

Advanced Injection Strategies for Controlling Low-Temperature Diesel Combustion and Emissions

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ABSTRACT

The simultaneous reduction of engine-out nitrogen oxide (NO_x) and particulate emissions via low-temperature combustion (LTC) strategies for compression-ignition engines is generally achieved via the use of high levels of exhaust gas recirculation (EGR). High EGR rates not only result in a drastic reduction of combustion temperatures to mitigate thermal NO_x formation but also increases the level of pre-mixing thereby limiting particulate (soot) formation. However, highly pre-mixed combustion strategies such as LTC are usually limited at higher loads by excessively high heat release rates leading to unacceptable levels of combustion noise and particulate emissions. Further increasing the level of charge dilution (via EGR) can help to reduce combustion noise but maximum EGR rates are ultimately restricted by turbocharger and EGR path technologies. Furthermore, at low loads, the low combustion temperatures lead to high levels of unburned hydrocarbon (HC) and carbon monoxide (CO) emissions. There is therefore a significant interest in trying to limit the EGR rates to the minimum necessary levels in order to achieve LTC (*i.e.* low NO_x and particulate emissions) by using alternative strategies to control combustion to address the HC/CO emissions at low load and the noise/particulates trade-off at higher loads. The present study investigates the potential of controlling combustion directly via fuel injection with the use of advanced injection strategies such as rate-shaping and/or closely-spaced double injections.

Experiments were conducted on a single-cylinder all-metal engine and an equivalent optically-accessible version of the same engine. The optical engine allowed for the application of several laser-based, optical

diagnostic techniques in order to study the in-cylinder processes of mixture preparation, combustion and emissions. Overall, the results of this study highlight the potential of controlling LTC directly via fuel injection using advanced injection strategies which could ultimately reduce the dependency on excessively high EGR rates in order to address the HC/CO emissions at low load and the emissions/noise/fuel economy trade-off at higher load LTC operating conditions without penalising fuel consumption.

INTRODUCTION

The increasingly stringent emissions legislation for Diesel engines has intensified research and development efforts within the automotive industry towards innovative combustion systems and exhaust gas after-treatment technologies. One promising approach which offers the potential to simultaneously reduce 'engine-out' NO_x and soot emissions is via low-temperature combustion [1, 2, 3]. Employing high levels of charge dilution via EGR (thereby reducing intake charge oxygen concentration) leads to a decrease of combustion temperatures thereby reducing thermal NO_x formation. Intake charge premixing can be enhanced by increasing the ignition delay and can be achieved for example by using advanced injection timings and reduced compression ratios. Increased premixing leads to improved mixture homogeneity and leaner local equivalence ratios thereby limiting soot formation [4, 5]. However, the low combustion temperatures and degree of premixing must be controlled to a certain extent in order to avoid the formation of over-mixed (*i.e.* fuel-lean) regions which can lead to increased unburned hydrocarbon (HC) and carbon monoxide (CO) emissions [6]. Furthermore, at higher load LTC operating

conditions, excessive pre-mixing tends to result in high heat release rates and subsequently, unacceptable levels of combustion-generated noise. Further increasing the EGR rates can help to alleviate combustion noise although this is detrimental to HC and CO emissions at low load and leads to high particulate emissions at higher load LTC operating conditions. It is therefore desirable to limit charge dilution to the minimum levels necessary to achieve low NOx and particulate emissions while investigating alternative strategies which might improve the HC/CO-noise trade-off at low loads and the particulate-noise trade-off at higher loads.

This study focuses on the potential of controlling combustion and emissions directly via injection. Indeed, the adoption of relatively late start of injection (SOI) timings during the compression stroke has the advantage that auto-ignition and combustion is closely coupled to the injection event. In particular, the adoption of advanced injection strategies using either injection rate shaping and/or closely-spaced double injection strategies have been explored. Although multiple injection strategies have been employed in conventional direct-injection (DI) Diesel combustion engines for many years, the vast majority of previous studies [7, 8, 9] are based on the use of pilot injections of small fuel quantities in order to limit the premixed burn associated with high combustion noise. Furthermore, post injection strategies are also used as it is claimed that this enhances post-combustion mixing and particulate oxidation. In [9], multiple injections corresponding to pilot-main-post and pilot-pilot-main strategies were studied with wide variations of injection timing and dwell times. For certain injection dwell times, the heat release phasing corresponding to the main injection was reportedly influenced by the cooling effect of fuel vaporisation. However, it was also reported that accurate control of the injected fuel quantity is difficult to achieve for such injection strategies due to pressure waves induced after successive injection events. A more recent study [10] examined the effect of split injection strategies with up to four injections per cycle on combustion and engine-out emissions. In particular, splitting the injection into multiple events was observed to increase the combustion duration, retard combustion phasing and reduce the peak heat release.

