

Imperial College London
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Partially Premixed Combustion in an Optically Accessible Compression Ignition Engine

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A thesis submitted in partial fulfillment of the requirements for the degree of
Doctor of Philosophy

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*To my Grandfather, Ioannis Katsivalis,
who introduced me and guided my first
steps into the fascinating world of science.*

Abstract

This thesis sets out the high-speed imaging and Laser Induced Incandescence (LII) work carried out on the Ricardo Hydra and the Volvo D5 single cylinder optical light duty diesel engines at Imperial College London. A literature review was presented on diesel engine operation strategies along with a review on the optical diagnostic techniques used for the purpose of the present thesis.

The experimental work on the Ricardo Hydra optical engine comprises three different experimental campaigns, which investigate three major engine operation strategies, commonly employed by engine manufacturers, namely inlet air dilution, pilot injection and post injection. The analysis identifies combustion patterns and characteristics and describes how these were affected when different operation strategies were employed. The investigation focused on soot emissions formation and their evolution throughout the combustion event. Most importantly, the employed operation strategies were evaluated in regard to their efficiency in reducing engine-out soot emissions.

The experimental work on the Volvo D5 optical engine focused on the effect of injection timing and of a close-coupled pilot injection on combustion stratification and soot formation at operating conditions that promote PPC operation.

Findings were analysed *via* means of pressure trace and apparent rate of heat transfer analyses, as well as a series of optical diagnostics techniques, namely high-speed flame natural luminosity imaging, CH^* and C_2^* line-of-sight chemiluminescence, as well as planar Laser Induced Incandescence (pLII).

Overall, results suggest that longer ignition delay is a key parameter in lower engine-out soot emissions, which can be achieved by several operating techniques. Longer ignition delay leads to enhanced in-cylinder mixing, which facilitates operation in the Partially Premixed Combustion regime, where fuel-rich areas can be eliminated, and soot formation weakens.

Declaration of Originality

The present PhD thesis is an own original work of Dimitrios Panagiotis Touloupis and all else is appropriately referenced.

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Time flies...

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Nomenclature

A	Cylinder wall area [m^2]
B	Bore [m] or Correction coefficient for Knudsen number [-]
c	Speed of light [$\text{m}\cdot\text{s}^{-1}$]
c_p	Specific heat capacity [J/K]
d	Diameter [m]
D	Diffusion coefficient [$\text{m}^2\cdot\text{s}^{-1}$]
E	Refractive index function [-]
f	Empirical Knudsen correction function [-]
h	Specific enthalpy [$\text{J}\cdot\text{kg}^{-1}$] or Planck constant [$\text{J}\cdot\text{s}$]
k_B	Boltzmann constant [$\text{J}\cdot\text{K}^{-1}$]
Kn	Knudsen number [-]
m	Mass [kg] or Complex refractive index [-]
M	Molecular mass [amu]
P	Pressure [Pa]
Q	Heat [J]
r	Radius [m]
R	Gas constant [$\text{J}\cdot\text{mol}^{-1}\cdot\text{K}^{-1}$] or Laser energy density [$\text{J}\cdot\text{m}^{-2}$]
S	Emission signal [V]
T	Temperature [K]
U	Internal energy [J]
w	Average cylinder gas velocity [$\text{m}\cdot\text{s}^{-1}$]

Greek symbols

α	Energy accommodation coefficient [-]
ε	Spectral emissivity [-]
κ	Thermal conductivity [$\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$]
λ	Wavelength [m] or Mean free path [m]
ρ	Density [$\text{kg}\cdot\text{m}^{-3}$]
σ	Stefan-Boltzmann constant [$\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-4}$]
φ	Equivalence Ratio [-]

Subscripts

0	Reference
10	Soot Particle Mean Diameter of 10 μm
2.5	Soot Particle Mean Diameter of 2.5 μm
abs	Absorption
B	Boltzmann
bb	Black-body
c	Continuum regime
$cond$	Conduction
g	Gas
h	Continuum heat flux
int	Internal
max	Maximum
p	Particle
rad	Radiation
S	Surface

<i>sub</i>	Sublimation
<i>T</i>	Translational
<i>tr</i>	Transition regime
<i>v</i>	Vapour
<i>w</i>	Wall

Abbreviations

AHTR	Apparent Heat Transfer Rate
aTDC	After Top Dead Centre
BSFC	Brake Specific Fuel Consumption
bTDC	Before Top Dead Centre
C ₂ [*]	Electronically excited dicarbon
C ₃	Tricarbon
CA10	Crank Angle where 10% of heat release has occurred
CA50	Crank Angle where 50% of heat release has occurred
CAD	Crank Angle Degree
CAHT	Cumulative Apparent Heat Transfer
CH	Methylidyne radical
CH [*]	Electronically excited methylidyne
CH ₂	Methylene
CI	Compression Ignition
CMOS	Complementary Metal-Oxide Semiconductor
CN	Cetane Number
CO	Carbon Monoxide
CO ₂	Carbon Dioxide

CWL	Central Wavelength
DI	Direct Injection
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation
EOC	End of Combustion
EOI	End of Injection
FACE	Fuels for Advanced Combustion Engines
FAME	Fatty Acid Methyl Ester
FWHM	Full Width at Half Maximum
GCI	Gasoline Compression Ignition
HCCI	Homogeneous Charge Compression Ignition
HD	Heavy-Duty
IMEP	Indicated Mean Effective Pressure
IR	Infrared
LD	Light-Duty
LIF	Laser Induced Fluorescence
LII	Laser Induced Incandescence
LTC	Low Temperature Combustion
Nd:YAG	Neodymium-doped Yttrium Aluminum Garnet
NO _x	Nitrous Oxides
NVO	Negative Valve Overlapping
OH*	Electronically excited hydroxyl
PIV	Particle Image Velocimetry
pLIF	Planar Laser Induced Fluorescence
pLII	Planar Laser Induced Incandescence

PM	Particulate Matter
PMT	Photomultiplier Tube
POD	Proper Orthogonal Decomposition
PPC	Partially Premixed Combustion
PPCI	Partially Premixed Compression Ignition
PRF	Primary Reference Fuel
PRR	Pressure Rise Rate
RAM	Random Access Memory
RPM	Rotations <i>Per</i> Minute
SD	Standard Deviation
SI	Spark Ignition
SINL	Spatially Integrated Natural Luminosity
SOC	Start of Combustion
SOI	Start of Injection
TC	Two-Colour
TDC	Top Dead Centre
TiRe	Time-Resolved
UHC	Unburned Hydrocarbons
UV	Ultra-Violet
WHO	World Health Organization

1 Introduction

1.1 Motivation

Compression ignition engines, commonly known as Diesel engines, account for a large percentage of applications in the transport sector, either as light-duty engines, found in passenger vehicles or as heavy-duty engines found in commercial vehicles as well as rail and marine applications. Predictions show that the demand for diesel fuel distillates will increase over the next decades, as seen in Figure 1.1. The reason behind compression ignition engine's wide range of applications can be located, mainly, in the high level of thermal efficiency and hence low fuel consumption, the capability of producing high torque in a wide range of operating conditions and the lower maintenance costs *per* operating hour compared to spark ignition (SI) engines. The main

drawback of the compression ignition engine is the exhaust emissions of particulate matter (PM) and nitrous oxides (NO_x), stemming from the nature of the combustion in this type of engine.

According to the World Health Organization (WHO), PM is an air pollutant consisting of a mixture of solid and liquid carbon particles suspended in the air. PM is commonly classified based on the mean diameter of the particles, with PM₁₀ referring to particles with mean diameter less than 10 µm and PM_{2.5} referring to particles with mean diameter less than 2.5 µm. The health effects of both PM classes are well documented and studies show that short-term exposure may lead to asthma, respiratory symptoms and an increase in hospital admissions, while long term exposure may lead all the way to mortality from cardiovascular and respiratory diseases and from lung cancer (World Health Organization, 2013). It is noted here that PM

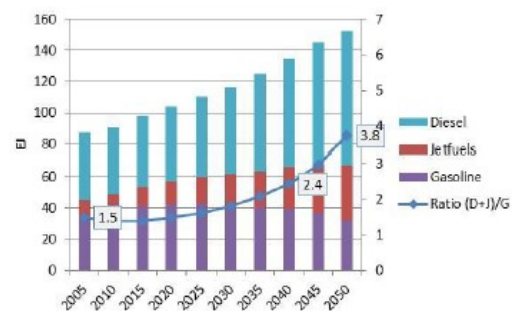


Figure 1.1 Distillates (diesel and jet) demand will increase. (World Energy Council and ExxonMobil Outlook 2012; obtained from Aramco, SAE HEE Symp, 4/13)

emissions, especially the $PM_{2.5}$ class, can also have significant impact on climate change, second only to carbon dioxide (CO_2) emissions (Bond *et al.*, 2013).

It becomes evident that there is a need to restrict PM emissions globally, whatever their source, and especially PM coming from commercial and passenger vehicles, because of the proximity and the daily exposure of human beings to such vehicles. Since the early nineties, both Europe and the US have established legislative authorities appointed to issue directives and rules on the permissible pollutant emissions from passenger and commercial vehicles, which aim to improving the air quality and ultimately, the life quality of individuals. An example of such legislation in Europe (the Euro emission standard), is presented in Figure 1.2. Apparently, there is a gradual reduction in permissible levels of PM and NO_x for passenger vehicles and, therefore, compliance with these limits is one of the main technical challenges for the motor industry.

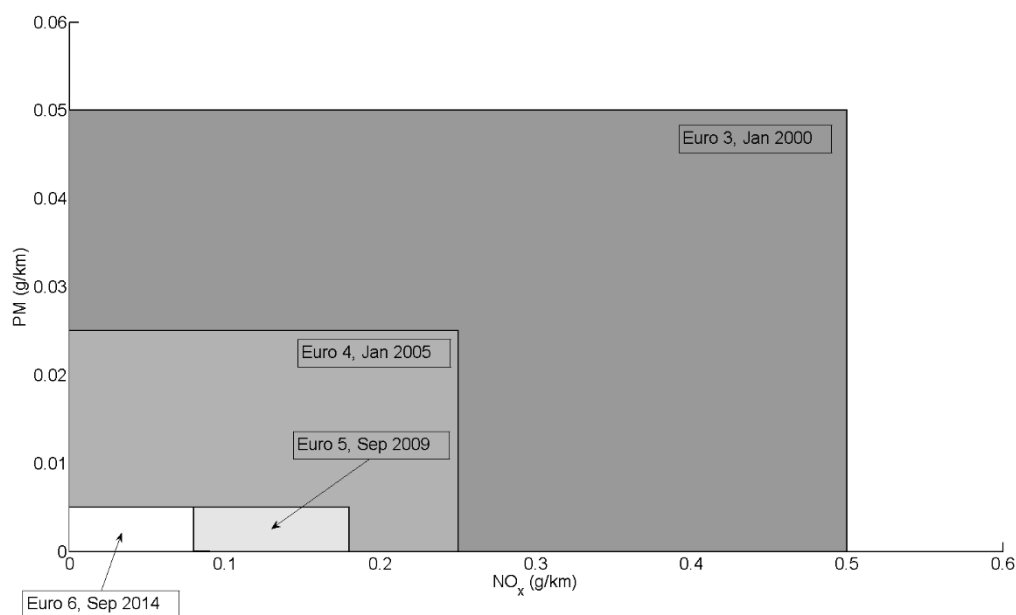


Figure 1.2 Tightening of European legislation on diesel passenger vehicles (Category M) of PM and NO_x (Source: European Commission)

In a compression ignition (CI) engine, the fuel is injected directly into the combustion chamber, using a high-pressure injector, where the air-fuel mixture is auto-ignited after being compressed by the piston. The nature of the conventional compression ignition combustion is such, that there is little time available between the injection of the fuel and the auto-ignition, not allowing for complete mixing of the fuel with the surrounding air. This leads to formation of areas which

are characterized by high values of fuel-to-air ratio, or, in other words, areas that are fuel-rich. Combustion of these fuel-rich areas in a high-temperature, high-pressure environment typically leads to carbon black (smoke/soot) emissions formation, which account for the largest part of PM emissions.

