

Optical Diagnostics of the Pollutant Formation in a CI Engine Operating with Diesel Fuel Blends

2011-37-0003

Published
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doi:10.4271/2011-37-0003

ABSTRACT

To meet the future stringent emission standards, innovative diesel engine technology, exhaust gas after-treatment, and clean alternative fuels are required. Oxygenated fuels have showed a tendency to decrease internal combustion engine emissions. In the same time, advanced fuel injection modes can promote a further reduction of the pollutants at the exhaust without penalty for the combustion efficiency. One of the more interesting solutions is provided by the premixed low temperature combustion (LTC) mechanism jointly to lower-cetane, higher-volatility fuels.

In this paper, to understand the role played by these factors on soot formation, cycle resolved visualization, UV-visible optical imaging and visible chemiluminescence were applied in an optically accessed high swirl multi-jets compression ignition engine. Combustion tests were carried out using three fuels: commercial diesel, a blend of 80% diesel with 20% gasoline (G20) and a blend of 80% diesel with 20% n-butanol (BU20). The fuels were tested at 70MPa injection pressure and different timings. At late injection timing coupled to high EGR rate (50%), the blends increased the ignition delay allowing to operate in partially premixed LTC (PPLTC) regime in which the fuel is completely injected before the start of combustion. Strong reduction of engine out emissions of smoke and NOx were obtained but with a little penalty on engine efficiency. This limitation was overcome operating at earlier injection timing in which a mixing controlled combustion (MCC) LTC regime was realized. In this regime, a good compromise between low engine out emissions and efficiency was achieved.

INTRODUCTION

The trade-off between NOx and particulate matter (PM) emissions from diesel engines is a well-known weakness of conventional diesel combustion. NOx and PM formation are influenced by in-cylinder local equivalence ratio and combustion temperature [1]. Therefore, if the combustion can be shifted outside the regions where the local equivalence ratio and combustion temperature are favorable for NOx and PM formation, the simultaneous reduction of NOx and PM emissions can be achieved. This is the concept of low temperature combustion (LTC) shown in Figure 1. It visualizes the progress of the combustion and emission formation/destruction processes in a ϕ -T plot [2]. The plot represents the equivalence ratio versus temperature showing the regions where soot and NOx formation occur for different combustion mechanisms. In this plot, the zones of soot and NOx formation are sketched according to the numerical results of Kitamura et al. [3] and upgraded by Kook et al. [4]. Starting from right, it is shown the LTC island around the $\phi=1$ line, the HCCI island is below $\phi=1$. Moving more towards left, the partially premixed LTC (PPLTC) is the typical regime, far from the high soot and NOx regions and above $\phi=1$ (fuel rich zone) the dotted line is representative of the mixing controlled combustion (MCC) mechanism. These two last regions are explored in the present work using a G20 blend (20% gasoline and 80% diesel fuel) and BU20 (20% n-butanol and 80% diesel fuel).

According to many authors [4][5][6][7][8], one major challenge of LTC is to prepare as much as possible premixed charge before the start of combustion (SOC). A fuel like gasoline, more resistant to autoignition and an appropriate injection pressure and timing, can be very useful to realize it. In fact, large amounts of exhaust gas recirculation (EGR) and high fuel injection pressure are common strategies to promote fuel and air mixing processes. High EGR reduces in-cylinder