Investigations into the use of multiple injection strategies under LTC operating conditions are quite limited. For example, developments of the UNIBUS (uniform bulky combustion system) concept reported in [11] exploited a double injection strategy which was claimed to offer more control over auto-ignition timing. The work reported in this paper is part of ongoing research at IFP [12 & 13] into the potential of using advanced injection strategies (rate-shaping and multiple injections) as a means of controlling combustion and emissions under highly-dilute LTC conditions. The experimental study reported here was performed on both a single cylinder all-metal engine and an equivalent optically-accessible version of the same engine enabling the use of several laser-based, optical diagnostic techniques in order to study the in-

cylinder processes of mixture preparation, combustion and emissions.

It is well known that operating at low loads under highly dilute LTC conditions leads to high levels of engine-out HC and CO emissions mainly as a result of low combustion temperatures which result in incomplete combustion. In view of this constraint, the authors have investigated the potential of injection rate shaping as a means of controlling the heat release rate and thus limiting combustion noise which is generally achieved via the use of aggressive EGR rates. If combustion control can be achieved directly via injection, lower EGR rates could potentially be employed which would be beneficial in terms of reducing HC and CO emissions. The trade-off is different when operating under LTC conditions at higher load (5 bar IMEP). In this latter case, engine-out HC and CO are no longer problematic since combustion and exhaust temperatures are sufficiently high and acceptable catalyst conversion efficiencies enable the after-treatment of these species. However, the noise and particulate emissions must be reduced while maintaining very low NOx emissions. In order to achieve this latter objective, combustion control via the use of double injection strategies has been investigated.

EXPERIMENTAL SETUP

ALL METAL AND OPTICAL ENGINES

The two single cylinder research engines used in the present study were identical in terms of engine hardware and combustion chamber geometry. As explained earlier, studies of engine in-cylinder processes using laser-based optical diagnostic techniques were performed on an optically accessible engine whilst parametric studies were carried out on a second, standard 'all-metal' version of the same engine. The engine used in this study was a single-cylinder version of a 2.0 litre light-duty, direct-injection (DI), 4-stroke Diesel engine with a compression ratio of 14:1. More details of the engine specifications are provided in Table 1. Heat release rates were calculated from the measured in-cylinder pressure traces using a 0-D thermodynamic analysis.

Table 1 : Engine specifications of the all-metal and optical-access engines.

	all-metal engine	optical engine
Engine base type	2 litre, DI Diesel	2 litre, DI Diesel
Number of cylinders	1	1
Number of valves	2 int. + 2 ex.	2 int. + 2 ex.
Bore/stroke	85/88 mm	85/88 mm
Displacement	0.5 l	0.5 l
Squish height	0.78 mm	0.78 mm
Compression ratio	14	14
Intake pressure	1 bar (abs.)	1 bar (abs.)
Swirl ratio	2.2	2.2
Fuel	Standard Diesel	Fischer Tropsch / Isane blend
Cetane number	~52	~52

The combustion chamber geometries of both engines were essentially identical as shown by the superposed cross-sections of the two piston bowls in Figure 1. Minor differences can be observed in the base of the piston bowl due to the need to maintain a flat surface on the optical piston bowl window (made from quartz). In order to compensate for this small difference and to maintain identical piston bowl volumes and therefore identical compression ratios between the two engines, the optical access piston bowl was modified very slightly in the re-entrant region (as can be seen in Figure 1). This modification was performed in such a way so as to maintain a similar combustion chamber shape with respect to the re-entrant wall region where fuel jet-wall impingement occurs. Indeed, the near-wall region within the chamber is of major interest as jet-wall interactions play a major role on mixture formation and the low-temperature pre-combustion reactions.

