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The Contribution of Different Oil Consumption Sources to Total Oil Consumption in a Spark Ignition Engine

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ABSTRACT

As a part of the effort to comply with increasingly stringent emission standards, engine manufacturers strive to minimize engine oil consumption. This requires the advancement of the understanding of the characteristics, sources, and driving mechanisms of oil consumption. This paper presents a combined theoretical and experimental approach to separate and quantify different oil consumption sources in a production spark ignition engine at different speed and load conditions.

A sulfur tracer method was used to measure the dependence of oil consumption on engine operating speed and load. Liquid oil distribution on the piston was studied using a one-point Laser-Induced-Fluorescence (LIF) technique. In addition, important in-cylinder parameters for oil transport and oil consumption, such as liner temperatures and land pressures, were measured.

Engine test data and modeling results were combined to separate and quantify the contributions of oil entrained in the blowby gas flow, oil evaporation, and oil flow past piston and valve guides into the combustion chamber. The results show that the contribution of each consumption source varies with engine operating conditions. At low load, oil flowing into the combustion chamber was found to be the major consumption source (90 percent), while the [1]contributions of oil evaporation and of blowby entrainment became more significant with increasing engine load. The variation of engine speed at full load increased the contribution of all three oil consumption sources. However, the relative importance of each source did not vary significantly with engine speed. The major contributions were from the oil transport (40 to 50 percent) and oil evaporation (30 to 40 percent).

INTRODUCTION

Oil consumption from the piston-ring-liner system significantly contributes to total engine oil consumption. Engine oil consumption is recognized to be a significant source of pollutant emissions in automotive engines. Unburned or partially burned oil in the exhaust gases contributes directly to hydrocarbon and particulate emissions. Moreover, chemical compounds in oil additives can poison exhaust gas treatment devices and can severely reduce their conversion efficiency. Oil consumption is also an important indicator for engine performance and, consequently, customer satisfaction. As a part of the effort to comply with increasingly stringent emission standards, engine manufacturers strive to minimize engine oil consumption. Moreover, to reduce development lead-time and costs, there exists increasing demand for analytical tools to predict oil consumption. This requires the advancement of the understanding of the characteristics, sources, and driving mechanisms of oil consumption. There is a general lack of oil consumption studies that connect comprehensive experiments and theoretical analysis.

Significant research effort has been devoted to the investigation of engine oil consumption phenomena in internal combustion engines. Five potential oil consumption sources have been suggested to contribute to total oil consumption during engine operation. The transport processes by which oil is consumed in are schematically illustrated in Figure 1.

A hypotheses supported by experimental data relate oil consumption to the mechanical transport of liquid oil into the combustion chamber due to inertia forces caused by acceleration and deceleration of the piston assembly (Figure 1a). The importance of this driving mechanism is believed to depend on the accumulated oil film on the top land and ring. Oil throw off from the top land was visualized in one-cylinder research SI-engines at low load conditions [1][2].

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In other non-quantitative studies, direct oil transport to the combustion chamber was found to depend on gas flow in the piston-ring-liner system. Gas pressures in the second land clearance, i.e. the volume between the top ring and second ring, can become greater than the combustion chamber pressure during some periods of the engine cycle. This pressure gradient will cause a reverse gas flow into the combustion chamber through the top ring gap and around the top ring groove if the top ring looses its stability in the groove. The reverse gas flow may transport oil in both liquid and mist form (Figure 1b) into the combustion chamber. This transport mechanism was also supported by visualization studies of the oil distribution in the piston-ring-pack, when the top ring was pinned. In these studies, oil flow through the top ring gap towards the combustion chamber was observed during low load conditions [3][4].

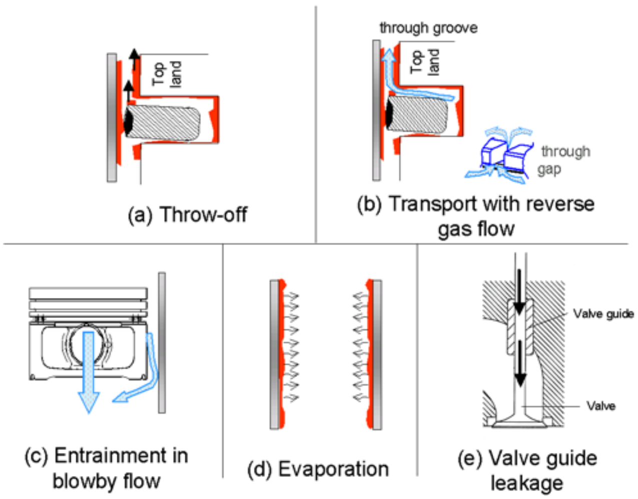

Figure 1. Schematic of oil consumption sources

Oil mist, also entrained in the recycled blowby gas flow (Figure 1c), has been found to enter the combustion chamber via the intake manifold system. Experimental studies on different engines quantified the contribution of oil in the crankcase ventilation gases to total oil consumption [5][6]. It was found that this oil consumption source could contribute significantly, depending on the specific engine, to total oil consumption.

Oil evaporation (Figure 1d) from the piston-ring-liner system was also found to contribute to total oil consumption, especially during severe operation conditions when the thermal loading of engine components is high. Several experimental results indicated that oil evaporation from the liner and piston might contribute substantially to oil consumption [7][8][9][10]. In addition, a number of purely theoretical approaches studied oil evaporation from the liner and found sensitivities in the evaporation process to oil composition and to component temperatures [11][12][13]. A recent experimental and theoretical study by the authors on the same test engine as used in this work demonstrated that oil evaporation from the liner was the main contributor to oil evaporation [14].

In earlier spark ignition engine designs, oil transport from the cylinder head through the valve guide into the intake port (Figure 1e) contributed significantly to oil consumption during part load conditions, when the intake manifold pressure is well below atmospheric. However, this oil leak path is effectively sealed in modern engines by employing positive valve stem seals [5]. Therefore, this oil consumption source is considered to contribute little to the total oil consumption in today's spark ignition engines.

In practice, all of the oil consumption sources described above contribute to total engine oil consumption. The relative importance of each source depends on different driving forces for oil transport that change with different design and engine operating parameters. Therefore, recognizing the contributions of different oil consumption sources enables engine manufacturers to solve oil consumption problems more effectively.

The purpose of this study is to characterize different oil consumption sources in a production spark ignition engine for a wide range of steady state engine operating conditions. It quantifies and analyzes the contribution of oil evaporation, oil entrained in the blowby gas flow, and oil flow into the combustion chamber passing by the piston system and valve guide, by combining comprehensive measurements and physics-based modeling work.

EXPERIMENTAL APPARATUS

A four-cylinder production spark ignition engine with an extensive diagnostic system was used as test engine to identify and quantify key oil consumption sources at different steady state speed and load conditions. The engine characteristics are listed in Table 1.

Engine Type	Port injected spark ignition engine
No. of cylinders	4
Compression ratio	10.4 : 1

Table 1. Test engine characteristics

All measurements were conducted with the baseline ring-pack design recommended by the manufacturer. This ring-pack consisted of a rectangular top ring with a barrel-faced running surface, a taper-faced Napier scraper ring, and a U-flex oil control ring, which is divided of about fifty ring segments with gaps between them. The details of the piston-ring-pack design are shown Figure 2.

The diagnostic system has the capability to measure real time oil consumption and blowby, as well as in-cylinder variables such as oil film thickness in the piston-ring-pack, land pressures, cylinder pressure, and liner temperatures.

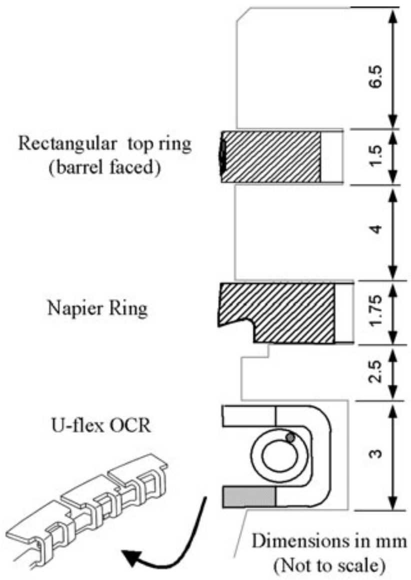

Figure 2. Piston and ring geometry

TOTAL OUTPUT MEASUREMENTS

A robust sulfur detector with its associated subsystems was implemented to measure real-time oil consumption. Further essential system elements to obtain reliable oil consumption measurements are high sulfur oil with a consistent sulfur concentration throughout the distillation curve, low sulfur fuel, and a heated sampling line to avoid condensation of unburned hydrocarbons. Table 2 summarizes the relevant fuel specifications.

Low Sulfur Fuel	
Sulfur	< 2 [ppm, wt.]

Table 2. Relevant fuel specifications

The sulfur content with distillation of the baseline oil is illustrated in Figure 2. A more detailed description of the sulfur tracer technique used in this work can be found in [15] [16] [17] [18] [19].