There are two ways to limit soot emission from a compression ignition engine; one way is to use exhaust after-treatment devices, such as Diesel Particulate Filters (DPFs), to capture the soot escaping the combustion chamber. Such devices, however, are able to capture only soot particles which are larger in size, letting the small particles escape into the atmosphere, and usually get blocked by soot deposits quite quickly, requiring frequent regeneration or cleaning. Alternatively, soot emissions can be reduced aborning, within the combustion chamber, by gaining better understanding of the in-cylinder soot formation and oxidation processes and by employing operation strategies which inhibit engine-out particulate emissions. This would have significantly beneficial effects not only on air quality and human health, but also with respect to minimization of the need for after-treatment in production vehicles (*e.g.* Li *et al.* 2015). Soot production in a diesel engine can be correlated to the ratio of the durations of premixed-burn to diffusion-burn (or mixing controlled) phases, with the diffusion regime being responsible for a greater portion of the soot

produced (see Figure 1.3). Higher temperatures and higher local equivalence ratio (ϕ) inside the cylinder should be avoided, otherwise some of the injected fuel will locally form fuel-rich “pockets” among the charge, prone to soot formation (*e.g.* Koci *et al.* 2009). A partially premixed combustion (PPC) operation mode may in principle exhibit suitable fuel/air mixing, limiting the conditions

susceptible to soot formation and thus has the potential to lead to reduced soot emissions (*e.g.* Musculus *et al.* 2013). The investigation of the mechanisms behind soot reduction stemming from the adoption of the PPC operation strategy is the springboard for the research conducted within the framework of the present thesis.

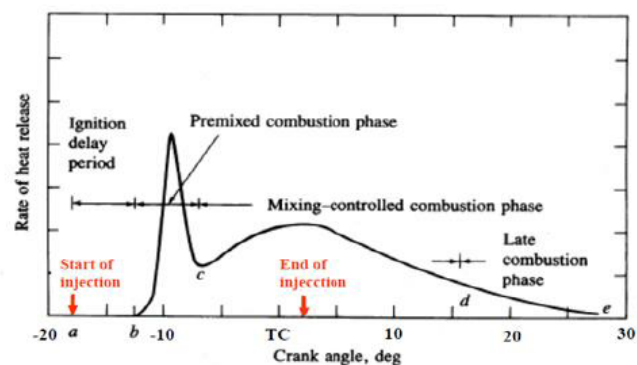


Figure 1.3 Combustion phasing in a heat release curve of a light-duty compression ignition engine (M. Salmani *et al.* 2015).

1.2 Thesis outline

Chapter 1 introduces the subject matter and contains a literature review for the compression ignition combustion strategies, as well as the optical research engines and the optical techniques used for the purposes of the present thesis. Chapter 2 describes the experimental apparatus and the methodology used for the experimental work.

Chapters 3 and 4 present and discuss the experimental work on the Ricardo Hydra and the Volvo D5 optical research engines respectively. Each chapter consists of independent experimental campaigns, with separate literature review, analysis of the acquired data and discussion on the results for the topic of each campaign. The aforementioned experimental campaigns focus on assessing the effect of various engine operation strategies on combustion characteristics and how effective they are on reducing the in-cylinder soot concentration.

The study is concluded in Chapter 5, with a general discussion on the obtained results and the motivation for future work, as occurs from the work conducted for the present thesis.

1.3 Compression Ignition Combustion Strategies

Engine technology is evolving in response to increasingly stringent emission limits imposed by regulatory authorities across the world (*e.g.* Johnson 2015). Compression ignition engine research focuses on the development of novel operating modes, such as Low Temperature Combustion (LTC), that have been found to reduce NO_x and to inhibit particulate emissions (*e.g.* Dec 2009; Reitz and Duraisamy 2015). The basic principle of the LTC mode lies in the reduced in-cylinder temperatures, mainly due to extensive use of exhaust gas recirculation (EGR), and in enhanced premixing, mainly due to the introduction of advanced injection schemes, in order to avoid, to a greater or lesser extent, conditions pertinent to the soot and NO_x formation regimes identified in the equivalence ratio-temperature (ϕ - T) space (see Figure 1.4).

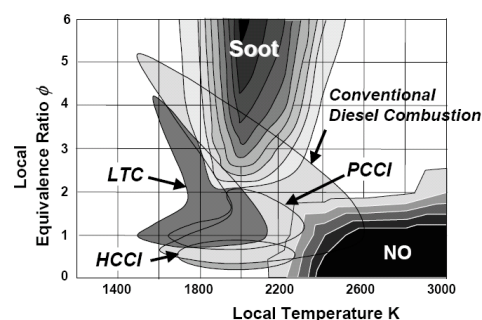


Figure 1.4 Modern diesel combustion strategies plotted in the ϕ - T space (Koci *et al.* 2009)

During conventional compression ignition operation, the main amount of the fuel is injected shortly before, or even after top dead centre (TDC) and this, combined with the high compression ratio, results in auto-ignition of the in-cylinder charge before the end of the fuel injection. Part of the fuel is injected after the start of combustion (SOC) and enters directly into a diffusion-burn (mixing controlled) phase, without having the time to mix with the surrounding air, forming, thus, local fuel-rich areas. Usually, these fuel-rich areas are characterized by high equivalence ratio (ϕ) values, which, combined with the high ambient temperature due to the ongoing combustion, results in soot formation (see Figure 1.4).

Efforts to implement the LTC concept in production engines have led to the technological development of new combustion strategies, such as the Homogenous Charge Compression Ignition (HCCI) and the Partially Premixed Combustion (PPC) (*e.g.* Stanglmaier and Roberts 1999; Bessonette *et al.* 2011; Yao *et al.* 2009; Saxena and Bedoya 2013). During HCCI operation standard port injection is utilized (like in SI engines) and the mixture of air and fuel is introduced inside the cylinder, usually *via* a preheating chamber. The mixture is homogeneous and lean ($\phi < 1$), therefore fuel-rich areas do not exist and thus, soot formation is suppressed to a very large extent. Additionally, the introduction of EGR lowers the combustion temperature, leading to lower NO_x formation inside the cylinder. The problem with HCCI lies to the auto-ignition and combustion of lean mixtures. The auto-ignition of lean mixtures is highly sensitive to temperature and therefore difficult to control, while the lean combustion leads to unacceptably high emissions of unburned hydrocarbons (UHC) and CO, and, at higher loads, significantly increased noise levels, due to high pressure rise rates (*e.g.* Epping *et al.* , 2002).

PPC operation can be thought of as a hybrid mode between HCCI and conventional CI combustion; the fuel is injected early enough in the compression stroke so as to provide more time than conventional CI for pre-mixing before ignition, while retaining a much greater degree of control over the ignition timing than HCCI does. By manipulating the timing of the fuel injection, the PPC operation can have different characteristics; a fuel injection at the beginning of the compression stroke leads to a fully Premixed Charge Compression Ignition (PCCI) operation, resembling HCCI operation, while a fuel injection later at the compression stroke will give the subsequent combustion characteristics that resemble a conventional CI operation.

The implementation of higher injection pressures in order partly to accelerate the atomization process and partly to increase turbulent mixing, together with a multiple injection strategy

which can help towards achieving the right level of fuel stratification inside the cylinder, is a common means for achieving a PPC operation mode (C. K. H. Hong, 2015). Higher injection pressures and multiple injection strategies, together with copious EGR, enhance premixing and act against the establishment of a flame. Depending on the injected fuel amount, the number of pre-injections, and the sequence of pilot(s) and main injections employed, the in-cylinder processes diverge from conventional diesel flames, which are collectively characterized by high temperatures and sooty behaviour, and shift towards LTC regimes featuring enhanced premixing and moderate heat release rates (Reitz, 2013). Contrary to HCCI, PPC allows for auto-ignition at low loads with high efficiency and low NO_x and PM emissions (*e.g.* Leermakers and Luijten 2011; Kalghatgi *et al.* 2013). However, UHC and CO emissions may still be present, depending on the global equivalence ratio of the mixture, combined with noise levels which are increased compared to conventional diesel combustion.

A PPC operation mode may in principle exhibit suitable fuel/air mixing, limiting the formation of conditions susceptible to soot formation and thus has the potential to lead to reduced soot emissions (*e.g.* Musculus *et al.* 2013).

1.4 Optical Research Engines

Optical research engines are reciprocating internal combustion engines, which have been modified to facilitate optical access inside the combustion chamber for the study of fuel sprays, in-cylinder charge formation, charge ignition, combustion evolution and pollutant formation. There are two main types of optical engines; the first one is based on endoscopic access inside the combustion chamber, while the second one is based on optical access through an optical window, embedded in the engine piston.

The endoscopic optical engines (see Figure 1.5) allow optical access through a small optical window, usually located in the cylinder head and a through-and-through hole to place the endoscope. The design and materials of the window are such to withstand pressure and temperature at full load engine operation. Although this type of optical engine offers robustness when it comes to engine operation, it offers limited optical access, because of the small size of the window.

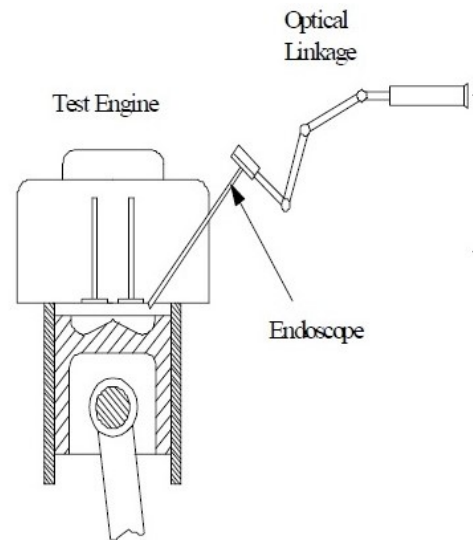


Figure 1.5 Schematic of an endoscopic optical research engine

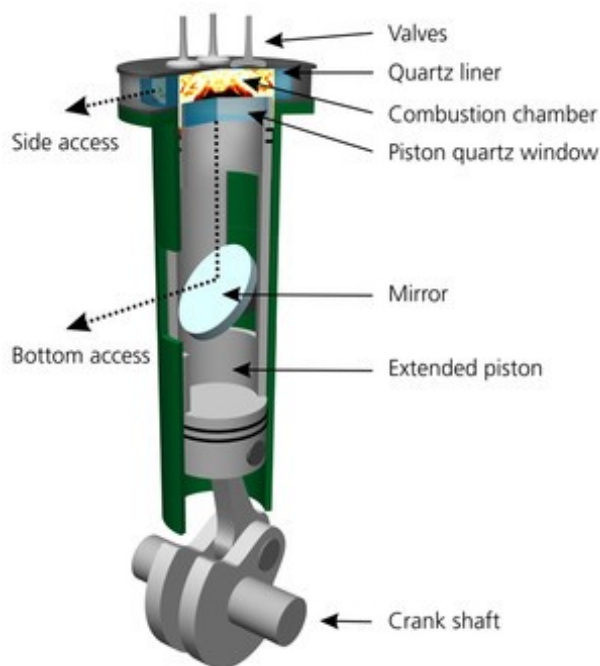


Figure 1.6 Schematic of a Bowditch-design optical engine.

The second type of optical engines feature a Bowditch design (Bowditch, 1961), where the engine block and cylinder head are separated and the original piston is elongated using two connecting rods, which link the metal and the optical piston (see Figure 1.6). An optical cylinder liner is usually employed to allow side optical access inside the combustion chamber. A 45° tilted elliptical mirror is placed between the connecting rods to reflect the inside of the combustion chamber. This type of optical engines offer wider optical access, however they suffer from a number of issues, with the most common being sensitivity issues of the

optical parts, issues stemming from different heat conduction behaviour because of the materials used to manufacture the piston and the cylinder liner, reduced compression ratio and increased blow-by (Kashdan & Thirouard, 2011). The aforementioned issues limit the range of engine operation to medium speed and medium load and require further actions to limit their

effect on the combustion characteristics, such as use of skip firing to avoid excessive thermal stress of the optical parts as well as introduction of boosted and heated air into the cylinder to compensate for the reduced compression ratio and the extensive heat losses. Nevertheless, the optical access offered by these research engines is unmatched and thus, this is the type of optical engines used for the purpose of the present thesis.

1.5 Optical Diagnostics

Throughout the years of studying combustion in optical engines, numerous optical diagnostic techniques have been employed, with the most common being Shadowgraphy, Schlieren Imaging, Mie and Rayleigh Scattering, Chemiluminescence, Laser Induced Fluorescence (LIF), Laser Induced Incandescence (LII) and Particle Image Velocimetry (PIV) (H. Zhao, 2010, Chapter 16). For the optical diagnostics employed for the purposes of the present thesis, more details are given below.

1.5.1 Natural luminosity of the combustion flame

In a compression ignition engine, the emitted light from the flame contains a wide range of wavelengths, from the infrared (IR) to the ultra-violet (UV) part of the spectrum. However, most of the luminosity of the flame stems from the incandescence of the soot particles, as they are heated during combustion. Consequently, the natural luminosity of the flame can be correlated with soot volume fraction in a flame, however, care must be taken when interpreting the natural luminosity signal to soot volume fraction (Hessel *et al.* , 2017; Mueller & Martin, 2002).