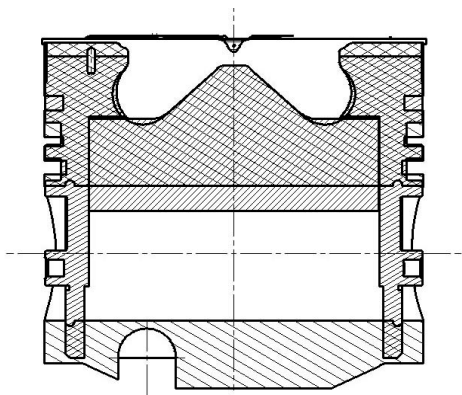


Figure 1 : Superposed cross-sectional drawings of the all-metal and optical-access piston bowls used in the study.

INJECTION SYSTEM

The common rail fuel injection system used in this study is the Delphi direct acting system technology. Opening and closing of the needle is controlled directly by a piezoelectric actuator. The operating principle of this injector differs from conventional servo hydraulic injectors. The principle of direct actuation is based on a large diameter piston and a smaller diameter needle that are part of a control chamber. The actuation of the piston by a piezoelectric actuator leads to a stroke amplified needle motion. The direct acting operating principle offers several advantages over servo hydraulic injectors and in particular the possibility to modify the injection rate profile. Lower needle opening speeds allow greater accuracy and repeatability over fuel metering particularly for small injected fuel quantities while at higher rail pressures, square injection rate profiles (high needle opening speeds) can be employed. The higher needle opening speeds that can be achieved using the direct acting principle means that the system can be used to perform closely-spaced multiple injections. Further details of the injection system are provided in Refs. [12 & 14]. The injector specifications are outlined in Table 2.

Table 2 : Injector specifications

Fuel injector type	Piezoelectric-directly actuated, common-rail
Number of holes	7, equally-spaced
Spray included angle	152°
Rail pressure	1000 bar
Injector fuel flow rate	350 cc/30 s/100 bar

OPTICAL DIAGNOSTICS

Several optical diagnostic techniques were employed in order to study the temporal and spatial development of the fuel distribution in addition to the auto-ignition and combustion characteristics. Simultaneous planar laser-induced fluorescence (LIF) imaging of the fuel distribution and auto-ignition was performed using tracer LIF and formaldehyde LIF respectively. Direct imaging of the naturally-occurring luminous emission from the combustion process was also performed, providing spatially and temporally resolved information with respect to soot forming regions within the piston bowl.

Optimisation of optical access within piston bowl

In addition to maintaining a fully representative combustion chamber geometry, the degree of optical access into the piston bowl was also optimised on the optical engine. Optical access was improved to enable visualisation of the near-wall regions where significant jet-wall interaction occurs. This was achieved by incorporating a quartz window housed within a section of the piston re-entrant as shown in Figure 2 (LHS image).

Furthermore, optical ray-tracing software was used in order to maximise optical access while minimising image distortion via the lateral piston window. Figure 2 (RHS image) shows a target image acquired within the piston bowl which not only reveals negligible image distortion but also confirms that visualisation within the near-wall region was possible (the red line on the target image corresponds to the extents of the re-entrant wall) due to the effects of light refraction across the window thickness. The reader should note that the target image was illuminated using a continuous light source. The intensity distribution does therefore not reflect the laser sheet intensity distribution. It should also be noted that laser-sheet illumination within the near-wall zone was not possible.

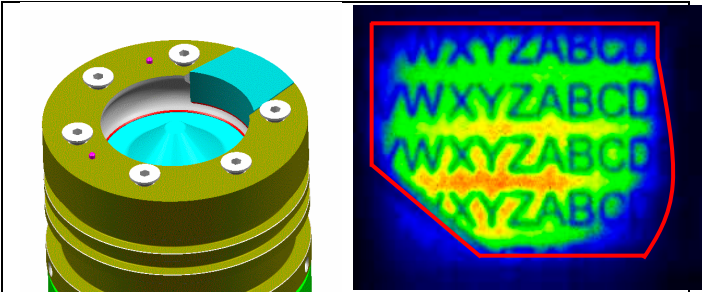


Figure 2 : 3D view of the optimised optical-access piston bowl (LHS). Target image showing field of view obtained within the piston bowl (RHS).