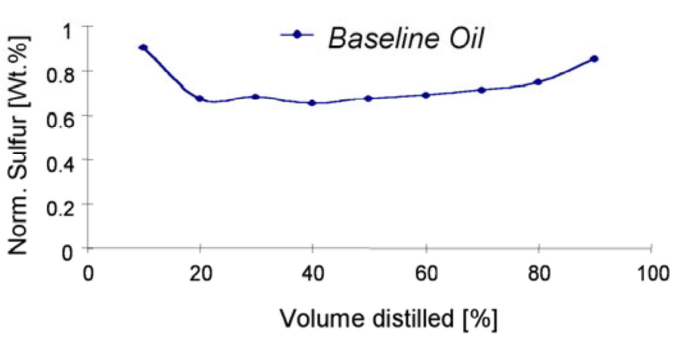

Figure 3. Baseline oil sulfur content with distillation

An accurate flow meter was used to measure the blowby gas flow rate into the crankcase. Details of the blowby flow meter are summarized in Table 3.

Principle	Von Kármán-vortex shedding
Range	4-100 [l/min]
Accuracy	< 1 [%]

Table 3. Blowby flow meter specifications

IN-CYLINDER MEASUREMENTS

The engine block was modified to allow for the measurement of in-cylinder variables such as oil film thickness along the piston, cylinder and land pressures, and liner temperatures in the third cylinder.

The one-point Laser-Induced-Fluorescence (LIF) technique was used to quantify the oil film thickness (OFT) between the piston rings and liner, and the oil accumulation on the piston lands during engine operation. Table 4 summarizes the principal specifications of the LIF technique. Figure 4 shows a schematic of the LIF setup on the test engine and sample LIF traces averaged over ten consecutive cycles. Additional details of the LIF technique and calibration can be found in [15] [20].

Laser	Helium-Cadmium
Power	~ 8 [mW]
Excitation Wavelength	442 [nm]
Dye	Coumarin 523
Fluorescence Wavelength	495 [nm]
Window transmission	90 [%]

Table 4. LIF specifications

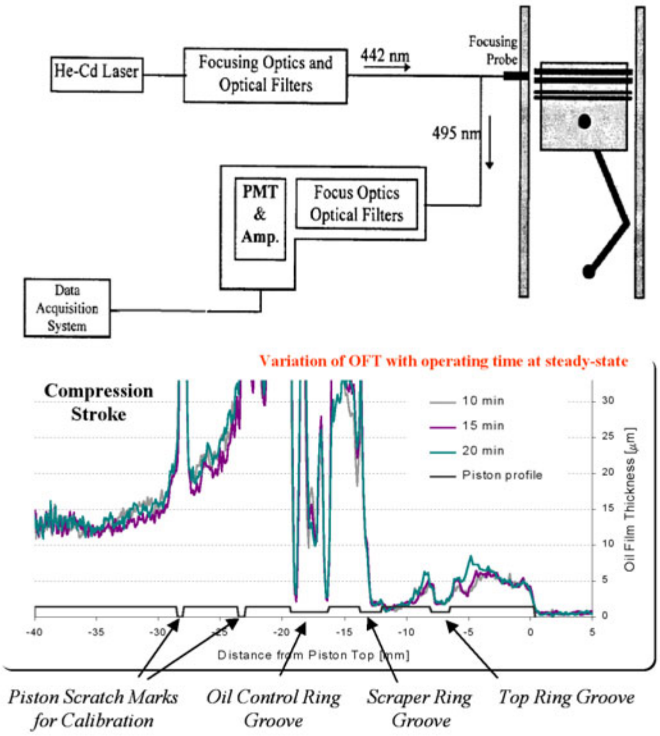

Figure 4. LIF setup and sample trace

Two pressure transducers were positioned along the cylinder liner at two different critical axial locations and inter ring land pressures were measured during the crank angle period of interest. An additional pressure transducer mounted on the cylinder head provided cylinder pressure traces.

Two temperature probes along the cylinder liner provided liner temperatures at the top center position of the scraper ring and at the bottom dead center position of the top ring. The positioning of the measurement probes implemented in the third cylinder of the test engine is shown in Figure 5. A more detailed description of the experimental apparatus can be found in [20].

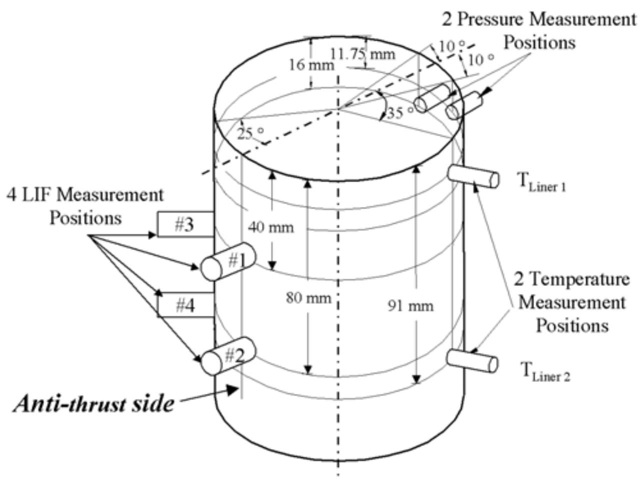

Figure 5. Measurement locations of in-cylinder variables

OIL SPECIFICATIONS

Oil consumption is directly related to physical oil properties, such as oil volatility and viscosity. The oil volatility directly governs the oil evaporation rate from hot surfaces during the engine cycle. Therefore, two lubricants with different volatility characteristics – different GCD curves (see Figure 6) – were used to investigate the impact of oil volatility on oil evaporation and consumption. The relevant oil properties are shown in Table 5.

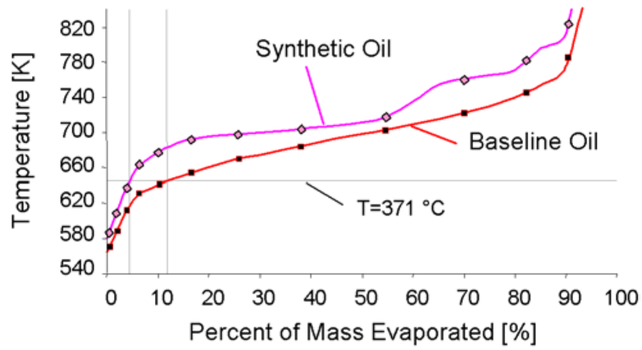

Figure 6. Distillation curves of baseline and synthetic oil

Baseline Oil (Mineral)		ASTM D
SAE Viscosity Grade	10W-30	
Sulfur [wt. %]	1.51	1552
Volatility: GCD % off @ 371° C	11.6	
Noack	16.8	5800
Viscosity @ 100°C [mm^2/s]	10.77	445
HTHS viscosity [cP]	3.04	4683

Synthetic Oil		ASTM D
SAE Viscosity Grade	0W-30	
Sulfur [wt. %]	1.68	1552
Volatility: GCD % off @ 371° C	3.7	
Noack	13.5	5800
Viscosity @ 100°C [mm^2/s]	11.22	445
HTHS viscosity [cP]	3.04	4683

Table 5. Relevant oil specifications of the test oils

It must be recognized that viscosity governs in large part complex oil transport mechanisms to the upper regions of the piston-ring-liner system and therefore affects oil consumption. The low shear viscosity affects different oil transport mechanisms in different regions on the piston surface such as inertia forces due to piston movement, gas flow dragging, and the interactions between the rings and grooves. The HTHS (High Temperature High Shear) viscosity is believed to influence ring and piston liner lubrication, and thus govern the oil transport on the liner.

As shown in Table 5, the two oils have been formulated with similar HTHS and low shear viscosity properties to minimize the difference in oil transport along the piston and liner. Even though the test oils low shear viscosities at 100°C are similar, they have different SAE oil classification numbers (SAE 10W-30 and SAE 0W-30). These oils must operate over a range of temperatures (from 100° up to 250°C) because of the temperature gradient along the piston assembly and because of the gradients' dependence on the operating conditions. Moreover, the oil viscosity decreases with increasing temperature. Consequently, any differences between the two oils viscosity at elevated temperatures could affect the oil transport mechanisms along the piston and thus oil consumption. Therefore, further lab analyses were performed on the test oils to examine their low shear viscosities at temperatures above 100°C (ASTM D 445 and SAVLAB LPYC density tests) and the Vogel equation was used to correlate the measured temperature-viscosity data. The obtained low shear viscosity-temperature relation for both oils is shown in Figure 7, which confirms the similarity of the oil viscosities at temperatures above 100°C. Therefore, any difference in oil consumption between the oils should be mainly a result of the difference in their volatilities.

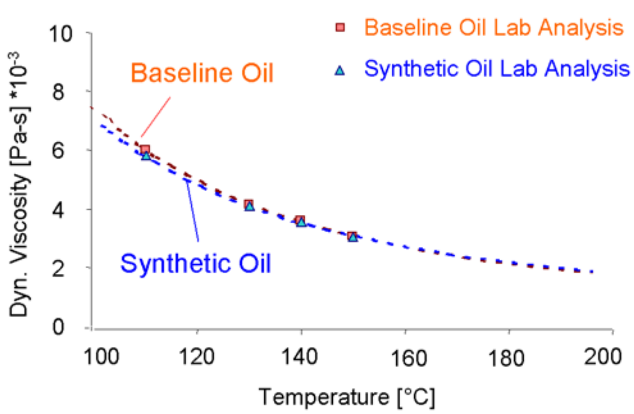

Figure 7. Low shear viscosity of both test oils

As discussed in [14], for both oils the average sulfur concentration by weight (ASTM D 1552) was used to determine the oil consumption rate.

