

Experimental and Numerical Investigations on the Mechanisms Leading to the Accumulation of Particulate Matter in Lubricant Oil

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ABSTRACT

The accumulation of particulate matter in lubricant oil can become an important issue in Diesel engines where large amounts of Exhaust Gas Recirculation (EGR) are used at medium to high load operating conditions. Indeed, the transport and subsequent accumulation of particulate matter in the engine oil can negatively impact the oil lubricant properties which is critical to ensure mechanical durability and limit the vehicle Total Cost of Ownership (TCO) by reducing the servicing intervals. The objective of this investigation was to gain an improved understanding of the underlying mechanisms that are responsible for the accumulation of particulate matter in the lubricating oil, and ultimately provide design guidelines to help limit this phenomenon.

The present study presents the development and validation of experimental and numerical tools used to investigate this phenomenon. Several advanced diagnostic techniques were developed and applied on an optically-accessible single cylinder Diesel engine to detect the presence of particulates and quantify their concentration in two particular zones: (1) in the upper part of the cylinder liner where particulate matter is believed to be absorbed into the oil film and (2) in the engine blow-by gases where particulates can be transported via the piston ring-pack to the oil sump reservoir. The accumulation of soot particulates on the surface of the upper part of the cylinder liner was characterized using a modified Laser Extinction Method (LEM), while the concentration of particulates in the engine blow-by gases was measured using a DMS500 soot sensor. In parallel to the experimental study, 3D numerical computations were performed and provided additional information to the experimental results. In the present paper, the various experimental and numerical methods are presented and first validation results are discussed.

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INTRODUCTION

Prior investigations performed on various heavy duty Diesel engines show that combustion strategies where retarded injection timing and high EGR ratios are employed with the aim of achieving vehicle exhaust emissions compliance can result in contamination of the lubricant oil by soot particles, up to levels of about 10% in weight [1]. This accumulation is particularly critical on engine operating points where high smoke levels are observed such as at mid to high loads (IMEP>12bar), where high EGR rates are employed or during regeneration strategies for exhaust gas catalysts.

The increased presence of soot in the crankcase oil is believed to aggravate engine wear [1]. This affects mainly the engine valve train, piston rings and cylinder liner wear [2] and imposes more frequent servicing intervals in order to prevent a reduction in engine

performance or, in extreme cases, engine failure. Accumulation of particulate matter in the lubricant oil therefore represents an interest for optimizing both engine reliability and Total Cost of Ownership (TCO). Even if an increase in engine wear which according to [2] is dependent on oil soot content and soot particle size, can potentially be limited by using specific oil formulations (base oil and additives) [2, 3], there is still a clear need to better understand the physical mechanisms explaining the accumulation of particulate matter in lubricant oil.

Different pathways can explain the accumulation of soot in oil. They can either be transported from the combustion chamber to the crankcase within the blow-by gases, or they can be deposited directly on the lubricant oil film on the liner and then subsequently scraped by the piston rings.

Soot deposition on the liner can be explained by several mechanisms, described in [4]: thermophoresis, brownian diffusion, turbulent diffusion, inertial impingement, electrophoresis or gravitational sedimentation, but the principal mechanism seems to be thermophoresis [5]. One previous study claims that the deposit mechanism accounts for more than 97% of the soot present in engine oil [9], but the relative weight of each of these mechanisms is still not well known.

Regarding potential solutions to this issue, a recent study shows a combustion chamber shape evolution aimed at improving combustion efficiency while reducing levels of in-cylinder soot and in particular the soot-liner interaction [8].

Based on the literature review, the two main mechanisms explaining the presence of soot in oil are the following:

- Soot brought into contact with the cylinder liner is absorbed by the oil film and subsequently scraped by the piston rings.
- The soot located in the vicinity of the piston top land zone is entrained directly by the blow-by gases into the engine crank case and subsequently absorbed in the oil sump.

The aim of this study was to gain a better understanding of the processes controlling soot accumulation in the lubricant oil, in order to subsequently identify new combustion chamber designs and/or combustion strategies which could help limit this phenomenon. This study focused first and foremost on establishing whether the two mechanisms were indeed responsible for the accumulation of particulate matter in the lubricant oil. Secondly, the influence of various engine parameters was also investigated such as Start of Injection (SOI) timing, EGR rate and engine load (IMEP). This requires the ability to measure or estimate:

- The quantity of soot present in the combustion chamber;
- The quantity of soot present in the blow-by gases;
- The quantity of soot deposited on the cylinder liner.