Fuel auto-ignition and spectroscopic considerations

Laser-based diagnostic techniques were employed in this study on the optical engine in order to complement the all-metal engine data and in particular provide a detailed insight into the physical phenomena occurring in-cylinder. Since this investigation was based upon a combined metal/optical engine approach, it was essential to verify that the auto-ignition characteristics and combustion phasing of the optical engine matched those of the metal engine. However, this is not a straightforward issue. For example, standard Diesel fuel could not be used on the optical engine in this study due to the strong LIF emission of aromatic and naphthenic compounds which are naturally present within the fuel (these compounds are generally considered as fuel 'impurities' from a laser diagnostic viewpoint) following ultra-violet (UV) laser excitation. The subsequent LIF emission from Diesel fuel would have made it impossible to separate spectrally the fuel-LIF signal from the formaldehyde-LIF signal. In order to overcome this problem, single component 'pure' (*i.e.* non-fluorescing) fuels such as iso-octane, n-heptane, dodecane or decane are generally used for optical engine studies. LIF imaging is achieved through the use of a fluorescent fuel tracer which is added to the fuel [6, 15]. For Diesel engine studies, single component fuels typically used are dodecane or decane due to their boiling points being reasonably close to the boiling points of the higher volatility components of standard Diesel fuel (dodecane : $T_b=216^\circ\text{C}$, decane : $T_b=174^\circ\text{C}$). However, matching the fuel volatility characteristics is only one part of the challenge. It was also necessary in this study to ensure that the auto-ignition delay of the fuel used on the optical engine matched that of standard Diesel fuel. Unfortunately, dodecane and decane fuels have significantly higher cetane numbers (dodecane : CN~80, decane : CN~77) compared to standard Diesel fuel (CN~52) resulting in significantly shorter auto-ignition delays. Within the context of this study, the application of LIF diagnostic techniques not only meant that a non-fluorescing fuel was required but furthermore, it was crucial that the non-fluorescing fuel had a cetane number comparable to that of standard Diesel fuel in order to ensure similar auto-ignition characteristics.

A fuel blend containing Diesel fuel derived from a Fischer Tropsch (FT) process and an isoparaffin solvent (Isane IP 185) was formulated. Previous studies by the authors [16] using FT fuel revealed that this fuel did not fluoresce following laser excitation at 355 nm, probably due to the low concentration of aromatic and naphthenic compounds. Laser excitation at 266 nm of this fuel did nonetheless result in noticeable LIF emission. However, the cetane number of the FT fuel (CN~76) was significantly higher than that of standard Diesel fuel (see Table 3). A low cetane number (CN~33) isoparaffin solvent (Isane IP 185) was blended with the FT fuel in order to obtain a cetane number comparable with standard Diesel fuel. Isane IP 185 contains neither aromatic compounds, nor benzene and was therefore chosen since it was not expected to fluoresce.

Subsequent LIF measurements indeed confirmed that the fuel blend did not exhibit detectable fluorescence emission following laser excitation at 355 nm.

Table 3 : Measured properties of standard Diesel, Fischer Tropsch and Isane IP 185 used in this study.

	Standard Diesel fuel	Fischer Tropsch	Isane IP 185
Cetane number	~52	76	33
Density (kg/m^3)	842	768	765
Boiling range ($^\circ\text{C}$)	201-364	206-301	187-206
Aromatic content (%)	27.1	0.3	<0.001
Sulphur content (ppm)	35	<10	<1
Viscosity at 40°C (mm^2/s)	3.7	2.1	1.3
Paraffin content (%)	39.1	95.9	100

Having verified the spectral characteristics of the fuel blend, subsequent tests were performed in a rapid compression machine in order to measure the cetane number. A blend of 55% (by weight) FT fuel and of 45% Isane IP 185 was found to have a cetane number of 52, comparable to the Diesel fuel used in this study. Finally, in order to fully validate the FT/Isane fuel blend, a back-to-back comparison of the fuel blend with standard Diesel fuel was performed on the optical engine.