EXPERIMENTAL CONDITIONS

TOTAL OIL CONSUMPTION, BLOWBY AND IN-CYLINDER PARAMETERS

All measurements presented here were taken during steady state speed and load conditions. The test engine could be operated between 2000 rpm to 5500 rpm and engine load could be varied by from 0% to 100% load. Total engine oil consumption, blowby, and in-cylinder variables were measured at five (0, 25, 50, 75, and 100%) load conditions at different speeds (see Figure 8). Engine load was defined by the engine brake torque compared to the torque measured at wide-open throttle (100% load), which is shown in Figure 9. Therefore, in Figure 8 the intake manifold pressure varied slightly for constant load conditions at different engine speeds. It has to be further noted that throughout this paper, 0% load represents the lowest possible load to operate the engine in a steady condition.

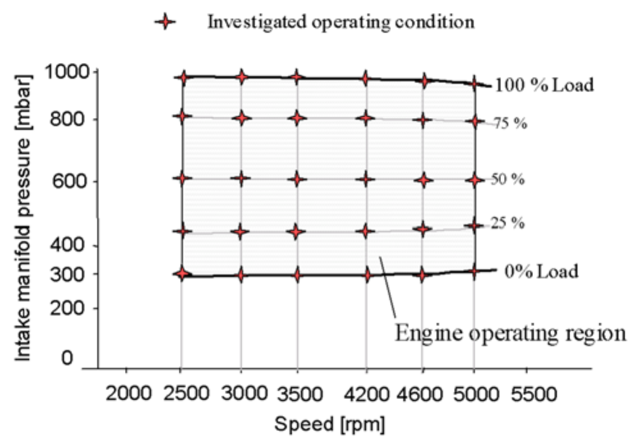

Figure 8. Investigated steady state engine operating conditions

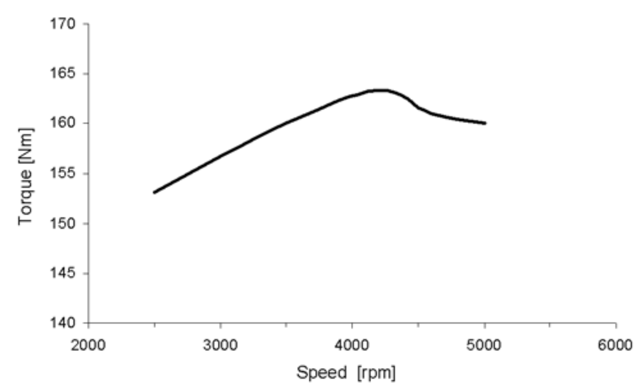

Figure 9. Torque at full load (100%) as a function of speed

BLOWBY OIL CONSUMPTION STUDY

As mentioned above the oil in the blowby gas flow can contribute significantly to total engine oil consumption. Along the blowby flow path from the combustion chamber through the piston-ring-pack and crankcase to the intake manifold system oil from different regions is entrained in the blowby gas. Initially the gases flow through various regions of the piston-ring-pack to the crankcase and entrain oil. On their flow path to the crankcase, the gases may flow through the ring gaps, through the ring groove during ring flutter, and through the clearance between ring and liner during ring collapse [21][25]. The oil entrainment in these regions is believed to be both in liquid and vapor form. More volatile oil compounds are assumed to evaporate from hot surfaces into the blowby gas stream, and the micron scale liquid oil film in the piston-ring-pack is atomized by locally high velocity gas flows. During most operating conditions, however, ring stability is preserved and the main leakage of consequence is the gas flow from ring gap to ring gap. This gas flow drags the oil on the piston lands toward the ring gaps due to interfacial shear stresses. It is believed that this oil atomizes in the ring gap as it flows through, dragged by the high velocity gas flow [46]. A large fraction of the oil is believed to be entrained in the gas stream. The oil transport to the ring gaps was found to depend on the average oil film thickness on the piston lands and on the cumulative volumetric blowby gas flow rate [23]. Therefore, both the oil distribution on the piston lands and blowby flow affect the oil entrainment in the piston-ring-pack. After passing the piston skirt, the oil-laden blowby gases enter the crankcase and mix with the crankcase gases.

In the crankcase additional oil entrainment can be expected, aided by oil splashing from moving parts, oil leaks from the bearings and the spray jet cooling of the piston. Spray jet cooling is employed in the test engine to reduce the piston temperature by enhancing heat transfer and to supply additional lubrication for the highly loaded major thrust side of the cylinder liner. Airborne oil in the crankcase from sources described above can reach significant velocities relative to the moving engine parts and the surrounding gas flow. The interaction of airborne oil with the gas flow may cause droplet break-up, forming smaller droplets due to the augmentation of surface tension; the impingement of oil droplets on moving surfaces may also generate smaller secondary droplets due to splash. In addition, vaporized oil in the blowby may condense along the flow path due to heat losses to the cooler environment, thus creating additional oil droplets.

The oil-laden crankcase gases are vented to the intake system. However, before entering the intake manifold the gases flow through an inertial oil separator to minimize oil entrainment and its impact on oil consumption and engine performance. As the gases flow through the separator, oil droplets are removed from the main gas flow and returned to the engine sump. The separation principle is based on inertial movements of the disperse phase, relative to the carrier gas streamline. However, not all particles can be efficiently separated from the carrier gas and droplet sizes smaller than a certain diameter follow the gas flow to the intake manifold system. From the above description, it is apparent that the amount of oil carried to the intake system depends not only on entrainment in the piston-ring-pack and crankcase, but also on the separator performance. All above described aspects for oil entrainment and separation vary with engine operating conditions, such as speed and load. Therefore, the oil transported with the blowby to the intake system, thus contributing to total oil consumption, varies with engine conditions, which emphasizes the complexity of these phenomena. It is appropriate to define the oil consumption source due to oil entrainment in the blowby gas flow as blowby oil consumption.

For the test engine, the contribution of blowby oil consumption was quantified for a range of steady state speed and load conditions. This contribution was obtained by determining the difference in oil consumption with and without blowby gas return to the intake system. In the latter case, the blowby gases were vented to the test cell exhaust trench after passing the oil separator. The investigated engine speed and load conditions are shown in Figure 10.

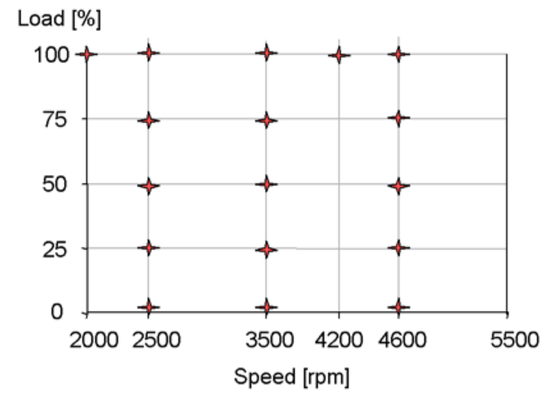

Figure 10. Investigated operating conditions of blowby contribution to total oil consumption.

OIL EVAPORATION STUDY

The importance of oil volatility and cylinder liner temperature on oil evaporation has been examined in an earlier paper [14], and are presented here to some extent for the sake of clarity. The authors believe that this will allow the reader to have a complete picture of the separation of the different oil consumption sources. Here the contribution of oil evaporation to total oil consumption of the test engine is quantified at different steady state speed and load by combining oil consumption measurements and predictions of the in [14] proposed multi-species liner evaporation model.

Oil consumption measurements were conducted for both test oils to understand and quantify the impact of oil volatility on oil consumption at different speed and load conditions. During these tests the coolant outlet temperature was controlled in the range 81 ± 1.5°C, since the engine is designed to operate at this standard thermal condition. In [14], it was demonstrated that oil evaporation from the liner was limited by gas side convection and could be adequately modeled as a mass convection process from the liner. Therefore, it is clear that changing operating conditions vary the mass transfer of oil from the surface and the engine component temperatures, which determine the oil vapor pressure on the surface. The mass transfer does not depend on oil composition and the temperature measurements showed little difference between the two oils. Consequently, any difference in oil consumption between the oils should be mainly a result of the difference in their volatilities.

Additional measurements were conducted with varying liner temperatures for both oils at different steady state speed and load conditions. A linear relationship between cylinder liner and coolant outlet temperatures was observed during all investigated conditions. Therefore, the liner temperature was easily varied by controlling the coolant outlet temperature. For the measurements, the coolant outlet temperature was varied between 55 and 90°C. Figure 11 shows the relationship between the liner temperature at the location of the scraper ring at TDC (TLiner 1) and coolant outlet temperature for different load conditions at 3500 rpm. It is evident that the engine thermal loading increases as the engine operating load increases. All investigated operating conditions for the evaporation study are summarized in Figure 12.