This paper presents experimental and numerical methodologies that were developed specifically for this study and preliminary results. A particular focus is given to the validation of the developed methodologies which enabled measurements or numerical predictions of these 3 quantities, together with their temporal evolution either during one engine cycle and/or during several consecutive engine cycles. The retained approach consisted in combining experimental (optical engine) and numerical (3D CFD) tools. Experimental results were used both for an in-depth analysis of the different physical phenomena and for the construction of a database which served to validate the 3D CFD simulations. Once validated, numerical simulations were performed and provided access to parameters that were not accessible by experimental measurements. Finally 3D CFD simulations were also used to perform variations and optimization of combustion chamber design

The advanced diagnostics are detailed in the following experimental setup section. The numerical tool is based on 3D CFD RANS computations. During this study specific data post-processings were

developed, they are exposed in the “Numerical setup” section along with the numerical models and the main assumptions used. Finally, a first set of results is presented, with a first attempt to propose a mechanism explaining the accumulation of particulate matter in the lubricant oil.

This study was performed within the framework of the Groupement Scientifique Moteur (GSM), a collaborative R&D program between IFPEN, and the French car manufacturers Peugeot-Citroën (PSA group) and Renault (RSA).

EXPERIMENTAL SETUP

Engine

An optical-access single cylinder engine was used in this study with a 0,5 liter displacement and is representative of a light duty direct-injection (DI) 4-stroke Diesel engine. A schematic view of this engine is presented in [Figure 2](#), and the general engine characteristics are listed in [Table 1](#).

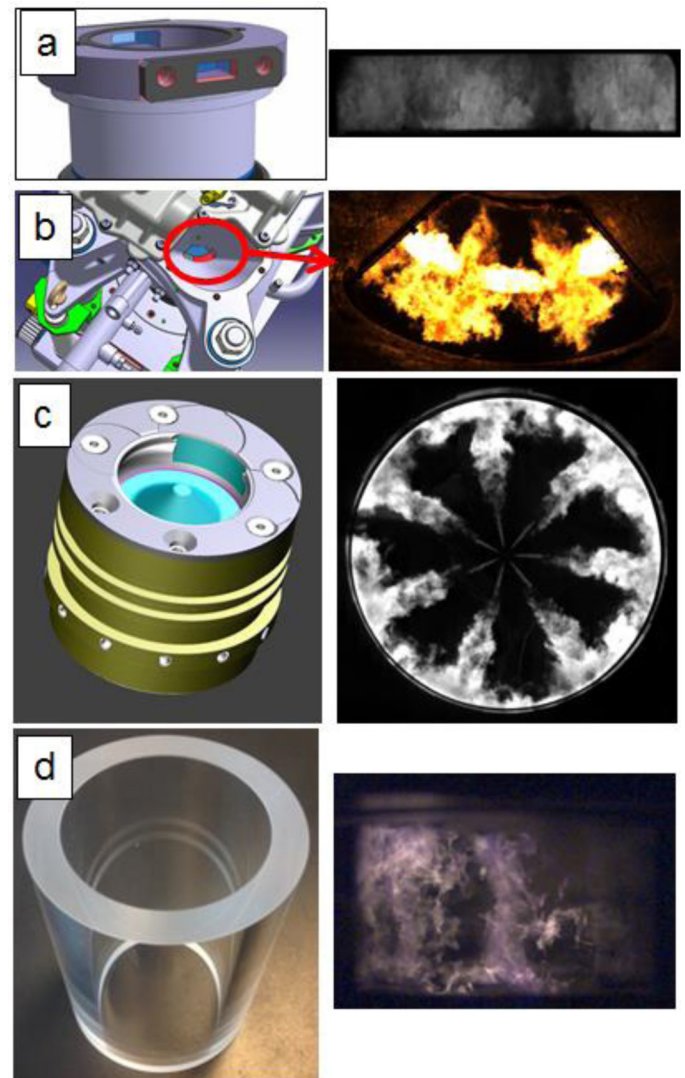


Figure 1. Left column: schematic view of the different optical accesses on the engine. Right column: an example of the corresponding experimental visualization of the combustion. Views from (a) the optical crown, (b) the exhaust valve port, (c) the optical piston, and (d) the optical liner.