temperature and extends ignition delay while high injection pressure promotes fuel atomization and vaporization; both are beneficial for mixing process [4] [9]. However, the application of high EGR reduces in-cylinder oxygen content, causing deterioration of combustion efficiency that leads to increased hydrocarbon (HC) and carbon monoxide (CO) emissions. To avoid the disadvantages of high EGR rates, a fuel with a lower cetane number and higher volatility can be considered. The resistance to autoignition of low-cetane fuels may provide an adequate ignition delay for mixing and faster vaporization that can increase mixing rate. In this way, a thorough mixing can be achieved without high EGR rates and worsening of combustion efficiency. Several researchers have investigated the relationship between LTC operational range and cetane number [10] [11]. In a light-duty diesel engine working at high loads, a low-cetane fuel allowed an homogeneous lean mixture with improved NO_x and smoke emissions joint to a good thermal efficiency. Regarding the fuel, it has been recently demonstrated that a blend of diesel and gasoline, termed “dieseline”, is beneficial for engine out emissions and combustion stability. [12][13][14] Nevertheless, the growing energy demand and limited petroleum fuel sources in the world have guided researchers towards the use of clean alternative fuels like alcohols for their better tendency to decrease the engine emissions [15]. Alcohols have less carbon, sulfur content and more oxygen than traditional fossil-based fuels. On the other side, alcohol fuels, generally, produce higher evaporative emissions due to higher vapor pressures while their low energy density causes a drop in engine performance. However, the very low cetane number limits the usage of neat alcohols in diesel engines; they should be blended with diesel fuel without any modifications in the engine fuel system. There are many studies performed to observe engine performance and exhaust emissions by alcohol fuels blended with diesel fuel. Most of them concern the use of methanol and ethanol. [16] [16] On the other, n-butanol has more advantages than ethanol and methanol. [11] [18] [19] Butanol is much less evaporative and releases more energy per unit mass than ethanol and methanol. Butanol has also a higher cetane number (12) than ethanol (8) and methanol (3) making it a more appropriate additive for diesel fuel. Butanol is less corrosive than ethanol and methanol and it can be blended with diesel fuel without phase separation. Finally, butanol can be produced by fermentation of biomass, such as algae, corn, and other plant materials containing cellulose that could not be used for food and would go, otherwise, to waste.

Even if the effect of butanol-diesel fuel blends on engine performance and emission of diesel engines was well studied [20][21][22][23][24], literature is poor about a detailed description of the in-cylinder thermo-physical phenomena. Same lacks can be also found for the gasoline-diesel fuel blend and for advanced combustion investigation such as low-temperature combustion.

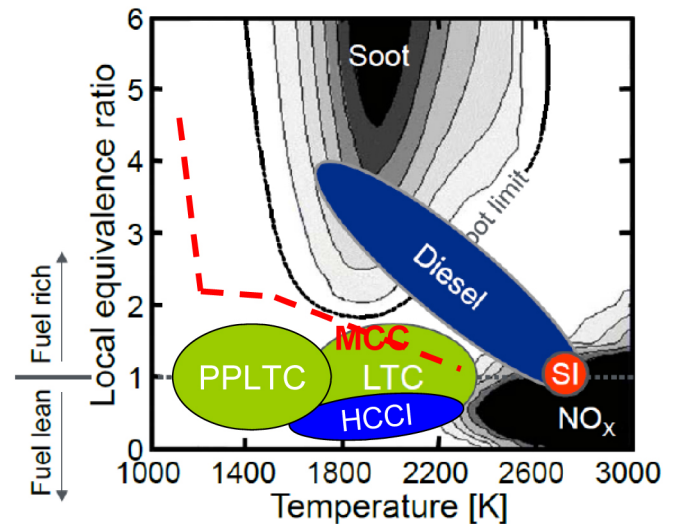


Figure 1. Plot of equivalence ratio (ϕ) versus temperature with the locations of the regions of different combustion mechanisms (Diesel, SI, HCCI, LTC, PPLTC, MCC)

In this paper two different low-temperature combustion (LTC) mechanisms (PPLTC and MCC), in an optical compression ignition engine, have been investigated. Two fuel blends with 80% of diesel and 20% gasoline and 20% n-butanol, respectively, have been tested. Optical diagnostics were applied to follow the combustion process changing injection timing and EGR rate. Their effect on the lift-off length and soot formation were studied. In-cylinder optical investigations were correlated with the engine parameters and with the engine out emissions, measured by conventional methods.

