 50 ns resolution, for time related functions.

To enable oil film visualisation during the test rig operation, the metal liner specimen was removed and replaced by a quartz glass window as shown in figure 6 and figure 7. The quartz liner has a 27 mm × 65 mm rectangular section contained in an aluminium frame witch has identical dimensions with the steel liner specimen.

![Figure 6. Metal liner](image1)

![Figure 7. Glass liner](image2)

The oil flow between the piston-ring and glass liner have a highly transient behaviour. Therefore, to fully understand the complexity of the physical phenomenon interacting during the formation of the cavity structure a high speed digital video system (7) (Fastcam-APX RS) was used. This arrangement allows capturing the development of the oil flow in the piston-ring liner conjunction. The video system allows frame rates of 5000 up to 250000 frames per second. However, higher resolution/frame rates require progressively higher computer memory allocation. For the current experiment it was satisfactory to use 10000 frames per second with a resolution of 512 × 512 pixels. The illumination of the test area was achieved by several strong halogen floodlights (8 in figure 1) which proved adequate to provide enough light for the Closed Coupled Device (CCD) video chip. The camera was triggered, synchronised with the TDC position pulse from the shaft encoder, and controlled with the image processor unit via a computer (6 in figure 1).

### 3. Results

The cavitation region is characterised by a significant drop in lubricant pressures below the surrounding atmospheric pressure and a consequent collapse of the intervening fluid film. For the contact between the glass and the ring specimen there is a region (in the diverging side of the contact) where the low pressure encourages formation of cavities. Arcoumanis et al. [19] proposed an experimental technique to measure the pressure in the lubricated conjunction. They have also proposed a theoretical model to predict the film thickness and the inception of the cavitation. However, the model (based on the integration of Reynolds’ equation using an “open end” boundary condition) can be successfully applied until the lubricating film in the conjunction is uninterrupted, and therefore, it can only predict the beginning of the cavitation area and not the oil film reformation.

To improve the current understanding of cavitation onset and evolution, it is important to correlate the local lubricant pressure with the amount of oil present in the conjunction during the cavitation zone. The LIF technique (described in the previous section) measures the amount of oil in a very
narrow control volume. Using the calibration method previously described, this amount of oil can be further converted into the thickness of the oil film. The advantage of this approach is that the technique does not measure the distance between the two sides of the conjunction, but rather the amount of oil encountered by the laser beam. Therefore, considering the liner perfectly flat and neglecting the local deflection of either side of the conjunction, the LIF measurement coincides with the ring profile in the regions where the film is continuous (before and after the cavitation region). If the oil film is partly depleted (because of cavitation) the method measures only the thickness of the oil film which adheres to either side of the conjunction, and therefore, these areas can be easily identified as they deviate from the ring profile.

Figure 8 shows the oil film pressure and the film thickness as a function of the crank angle degree, for “downstroke” and “upstroke” movement of the liner specimen at a corresponding motor speed of 600 rpm and a load of 977 N/m. To understand these figures it should be taken into consideration that both sensors are rigidly mounted on the liner, which is rapidly sliding, moving the measuring point along the profile. It should also be noted that the rig profile is not symmetrical, having different curvature radius on each side. This construction of the test rig (also characteristic for an automotive type piston ring) can highlight the importance of correctly understanding the ring geometry. The dotted blue line represents the ring profile, and the dotted red line the atmospheric pressure.

It was observed that the shape of the cavity strings is consistent over a large number of cycles; however, their exact location along the ring profile cannot be currently predicted. Therefore, it is possible that in successive tests the laser beam measures the film thickness either within one of the empty areas of the cavity or in the thicker string between two cavities (see figure 10 below). To overcome this shortcoming, in the current stage of the research the oil film signal was averaged over 100 successive cycles. This approach can predict accurately the location of the cavitation boundaries, but it cannot predict the exact thickness of the surviving film. For this a separate test will be conducted, to simultaneously measure the oil film parameters (pressure and LIF) while visualising the film with the high speed camera.

To study the influence of entrainment speed and load on cavitation formation, the fast speed camera was used to visualise the inception as well as the subsequent evolution of individual cavities. Figure 9 shows the liner velocity and load variation along a section of the downstroke. To explain some particularities of this process, the pictures taken in four individual positions (a-d in figure 9) are shown out in figure 10.
Although it is expected that in each one of the selected positions the pressure curve has a negative region in the divergent part of the conjunction, it is observed that the cavitation does not start simultaneously in the entire area. It starts separately in several nucleation spots (figure 10 a) where it is believed that sub-micrometer scale features on either sides of the conjunction lead to localised improved cavitation conditions. From these initial nucleation areas, the cavitation spreads rapidly in a fractal-like structure. Dellis and Arcoumanis [15] used a still frame camera to visualise the cavitation development between a quartz liner and a ring specimen. They observed these structures and called them “fern cavities”. The initial ferns extend rapidly over the entire cavitation area, generating finger like structures known as “fissure cavities” (figure 10 c). The fissures are quickly replaced by the more stable structures “string cavities” (figure 10 d). The later structures are very similar with the ones encountered in journal bearing cavitation. The main difference is that in the current conditions, due to the high transience of the piston reciprocal motion, these strings are short lived. Finally, when the entrainment velocity diminishes, each cavity between two strings becomes a bubble which is released in the oil behind the contact. These bubbles could become particularly hazardous for combustion engine, as they tend to “foam” the oil and reduce the efficiency of the oil circulation. In engineering applications, to avoid this foam, the oil is doped with specific anti-foam additives. It is hoped that a fundamental understanding of the mechanisms governing the cavitation formation could ultimately contribute towards diminishing the need of such additives.

The advantage of using a high speed camera technique is well emphasized by figure 10 (b). This shows the transition between the initial “ferns” towards later “fissures”. Figure 10 a) shows that the initial nucleation points were located in the lower side of the viewing window. Therefore in this section of the contact the ferns developed earlier in the cycle. Consequently, the transition towards fissures also started earlier in this area. Meanwhile, the upper side of the viewing window was still dominated by the ferns cavitation. Considering that the cavitation area has an important role in the load carrying capacity of the contact, uneven development of the cavitation generates uneven distribution of the load. This can slightly tilt the ring and induce an even greater difference in the cavitation in different zones of the contact. Therefore, a model which could fully describe the cavitation behaviour of the ring-liner conjunction should include not only ring tribology, but also ring the ring dynamics.
4. Conclusions

The cavitation regime between the ring and the liner occurs in the diverging section of the conjunction, when the pressure drops below the atmospheric level. The phenomenon is highly dependent on the localised kinetics within the contact region as well as the dynamics of the ring and piston. The cavitation starts as nucleation and rapidly evolves into ferns, fissures, strings and finally bubbles.

The current research proposed a set of integrated experimental techniques to investigate cavitation onset and development. The oil film pressure and the film thickness are simultaneously measured, then compared with the images obtained by a fast speed camera. The advantage of the high collection rate of the camera is emphasised by the ability of analysing the evolution of individual cavitation structures.
5. References

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