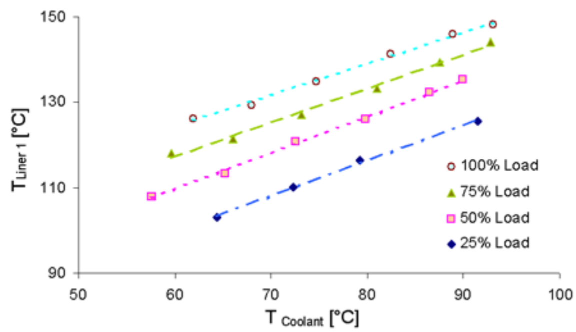

Figure 11. Effect of coolant outlet temperature on liner temperature at the TDC location of the scraper ring at different load, 3500 rpm

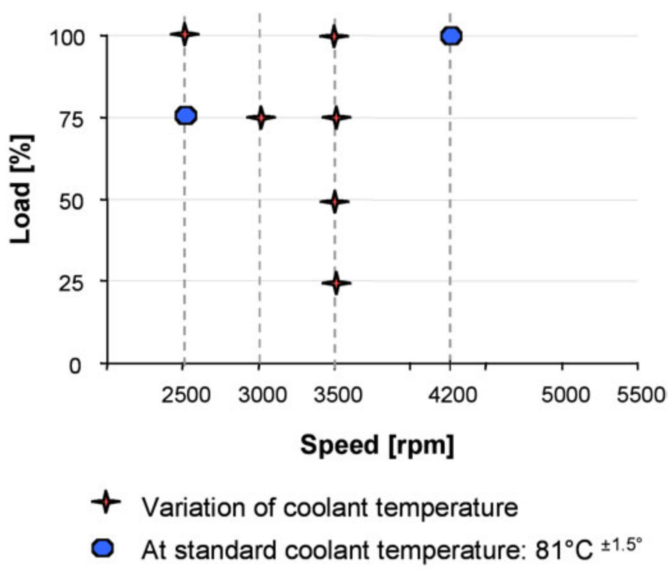

Figure 12. Investigated operating conditions (evaporation study)

RESULTS AND DISCUSSION

MEASUREMENTS OF TOTAL OIL CONSUMPTION, BLOWBY, AND OIL FILM THICKNESS

First, steady state oil consumption measurements were conducted to gain information about the oil consumption pattern of the test engine with changing speed and load, and an oil consumption map (illustrated in Figure 13) was created from the measurements. The oil consumption map has several important characteristics. For instance, the measurements showed a clear dependence on engine speed and load. Oil consumption increased with increasing engine speed and load, which is typical for internal combustion engines. Moreover, the consumption rates become more significant at higher speed and load. Severe oil consumption was noticeable at high speed (> 4500 rpm) and high load (≥ 75%) conditions. However, an exception to this general trend was found at low load conditions, where the oil consumption was higher than expected. The dependency of oil consumption on load was found to disappear, as the load was decreased from 25% to 0% load. At 3500 and 4200 rpm, oil consumption was found to increase with a decrease in load from 25% to 0%.

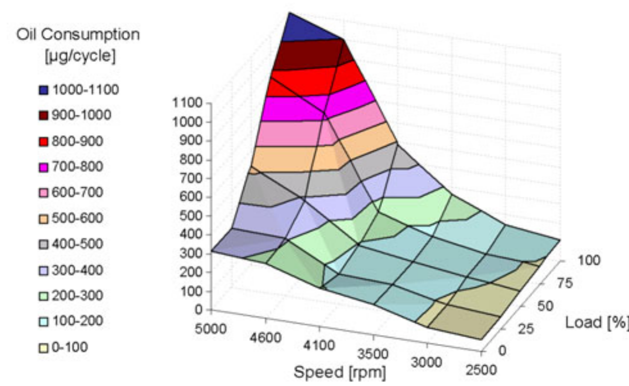

Figure 13. Oil consumption at different engine speed and load

Having determined the oil consumption pattern, the next step was to investigate the impact of engine operating conditions on blowby, and oil transport and distribution. Blowby measurements were taken at different steady state speed and load conditions. Figure 14 illustrates the variation of the average volumetric blowby flow rate (cm^3 per cycle) with engine speed and load. It is evident that blowby increased with increasing engine load at all investigated speed conditions. The increase of the cylinder pressures history with engine load causes a greater pressure gradient between the combustion chamber and crankcase, which enhances the blowby gas flow through the piston-ring-pack. On the other hand, blowby was found to decrease with increasing engine speed at constant load. This blowby characteristic can be explained as follows. The increase of engine speed reduces the available time for the gases to flow through the piston-ring-pack during one engine cycle. As a result, the amount of gases leaking through the ring-pack, and thus blowby, decreases with increasing engine speed. Consequently, the greatest increase of blowby between the 0% and 100% load conditions was observed at 2000 rpm (the lowest speed investigated). This increase of blowby with engine load was found to be less significant with increasing engine speed.

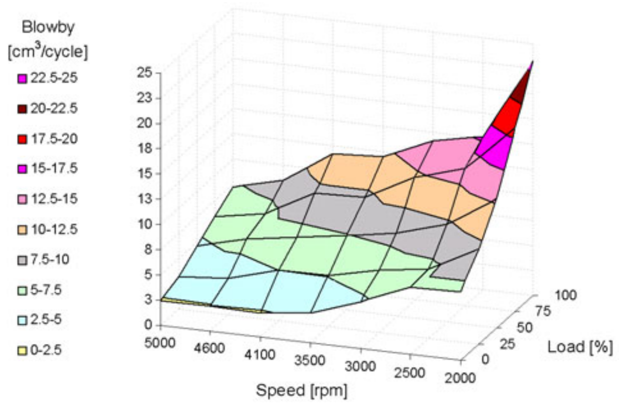

Figure 14. Blowby (cm3/cycle) dependency on engine speed and load

Oil transport on the piston is governed by, among other mechanisms, dragging of the gas flow due to interfacial shear stress. Increasing blowby with increasing speed and decreasing load should increase the oil flow rate towards the crankcase. Therefore, the variation of the blowby flow and the associated driving forces for oil transport, along with engine operating conditions, surely will also have an impact on the oil distribution on the piston. Of great interest for oil consumption is the amount of oil on the upper part of the piston. This region of the piston, in particular the region above the second ring (see Figure 2), is believed to be a major source for oil consumption. Oil can be consumed by throw off and evaporation from the top land, and by the transport of oil from the second land in the direction of the combustion chamber with the reverse blowby gas flow. The presence of more oil in these upper piston regions should increase the contribution of both oil consumption sources. Therefore, LIF measurements were taken to analyze the dependence of the oil film thickness in the upper piston regions on different speed and load conditions. The average oil film on each piston land was quantified by integrating the oil film thickness trace along each land, dividing this oil volume by the corresponding land's length for each stroke, and averaging the obtained value for the oil film on each land over all engine strokes. Figure 15 illustrates a sample LIF trace and the piston lands for which the average oil film thickness was determined.

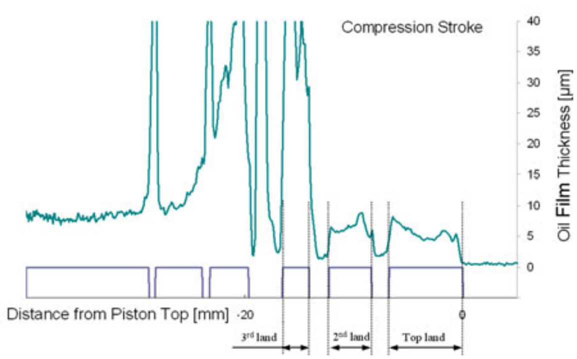

Figure 15 Sample LIF trace with control volumes of piston land for averaging

Figure 16 shows the measured average oil film thickness on the top two piston lands with engine speed and load. The shown results are from the window on the anti thrust line at mid stroke (measurement position #1 in Figure 5). It is apparent that the average oil film on the lands varies with engine conditions. The oil film decreases on both lands with increasing engine load, which should be mainly related to the variation of the blowby flow.

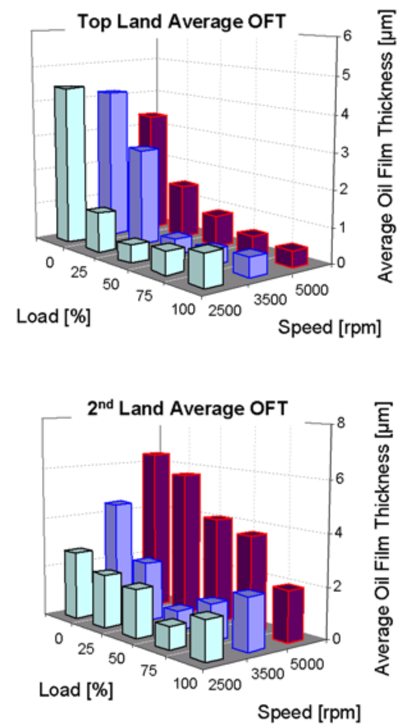

Figure 16 Effect of engine speed and load on the oil accumulation on top and 2nd land

BLOWBY OIL CONSUMPTION

Figure 17 shows measurements of the oil consumption rate due to blowby at different speed and load. A close dependence on engine conditions was found when blowby oil consumption increased with engine speed and load. It should be noted that for the measurements during 0% load at 2500 and 3500 rpm, there was no measurable difference in oil consumption when testing with and without blowby gas return to the intake system. Therefore, it was concluded that at these conditions contribution of blowby oil consumption was negligible.