1.5.2 Species Chemiluminescence

Chemiluminescence is the emission of light (luminescence), as the result of an exothermic chemical reaction. The emission of light occurs when part of the heat from the exothermic reaction is translated into excitations of the electronic state of intermediate chemical species (marked by the symbol $*$), which then decay to into an electronic ground state and emit photons (light). Combustion is an oxidation (thus exothermic) process, *i.e.* a reaction of the fuel molecules with oxygen, triggered by an increase of the energy of the reactants (in the case of

compression ignition engines this energy is transferred to the reactants *via* the compression of the in-cylinder charge), in order to overcome the activation energy barrier. As an exothermic process, combustion produces several intermediate chemical species which have chemiluminescent properties and emit light at different wavelengths. The most common and easily detectable chemiluminescent species, due to their narrowband and intense light emission are the OH^* , CH^* and C_2^* species.

Each of the chemiluminescent species is a proxy of a fundamental combustion property. OH^* has been found to be a good proxy of the heat release zone and the flame lift-off location during second-stage combustion and is used to determine spatial and temporal ignition. CH^* is related to the reaction zone and the flame front, while C_2^* is indicative of fuel-rich reactive areas. The C_2 and C_2H radicals are the precursor species for the CH^* radicals (Bozkurt *et al.*, 2012) and are formed within a narrow reaction zone with high temperature gradients. The short-lived CH^* radicals are hence considered as a proxy for the flame front and can also be used to support arguments on the reactivity. On the other hand, CH_2 and C_3 species play a prominent role on the formation of the excited C_2^* species, which become more intense in the locally formed fuel-rich pockets. Therefore, C_2^* chemiluminescence is correlated to the reacting fuel-rich zones (*e.g.* Kohse-Höinghaus 1994; Hultqvist *et al.* 1999; Kathrotia *et al.* 2012). OH^* is widely used as a second-stage heat release zone indicator and thus can be used to characterize the level of combustion stratification (*e.g.* I. Najafabadi 2017; M. I. Najafabadi *et al.* 2017). Excited OH^* species are mostly formed through reaction of molecular oxygen with CH radicals (*e.g.* Panoutsos, Hardalupas, and Taylor 2009; Kathrotia *et al.* 2012) and are not, in general at least, a proxy of OH radicals.

It is important to note at this point that, when using optical filters for chemiluminescence detection, the captured signal contains interference by other sources, mainly broadband CO_2^* chemiluminescence and thermal radiation, stemming from black-body radiation of soot particles which are heated to flame temperatures during the combustion (M. I. Najafabadi *et al.*, 2017). This interference may affect the observed trends, even when the chemiluminescence signal is used to describe combustion behaviour qualitatively. In order to enhance the validity of the observed trends when using chemiluminescence measurements, it is advisable to remove the background interference, as described in M. I. Najafabadi *et al.*, 2017. In the present thesis an attempt to remove the background interference has been made, following the methodology described in Section 2.4.2. The results of this work are presented in Section 4.2.

1.5.3 Laser Induced Incandescence

Laser Induced Incandescence is an optical technique which enables the detection of soot particles during and after the combustion process. The technique is based on the measurement of the black body radiation emitted by soot particles when heated up by a source of energy, most commonly a laser. LII signal stems from the blue-shifted light emitted from soot particles as they are heated up to a temperature much higher than that of the surrounding gases.

The LII technique can be used either qualitatively to investigate the trends in signal intensity and distribution, or quantitatively, using the appropriate calibration method, to measure soot volume fraction and soot primary particle sizes. Soot volume fraction is proportional to both the integrated signal and the peak signal (*e.g.* Melton 1984), while the signal decay rate provides information about the size of the primary particles.

An overview of the research on LII and its applications has been conducted by a number of researchers (Burtscher 2005; Tropea *et al.* 2007; Zhao 2010; Skeen *et al.* 2016; Michelsen 2017).

1.5.3.1 LII theoretical model

Weeks and Duley in 1974 first noticed that laser pulses can be used in diagnostic studies of aerosols. Melton in 1984 was the first researcher to come up with the mass and energy balance equation for quantitative soot measurements. Through the years, different models were developed by researchers, varying mainly in the ambient temperature and pressure at which they can be applied (Bladh and Bengtsson, 2004; Dreier *et al.* , 2005; Hofeldt, 1993; Michelsen *et al.* , 2003; Snelling *et al.* , 2000; Vander Wal and Weiland, 1994; Will *et al.* , 1998). H. A. Michelsen *et al.* in 2007 reviewed and compared the aforementioned models, while Dreier *et al.* in 2005 reviewed models which are applicable for high pressure environments. The model presented indicatively below is from Kock *et al.* 2006, which applies to high pressure environment. This model was also used by Boiarciuc *et al.* in 2007 for in-cylinder soot modelling in a diesel engine.

The energy balance equation of a soot particle which is heated up by a laser source is shown in equation 1.1.

$$\dot{Q}_{int} = \dot{Q}_{abs} - \dot{Q}_{sub} - \dot{Q}_{cond} - \dot{Q}_{rad} \quad (1.1)$$

In the equation above $\dot{Q}_{int}$ is the rate of internal energy change of the particle, $\dot{Q}_{abs}$ is the rate of laser energy absorption, $\dot{Q}_{sub}$ is the rate of heat loss due to sublimation, $\dot{Q}_{cond}$ is the rate of heat loss through conduction and $\dot{Q}_{rad}$ is the rate of heat loss through radiation. Energy fluxes due to particle annealing, oxidation, or thermionic emission of electrons are neglected.

The rate of internal energy change $\dot{Q}_{int}$ is given by equation 1.2 below:

$$\dot{Q}_{int} = \frac{d}{dt} (m_p c_p T_p) \quad (1.2)$$

$$\text{where} \quad m_p = \frac{\pi d_p^3}{6} \rho_p \quad (1.3)$$

For the heating phase of the particle it is important to determine the temperature T_p^0 that the particle reaches when it is heated up by the laser. For this reason, the particle is considered spherical and thus, the absorption of electromagnetic radiation is described by the Mie theory, as shown in equation 1.4.

$$T_p^0 = T_g - \frac{6\pi R_0}{\rho_p c_p \lambda_{Laser}} \text{Im} \left(\frac{m^2 - 1}{m^2 + 2} \right) \quad (1.4)$$

In equation 1.4 R_0 is the laser energy density, T_g is the ambient gas temperature, ρ_p is the particle density, c_p is the particle heat capacity and λ_{Laser} is the wavelength of the laser. The key parameter in equation 1.4 is m , which represents the complex refractive index. For nanoparticles, there are no reliable values for m in the literature and this makes the calculation of T_p^0 using equation 1.4 tricky.

In order to avoid using the complex refractive index of soot, two-colour pyrometry can be employed to determine T_p^0 . In two-colour pyrometry, the particle emission signals S_p is recorded at two different wavelengths, λ_1 and λ_2 , and the particle temperature is calculated using equation 1.5 below.

$$T_p = \frac{hc_0}{k_B} \left(\frac{1}{\lambda_2} - \frac{1}{\lambda_1} \right) \times \left\{ \ln \left[\frac{S_p(\lambda_1, T_p)}{S_p(\lambda_2, T_p)} \frac{S_{bb}(\lambda_2, T_{bb})}{S_{bb}(\lambda_1, T_{bb})} \frac{\varepsilon_p(\lambda_2)}{\varepsilon_p(\lambda_1)} \right] + \frac{hc_0}{k_B T_{bb}} \left(\frac{1}{\lambda_2} - \frac{1}{\lambda_1} \right) \right\}^{-1} \quad (1.5)$$

In equation 1.5 h is the Plank constant, c_0 is the speed of light, k_B is the Boltzmann constant, S_{bb} and T_{bb} are the measured signal and temperature from the calibration process using a black-body source and ε_p is the particle spectral emissivity. For small particles in the Rayleigh limit the following simplification can be made:

$$\frac{\varepsilon_p(\lambda_2)}{\varepsilon_p(\lambda_1)} \approx \frac{\lambda_1}{\lambda_2} \quad (1.6)$$

The rest of the terms in equation 1.1 are responsible for the cooling of the particle and can be calculated using different models. These models can be classified by the Knudsen number (Kn), which is the ratio of the mean free path of the ambient gas molecules (λ_g) to the particle radius (r_p). For diesel engines the Knudsen number is around one ($Kn \cong 1$), which means that the heat and mass transfer are within the so-called transition region (B. F. Kock *et al.* , 2002).

In the transition region the conduction term can be expressed as in equation 1.7:

$$\dot{Q}_{cond,tr} = \dot{Q}_{cond,c} f_h(Kn_h) = 2\pi d_p \kappa_{cond} (T_p - T_g) f_h(Kn_h) \quad (1.7)$$

In the equation above κ_{cond} is the thermal conductivity of the gas, while the subscript tr indicates the transition regime and c the continuum regime, f_h is an empirical function correcting the Knudsen affected continuum heat flux, which is shown in equation 1.8, while Kn_h is defined in equation 1.9.

$$f_h(Kn_h) = \left[1 + Kn_h \frac{1.9234A + 1.3026}{1.9234Kn_h + 1} \right]^{-1} \quad (1.8)$$

$$\text{with} \quad A = \frac{\dot{Q}_{cond,c}}{\dot{Q}_{cond,fm}} = 5\sqrt{\pi} / \alpha_T \left(\frac{\gamma-1}{\gamma+1} \right) Kn_h \quad (1.9)$$

where γ is the heat capacity ratio and α_T is the translational energy accommodation coefficient, which is in the region of 0.07 and 1.00 (Kock *et al.*, 2005).

$$Kn_h = \frac{4\lambda_{cond}T_g}{5r_p p_g} \left(\frac{M_g}{2k_B T_g} \right)^{1/2} \quad (1.10)$$

In equation 1.10 p_g and M_g are the ambient gas pressure and molecular mass respectively. The sublimation term is given by equation 1.11:

$$\dot{Q}_{sub} = \dot{U}_{sub} \Delta h_v \quad (1.11)$$

In equation 1.11 Δh_v is the evaporation enthalpy of the particles, which, using the Clausius-Clapeyron equation was found to be 21.946 kJkg⁻¹ and $\dot{U}_{sub}$ can be expressed in the transient phase by equation 1.12 below.

$$\dot{U}_{sub} = \dot{U}_{sub,c} f_c(Kn_c) = 2\pi d_p D (\rho_s - \rho_\infty) f_c(Kn_c) \quad (1.12)$$

In equation 1.12 D is the diffusion coefficient of the vapour into the gas, ρ_s and ρ_∞ are the vapour density of the carbon at the particle surface and at infinity respectively, with the latter being considered negligible.

Just like in the calculation of the conduction term, the function f_c is used to correct the sublimation rates under continuum conditions, due to Knudsen effect, only this time for particles in vapour phase. The function is shown in equation 1.13 and the Knudsen number for vapour phase is shown in equation 1.15.

$$f_c(Kn_c) = \left[1 + Kn_c \frac{1.3333B + 1.0161}{1.3333Kn_c + 1} \right]^{-1} \quad (1.13)$$

$$\text{with} \quad B = \frac{\dot{U}_{sub,c}}{\dot{U}_{sub,fm}} = Kn_c \sqrt{\pi} / \alpha_v \quad (1.14)$$

where α_v is the accommodation coefficient for evaporation used to account for non-equilibrium evaporation kinetics and real gas effects.

$$Kn_c = \frac{2D}{r_p} \left(\frac{M_v}{2k_B T_v} \right)^{1/2} \quad (1.15)$$

Here, M_v denotes the molecular mass of the vapor and T_v is the vapor temperature, which is assumed to be equal to the particle temperature T_p .

Finally, the radiation heat flux is given in equation 1.14, where ε_{pt} is the total emissivity of the particle and σ is the Stefan-Boltzmann constant. The total emissivity can be assumed equal to one ($\varepsilon_{pt} \cong 1$). We should note here that the heat loss due to radiation is minor, compared to conduction and sublimation heat losses.

$$\dot{Q}_{rad} = \pi d_p^2 \varepsilon_{pt} \sigma (T_p^4 - T_g^4) \quad (1.16)$$

Equations 1.1, 1.2, 1.7, 1.11 and 1.16 are coupled and must be numerically integrated in order to obtain the time dependent temperature and diameter of a spherical carbon particle under a particular excitation wavelength.

Although the radiation term plays only a marginal role in the energy balance of the particle, it is responsible for the emitted LII signal. The signal generated by a single particle in the detection volume, according to the Planck function integrated over all solid angles is given by equation 1.17 (Michelsen *et al.*, 2007).

$$S = \varepsilon_{\lambda_s} \frac{2\pi^2 D^2 h c^2}{\lambda_s^5 \left[\exp\left(\frac{hc}{\lambda_s k_B T}\right) - 1 \right]} \quad (1.17)$$

In equation 1.17 S is the signal from the LII, D is the diameter of the particle, h is the Planck constant, c is the speed of light, λ_s is the emission wavelength, k_B is the Boltzmann constant and T is the temperature of the particle.

The emissivity is assumed to be

$$\varepsilon_{\lambda_s} = \frac{4\pi DE(m)}{\lambda_s} \quad (1.18)$$

which yields

$$S = \frac{8\pi^3 D^3 hc^2 E(m)}{\lambda_s^6 \left[\exp\left(\frac{hc}{\lambda_s k_B T}\right) - 1 \right]} \quad (1.19)$$

The function $E(m)$ found in the equations above is the refractive index function. Indicative values for this function, when using a 532 nm excitation and a 500 nm detection, can be found in (Michelsen *et al.* , 2007).