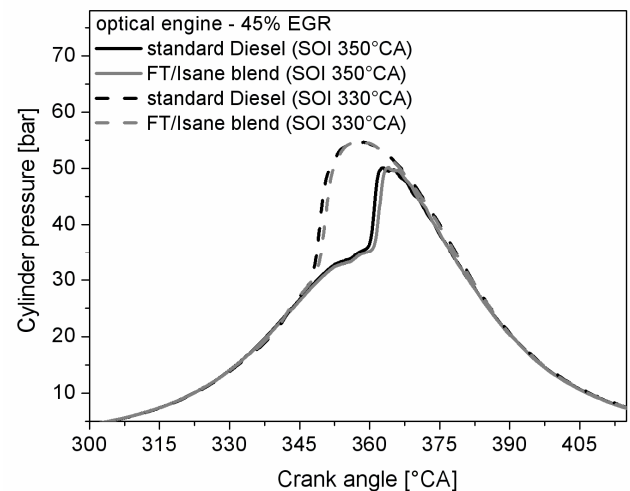


Figure 3 : Comparison of measured in-cylinder pressure traces obtained using standard Diesel fuel and Fischer Tropsch/Isane fuel blend for SOI timings of 330°CA (dashed lines) and 350°CA (solid lines) under LTC conditions, optical engine tests, 1200 rpm, 2 bar IMEP.

Measured in-cylinder pressure traces acquired at two different SOI timings (SOI at 330 and 350°CA) are shown in Figure 3 and reveal excellent agreement between the two fuels confirming that the FT/Isane fuel blend was representative of standard Diesel fuel in terms of its auto-ignition characteristics and was therefore

used as a surrogate fuel to represent standard Diesel fuel on all experiments performed on the optical engine.

Matching combustion phasing between all-metal and optical engines

Prior to performing visualisation studies on the optical engine, it was necessary to ensure that the combustion phasing between the all-metal and optical engines was comparable. This is not a straightforward issue since optical engines differ from standard all-metal engines in several ways. Firstly, the combustion chamber heat transfer characteristics differ significantly on optical engines compared to metal engines due to the use of optical (quartz and/or sapphire) windows. The significantly lower thermal conductivity of quartz and sapphire compared to metal (aluminium) leads to much higher piston surface temperatures. This leads to shorter auto-ignition delays and correspondingly, over-advanced combustion phasing [17, 18]. In addition to thermal effects, geometrical differences can occur due to dynamic (compressive/tensile) loading of the extended piston assembly of optical engines. This may result in variations in the effective compression ratio [19]. As reported in [17, 18, 19], recent efforts have been made with the objective of understanding the differences between optical and all-metal engines and the effects of these differences on the combustion and emissions behaviour. Since this study was based on exploiting experimental data obtained on both an all-metal and equivalent optical-access engine, it was important to ensure similar combustion phasing between the two engines. This was achieved by matching the motored pressure-trace of the optical engine to that of the metal engine, by adopting an appropriate skip-firing strategy and adjusting slightly the intake air temperature. A more detailed description of the experimental methodology employed is provided in [17].

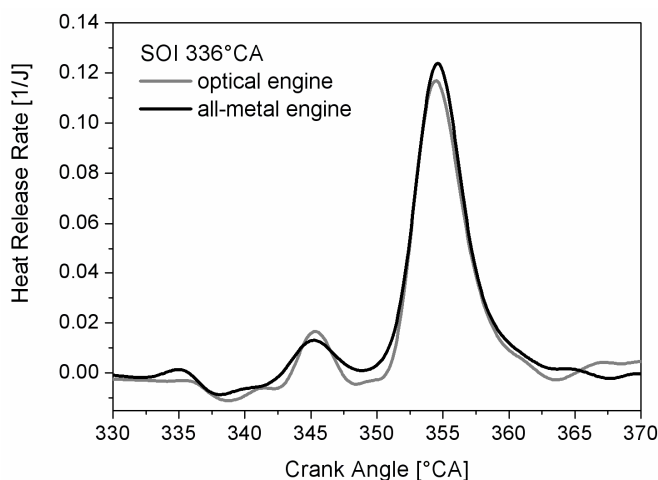


Figure 4 : Comparison of heat release rates obtained on optical and all-metal engines confirms similar combustion phasing, SOI 336°CA, 1200 rpm, IMEP 2 bar.