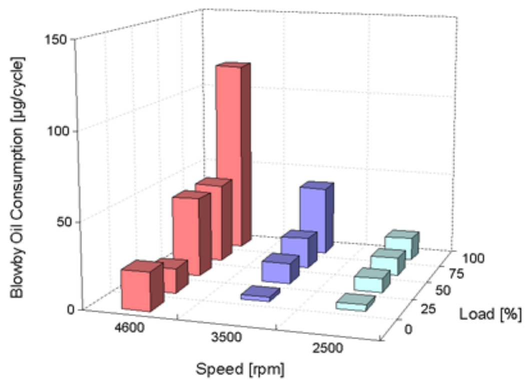

Figure 17. Oil consumption due to oil entrainment in the recycled blowby gas flow at different speed and load

As discussed above, steady state total oil consumption and blowby flow rate vary with engine speed and load. Therefore, the data in Figure 17 was used to determine the contribution blowby oil consumption to total oil consumption. The relative importance of this oil consumption source was found to increase with engine load reaching significant levels at high load conditions. For the investigated operating conditions, the maximal contribution of blowby to total oil consumption was found to be about 16 percent of the total oil consumption at full load (100%) and 3500 rpm. However, there was no consistent trend found with the increase of engine speed at constant load. Figure 18 illustrates the blowby contribution to total oil consumption at different speed and load conditions.

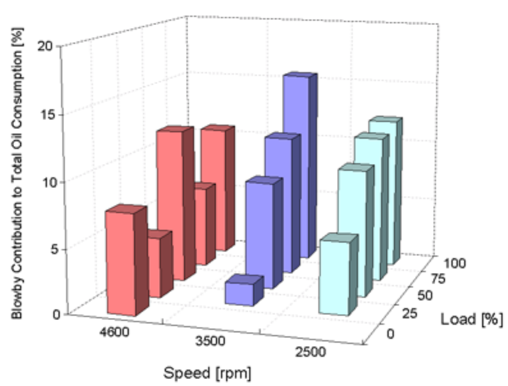

Figure 18. Contribution of blowby oil consumption to total oil consumption at different speed and load

ANALYSIS OF THE CONTRIBUTION OF OIL EVAPORATION

Oil Volatility — Oil consumption measurements were conducted for both test oils to examine the impact of oil volatility on oil consumption at different steady state speed and load conditions. For these tests the engine was operated at the standard thermal condition (81°C ± 1.5°C). As oil volatility was reduced, oil consumption decreased at most engine speed and load conditions. Figure 19 shows the measured difference of oil consumption (Δ OC) between the baseline oil and the lower volatility synthetic oil at different speed and load conditions. This difference in oil consumption represents mainly the effect of volatility, since the viscosities of the two oils are comparable. It is clear that the impact of oil volatility on oil consumption increases with engine speed and load.

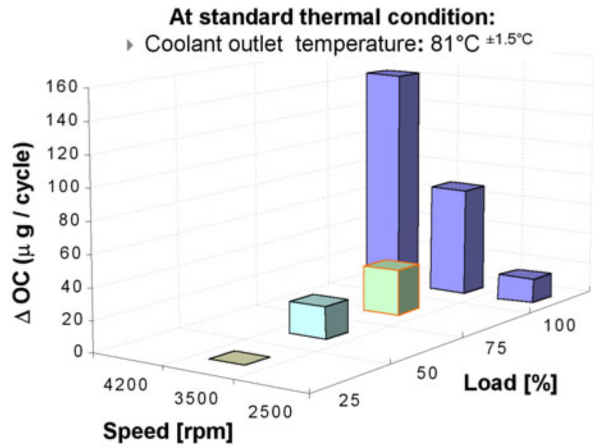

Figure 19. Difference in oil consumption due to oil volatility at different speed and load

To further demonstrate the effect of oil volatility, the same data was used to determine the improvement in oil economy due to the reduction in oil volatility – as a percent of the baseline oil consumption. Figure 20 illustrates this improvement at different speed and load conditions. Again, a close dependence on speed and load is evident. For example at 4200 rpm and full load (100%), oil consumption was improved by approximately 30 percent due to the reduction in oil volatility, implying a substantial contribution of oil evaporation to total oil consumption at high speed and load conditions.

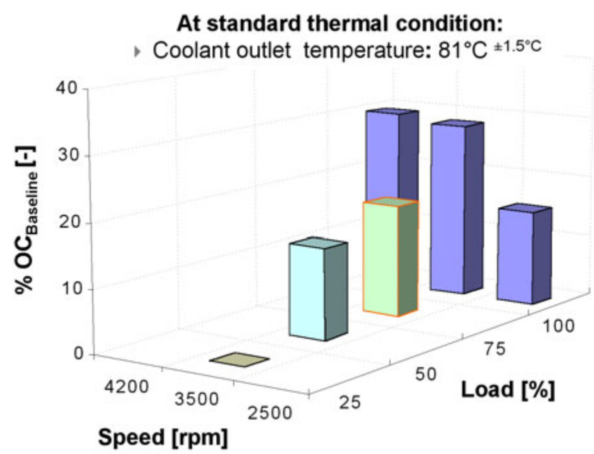

Figure 20. Decrease of oil consumption (as percent of the baseline oil consumption) due to reduction of volatility at different speed and load

Cylinder Liner Temperature — Figure 21 shows the experimentally obtained total oil consumption results along with liner oil evaporation predictions for both oils as a function of liner temperature under different load conditions at 3500 rpm. It should be noted that the axis that represents the evaporation results was shifted to compare it with the total oil consumption data.

At all engine loads and for both lubricants, a linear correlation between oil consumption and liner (coolant) temperature was observed when coolant outlet temperatures were increased from 55 to 90°C. For medium load (50 %), oil consumption increased by about 50% when the liner temperature at TDC of the scraper ring increased from 105°C to 130°C. However, there was little difference found in the oil consumption between the two lubricants at the investigated lower liner temperatures (< 110°C). As the liner temperature was increased, a difference of oil consumption between the two oils was observed. At 75% load, and 100% load the baseline oil consumption rate was higher and its dependence on liner temperature was stronger. Therefore, the difference in oil consumption between the two oils increased as liner temperatures were increased at constant engine load and speed.

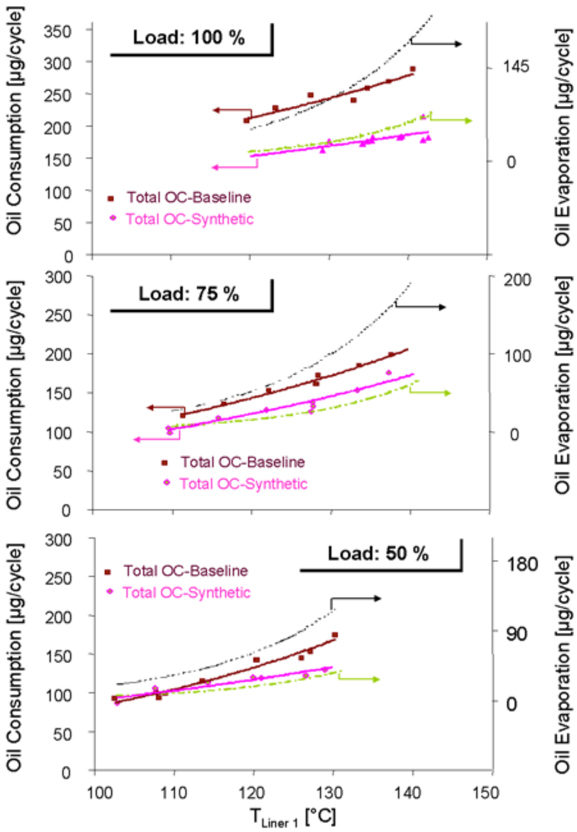

Figure 21. Total oil consumption and predicted liner evaporation dependence on liner temperature at different load, 3500 rpm

For the baseline oil, the dependencies of the measured oil consumption rate and the predicted liner evaporation rate on liner temperatures are comparable when the liner temperatures are low. Moreover, at low liner temperatures, the difference between the liner evaporation rates of the two oils is comparable to the measured difference in oil consumption for the 100% load and 75% load conditions. Under the 50% load condition when component temperatures become lower, the oil consumption rate approaches the same value for the two oils. The model, however, predicts a small difference between the liner evaporation rates of the two oils. At higher liner temperatures, the evaporation predictions for the baseline oil show a stronger dependence on liner temperature than the oil consumption measurements. This discrepancy between the model and experimental results for the baseline oil can be attributed to the assumptions of the model on oil supply.

The evaporation model assumes an infinite supply of oil species on the entire liner when the piston passes by during compression and exhaust strokes. However, in reality the oil supply to the liner is limited. In particular, the top of the liner in the region between the TDC location of the oil control ring and the TDC location of the top ring [24]. Since the oil control ring and piston skirt never travels in this region the oil supply can be either from the piston lands or along the liner due to ring-liner interaction. As discussed in [20], LIF measurements indicate that there is little oil accumulation on the piston top and second land at the investigated operating conditions. Therefore, oil flow from the piston lands to the top of the liner is unlikely, and the ring-liner interface is probably the only oil source of consequence. Therefore, the oil supply to the top of the liner may depend mainly on the lubrication conditions of the top two rings and the availability of oil before they enter this region, as described in [24]. In addition, the lubrication conditions are affected by the piston dynamic tilt, because it determines the position of the rings relative to the liner and thus the ring running surfaces. Consequently, this variation of the ring lubrication conditions causes a variation of the oil supply along the liner circumference.