1.5.3.2 Variations of the LII technique

There are several variation of the LII technique which can be applied to investigate soot formation and distribution during combustion. The variations can be characterised as spatially resolved (planar method) or time resolved and as single wavelength detection or double wavelength detection (two-colour method). The combination of these characteristics results to the variations of the LII technique. The most commonly used techniques are described below.

Planar LII

The planar LII is a spatially resolved technique which employs a laser sheet, passing through the volume of interest in order to excite the soot particles. The laser sheet defines the measuring plane. The LII signal is recorded by an intensified CCD camera. The advantage of the planar technique is that it provides measurements of the spatial distribution of soot within the area of interest. When the LII signal is calibrated to a known source, the technique can yield quantitative information about the soot volume fraction within the area of interest. This technique has been used by a large number of researchers, as it is easier to implement and it provides valuable information about the spatial distribution of the soot particles (Cenker *et al.* , 2013; Crua *et al.* , 2003; J E Dec & Espey, 1992; J E Dec *et al.* , 1991; Hardalupas *et al.* , 2016; C. Hong *et al.* , 2015; C. A. Idicheria and Pickett, 2011; Kook and Pickett, 2012; B. Menkiel, 2012; Morgan *et al.* , 2003; O'Connor & Musculus, 2014; Singh *et al.* , 2007).

Special consideration has to be given to the choice of the detection wavelength, by using the appropriate optical filter to avoid interference with excited species such as OH^* , CH^* and C_2^* .

An example of such a filter can be found in Miles *et al.* 2007, where a short-pass filter with a central wavelength (CWL) of 450 nm was used.

Two-Colour Time-Resolved LII (TC-TiRe LII)

This technique relies on the fact that soot particles, when heated up, radiate due to high temperature, in a black-body manner. According to Planck's law, the wavelengths emitted vary with temperature and particle size and this technique takes advantage of this property.

The LII signal is recorded using a probe in the volume of interest and then filtered using two different short-pass filters, usually one in the blue part of the spectrum and one in the red part of spectrum. The two signals are recorded by two photomultiplier tubes (PMT). The ratio between the signals is used to deduce the temperature of the soot particles at the tip of the probe. Additionally, the decay rate of the signals can provide information about the primary soot particle size. This technique has been successfully applied by some research groups and gave promising results (Axelsson *et al.* , 2000; B.F. Kock *et al.* , 2002; Lehre *et al.* , 2003; Menkiel *et al.* , 2012; Wal *et al.* , 1999; S Will *et al.* , 1998; Will *et al.* , 1996; Woiki *et al.* , 2000).

The main benefit of this technique is that it is self-calibrating, as it requires no calibration against a known source to produce quantitative results, unlike the case of the planar LII. The disadvantage of the technique lies to the fact that the probe volume is small and thus unable to provide information about the soot distribution in a larger volume, such as the engine cylinder.

The combination of pLII and TC-TiRe LII can help overcome the drawbacks of both methods, as the TC-TiRe LII can provide information for the temperature and the soot volume fraction for a small probe volume, which can then be translated to the whole measurement plane of the pLII (Boiarciuc *et al.* , 2006; Boiarciuc *et al.* , 2007).

1.5.3.3 LII applications in the literature

The LII technique and its variations as described above are applied in a wide range of experimental work, when characterization of the distribution, the volume fraction and the size of soot particles is required. Since Weeks and Duley in 1974 first observed the heating up of soot particles with a CO₂ laser, the technique has been applied in both laminar and turbulent flames (Appel *et al.* , 1996; Axelsson *et al.* , 2000, 2001; Bladh *et al.* , 2008; Bockhorn *et al.* ,

2002; Bryce *et al.* , 2000; Geitlinger *et al.* , 1998; Meyer *et al.* , 2005; Ni *et al.* , 1995; Quay *et al.* , 1994; Shaddix and Smyth, 1996; Vander Wal and Weiland, 1994; Witze *et al.* , 2001; H. Zhao and Ladommatos, 1998), constant volume combustion vessels (Cenker *et al.* , 2015; Cenker *et al.* , 2015; Idicheria and Pickett, 2011; Idicheria and Pickett, 2006; Idicheria and Pickett, 2005; Vogel *et al.* , 2013) as well as in-cylinder measurements (Aronsson *et al.* , 2010; Boiarciuc *et al.* , 2007; Bougie *et al.* , 2006; Chartier *et al.* , 2009; Dec and Espey, 1992; Dec *et al.* , 1991; Kock *et al.* , 2002; Lilik *et al.* , 2015; Menkiel *et al.* , 2012; Miles *et al.* , 2007; Musculus, 2006; O'Connor and Musculus, 2014; SINGH *et al.* , 2007). The following literature review on the LII applications is indicative and focuses on highlighting the findings which are more relative to the content of this thesis.

Melton in 1984 investigated two strategies to detect soot particles; one based on the laser modulated incandescence of soot particles and the other based on the evaporation of C₂ molecules. The author concluded that both strategies can yield quantitative information about soot particles, such as size distribution and volume fraction, provided the laser wavelength is near the graphite absorption band. Melton's theoretical model on the LII technique based on the energy balance of a single soot particle was the cornerstone for the development of the method.

One of the first applications of the LII technique in flames was by Quay *et al.* in 1994, who used the fundamental wavelength of an Nd:YAG laser at 1064 nm to study soot distribution in a co-annular laminar diffusion flame burner. It was observed that the LII signal increased non-linearly with laser fluence and that it featured a profile with a rapid rise followed by a long decay period at the order of a few hundred nanoseconds, which corresponded to the cooling phase of the heated particles, while it was mentioned that the used laser pulse was too long to use it for particle sizing. Along the same lines, Ni *et al.* in 1995 used the same experimental setup with a frequency-doubled Nd:YAG laser at 532 nm, firing at a laminar diffusion flame, to come to the conclusion that when the soot particles were smaller in size, the detected LII signal decayed faster. It was pointed out that the soot volume fraction could be underestimated in areas where the mean soot particles diameter is small, although this biased measurements could be avoided with the use of short gate times immediately after the laser pulse. Shaddix and Smyth in 1996 studied a laminar diffusion flame as well, however in their experiment, the flame was flickering. The flickering flame resulted in larger peak soot particle diameters, compared to the particles of the steady flame and this demonstrated the influence of radiative

cooling on soot oxidation and breakout. Another study in a laminar premixed flame was conducted by Axelsson, Collin, and Bengtsson in 2000, using the 532 nm wavelength of an Nd:YAG laser on a McKenna burner. The researchers combined the LII technique with scattering and extinction measurements to conclude that the primary soot particle size increases with the vertical distance from the burner, due to agglomeration. However, errors in the particle size calculations emanated from uncertainties in the ambient gas temperature.

Because the combustion inside a compression ignition engine is turbulent, the successful application of the LII technique in turbulent flames opened the way for the application of the technique in research engines. One of the first documented attempts of studying a turbulent flame using the LII technique was that of Ni *et al.* in 1995, where a freefalling burning droplet was used to create a turbulent flame. In this work it was confirmed that the technique could indeed be applied in such flames. Following this first attempt, Geitlinger *et al.* in 1998 compared soot volume fractions and particle densities between a laminar diffusion flame and a turbulent diffusion flame, using an Nd:YAG laser at 532 nm and the RAYLIX technique. The results revealed that both the laminar and the turbulent flames exhibited the same behaviour between soot volume fraction and soot particle density and that the soot formation timescales are smaller than the turbulent timescales and thus, flamelet modelling could be used. The same research group in 2002 (Bockhorn *et al.* , 2002) took their previous work a step further and showed that correlations between soot volume fraction, number density, and particle size found in laminar flames exist in turbulent flames as well. This was a basic finding indicating that in turbulent flames soot formation and oxidation are taking place in laminar flame filaments. Additionally, in this work it was mentioned that the spectra of soot volume fractions in the turbulent flame showed that the large scales were dominated by the flow, while structures smaller than 1mm were affected by chemical reactions. Another important remark on soot formation in turbulent flames was made by Meyer *et al.* in 2005, who applied simultaneously pLII and OH pLIF in a swirl stabilized combustor to study the flame structure and its effect on soot formation. The results showed that soot formation originated from fuel-rich premixed regions of the flame where the OH pLIF signal was low.

Taking the fundamental research on soot formation a step further, a number of researchers have applied the LII technique to study combusting fuel sprays inside constant volume vessels, under engine-like conditions. Cherian A. Idicheria and Pickett in 2005 studied the behaviour of a diesel-like spray under high EGR conditions inside an optically accessible constant volume

vessel. For that purpose they used quantitative pLII to determine the level and the distribution of soot as well as OH* chemiluminescence to determine the lift-off length. The laser used for the pLII was an Nd:YAG at 532nm and the surrogate fuel used was n-heptane to avoid attenuation of the pLII signal. The results indicated that when the EGR levels increased, soot volume fraction decreased, as a result of the lower combustion temperature leading to lower rates of soot formation. The researchers highlight the need for EGR cooling in order to control the ambient temperature of the spray. Idicheria and Pickett in 2011 used the same experimental setup and the same apparatus for the pLII measurements to study the ignition characteristics, soot formation and the end-of-injection transients of a diesel spray under high EGR conditions. Instead of OH* chemiluminescence, this time formaldehyde chemiluminescence was used to detect the cool flame regions. The main findings of this work were that the first soot formation occurs a finite time after start of ignition and high-temperature reactions and that the time delay to soot formation from ignition increases with increasing EGR. Additionally, ending injection prior to the time of first soot formation significantly reduces the amount of soot formed. The application of the LII technique goes beyond the standard diesel fuel and its surrogates, as demonstrated by Vogel *et al.* in 2013. The researchers studied the ignition processes and the soot formation of three different fuels, namely commercial diesel fuel, pure rapeseed FAME and reference fuel CEC RF-06-03, injected through a diesel injector in an optically accessible constant volume chamber. The results showed increased combustion speed of the commercial diesel fuel 1000 μ s after the start of injection, leading to intensive soot production compared to the other two fuels. As a general remark, fuels with higher CN number feature higher heat release in the later stages of the combustion, stemming from the increased combustion speed and resulting in enhanced soot formation.

The LII technique has been proven to be a useful diagnostic tool for both qualitative and quantitative measurements in optical engines. As early as in 1991, Dec *et al.* applied the pLII technique using an Nd:YAG laser at 532 nm to investigate the soot distribution inside an optically accessible 4-stroke direct injection (DI) heavy duty (HD) engine based on a Cummings NH/NT production engine, operating at the low speed, low load regime. The researchers highlighted the competency of the technique to detect soot particles inside the engine cylinder during and after the combustion. A year later, Dec and Espey in 1992 used the same experimental setup, utilizing, additionally, Mie scattering and fuel tracer LIF, to depict the fuel distribution inside the cylinder, for different engine speed settings, varying from 600 to 1800 RPM. The results indicated that when the fuel jet burned, soot occurred nearly

everywhere there was vapor fuel toward the leading edge of the fuel jet and that the soot distribution extended upstream into the vapor fuel to a distance which decreased with increasing engine speed.

A different variation of the LII method, the Time Resolved (TiRe) LII, was applied in an attempt to get a quantitative estimate of the primary soot particle size during and after combustion. One of the first applications of the TiRe LII in optical engines was that of Kock *et al.* in 2002. The researchers employed a single cylinder CI optical engine and an Nd:YAG laser to study the evolution of the LII signal at a certain crank angle and extract particle sizing information. They also compared the primary soot particle size from the LII technique with DMPS measurements to conclude that there was qualitatively good particle emission signals after heat-up by a laser and a quite high reproducibility for four different crank angles and various engine loads. Kock *et al.* in 2006 used the same engine setup to perform two-colour TiRe LII (TC-TiRe LII), which can yield information on both the temperature and the size of soot particles. The researchers used an Nd:YAG laser tuned at 1064 nm to heat up the soot particles and recorded two different wavelengths from the emission spectrum, at a range of engine load settings. The resulting particle sizes showed an increase and a subsequent decrease during a single combustion cycle, which can be explained by particle formation and oxidation. Additionally, when the engine load increased, the primary particle size increased consequently. The work of Bougie *et al.* in 2006 yielded similar results on the evolution of the particle sizes during the combustion. The researchers used the TiRe LII technique to measure the primary soot particle size inside the cylinder of a modified six-cylinder DI HD CI engine, operating at the low load regime. They found that the earliest detectable soot was with a particle radius of about 26 nm, which increased to about twice this size around 60 CAD aTDC, after which it gradually declined again. Boiarciuc *et al.* in 2007 combined the pLII and TC-TiRe LII techniques to study the effect of EGR levels on soot diameter and volume fraction in an optically accessible four-stroke four-cylinder DI CI engine from PSA. The researchers found stronger increase of soot concentration in the exhaust gases, compared to soot fraction evolution inside the combustion chamber when raising the EGR rate, and this behavior suggested a lower oxidation potential as the dilution increased. It was mentioned that for all EGR rates, the evolution of the local primary particle diameter along the cycle is very similar, reflecting generally the dominance of soot formation or oxidation. In another study by Menkiel *et al.* in 2012, the TiRe LII technique using an Nd:YAG at 1064 nm was employed to determine the primary soot particle size in a single cylinder Ricardo Hydra optical engine, operating in

the low speed, low load regime, under a single injection scheme with variable timing and fuel quantity. A general trend of soot particles sizes decreasing with crank angle was observed, which was attributed to soot oxidation, while, later at the expansion stroke, slightly larger soot particles were observed, due to agglomeration. The researchers found that the fuel injection timing had no effect on the measured primary particle size, while a larger fuel quantity led to an increase in soot particle size. Additionally, it was pointed out that a shorter ignition delay led to increased levels of soot, due to the limited pre-mixing time.