As shown in Figure 4, fine tuning of the various parameters meant it was possible to achieve similar combustion phasing (in terms of the heat release rate) between the all-metal and optical engines. Such tests were performed on a systematic basis for all engine operating conditions subsequently studied on the optical engine.

Fuel-tracer planar laser-induced fluorescence (LIF)

Having formulated and validated the auto-ignition and spectroscopic characteristics of the FT/Isane fuel blend on the optical engine, a small quantity (2% by weight) of the fuel tracer naphthalene was added to the fuel blend. As previously reported in [15], this particular fuel tracer not only exhibits strong fluorescence upon excitation at 266 nm but seems relatively insensitive to oxygen-quenching effects. Furthermore, naphthalene has a boiling point ($T_b=218^\circ\text{C}$) which lies within the boiling range of the FT/Isane blend so that this tracer should track fuel vapour reasonably well. As shown in Table 3 above, the FT/Isane blend has a boiling range which is within the boiling range of standard Diesel fuel. As explained earlier, fluorescence of the FT fuel could be detected upon 266 nm laser excitation however the signal strength was observed to increase significantly following the addition of naphthalene tracer.

Since the tracer LIF technique was applied under reactive (*i.e.* combustng) conditions, collisional quenching effects due to the presence of oxygen might be significant. However, attempts to correct for such quenching effects have not been performed here since this is complicated by the highly stratified nature of LTC and in particular the non-uniform spatial variations of both the fuel and oxygen distributions. Although background fluorescence and laser elastic scattering effects were accounted for, it should be noted that the tracer LIF signal was not corrected for pressure or temperature effects. As a result, the tracer LIF measurements presented in this paper are qualitative rather than quantitative but nevertheless provide valuable information with regards to the fuel distribution within the piston bowl on both a spatial and temporal basis.

Tracer-LIF was achieved by 266 nm excitation using a frequency-quadrupled Nd:YAG laser (50 mJ laser energy per pulse). The output beam was formed into a two-dimensional sheet which entered the combustion chamber via the 45° mirror and piston crown window (Figure 5). The laser sheet illuminated a vertical plane coincident with the cylinder axis of symmetry corresponding to almost one half of the piston bowl as shown by the target image in Figure 2. The LIF images were collected at 90° across the quartz window (housed within a section of the piston bowl wall) and the transparent sapphire ring (Figure 2 and Figure 5) onto a 16-bit intensified, UV-sensitive CCD camera (Roper Scientific). A 320 nm high pass filter combined with a 400 nm low pass filter were used to capture the fuel-tracer LIF signal.

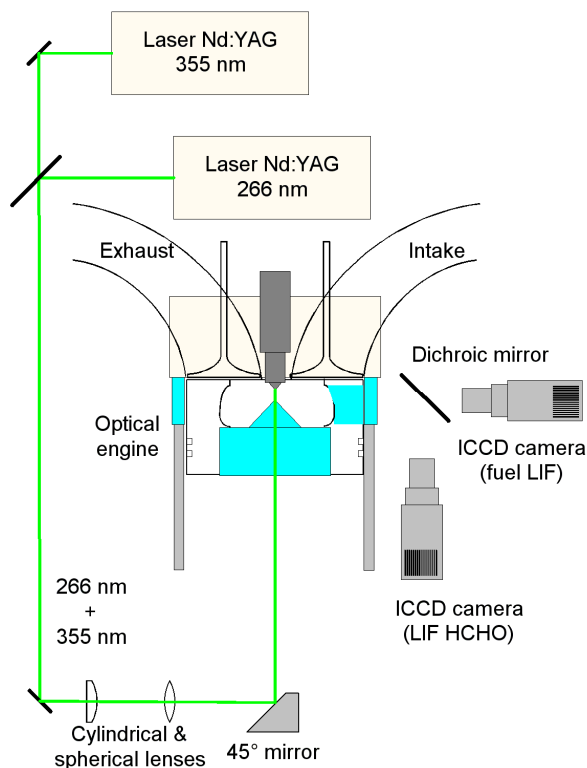


Figure 5 : Schematic of the optically accessible DI Diesel engine and optical diagnostics setup.