On the other hand, the region above the oil control ring at TDC displays the highest evaporation rates since it experiences the highest thermal loading as it is exposed to the cylinder gases for the longest period during the cycle. For example, the evaporation model results at 3500 rpm, 100% load, and high liner temperatures show that roughly 50 percent of the oil evaporation from the liner occurs in this region. Furthermore, because the evaporation process depends on the oil species partial pressures, the greatest contributors to total evaporation are the species from the light end of the distillation curve. Figure 22 shows the predicted cumulative total evaporation and the cumulative evaporation of the four lightest oil species. The results are normalized by the total oil evaporation rate during one cycle. The lightest oil species was found to contribute around 35 percent to the total oil evaporated. The four lightest constituents, which represent the first 6 percent on the oil distillation curve (see Figure 6), contributed almost to 80 percent of the evaporated oil.

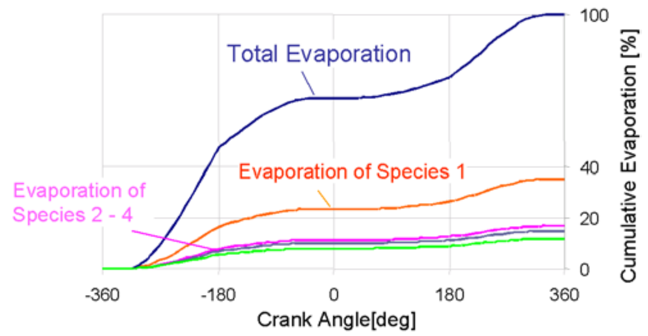

Figure 22. Cumulative total oil evaporation and proportions of the lightest four species during one engine cycle at 100% load, 3500 rpm, and 93°C coolant outlet temperature

To illustrate the impact of oil composition variation along the liner, the liner evaporation rates were recomputed for the baseline oil at 3500 rpm and 100% load. At low liner temperatures, the fresh oil composition was assumed to hold for the entire liner. At high liner temperatures, the liner was divided into two regions with different oil compositions. Below the TDC location of the oil control ring, the fresh oil composition was used. Above the TDC location of the oil control ring, however, a depleted oil composition was assumed. Figure 23 shows the light ends of both distillation curves that were used for the two liner regions. The new evaporation predictions are also shown in Figure 23 along with the total oil consumption measurements at different liner temperatures. Unlike the results in Figure 21, here the predicted rate of change of liner evaporation with liner temperature clearly follows the measured rate of change of oil consumption. This result shows that a depletion of the light ends in the region above the oil control ring at TDC may indeed change the slope of the evaporation curve at high liner temperatures. In summary, the results presented in this section strongly imply that the variation of oil consumption with liner temperature at constant speed and load conditions is due to the increase of the oil evaporation rate from the liner.

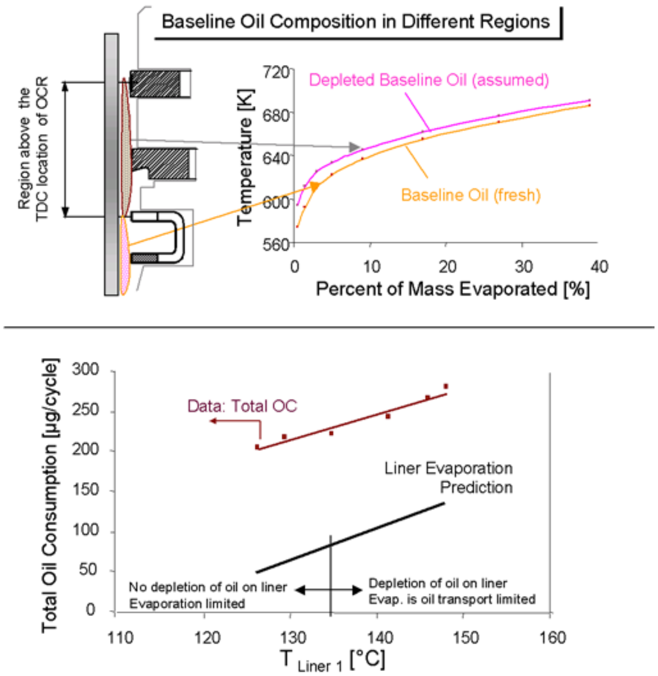

Figure 23. Effect of the liner oil composition variation (baseline oil) on the evaporation at 100% load, 3500 rpm

SEPERATION OF OIL CONSUMPTION SOURCES

As pointed out before, all oil consumption sources that are illustrated in Figure 1 contribute to total oil consumption. The relative importance of these sources at different operating conditions depends on the variation of the driving forces for each source. Steady state total engine oil consumption measurements were used in conjunction with the results of the studies on the blowby contribution and oil evaporation to separate and assess three different oil consumption sources. The contributions of oil evaporation, blowby oil consumption, and oil transport into the cylinder through the piston system and valve guide were quantified. The variation of these oil consumption sources under different steady state engine speed and load conditions was analyzed. This was done for the baseline oil at the standard thermal operating condition (~81°C coolant outlet temperature). Figure 24 illustrates a schematic of the separated oil consumption sources. In the following a description is provided as to how the contributions of the three major oil consumption sources were separated and quantified at different engine operating conditions.

For each investigated engine operating condition, it was assumed that the sum of all three oil consumption sources would result in the experimentally obtained total oil consumption. As discussed above, total oil consumption measurements were conducted at a wide range of engine speed and load conditions. The results shown in Figure 17 and Figure 18 were used for the contribution of blowby to total oil consumption. The procedure used to determine the contributions of oil evaporation and of oil transport into the cylinder for each operating condition follows.

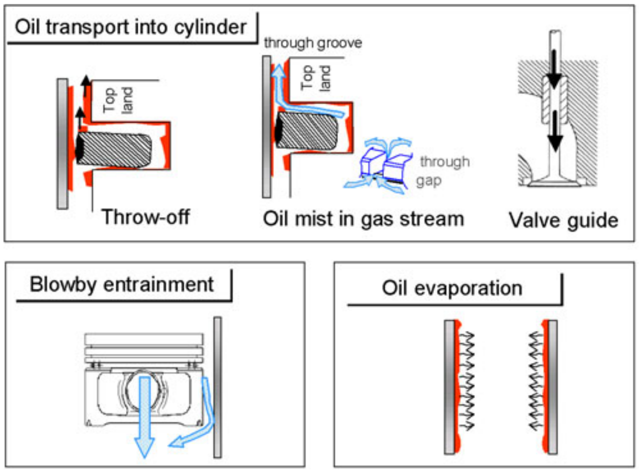

Figure 24. Illustration of the separated oil consumption sources

<u>Contribution of oil evaporation</u> — The contribution of oil evaporation to total oil consumption for the baseline oil was determined by combining oil consumption measurements and predictions of the in [14] proposed liner evaporation model. As discussed previously, the evaporation model overestimated the evaporation of the baseline oil. However, good agreement was obtained between the evaporation predictions and the oil consumption measurements at the different engine operating conditions that were investigated for the low volatility synthetic oil. Consequently, it was assumed that the liner evaporation model holds for the synthetic oil. The evaporation rate for the baseline oil was estimated by adding the measured difference in oil consumption (Δ OC) between the baseline and the lower volatility synthetic oil to the evaporation predictions for the synthetic oil. This difference in oil consumption (Δ OC) accounts for the additional evaporation of the baseline oil due to its higher volatility than that of the synthetic oil. This approach is possible because the oil consumption due to oil transport is assumed to not vary between the two oils since their viscosities are comparable (see Figure 7). The difference in oil consumption (Δ OC) is mainly due to the difference in liner evaporation between the two oils. There may also be small differences in evaporation from other regions, such as within the piston and crankcase. However, as described above and in [20], the latter sources for oil evaporation seem to have little impact on the difference in oil consumption between the two oils. Therefore, at a given steady state engine operating condition, the total evaporation rate of the baseline oil was determined by the sum of the experimentally obtained difference in evaporation between the baseline and synthetic oil (Δ OC) and the estimated evaporation for the synthetic oil.

CONTRIBUTION OF OIL TRANSPORT — The amount of oil transported directly into the combustion chamber through the piston ring-pack and the valve guides was deduced by subtracting the contributions of oil entrained in the blowby flow and oil evaporation for the baseline oil determined by the above described procedure from the measured total engine oil consumption rate. A schematic of the described separation of the different oil consumption sources is shown in Figure 25.