The combination of the LII technique with other optical diagnostics can yield interesting results not only regarding soot distribution and particle size, but also regarding soot precursors, formation and oxidation. Musculus in 2006 combined the pLII technique with a number of other optical techniques to study the combustion of the fuel spray in a single-cylinder DI four-stroke diesel engine based on a Cummins N-series production engine, operating in the low speed, low load regime, under high levels of charge dilution and intake boosting. More specifically, the researcher employed Mie scattering to detect liquid-fuel penetration, fuel fluorescence to depict the leading edge of the vapor jet, chemiluminescence imaging to identify the location of ignition, OH pLIF to detect hot second-stage combustion as well as soot luminosity and soot laser-induced incandescence to measure the development of in-cylinder soot. Results indicated that soot was formed in the in the fuel-rich head vortex, where mixing is slowest and suggested that if the head vortex could be disrupted, or its mixing rate accelerated, the pre-combustion mixing requirements necessary to reduce soot formation might be significantly relaxed, so that fuel injection could be phased closer to TDC for better controllability and higher thermal efficiencies. SINGH *et al.* in 2007 used the same experimental apparatus and combination of techniques to study a double injection scheme and the combustion processes related to each injection event. It was found that soot formation for the first ignition occurred only in tiny pockets, indicating that the most of the jet regions were below sooting limits while for the second ignition, soot was formed in the less well- mixed jet cross-section. Chartier *et al.* in 2009 used a very similar combination of optical diagnostics to study smokeless spray combustion in a modified, optically accessible six-cylinder Scania D12 HD DI CI engine. For the LII measurements an Nd:YAG laser tuned at 1064 nm was used, while the engine operated ant the low speed, low load regime, with medium levels of inlet boosting and a single, high-pressure injection scheme. The case studied produced no engine-out smoke and no soot was detected in the laser-illuminated section of the jet. Soot was, however, observed in the recirculation zones close to the bowl perimeter, indicating very slow

soot formation. The absence of soot in the jet was attributed to a significant rate of air entrainment into the jet.

Miles *et al.* in 2007 employed the pLII technique in conjunction with formaldehyde pLIF and PIV to characterize the flow structures and their effect on fuel mixing and soot formation. The work was performed on a Volvo D5 optical four-stroke five-cylinder DI LD CI engine with the help of a Multi YAG laser system developed in Lund University, Sweden. The engine operated at the low speed, low load regime, with low levels of inlet boosting and high levels of charge dilution. The investigation showed that later in the cycle, soot and partially oxidized fuel were concentrated in the central region of the cylinder, while there were no bulk flow structures to transport this fluid to other regions of the cylinder for further oxidation and thus, soot oxidation froze. O'Connor and Musculus in 2014 used the pLII technique along with high speed natural luminosity imaging to study the in-cylinder mechanisms of soot reduction by a close-coupled post injection in a single-cylinder four-stroke optical engine based on a Cummins N-series production engine. The operating conditions were at the low speed, low load side, with low inlet boosting and low levels of EGR. Engine-out soot measurements unequivocally indicated that there was an interaction between the post jet and the main- injection soot, leading to a 20% reduction of engine-out soot compared to main-injection-only operation. Finally, Lilik *et al.* in 2015 used the same experimental engine to study in-cylinder soot evolution late at the combustion cycle, employing the pLII technique with a vertical laser sheet generated from an Nd:YAG laser tuned at a 1064 nm wavelength. The engine operated at the low speed, low load regime, with moderate inlet boosting and various levels of charge dilution. The researchers also employed high speed chemiluminescence imaging to conclude that regardless of the dilution level, the soot mass was increasing as long as primary heat release occurred, *i.e.*, net soot oxidation did not exceed net soot production until after the end of primary heat release.

1.5.3.4 Experimental Issues of the LII technique

Throughout the years, in the numerous applications of the LII technique, researchers have stumbled upon various issues and limitations. Quay *et al.* 1994 argued that a long laser pulse would bias the particle size distribution measurements towards the larger particles, while Ni *et al.* 1995 observed that in laminar flames, the soot volume fraction measurements were affected by the mean size of the particles, in a way that areas with small mean particle size would exhibit lower soot volume fractions. Axelsson *et al.* 2000 pointed out that when the TiRe-LII method

was applied, errors in the primary soot particle size emanated from uncertainties in the ambient gas temperature, which did not allow the proper fitting of the experimental LII signal curve to the theoretical curve. The same research group a year later (Axelsson *et al.* , 2001) observed the same issue in the gate-ratio technique which they tested, because the conversion of the recorded LII signal to particle size relied on a theoretical model of the particle cooling, which relies on the ambient gas temperature.

The work of Cenker *et al.* 2015 recites the main issues of the LII technique. The researchers applied a combination of the pLII and TiRe-LII techniques on a reacting fuel spray inside an optically accessible vessel, using an Nd:YAG laser tuned at 1064 nm with 7 ns pulse duration. One major complication of TiRe-LII at high pressure, according to the researchers, was the shorter signal lifetime due to fast heat exchange of the particles with their environment. Therefore, each particle-size class reached a different peak temperature causing a bias of the total signal toward large particles. The TC-TiRe LII method was also affected by this effect and therefore did no longer provide the required information to initiate the modeling of the cooling process. In pLII, the contribution of small particles was weak, because of the reduced heat-up temperature, and vanished quickly, because of the enhanced convective cooling already during signal collection. As a result, the measured soot volume fractions systematically overlooked the contribution of small particles. The same research group in a subsequent study using the same experimental apparatus (Cenker *et al.* , 2015), pointed out that at elevated ambient pressures, there was a substantial difference on the decay rate of the LII signal between isolated and aggregated soot, while soot clouds with significant optical density could cause non-uniformity in the spatial distribution of the heat-up temperatures, due to laser attenuation. They also mentioned that in measurements where the signal lifetime and gate duration were short, timing jitter could influence the results strongly.

1.6 Chapter Summary

This chapter introduced the subject matter and the motivation for the conducted work. The various CI engine operation strategies and their effect on efficiency and emission were laid out, providing the theoretical basis for the experimental work carried out. The optical engine and the optical diagnostic techniques used for the purposes of the present thesis were reviewed.

2 Experimental Apparatus and Methodology

2.1 Ricardo Hydra Optical Engine

Engine specifications

The Ricardo Hydra optical engine (see Figure 2.1) is a single cylinder light duty direct injection compression ignition experimental engine, which was modified to facilitate optical access inside the combustion chamber. Ricardo PLC was responsible for the initial engine design and manufacturing of the optical engine, which featured an elongated Bowditch piston design. The engine was modified and fitted with a Ford York piston and a Ford cylinder head to match the Ford Puma I4 (4-cylinder, 2.2L, 155HP, Euro 5) metal engine housed in a test cell of the same laboratory. This modification enabled the visualization of the combustion in the optical engine in order to gain deeper insight in the combustion processes and explain the emission trends observed in the metal engine. Table 2.1 summarizes the engine specifications.

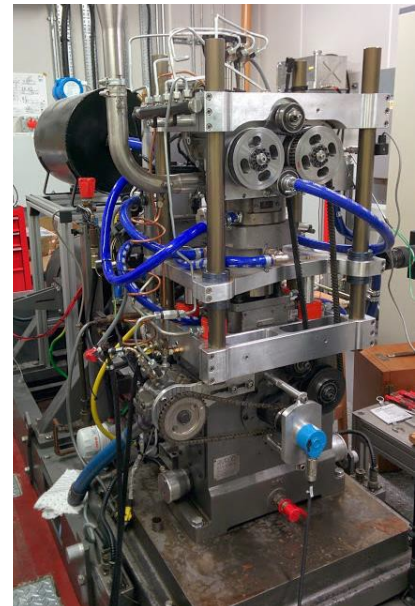


Figure 2.1 Actual image of the Ricardo Hydra optical engine.

Optical access inside the combustion chamber from the bottom side was facilitated by a 45° tilted mirror and an optical piston, which featured a simple cylindrical bowl design and which, at the bottom of the bowl, housed an optical window, as seen in Figure 2.2. The piston diameter was 92.8 mm, while the bowl diameter and, consequently, the width of the optical window was 46.5 mm. The bowl depth was 14.2 mm. Side optical access was facilitated by four equally spaced optical windows inserted in the metal cylinder liner, as seen in Figure 2.3. The windows had a width of 30 mm and a height of 17 mm and were placed in such a distance from the engine head, that they allowed optical access inside the chamber before 31 CAD bTDC and after 31 CAD aTDC, thus limiting the side optical access around TDC.



Figure 2.2 Actual view of the Ricardo Hydra optical piston (C. K. H. Hong, 2015).

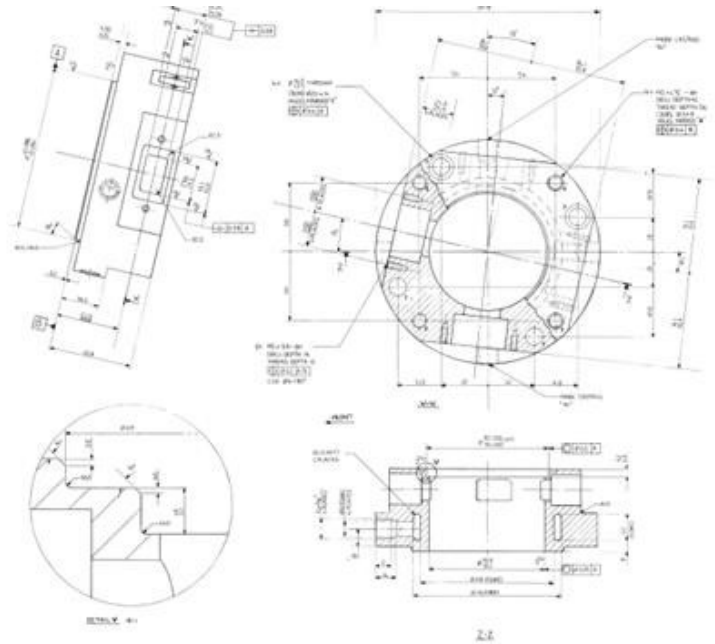


Figure 2.3 Drawings of the Ricardo Hydra cylinder liner. The optical window inserts can be distinguished (C. K. H. Hong, 2015).

Another modification on the original design of the engine was the retrofitting of a hydraulic system, which allowed for the cylinder liner to be removed easily for cleaning. Figure 2.4 shows the main parts of the hydraulic system. The free moving platform sat on four hydraulic jacks, which were connected to a hydraulic hand pump. When the pressure of the hydraulic system was released, the moving section dropped, allowing for the removal of the cylinder liner. To reinstall the liner, the hand pump was operated to lift the moving plate upwards and secure the liner in place, ensuring a tight fit to eliminate the blow-by during engine operation.