EXPERIMENTAL SET-UP

ENGINE

The experiments were carried out in an external high swirl optically accessed combustion bowl connected to a single cylinder 2-stroke high pressure common rail compression ignition engine as shown in Figure 2. The main engine specifications are reported in Table 1. The external combustion bowl (50 mm in diameter and 30 mm in depth) is suitable to stabilize, at the end of compression stroke, swirl conditions to reproduce the fluid dynamic environment similar to those within a real direct injection diesel engine. The implication of “cylindrical bowl” is related to the peculiar design of the prototype engine that has a large displacement as an air compressor.

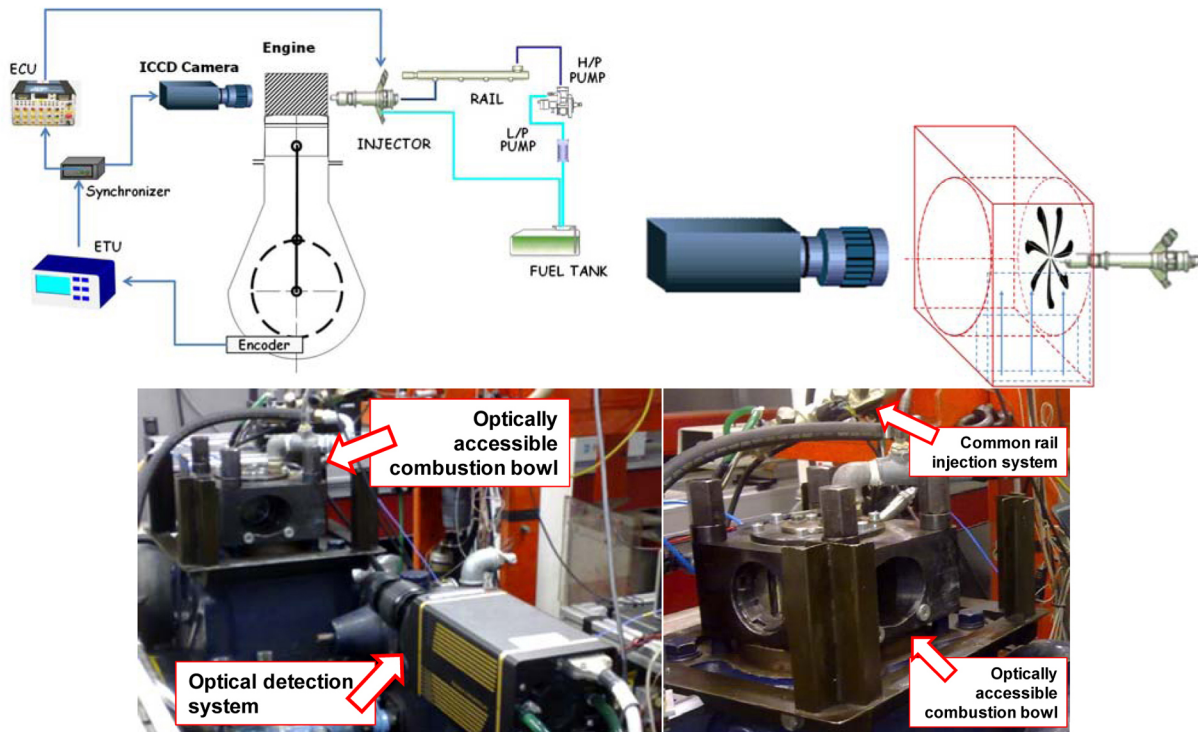


Figure 2. Sketch of the experimental set-up and details of the optically accessible engine