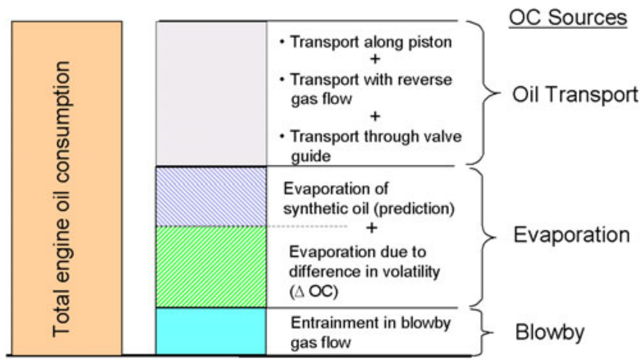

Figure 25. Schematic illustrating the separation of the three oil consumption sources

EFFECT OF ENGINE LOAD ON SEPERATED OIL CONSUMPTION SOURCES

Figure 26 shows the variation of the three oil consumption sources for the baseline oil with different steady state engine load conditions at 3500 rpm as determined by the procedure that was described above. In the figure, the oil consumption rate of each source during one engine cycle along with the total oil consumption rate is shown.

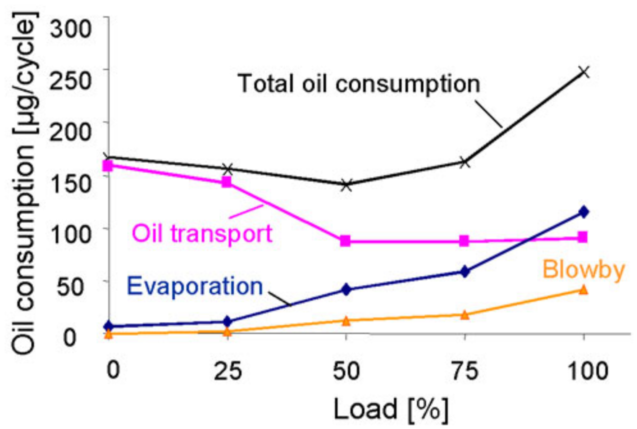

Figure 26. Effect of engine load on different oil consumption sources and on total oil consumption at 3500 rpm for the baseline oil

As pointed out above, total engine oil consumption initially decreases as engine load is increased from 0% load to 50%. Oil consumption then increases to a maximum as the engine load is further increased. This characteristic of oil consumption to exhibit two operating regions with distinct trends is a result of the variation of the three oil consumption sources, which show significantly different characteristics at different loads. The impact of the variation of engine load on the separated oil consumption sources is described below.

Load Effect on Oil Transport — The rate of oil transport through the piston-ring-pack into the cylinder is highest at 0% load. This consumption source decreases progressively as engine load is increased to 50% load. However, there is no noticeable variation of this source as load is further increased from 50% to 100% load.

Engine load affects cylinder pressures during the intake stroke and the peak cylinder pressures, which govern the gas flow structure through in the piston-ring-pack and blowby. Decreasing engine load generally decreases the blowby flow, and therefore the oil removal from the piston to the crankcase due to interfacial shear stresses also decreases. This results in more oil accumulation at various regions in the piston-ring-pack. The dependence of blowby and of oil accumulation on engine load in the top two piston lands was demonstrated above. The presence of more oil on the second land with the decrease of engine loads may result in an increase of the oil transport along the piston into the cylinder via different paths. More oil may flow to the top ring groove and from there to the top land. The oil could also be transported through the top ring gap with the reverse gas flow from the second land, when the second land pressure becomes greater than the cylinder pressure. The variation of engine load has a considerable influence on the latter transport mechanism, because it determines both cylinder and land pressures.

Physics-based models, developed at MIT [25], were used to predict the volumetric reverse gas flow rates from the second land through the top ring gap for different load conditions at 3500 rpm. Figure 27 shows the results for the gas flows through the top ring gap during one engine cycle. The positive flow direction is defined as the flow from the top land to the second land and thus negative gas flow rates indicate phases of the engine cycle when gases flow from the second land to the top land. There exist two periods during the engine cycle when gases may flow from the second land to the top land due to negative pressure gradients across the top ring. The cylinder pressure may become smaller than the second land pressure during the intake stroke and during the late part of expansion and early exhaust strokes. Decreasing engine load causes an increase of the reverse gas flow through the top ring gap during the intake stroke due to the lower cylinder pressures. In addition, decreasing engine load also increases the crank angle period during which gases flow into the cylinder due to the pressure difference across the top ring. The reverse flow during late expansion and early exhaust stroke is less prominent and the impact of engine load is small. Therefore, it can be concluded that the overall reverse gas flow rate through the top ring gap increases as engine load is decreased. The combination of increased oil accumulation on the second land, combined with the increase of reverse gas flows through the top ring gap with decreasing engine load, could result in greater oil flow rates into the cylinder due to interfacial shear stresses. The higher oil flow through the top ring gap could explain the increase of oil transport as engine load is decreased.

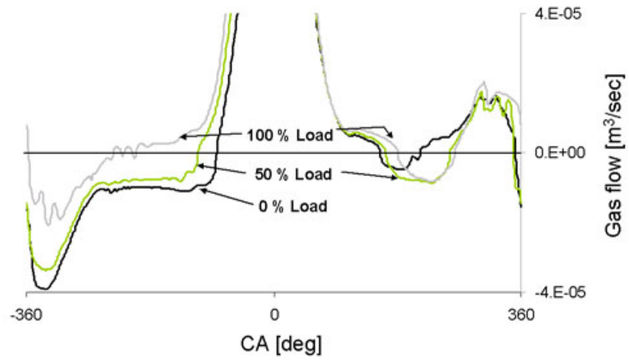

Figure 27. Reverse gas flow through top ring gap at different engine load

A one-dimensional model for gas-flow driven oil flow through ring gaps as described in [23] was used to estimate this oil flow into the cylinder. This oil flow through the top ring gap was estimated for different engine loads at 3500 rpm using the computed average reverse gas flow rates through the top ring gap, the estimated average clearance between land and liner, and the steady state LIF oil film thickness measurements. 2-D LIF observations of the top ring gap area during low load show that the oil accumulation adjacent to the gap is thicker than the average oil film thickness results [23]. This variation of the oil film distribution can affect the oil dragged through the top ring gap by the reverse gas flow. To illustrate the effect of the increased oil film thickness near the gap, the oil flow through the top ring gap was calculated using estimated elevated average oil film thickness (the measured average oil film thickness was multiplied by a factor of two). Figure 28 shows the results for the oil flow rate through the top ring gap for the two different average oil film thickness values that were used along with the obtained values for the oil transport into the cylinder (from Figure 26). The results show that the highest oil flow rates through the top ring gap occur at 0% load. This oil flow decreases as engine load is increased to 50% load. As engine load is increased further, there is little change in the oil flow through the ring gap to the cylinder, because at higher loads little oil is present on the second land to be transported. It is also for this reason that the two average oil film thicknesses used for the calculation only show a difference in the reverse oil flow through the gap at low load conditions. It is clear that the computed oil flow rate through the top ring gap (using $\hat{h}_{Oil} = 2 * \tilde{h}_{Oil}$) exhibits a similar response to increasing engine load as the total oil transport rate into the cylinder. As engine load increases, both the obtained oil transport and the calculated oil flow decrease by the same amount. Therefore, it is plausible that the increase of oil transport as engine load is decreased form 50 % to 0% load is due to the reverse oil flow through the top ring gap.

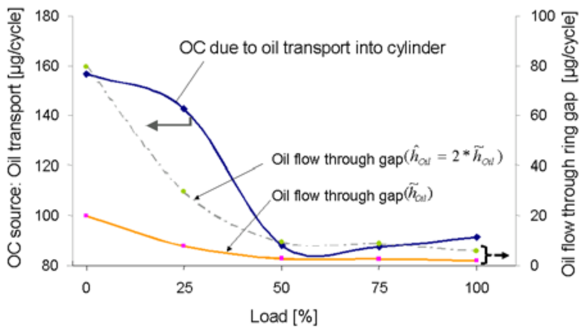

Figure 28. Computed oil flow with reverse gas flow through top ring gap and separated oil transport rate at different engine load

Load Effect on Oil Evaporation and Blowby — As discussed earlier, oil evaporation from the liner is the most significant contributor to total evaporation during engine operation. The liner oil evaporation rate into the cylinder is smallest at 0% load and increases continuously with increasing engine load to reach its maximum at 100% load. This increase of oil evaporation with engine load is attributed to the increase of the engine thermal loading and thus to increased liner temperatures.

As shown above, the blowby oil consumption rate increased continuously with increasing engine load from negligible contributions at 0% load to considerable consumption rates at full load.

The increases of both oil evaporation and blowby oil consumption compete with the decrease in the oil transport rate into the cylinder as engine load is increased from 0% load to 50%. Consequently, all three oil consumption sources influence the oil consumption pattern in the operating region between 0% to 50% loads. The decrease in oil transport into the cylinder has a higher impact on oil consumption than the increase of oil evaporation and blowby contribution, as engine load is increased 0 to 50%. As load is further increased to 100%, the oil consumption rate is governed only by the oil evaporation and blowby contributions.

Load Effect on the Relative Importance of Oil Consumption Sources — Figure 29 quantifies the relative contribution of each of the three separate oil consumption sources to total engine oil consumption for different load conditions at 3500 rpm. The values displayed are the ratio of the oil consumption rate due to one source, normalized by the total engine oil consumption at each operating condition. At 0% load, oil transport into the cylinder contributes substantially to total oil consumption (~90 percent). As engine load increases, the contributions of oil evaporation and blowby oil consumption rise continuously, while the contribution of oil transport decreases. At 100% load, all three sources contribute to total oil consumption. While the relative importance of oil transport decreases to 40 percent, oil evaporation was found to increase to about 45 percent, and roughly 15 percent of oil consumption is due the oil entrainment in the blowby gas flow.