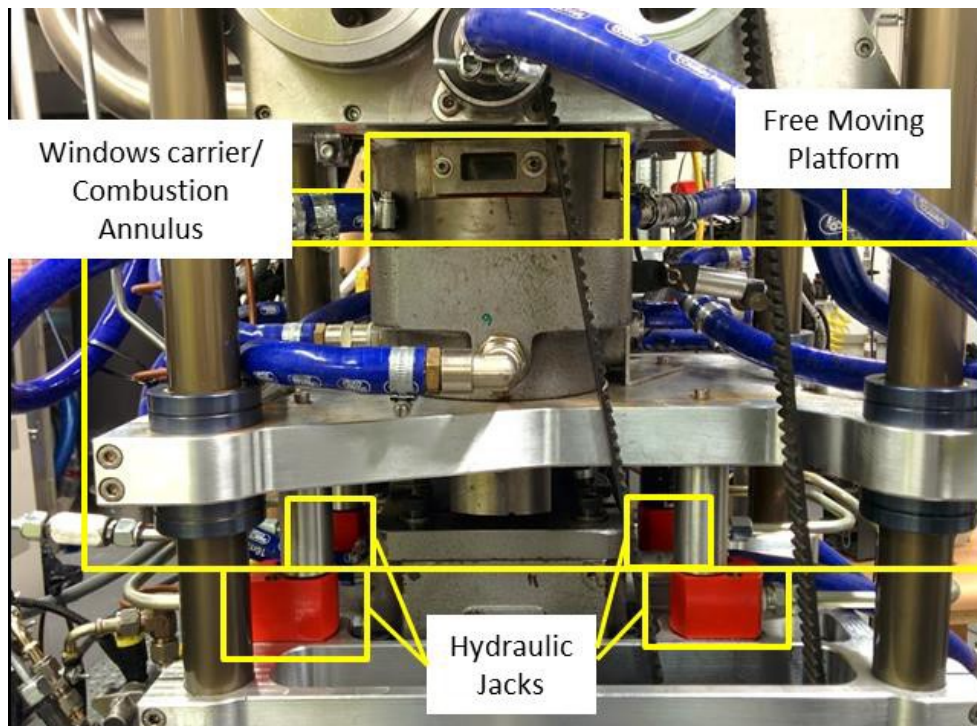


Figure 2.4 Actual image of the main parts of the hydraulic system on the Ricardo Hydra engine (Hong, 2015).

Table 2.1 Ricardo Hydra Optical Engine specifications

Displacement Volume	623 cc
Stroke	90.55 mm
Bore	93.62 mm
Combustion bowl diameter	46.5 mm
Combustion bowl depth	14.2 mm
Connecting Rod	154 mm
Compression Ratio	16.9
Number of Valves	4
In-bowl Swirl Ratio	3.4

Fuel delivery system

The fuel delivery system is divided into the low pressure and the high-pressure systems. The low-pressure system includes all the components upstream the high pressure pump, whereas the high pressure system includes the component downstream the high pressure pump, including the high pressure pump. At the low-pressure side, fuel is drawn from the fuel tank with the help of a low pressure lifting pump, goes through a fuel filter and then fed into the high pressure pump, which is attached to the engine. At the high-pressure side, the high pressure pump, which is powered by the rotational movement of the inlet valves camshaft, feeds fuel into the common rail, where the pressure is regulated by a return valve and an electronic control circuit to the desired pressure value. The high-pressure fuel is guided from the common rail to the fuel injector through a small diameter tube, where an electronic signal is sent from the control unit to open the injector needle and spray fuel into the combustion chamber at the desired timing.

This particular setup featured a Siemens[®]/Continental[®] K10_16 fuel pump feeding a Siemens VDO K10_14 common rail. The injector nozzle had 8 holes equally spaced around the circumference, with 120 μm hole diameter, a 140° spray umbrella angle and a static flow rate of 350 cm³/30'' at 100 bar. The fuel used for the purpose of this thesis was commercial ultra-low sulphur diesel with a cetane number of CN51. Table 2.2 summarizes the fuel system specifications. One of the four injectors injected fuel into the engine while the other three injected into a collection volume so the pressure wave effect of injections in the high-pressure lines can be simulated and the collected fuel could be returned and measured.

Table 2.2 Ricardo Hydra fuel delivery system specifications.

Fuel	Ultra-low sulphur diesel fuel (CN=51)
Fuel injection system	Siemens/Continental common rail
Injector type	Solenoid
Injector holes	8, equally spaced
Injector cone angle	140°
Injector hole diameter	120 μm
Static fuel flow rate	350 cc/30'' @ 100 bar

Auxiliary Systems

The Ricardo Hydra optical engine featured various auxiliary systems which controlled the inlet air stream, as well as the oil and the coolant. The temperature of the inlet air was regulated by a 3.6 kW Leister[®] LHS System 41S and it was connected directly upstream of the inlet manifold of the engine. The air heater had a minimum air flow requirement of 240 l/min (17.7 kg/hr). The engine setup did not feature any air boosting device, so the engine was naturally aspirated. The air mass flow was measured by a Valumass[®] 500 Series in-line thermal mass flow meter, coupled to a plenum chamber, which smoothed the airflow. The oil and coolant of the engine were heated using heat exchangers and their temperature was regulated by two thermostatic valves.

Data Acquisition

In order to capture the fast-evolving phenomena in a reciprocating internal combustion engine it is necessary to have a high-speed acquisition system. National Instruments[®] CRIO Real-Time PowerPC with an embedded FPGA is an excellent alternative for this kind of data acquisition. The main hardware was a National Instruments cRIO-9022, Real-Time PowerPC embedded controller for CompactRIO, together with a cRIO-9114 8-slot reconfigurable chassis. There was an on-board FPGA, a programmable hardware chip with a high acquisition rate.

The signal from the crank angle encoder and the amplified signal from the in-cylinder pressure sensor are sent to a NI 92018-channel fast acquisition module to ensure that there is enough resolution to the recorded dataset. The in-cylinder pressure was recorded using a Kistler[®] type 6056A41 Quartz Pressure Sensor, M5x0.5, for combustion engines pressure transducer, mounted in a type 6542Q313 glow plug adaptor and was logged every 0.2 CAD. The pressure transducer gave 20.87 pC/bar which was connected to a Kistler Type 5001 charge amplifier. The NI 9201 module was also responsible for measuring the engine torque, the air flow rate and the current clamp reading for injection timing.

The NI9213 16-channel thermocouple module was responsible for measuring the temperature from various spots on the engine and the auxiliaries, namely the ambient air temperature (TC1), the coolant temperature (TC8), the oil temperature (TC7), the inlet air temperature (TC0) and the fuel temperature (TC9).

The recorded data from the CRIO Real-Time module were sent to the Host PC, where they could be displayed and saved using National Instruments Labview® software.

Engine Control Hardware and Software

The engine was coupled to a Cussons® 30 kW dynamometer which was controlled by a Cussons control unit and ensured that the engine speed was kept constant by balancing the torque and power produced by the engine. The unit also controlled the oil, fuel and coolant temperatures.

The fuel injection system was controlled by a custom-made ECU, which came with its own user interface; the ATI Vision 3.5.1. The ECU allowed for up to 6 injections *per* cycle, skip firing operation and a fuel pressure up to 1600 bar. Common rail pressure, injection timings and fuel amounts were fully adjustable. Figure 2.5 shows the ATI Vision user interface.

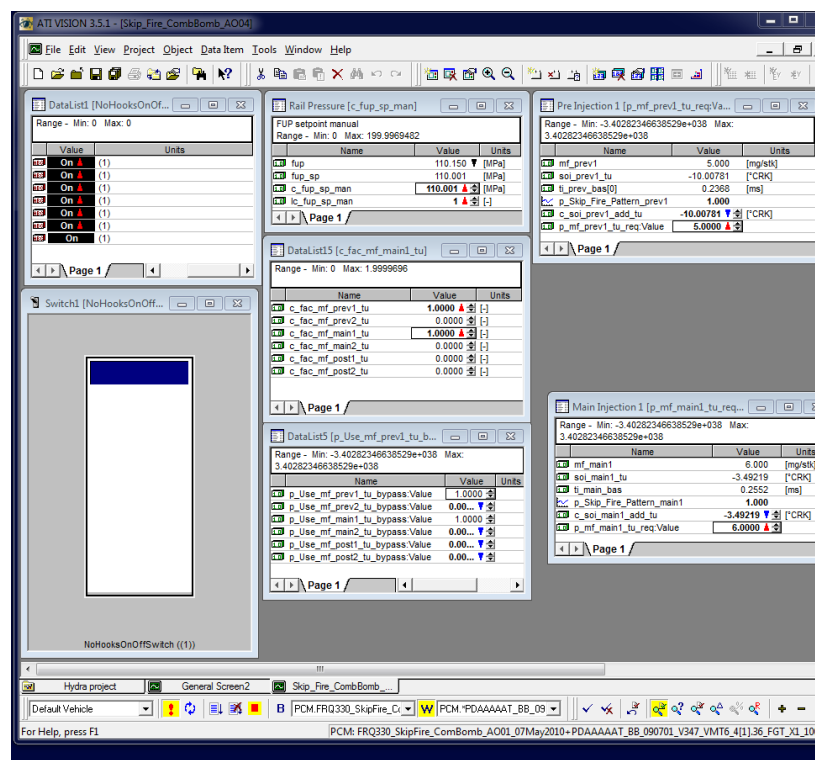


Figure 2.5 ATI Vision 3.5.1 user interface used to control the fuel delivery system in the Ricardo Hydra engine.

The top level of the user interface was written in Labview and is shown in Figure 2.6. The interface displayed various temperature and pressure readings as well as the real-time recorded in-cylinder pressure and the calculated heat release rate. It also contained control buttons and indicators.

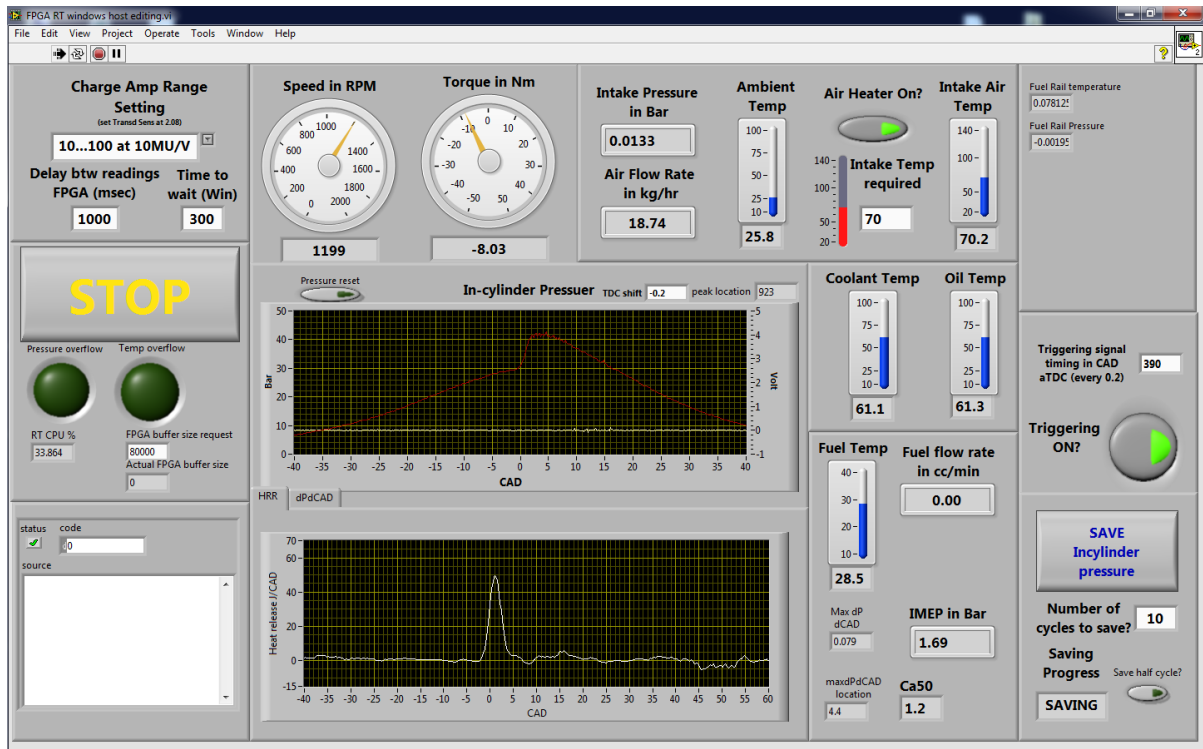


Figure 2.6 Ricardo Hydra engine control interface in Labview.

2.2 Volvo D5 Optical Engine

Engine specifications

The Volvo D5 optical engine is based on a commercial Volvo D5 five-cylinder light-duty direct injection compression ignition engine. One of the cylinders was modified to facilitate optical access according to the Bowditch design, while the other four cylinders were drilled through in order to avoid compression and expansion work. In order to balance the modified engine, additional masses were added to the motored pistons and the flywheel in the form of tungsten allow weights. Because of the extended piston design, it is necessary to separate the engine head from the cylinder block using spacers in the form of I-beams. The bottom face of the beams was bolted directly onto the

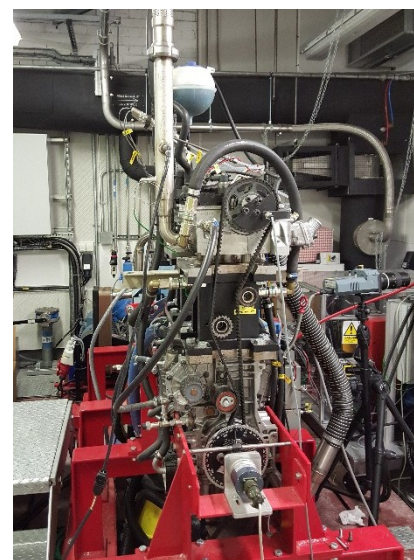


Figure 2.7 Actual image of the Volvo D5 optical engine.

cylinder block, while at the top of the beams bolted a plate with a circular section at the middle to allow piston movement. Between the plate and the cylinder head there was the quartz cylinder liner with a height of 25 mm, which allowed for side optical access to the combustion chamber. The crown of the elongated piston consisted of the piston holder and the optical piston, with the latter being housed in the former. The rim of the piston holder was narrow, to achieve the widest possible optical access inside the combustion chamber. The optical piston used in this study had a flat design, however, it is possible to fit a fully optical omega-shaped bowl design on the elongated piston, using the appropriate piston holder. In order to obtain optical access inside the combustion chamber from the bottom of the optical piston, a stationary 45° tilted mirror was placed inside the slot of the elongated piston. A schematic of the engine is shown in Figure 2.8 and an actual image of the engine is shown in Figure 2.9. Engine specification are presented in Table 2.3.