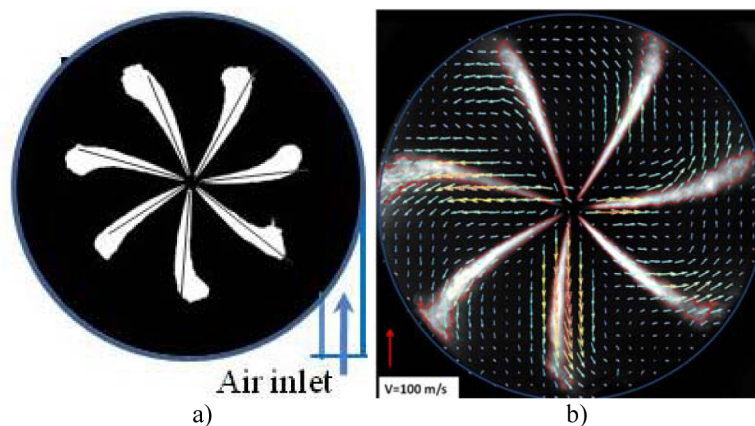


Figure 3. (a) High swirl combustion bowl and jet axis locations. (b) Ensemble-averaged air velocity vector distribution and fuel jet

Table 1. Specifications of the engine

2-stroke single cylinder ci engine	
Cylindrical Bowl (mmxmm)	50 x 30
Bore (mm)	150
Stroke (mm)	170
Connecting Rod (mm)	360
Compression ratio	10.1:1
Air supply	Roots blower
Abs. intake air pressure (MPa)	0.217
Bosch Injector nozzle	7/0.141/148°
Injection pressure (MPa)	70

As shown on the left of Figure 3, the main cylinder, connected to the external “swirled bowl” through a tangential duct, allows to supply compressed air flow to the bowl as the piston approaches TDC. The air flow, coming from the engine cylinder, is forced within the combustion chamber by means of a tangential duct. In this way, a counter clockwise swirl flow, with the rotation axis about coincident to the symmetry axis of the chamber, is generated, as shown on the right side of Figure 3. The injector was mounted within this swirled chamber with its axis coincident to the chamber axis; in this way the fuel, injected by the nozzle, is mixed up through a typical interaction with the swirling air flow. The

combustion process starts and mainly proceeds in the chamber. As soon as the piston moves downward, the flow reverse its motion and the hot gases flow through the tangential duct to the cylinder and finally to the exhaust ports. An example of the fuel jet penetration within the chamber is illustrated in [Figure 3](#) (left plot), whereas the right plot shows the interaction between the gas swirling flow and fuel droplets velocity distribution vectors taken by the Particle Image Velocimetry (PIV), injecting the fuel at 120 MPa. The combustion chamber also provides a circular optical access (50 mm diameter), on one side of it, used to collect images and a rectangular one (size of 10×50 mm) at 90° , outlined on the cylindrical surface of the chamber, used for the laser illumination input. The injector was located on the opposite side of the circular optical access with the axis coincident to that of the combustion chamber. The common rail injection system was arranged by a solenoid controlled injector with a micro-sac 7 hole, 0.141 mm diameter, 148° spray angle. An external roots blower provided an intake air pressure of 0.217 MPa with a peak pressure within the combustion chamber of 4.9 MPa under motored conditions.

OPTICAL DIAGNOSTIC TECHNIQUES AND EXPERIMENTS

In the preliminary phase of the work, cycle resolved visualization of the spatial and temporal spray evolution were performed. The visualization was obtained using an 8-bit high-speed camera (Optronis CamRecord 5000 – 512×512 pixel) equipped with a 50 mm focal Nikon lens. The spectral range of the high speed camera ranged from 390 nm to 900 nm. A camera region of interest was selected (440×440 pixel) to obtain the best match between the spatial and temporal resolution. This optical assessment allowed a spatial resolution around 0.15 mm/pixel and a frame rate of 5882 fps. The exposure time was fixed at 50 μ s corresponding to 0.15 CAD at the engine speed of 500 rpm. The spray was back-illuminated by two 250 W halogen lamp in order to visualize the fuel jets still before the start of combustion.