This engine can be equipped in several optical configurations to enable the application of laser based diagnostics for in-situ measurements:

- Fully/partially optical crown: this corresponds to the upper part of the liner, in which transparent windows can be installed. This optical crown can also be fully optical made of sapphire in order to have a full optical access to the upper part of the combustion chamber.
- Replacement of one of the two exhaust valves by a window providing a view from the cylinder head down into the combustion chamber;
- Optical piston: an optical bowl window is used;
- Optical liner: the entire liner is transparent.

There are several differences between optical and all-metal engines that have been studied in [10]. Given the context of the present study, probably the most important one is the position of the piston ring pack. In the optical engine, the rings are positioned about 17mm below the upper surface of the piston, whereas this distance is typically between 8 and 5mm in a production series engine. As soot potentially located in the piston top land area and the transport of soot in the blow-by gases are the topic of interest of this study, it is important to keep this difference in mind since it can potentially have an effect on the observed phenomena and especially on the interactions between the flame or the burnt gases and the ring pack. Also, the optical engine is not lubricated which means that there is no lubricant oil on the wall of the combustion chamber. This can modify the interaction between the soot and the wall in terms of the propensity of soot present in the bulk gases coming into contact with the liner and being subsequently deposited on the combustion chamber walls.

The engine was equipped with an intake heater allowing pre-heating of the fresh intake air, and with a simulated EGR (nitrogen) circuit allowing to reproduce the oxygen concentration of the intake air of real engines for varying levels of charge dilution.

An optical ring (upper part of the liner) enabled combustion visualization in the upper part of the combustion chamber as presented in [Figure 3 \(left\)](#). Two fused silica windows inserted into a metal ring provided the necessary optical access coupled with excellent mechanical resistance to enable part load engine operation up to 12 bar IMEP and for in-cylinder pressures up to 130 bar. The engine operating point investigated in the present study is detailed in [Table 1](#).

For this specific investigation, the extended piston assembly was modified in such a way as to ensure an airtight seal to isolate the volume comprised between the upper section (optical piston assembly) and the lower piston (lubricated). Hereafter this volume will be referred to as the dry crankcase volume. This modification was implemented in order to enable measurements of soot concentration in the blow-by gases as well perform measurements of the optical engine blow-by mass flow rate. This cover is a metal cylinder shown in [Figure 3 \(right\)](#). Four ports allowed measurements on the gases located in this volume.

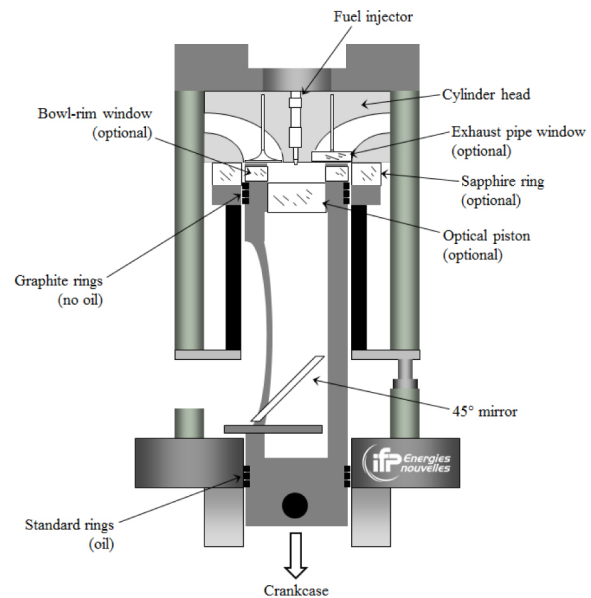


Figure 2. Schematic diagram of the IFPEN Diesel optical engine.,

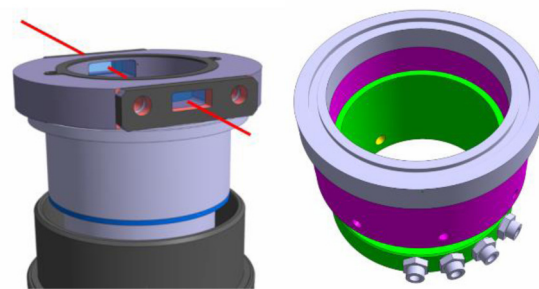


Figure 3. Left: Design of the specific optical crown providing optical access in the upper part of the combustion chamber. - Right: Airtight cover to isolate the volume located between the upper (graphite) and the lower (lubricated) rings.