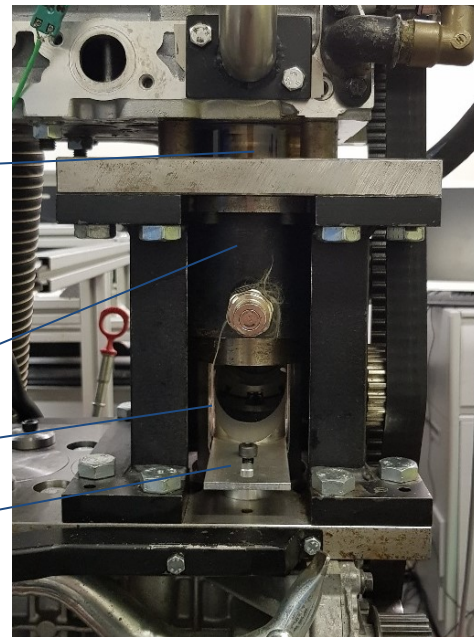
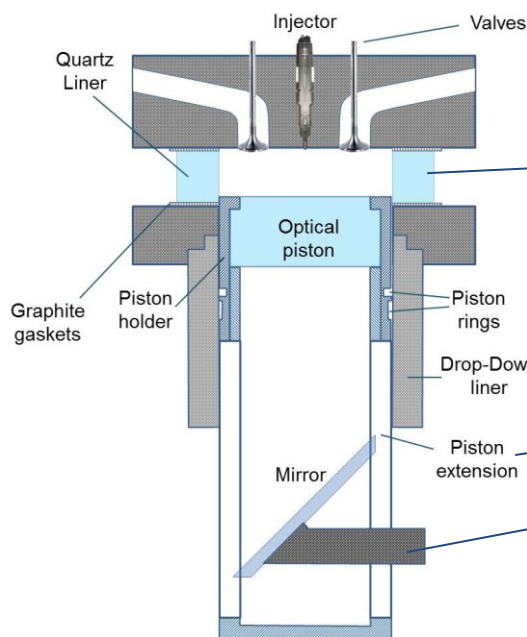


Figure 2.8 Schematic of the Volvo D5 optical engine. (Najafabadi, 2017)

Figure 2.9 Actual side view of the Volvo D5 Optical Engine

Table 2.3 Volvo D5 Optical Engine specifications

Displacement Volume	480 cc / cylinder
Stroke	93.2 mm
Bore	81 mm
Connecting Rod	147 mm
Compression Ratio	14.4
Number of Valves	4
Swirl Ratio	2.6

Fuel Delivery System

The fuel delivery system of the Volvo D5 engine featured the same layout as the one in the Ricardo Hydra engine, with the low and the high-pressure sides having the same components as described earlier in section 2.1.

The Volvo D5 engine setup featured a Bosch[®] common rail system with a solenoid fuel injector. The injector nozzle had 7 holes equally spaced around the circumference, with 140 μm hole diameter and a 140° spray cone angle. The fuel used for the purpose of this thesis was commercial ultra-low sulphur diesel with a cetane number of CN51. Table 2.4 summarizes the fuel system specifications.

Table 2.4 Volvo D5 fuel delivery system specifications.

Fuel	Ultra-low sulphur diesel fuel (CN=51)
Fuel injection system	Bosch common rail system
Injector type	Solenoid
Injector holes	7, equally spaced
Injector cone angle	140°
Injector hole diameter	140 μm

Auxiliary Systems

The Volvo D5 Optical Engine featured various auxiliary systems which control the inlet air, the exhaust gases as well as the oil and the coolant. The temperature of the inlet air was regulated by a 7.5 kW Leister Air Heater, while the inlet pressure was boosted using a positive displacement air compressor, driven by an electrical motor. The oil and coolant of the engine were heated using heat exchangers and their temperature was regulated by two thermostatic valves. Finally, the exhaust gases could be diverted back inside the engine (Exhaust Gas Recirculation – EGR) after they had passed through an intercooler and mixed with fresh air. Figure 2.10 shows the air compressor setup, while Figure 2.11 shows the EGR system installation.

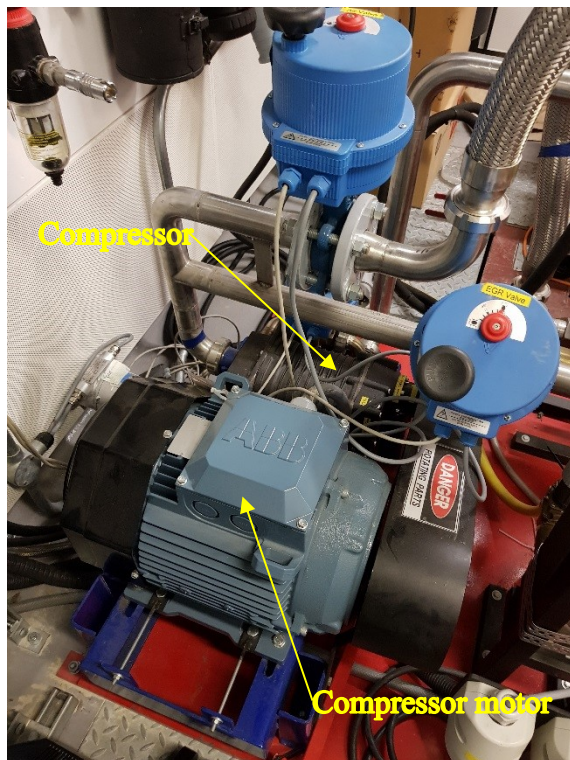


Figure 2.10 Actual image of the air compressor system.

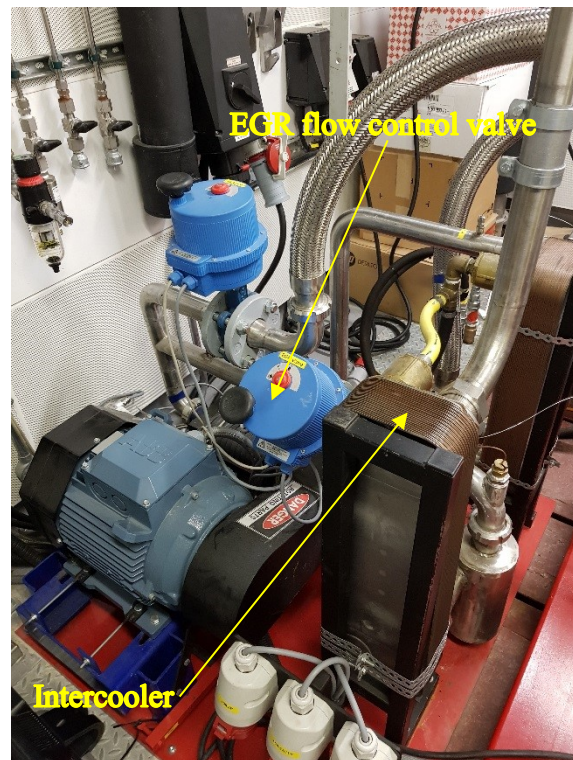


Figure 2.11 Actual image of the EGR system.

Data Acquisition

Just like it was described in section 2.1, for the high-speed data acquisition from the engine a National Instruments[®] Real-Time PowerPC with an embedded NI PCIe-7842R card, which supports FPGA system, was used. The signal from the crank angle encoder and the amplified signal from the in-cylinder pressure sensor were sent to the aforementioned module to ensure that there is enough resolution to the recorded dataset. The in-cylinder pressure was captured using an AVL[®] GU13P pressure transducer coupled to a KISTLER 5011 charge amplifier.

However, not all the operational parameters of the engine have to be recorded using a fast acquisition module. For instance, all the signals from the thermocouples, measuring temperature at various locations on the engine, as well as the signals from the pressure transducers outside the engine cylinder were sent to an Agilent[®] 34970A Data Acquisition/Switch Unit and then to the Host-PC *via* an RS232 port, using the NI-VISA communication protocol.

The recorded data from the Real-Time system were sent to the Host PC, where they would be displayed and recorded, along with the RS232 data, using Labview software.

Control Software

The user interface was a Labview-based code which allows the display and manipulation of the various operational parameters of the optical engine. Figure 2.12 shows the interface related to the control of the engine, while Figure 2.13 shows the interface related to the logging of the operational parameters.

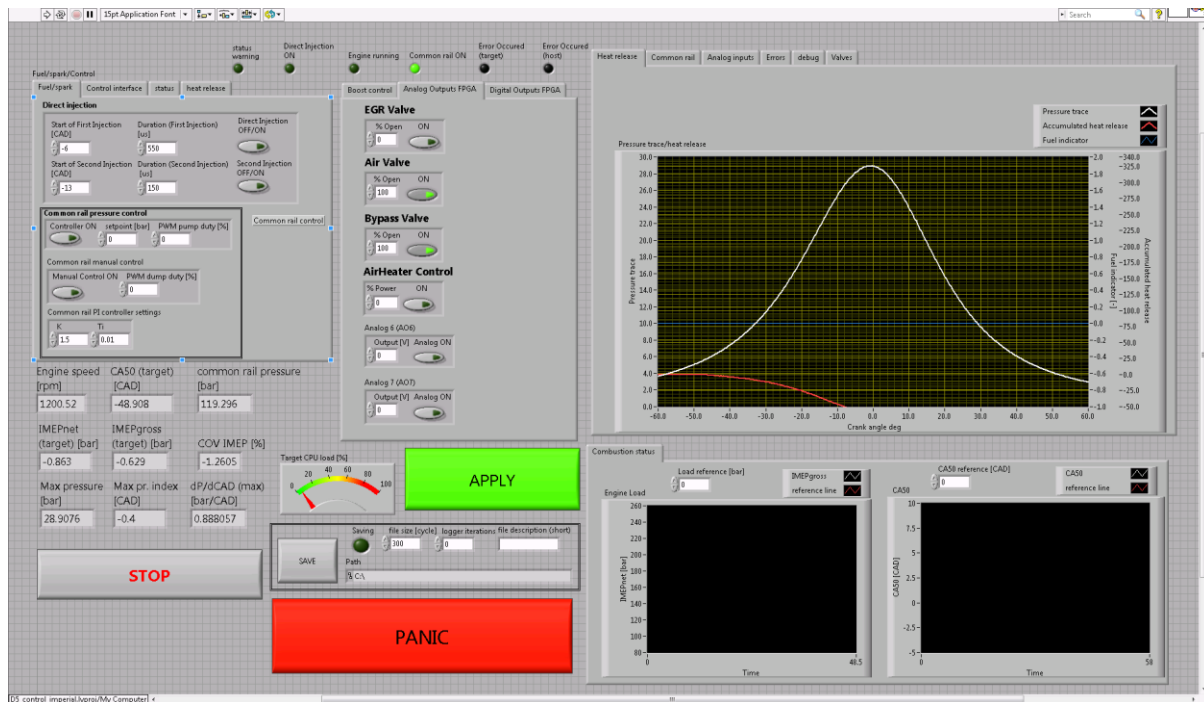


Figure 2.12 Engine control interface window in Labview.