To reach the target of the paper, UV-Visible digital imaging and chemiluminescence were applied. Previous works demonstrated that images of flame luminosity in the UV can be used to measure the lift-off length of diesel flames [25][26][27]. Even if several sources, including soot emission, may contribute to the total UV-VIS luminosity, the first exothermic reactions occur in the UV spectral range (<400 nm). [25] Moreover, as reported in [28], the UV flame emission, below 350 nm, is governed by OH chemiluminescence. [29] The energetic reactions and high temperatures, which occur during stoichiometric combustion of typical hydrocarbon fuels, form excited state species that include excited state OH*. Once formed, OH* returns fast to its ground state, a fraction through chemiluminescent emission and through collisional quenching. OH

chemiluminescence provides an excellent marker of the high heat release regions where it was generated.

Optical measurements have been performed, using an intensified CCD camera equipped with a quartz lens (UV-Nikon 78-mm), collecting the light emission that passes through the optical access of the combustion chamber. The electronically gated ICCD camera had an array size of 512×512 pixels with a pixel size of 19×19 μ m and 16-bit dynamic range digitization at 100 kHz. The match between the ICCD and the lens allowed 185 μ m spatial resolution. The camera spectral range spread from UV (180 nm) until visible (700 nm). The line-of-sight light emission measurements were performed in the whole ICCD spectral range. The ICCD is not a cycle resolved detector and each acquisition was carried out at a fixed crank angle of different engine cycles setting the exposure time at 5 μ s. The temporal difference between two images was 50 μ s. The intensifier gain was adjusted so that the brightest region of images was on the threshold of saturating the detector and it was the same for all the engine tests.

Single-jet experiments in well-defined ambient environments have provided valid relationship between flame lift-off length and soot formation. [25][27] In multi-jet realistic engine environments, the determination of lift-off length becomes more difficult due to in-cylinder swirling flows, the proximity of in-cylinder surfaces and interaction between adjacent jets. [26] To measure lift-off length from UV-VIS chemiluminescence, the luminosity along the axis of each liquid fuel spray was considered. The flame emission occurred spatially outside the envelope of the liquid scatter, appearing as a “bulge”. [25][26][28] A 15% threshold in the luminosity, along the spray axis, was fixed in order to remove liquid scattering and evaluate the flame lift-off. Along each fuel jet axis an automated computer-based image-processing routine evaluated the distance from the combustion chamber center to the first treated UV-VIS signal. The flame lift-off was obtained by the average on the seven directions. [Figure 4](#) shows a sketch of the procedure for the optical data retrieving. 2D chemiluminescence measurements were also performed by band-pass filters with 10 nm half height width, mounted in-front the UV Nikon lens. The filters were centered at 532 nm. The exposure time was set at 50 μ s while the temporal difference between two images was 50 μ s.

Visible wavelength flame emission is dominated by soot incandescence. It occurs farther downstream than the shorter wavelength UV emission [27]. Due to this spatial separation, soot flame lift-off was evaluated. The procedure was the same described for the UV-VIS emission. Finally, a post-detection processing of the optical data was realized. For each 16-bit image only the data in a selected region of interest corresponding to the combustion chamber were retrieved. Each image was converted in a numerical matrix. In this way it was possible to evaluate the luminous signals as integral on

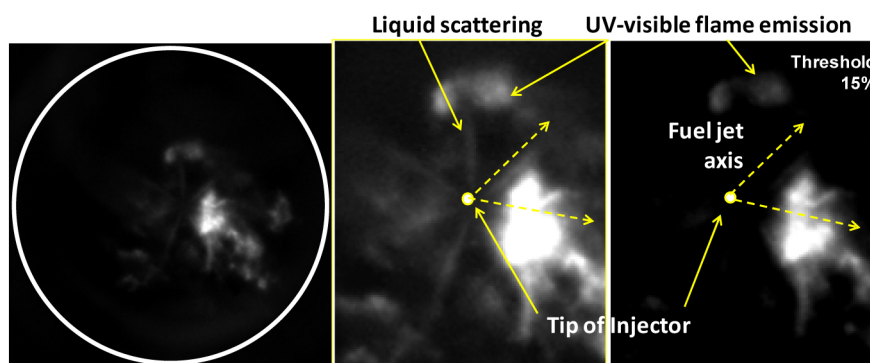


Figure 4. Steps of the procedure for the lift-off evaluation.

the whole combustion chamber. This procedure allowed also to identify the optical start of combustion for all the selected engine conditions. The procedure was applied to the filtered optical data in the visible (532 nm). The crank angle encoder signal synchronized the cameras and the engine, through a delay unit. The AVL Indimodul recorded the TTL signal from camera acquisitions together with the signal acquired by the pressure transducer. In this way, it was possible to determine the crank angles where optical data were detected.