Table 1. Characteristics of the IFPEN single-cylinder Diesel optical engine.

Characteristics	Value
Number of cylinders	1
Cycle	4-Stroke
Number of intake valves	2
Number of exhaust valves	1
Combustion chamber	Flat optical bowl
Cylinder head type	Flat
Displacement	500cc
Bore	85 mm
Stroke	88 mm
Connecting Rod	145 mm
Geometrical compression ratio	15.7:1
Swirl at Bottom Dead Center	1.1
Number of Valves	4
Exhaust Valve Open	34° BBDC @ 0.15 mm lift
Exhaust Valve Close	6° BTDC @ 0.15 mm lift
Inlet Valve Open	2° BTDC @ 0.15 mm lift

Fuel Injection System

The engine was equipped with a common rail fuel injection system (Bosch CP2) capable of supplying fuel at nominal rail pressures up to 1600 bar. A six-hole Bosch injector nozzle with a 145° included angle was used, mounted on a Bosch CRI2 solenoid injector body. The characteristics of the injector are listed in [Table 2](#). The fuel used in the present study was standard European Diesel fuel whose principal physical properties are presented in [Table 3](#).

Table 2. Characteristics of the Bosch CRI 2 injector

Characteristics	unit	value
# holes	-	6
Included angle	°	145
Mass flow rate / 30s (@ 100bar discharge pressure)	cm ³	395
Nozzle Tip Protrusion	mm	3.35
Orifice diameter	µm	147

Table 3. Properties of the Diesel used in this study.

Density @ 20°C	kg/m ³	831
Viscosity @ 40°C	mm ² /s	2.5
Cetane number (ISO 5165-98)	-	52.1
%C - %H - %O	%mass	86.1-13.3-0.6
Lower Heating Value (ASTM D 4868)	MJ/kg	42.94

Engine Operating Condition

Table 4. Characteristics of the selected reference operating point.

Characteristics	Unit	Value
Speed	Rpm	1200
IMEP	Bar	10
Intake manifold pressure	Bar	1.6
Equivalence ratio	-	0.82
Injection pressure	Bar	600
Pilot injection duration (DSE ¹)	µs	275
Pilot injection phasing (SSE ²)	CAD	348
Pilot injection mass	Mg	1.8
Main injection duration (DSE ¹)	µs	1800
Main injection phasing (SSE ²)	CAD	358
Main injection mass	Mg	39.4
Simulated EGR rate (N ₂)	%mass	15
Equivalent real EGR rate	%mass	18

One of the challenges of this study was to reach sufficiently high engine load conditions whilst being able to perform in-situ, optical measurements. It was necessary that the operating point targeted for

this study be representative of a low engine speed-part load operating condition (high torque condition), where the phenomena of transport of soot-in-oil are believed to be the most problematic. This is the case of mid-high load engine loads, where high EGR rates are employed. However, because of the absence of lubricating oil in optical engines (graphite rings are used rather than a standard piston ring pack and lubricating oil, see [Figure 2](#)), optical engines are limited in terms of maximum load and engine speed. For the IFPEN optical engine, the maximum attainable engine speed is 2000rpm, and the maximum load is approximately 12-14bars. These limitations are dependent on the degree of optical access required. A summary of the characteristics of the selected operating point for this study is presented in [Table 4](#). The corresponding in-cylinder pressure trace and Apparent Heat Release Rate (AHRR) curves are shown in [Figure 4](#).