Formaldehyde (HCHO) and polycyclic-aromatic hydrocarbons (PAH) planar laser-induced fluorescence (HCHO/PAH LIF)

Formaldehyde (HCHO) is formed as a result of fuel decomposition during the initial low temperature heat release and is considered to be an auto-ignition precursor [20]. Formaldehyde LIF imaging was performed by using a frequency-tripled (355 nm) Nd:YAG laser (65 mJ laser energy per pulse). The 355 nm and 266 nm laser sheets were aligned to ensure that the same vertical plane within the piston bowl was illuminated. The 355 nm wavelength only allows excitation of relatively weak transitions in the HCHO molecule [23] and unlike selective wavelength excitation [24], does not allow the discrimination of the formaldehyde signal against non-resonant signal contributions, notably from the fluorescence of polycyclic-aromatic hydrocarbons (PAH) [25, 26]. It is therefore possible that PAH fluorescence occurs under certain conditions which could potentially make it difficult to distinguish between formaldehyde LIF and PAH LIF signals. However, as will be seen later in the paper, an analysis of the timing (*i.e.* appearance of the LIF signal within the engine cycle relative to in-cylinder pressure and fuel injection phasing for example), spatial distribution and intensity of the collected LIF signal, it is in general possible to distinguish formaldehyde fluorescence from PAH fluorescence. The formaldehyde/PAH LIF signal was detected by a 16-bit, intensified CCD camera (Roper Scientific). A 390 nm

high pass optical filter combined with a 465 nm band pass filter (70 nm FWHM) were used to capture the HCHO/PAH LIF signal.

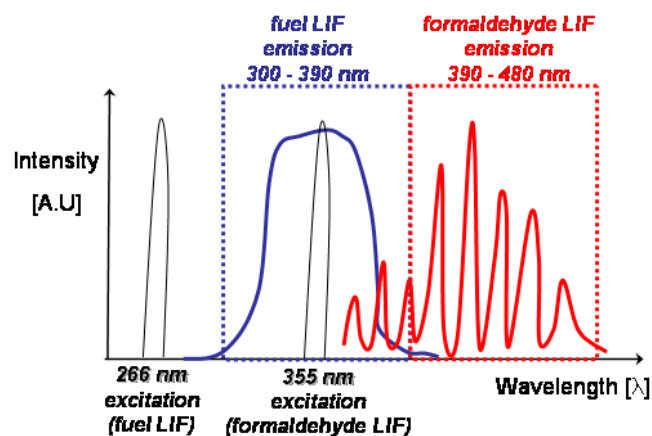


Figure 6 : Schematic representation showing LIF excitation (266 nm and 355 nm) and emission spectra for simultaneous fuel LIF and formaldehyde LIF measurements.

Figure 6 presents schematically the tracer and formaldehyde/PAH excitation wavelengths (266 and 355 nm) and corresponding LIF emission spectra. Fuel-tracer LIF emission was collected between approximately 320 and 390 nm whilst HCHO/PAH fluorescence was collected between 390 and 480 nm. Due to the limited repetition rate (10 Hz) of the pulsed lasers, only one tracer and HCHO/PAH image pair could be acquired for each engine cycle. Therefore, in order to generate a statistical dataset, a total of 30 images (corresponding to 30 different engine cycles) were acquired at each engine crank angle of interest. The two Nd:YAG lasers used for simultaneous LIF imaging were temporally separated by 50 μ s so as to avoid potential interference.

Direct combustion imaging

In addition to the simultaneous application of fuel-tracer and formaldehyde/PAH LIF imaging techniques, direct imaging of the combustion process was also performed using a 16-bit intensified CCD camera. The naturally-emitted light is due to chemiluminescence of species such as HCHO^* , HCO^* , CH^* , OH^* , C_2^* and CO_2^* [27] which predominantly occurs at wavelengths shorter than 500 nm and/or soot incandescence which is characterised by black body radiance emission above 500 nm. As will be shown later in the paper, images of the combustion process provided complementary information to the LIF diagnostics and in particular with regards to soot-forming regions within the piston bowl. It should be noted that the signal corresponding to combustion luminosity is integrated along the line-of-sight which is in contrast to the LIF images. In the latter case, the laser sheet thickness ultimately defines the effective depth-of-field, thus providing higher spatial image resolution.