OPERATING CONDITIONS

Combustion tests were carried out using three fuels. The baseline fuel was the European low sulphur (10 ppm) commercial diesel with a cetane number of 52. Moreover, two blends were tested; the first was obtained by blending 80% of the baseline diesel and 20% of commercial 98 octane gasoline by volume, denoted as G20 in the following. The second blend was composed by 80% of the baseline diesel and 20% of n-butanol by volume and denoted as BU20. The main properties of the tested fuels are reported in Table 2.

In the combustion tests data were collected using the AVL INDICOM driven by an optical encoder with 0.2 crank angle degree resolution. The pressure within the swirl chamber was acquired by a quartz pressure transducer AVL QC34C and the rate of heat release was computed by the pressure signal, assuming no pressure difference between the swirl chamber and the cylinder. Results of the in-cylinder pressure have been computed averaging 300 consecutive engine cycles. Exhaust gaseous emissions have been acquired by the AVL DiGas 4000 analyzer for NO_x (1 ppm resolution), the Smoke Meter AVL 415S was used for FSN and soot concentration (0.01 mg/m³ resolution) measurements. All combustion tests were carried out running the engine at the fixed speed of 500 rpm, injecting a fuel amount of 30mg ±1% at the pressure of 70 MPa. Tests were carried out setting the electronic injection timing (SOI) of 11 CAD BTDC, 3 CAD BTDC, 1CAD ATDC and 5 CAD ATDC at two EGR rates, 0% and 50% corresponding to oxygen concentration at intake of O₂=21 and 17%, respectively.

Table 2. Properties of tested fuels

Fuel Properties	Diesel	G20	BU20
Density @ 15 °C [kg/m ³]	840	835	830.4
Viscosity @ 40 °C [mm ² /s]	3.2	2.9	2.2
Cetane Number	52.0	44.2	44.0
Net Heat Value [MJ/kg]	42.5	42.4	41.0
Carbon content [%]	87.0	86.8	80.2
Hydrogen content [%]	12.6	13.0	15.4
Distillation 50% Vol. [°C]	280	250	233
Distillation 90% Vol. [°C]	338	328	330

RESULTS AND DISCUSSION

EFFECT OF ENGINE PARAMETERS ON COMBUSTION AND EMISSIONS

This paragraph describes the behavior of combustion and emissions using conventional diesel fuel, the blend G20 and BU20 with 50% EGR and 0% EGR, at different injection timings. Figure 5 displays the in-cylinder combustion pressure trace versus crank angle for the three fuels at different SOI. The plots report the real start of fuel injection (FSOI) and the end of fuel injection (FEOI), as well as the pressure start of combustion (SOC). The FSOI and FEOI have been computed from the instantaneous fuel flow rate. Looking at the in-cylinder pressure signals, it can be observed that the combined effects of lower cetane number of the G20 and BU20, which gives a longer ignition delay with respect to diesel fuel, and their higher volatility contribute to increase the mixing time joined to a better mixed charge. Moving the SOI from 11 CAD BTDC towards later start of injection a longer ignition delay may be observed. This trend is confirmed both at EGR=0 and 50% suggesting that the decrease in the fuel cetane number increases the ignition delay for the low and high temperature conditions. A summary of SOC and ignition delay is reported in Table III for the different fuels and operating conditions. The test at SOI=-11 cad btcd gives a moderate higher peak in the cylinder pressure for G20 and BU20 with respect to the