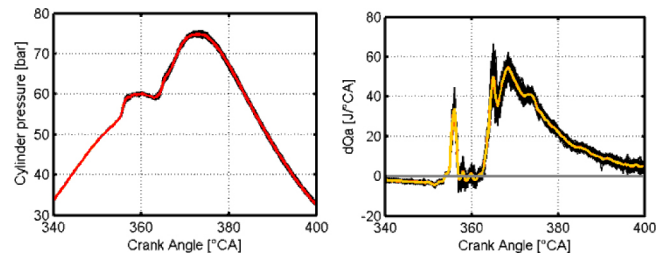


Figure 4. In-cylinder pressure (left) and Apparent Heat Release Rate (AHRR, right) corresponding to the reference operating point described in [Table 4](#).

Measurement Facilities

This section describes all the measurement procedures applied in this study, and the way these procedures have been validated.

Blow-By Mass Flow Rate Measurements

One of the four ports of the airtight housing (see [Figure 3](#)) on the extended piston assembly was used to measure engine blow-by mass flow rate. This port was connected to a calibrated sonic nozzle. The upstream and downstream pressures on both sides of the nozzle were measured. Measurement of the pressure drop and a prior calibration of the nozzle allowed determination of the mass flow rate.

The mass flow rate measurements were first calibrated in a steady-state flow condition, with the engine stationary. The crankshaft was positioned such that the intake valves were open and the exhaust valves were closed. A constant mass flow rate was imposed through the intake valves, using a sonic flow nozzle and ensuring precise control of the upstream pressure of this sonic nozzle. The piston was firstly mounted without any piston rings, so that the intake flow went through the piston ring crevices into the dry crankcase. The imposed mass flow rate was 1.15kg/h and the measured flowrate was 1.09kg/h. This yields a 5% measurement error for the steady-state flow. Subsequently, measurements were then performed with the engine in operation and the piston equipped with the full set of piston rings. The blow-by mass flow rate was measured once per engine cycle for the reference operating point described in [Table 4](#). The blow-by mass flow rate measurements oscillated between a minimum of 0.6 and a maximum of 2kg/h, with an average value of 1.1kg/h, which represents 3.6% of the trapped air. Even if this mass flow rate is high,

1. DSE is the Duration of Solenoid Energizing, i.e. the duration of the electrical command of the injector.

2. SSE is the Start of the Solenoid Energizing, i.e. the start of the electrical command of the injector.

it is of the same order of magnitude as the blow-by mass flow rate observed (about 0.7kg/h) for similar operating points on (standard (*i.e.* non optical) engines. Lastly, in order to further validate the relevance of the blow-by measurements on our optical engine, a correlation between the maximum in-cylinder pressure and the blow-by mass flow rate was analyzed and is shown in Figure 5. The results show that these two values were, as expected, well correlated. This means that the observed decrease in maximum in-cylinder pressure can be attributed to a decrease in the trapped mass in the cylinder.

Based on these 3 different sets of validations, it was concluded that the selected methodology for the measurement of the blow-by mass flow rate was relevant.

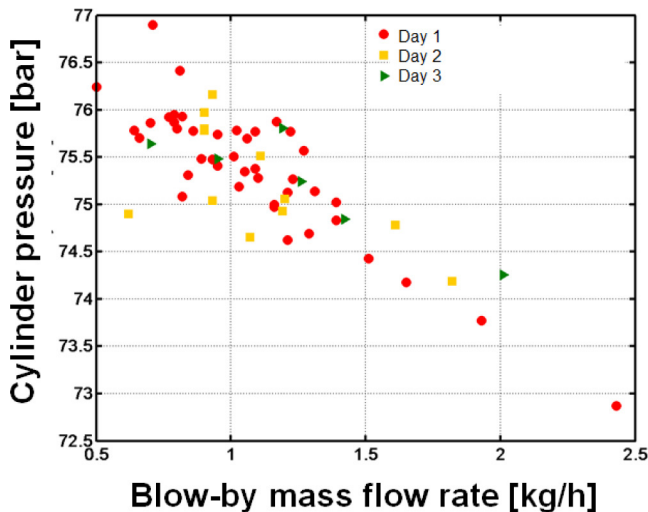


Figure 5. Correlation between maximum in-cylinder pressure and blow-by mass flow rate..