OPERATING CONDITIONS

The results presented in this paper correspond to two highly-dilute, LTC operating conditions. A low engine load condition of 2 bar IMEP and a higher load, 5 bar IMEP condition were investigated. Table 4 summarises the objectives and corresponding methods that were studied as a function of the Diesel LTC operating condition. As explained earlier, the objective at low load was to study the potential of controlling combustion either via injection rate shaping or adopting closely-spaced double injections in order to adapt the heat release and to limit combustion noise which would alleviate the use of excessively high EGR rates and ultimately allow for a reduction in engine-out HC and CO emissions. At the higher load LTC condition, limiting excessive combustion noise is the major challenge. Advanced injection strategies are investigated here in order to evaluate their potential for controlling combustion (*i.e.* heat release rate) and improve the noise/emissions/fuel economy trade-off. This would allow for a reduction of the required EGR rates and might prove beneficial in terms of decreasing particulate emissions. The tests were performed at 1200 rpm and 2500 rpm and a fuel rail pressure of 1000 bar (Table 2). For the 1200 rpm/2 bar IMEP operating condition, a single injection strategy was adopted with the aim of evaluating the potential of injection rate shaping on controlling to some extent the heat release rate. In particular, varying the injection rate profile (*e.g.* from fast to slow injection rates) might have a significant influence on fuel mixing, auto-ignition and therefore combustion and emissions. In these tests, a 'late SOI timing' of 349°CA (*i.e.* 11°CA before TDC) and an 'early SOI timing' of 336°CA (*i.e.* 24°CA before TDC) were investigated. For the higher load condition corresponding to 5 bar IMEP, several double injection strategies were investigated with the objective of evaluating the potential of splitting the injection event in order to control the heat release rate. In these experiments, the first SOI timing (SOI₁) was maintained constant (351°CA at 1200rpm, and 340°CA at 2500 rpm) while an injection timing sweep of the second injection (SOI₂) was performed. For the double injection strategies, a parameter hereafter defined as the injection ratio (R) was defined which corresponds to the ratio of the injected fuel mass of the first injection (m_1) relative to the total injected fuel mass (m_{total}):

$$R = \frac{m_1}{m_{total}} \quad (1)$$

At 1200 rpm, the injection ratio, R was 0.6 (*i.e.* 60% of the fuel injected during the first injection and the remaining 40% injected during the second injection). An injection ratio R of 0.7 was used for the 2500 rpm/5 bar IMEP operating condition.

Two series of tests were performed:

- Tests were initially performed for a constant injected fuel mass and EGR rate. In this case, slight IMEP variations might occur however these tests are useful for an improved understanding of the direct effects of the injection strategy on mixing and combustion. Such tests were performed on both the metal and optical engines.
- Tests were also performed under constant IMEP conditions (the fueling rate was varied slightly) while respecting predefined noise and emissions levels in order to quantify the potential of the various injection strategies adopted. These tests were limited to the metal engine only.

For all operating points the engine water and oil temperature were regulated at 90°C, and the plenum temperature were maintained constant at 70°C.

Table 4 : Outline of methods adopted in order to achieve LTC objectives according to operating condition.

LTC operating condition	Objectives	Approach/method
'Low-load' (2 bar IMEP)	Reduce HC/CO without compromising NOx emissions and noise levels	Potential control of heat release rate via injection rate shaping allowing for a reduction of EGR levels
'High-load' (5 bar IMEP)	Reduce combustion noise without compromising engine-out emissions	Control heat release rate via injection in order to maintain acceptable noise limits while limiting soot emissions through reduced EGR levels

RESULTS

SINGLE INJECTION : INJECTION RATE SHAPING

A low load operating condition of 1200 rpm and ~2 bar IMEP was studied and in particular, the potential of controlling combustion via injection rate shaping using a single injection strategy. It should be noted that the injected fuel quantity was maintained constant for the experiments in which the rate shape was varied and so the IMEP varied slightly around 2 bar. As a result, the injection duration is therefore longer for the 'slow' injection rate compared to the 'fast' injection rate. The injection profiles for the fast and the slow injection rates are illustrated schematically in Figure 7.