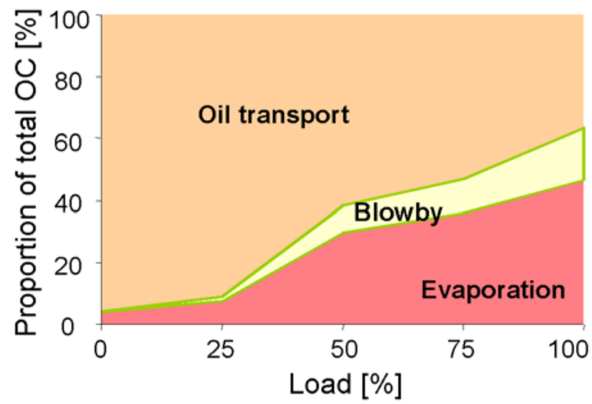

Figure 29. Effect of engine load on the relative importance of different oil consumption sources to total oil consumption at 3500 rpm for the baseline oil

VARIATION OF THE OIL CONSUMPTION SOURCES WITH ENGINE SPEED

The influence of engine speed on the different oil consumption sources was studied at 3 different speeds and 100% load. Figure 30 shows the oil consumption rates of each source during one engine cycle along with total engine oil consumption. Changing engine speed affects all three oil consumption sources. Blowby oil consumption, oil evaporation, and oil transport all increase as engine speed is increased from 2500 to 4200 rpm. Consequently, all three oil consumption sources contribute to the increase of total oil consumption with increasing engine speed.

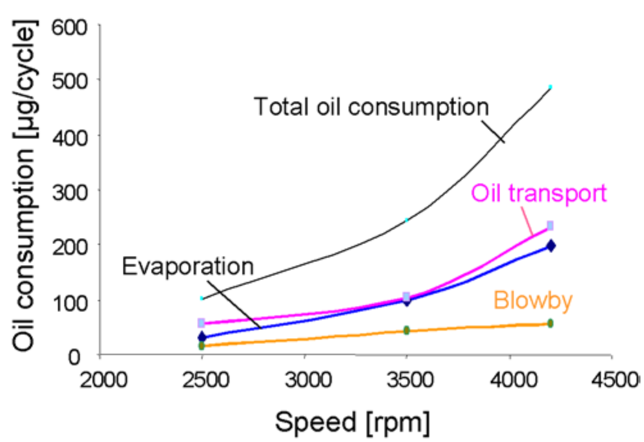

Figure 30. Effect of engine speed on different oil consumption sources and on total oil consumption at 100% load for the baseline oil

The variation of engine speed affects three different mechanisms that control oil transport and distribution in the piston-ring-liner system and therefore the oil consumption sources:

- Increasing engine speed results in a decrease in the average blowby gas flow rate per cycle, and thus a decrease in interfacial shear stresses that drive the oil layer on the piston surface. Therefore, the removal of oil from the piston to the crankcase decreases with increasing speed.

- Increasing engine speed causes higher inertial forces acting on the oil volume that is accumulated on the piston due to the alternating motion of the piston. The increase of inertia is expected to increase oil flow in the axial direction to the upper piston regions. However, at high load conditions (as is the case here), inertial oil transport is believed to be important only in regions on the piston with significant oil accumulation, such as on the third land. This region is in particularly important because the oil has only a short distance to travel to reach the second ring groove.

- Increasing engine speed also increases the thermal loading on the engine. Higher piston temperatures reduce the oil viscosity, and therefore may change the oil transport along the piston. Moreover, higher liner temperatures will increase oil evaporation.

With the increase of engine speed, more oil accumulation is expected in the upper piston regions due to the decrease of blowby and the increase of inertial oil transport on the piston. Therefore, the increase of oil consumption due to oil transport with engine speed could be attributed to higher oil accumulation on the upper piston regions. It is believed that increased oil accumulation in the region between the second land and top ring groove may result in more oil flowing through the top ring groove and gap into the cylinder.

As speed is increased, more oil must be entrained in the blowby because the average blowby gas flow per cycle decreases, but blowby oil consumption increases.

Oil evaporation is mainly governed by engine liner temperature, mass transport in the gas phase, and the available time for evaporation. Increases in engine speed increase evaporation due to the enhancement of the mass transport in the gas phase and due to the increased liner temperatures. The available time for evaporation decreases per cycle as engine speed increases, which competes with the two above effects. Since the contribution of oil evaporation increases with engine speed, the effect of higher liner temperatures and increasing mass convection coefficients must have a higher impact than the lower time available for evaporation.

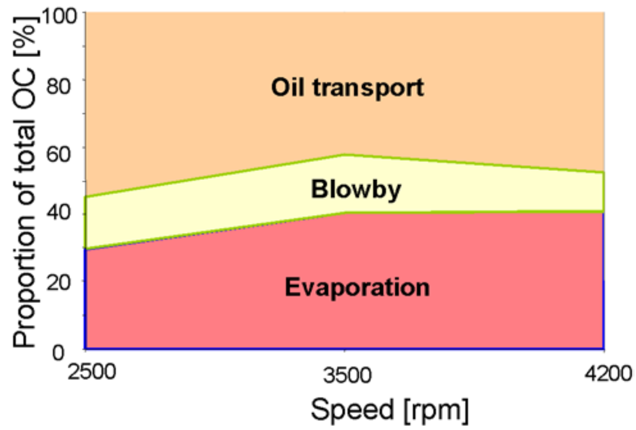

Figure 31. Effect of engine speed on the relative importance of different oil consumption sources to total oil consumption at 100% load for the baseline oil

Figure 31 quantifies the relative contribution of each of the separated oil consumption sources to total engine oil consumption for different speeds at 100% load. The values displayed are the oil consumption rates due to each source, normalized by the total engine oil consumption at each operating condition. The importance of each oil consumption source does not change significantly as engine speed is increased from 2500 rpm to 4200 rpm. The major contribution in this range is from the oil transport, which varies between 40 to 50 percent. Oil evaporation, also a significant contributor to oil consumption at high load, exhibits a consistent trend as its relative importance increases gradually from 30 percent at 2500 rpm to 40 percent at 4200 rpm. The blowby contribution varies between 10 to 15 percent as the engine speed is increased.

CONCLUSIONS

This study has systematically analyzed for the first time three major oil consumption sources and their dependence on steady state speed and load conditions in a production spark ignition engine. Engine test data and modeling results were combined to separate and quantify the contribution of oil evaporation, oil entrained in the blowby gas flow, and oil flow through piston and valve guide into the combustion chamber.

Total engine oil consumption, blowby, and in-cylinder parameters affecting oil transport were measured at different engine operating conditions. Steady state oil consumption and blowby measurements exhibited typical trends for spark ignition engines. Oil consumption increased with increasing engine speed, and load and blowby (per cycle) increased with engine load and decreased with speed. The contribution of oil entrained in the blowby gas flow to total oil consumption was quantified at several engine load and speed conditions. Blowby oil consumption was found to increase with engine load. An insignificant amount of oil was consumed due to blowby at low load but this contribution reached significant levels at high load conditions.

Oil evaporation from the liner was analyzed and characterized. A previously developed multi-species liner evaporation model [14] was verified by testing two oils with different volatility at varying cylinder liner temperatures and steady state engine speed and load. The results show that liner evaporation substantially affects total engine oil consumption and imply that liner evaporation is the main contributor to total evaporation. Liner evaporation, however, was found to be sensitive to liner temperatures, oil composition, and oil transport to regions of high evaporation. The contribution of oil evaporation to total oil consumption for the baseline oil was determined by combining oil consumption measurements and liner evaporation predictions. The amount of oil transported directly into the combustion chamber through the piston ring-pack and the valve guides was deduced by subtracting the contributions of oil entrained in the blowby flow and oil evaporation for the baseline oil from the measured total engine oil consumption rate. The importance of the three quantified oil consumption sources to total engine oil consumption was analyzed for different speed and conditions. The impact of engine load on oil consumption and its sources show several important characteristics. The consumption due to the oil transport through the piston into the cylinder evinced two trends. At 0 to 50% load, oil transport was found to increase with reduction in engine load that may be related to the decrease of blowby flow and the associated increase of oil accumulation on the upper piston regions. Above 50% load, the oil consumption due to oil transport through the piston and valve guide was found to vary little with engine load. The oil evaporation rate and blowby oil consumption rate increase with increasing engine load. Also the relative importance of these oil consumption sources varied with engine load. At low load, oil transport through the piston and valve guides was found to be the major consumption source (90 percent), while the contribution of oil evaporation and blowby oil consumption became more significant with increasing engine load. The increase of engine speed (from 2500 to 4200 rpm) at full load (100%) increased the contribution of all three oil consumption sources. The relative importance of each source to total oil consumption varied little in this engine speed range. The major contributions were from the oil transport (40 to 50 percent) and from oil evaporation (30 to 40 percent). The importance of blowby oil consumption varied between 10 to 15 percent.

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