Soot Concentration and Size Distribution

A Cambustion DMS500 device was used to measure the particle concentration and particle size distribution in the blow-by gases. This device was connected to another port of the airtight housing of the dry crankcase, and was also connected to the exhaust line (at 2m from the engine exhaust valves). Therefore, soot measurements were performed both in the exhaust gases and in the blow-by gases. In addition to the Cambustion DMS500 measurements, standard AVL smoke-meter measurements were performed both in the exhaust and the blow-by gases. During the validation phase, soot measurements were performed with both measurement systems, in the exhaust and in the blow-by gases.

A first validation consisted in comparing the size distribution of the soot particles measured by the DMS500 in motored (*i.e.* without fuel injection) and in fired (injection and combustion) conditions. The results are shown in Figure 6. This shows that in the fired operating case, the number of measured particles was about 2 orders of magnitude higher than in the motored case. This proves that the DMS500 measurements in the blow-by gases effectively measures soot, and not graphite particles which could be generated as a result of wear of the graphite piston ring pack.

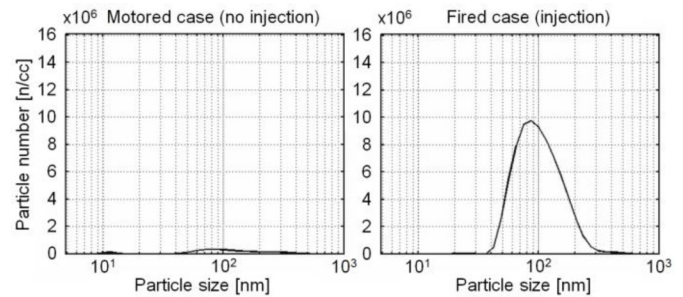


Figure 6. Comparison of particle size distribution in motored (left) and fired (right) engine operation. The same scales are used in both figures.

Subsequently a comparison of the soot levels measured by the AVL smokemeter and the Cambustion DMS500 was performed, for both measurement locations corresponding to gases in the exhaust line and the engine blow-by. The results are presented in Figure 7. It shows that the number of particles measured by the DMS500 was proportional to the smokemeter measurements, both in the exhaust and blow-by gases. However, it must be noted that the proportionality coefficients are different: $2 \cdot 10^8$ for measurements performed in the exhaust line and $3 \cdot 10^7$ for measurements in the blow-by gases. One explanation for this could be linked to the difference in size distribution of the particles present in the exhaust and blow-by gases. As shown in Figure 8, nucleation (small particles) and accumulation modes (large particles) are observed for the measurements performed in the exhaust line, whereas only an accumulation mode is apparent in the blow-by gases. Thus, the particle size distribution for measurements performed in the blow-by gases is shifted towards larger particles. Since the smokemeter measurements are sensitive to the projected area of the particles, for the same total number of particles, large particles will give a higher smoke value than small particles, and subsequently, a lower ratio (Value DMS500)/(Value Smokemeter).

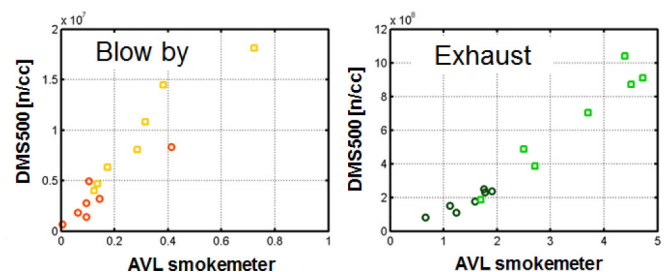


Figure 7. Comparison of the soot content of blow-by (left) and exhaust (right) gases measured by the DMS 500 and the smoke-meter.

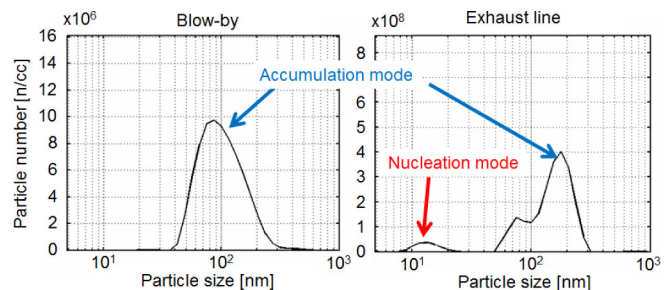


Figure 8. Size distribution of the particles in the blow-by (left) and exhaust (right) gases.