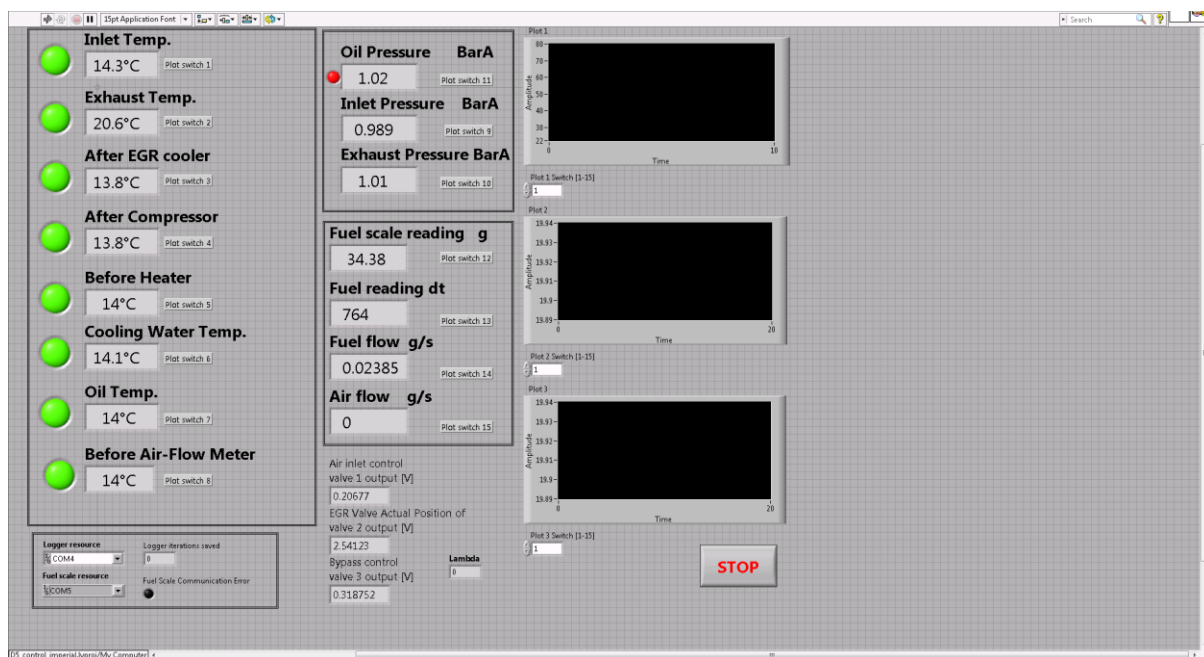


Figure 2.13 Engine operation logger window in Labview.

Comparison with the Ricardo Hydra engine

The Volvo D5 optical engine offered wider optical access inside the combustion chamber, compared to the Ricardo Hydra engine, thanks to the design of the cylinder liner and of the elongated piston. The lower compression ratio of the Volvo D5 engine, compared to the Hydra engine makes it more suitable for PPC operation, as it allows for longer ignition delay and longer mixing time of the charge. The use of boosted inlet and EGR offered great versatility and a wide range of operation points available for testing and can also be used to compensate for the lower compression ratio if necessary. Nevertheless, due to the extended use of optical parts, the Volvo D5 was more sensitive to high in-cylinder pressure levels and thermal stresses than the Ricardo Hydra and thus, skip-firing operation was necessary, in order to prevent the optical parts from failing.

2.3 Thermodynamic measurements

The thermodynamic variables of the combustion derive from the recorded in-cylinder pressure. From the recorded pressure signal one can extract useful information regarding the combustion event, such as the engine load, as expressed by the Indicated Mean Effective Pressure (IMEP), the peak in-cylinder pressure and the peak pressure rise rate (PRR) which is shown to correlate well with combustion noise (*e.g.* Busch *et al.* 2015). It is important to note here that the cycle-to-cycle variability of the IMEP values is below 2%, as measured using the coefficient of variation.

The net Apparent Heat Transfer Rate (AHTR – can also be less accurately described as Heat Release Rate or HRR) was calculated using equation (2.1) and involves a single-zone thermodynamic model, factoring in the heat losses (eq. (2.2)) to the cylinder surroundings according to the Woschni correlation, shown in equation (2.3) (Woschni & Fieger, 1979).

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{dt} + \frac{1}{\gamma - 1} V \frac{dp}{dt} - \frac{dQ_{hl}}{dt} \quad (2.1)$$

where

$$Q_{hl} = A_w h_c (T_g - T_w) \quad (2.2)$$

and

$$h_c = 3.26B^{-0.2}p^{0.8}T_g^{-0.55}w^{0.8} \quad (2.3)$$

In equation (2.2) A_w is the total surface area in contact with the gases, T_g and T_w the instantaneous gas and wall temperatures respectively, while in equation (2.3) B is the bore, p is the pressure and w is the average cylinder gas velocity.

For the numerical calculation of the AHTR a simple three-point estimation of the pressure and the volume derivatives was used. In order to avoid derivation of the pressure signal noise, a second-order Butterworth filter was applied with a cut-off frequency of approximately 2 kHz (6% of the sampling frequency).

The Cumulative Apparent Heat Transfer (CAHT) was calculated by integrating the AHTR from the Start of Injection (SOI) until the End of Combustion (EOC), as shown in equation (2.4).

$$CAHT = \int_{t_0=SOI}^{t_1=EOC} \frac{dQ_{aht}}{dt} dt \quad (2.4)$$

2.4 High-Speed imaging

The study of the combustion process in reciprocating internal combustion engines requires measurements with high temporal resolution in order to capture the fast-evolving phenomena. This applies to combustion imaging as well, thus a high-speed camera is necessary to capture the natural luminosity of the flame and the chemiluminescence emitted by the excited radicals of CH and C₂ species. The high-speed camera used throughout the present study was a Photron Fastcam APX-RS high-speed CMOS camera, which was controlled by a LaVision HighSpeed Controller and was fitted with a 105 mm (1:1.8) UV Nikkor lens. The camera pointed at the 45° tilted Newport 18E20AL.2 elliptical broadband metallic mirror and captured line-of-sight snapshots of the combustion process through the optical piston. Figure 2.14 shows a schematic of the setup for the high-speed imaging, while Figure 2.15 and Figure 2.16 present the spectral response of the camera and the mirror respectively.

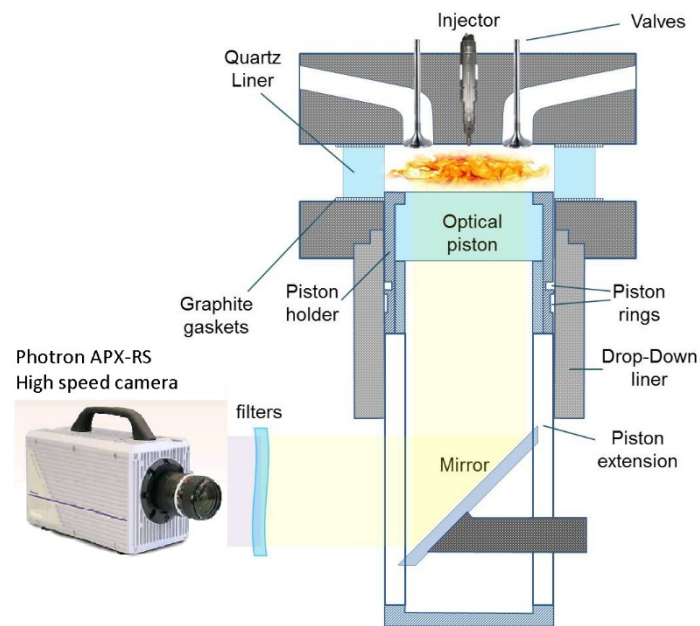


Figure 2.14 Schematic of the high-speed imaging setup.

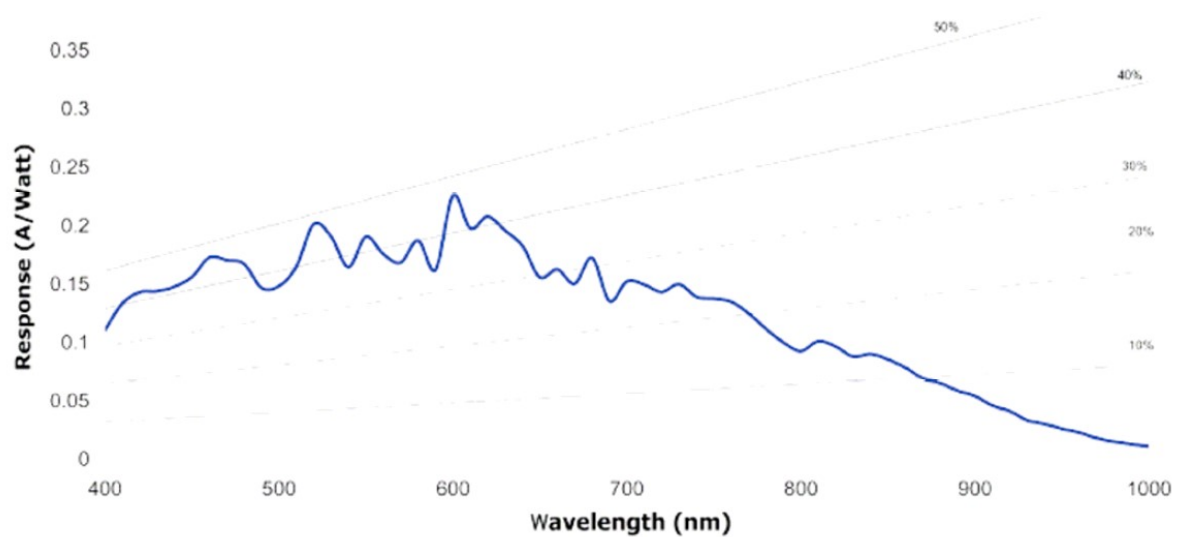


Figure 2.15 Response curve of the Photron APX-RS high speed camera

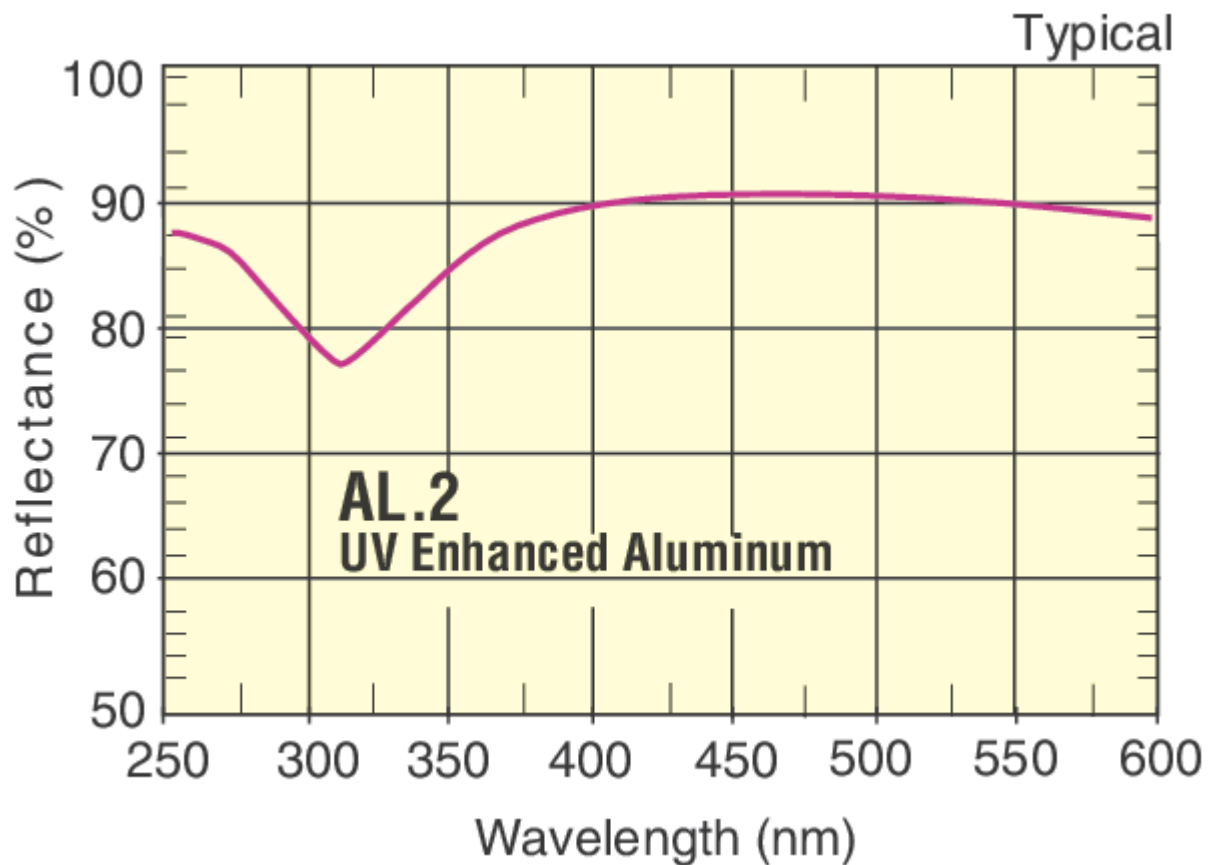


Figure 2.16 Mirror spectral reflectance.

The camera was utilized with an acquisition frequency of 10 kHz, corresponding to approximately 3 images *per* 2 CAD at 1200 RPM, exposure time 100 μ s and a resolution of 512x512 pixels. The image acquisition took place for 10 consecutive cycles, due to size limitations of the on-board RAM capacity of the camera. The cycle-to-cycle variation of the captured images has been assessed using both Standard Deviation (SD) and Proper Orthogonal Decomposition (POD) analyses. Results indicate that the captured images exhibit significant variability at the start and end of combustion, as the energy of the first POD mode of a snapshot at a given crank position is, in average, around 50% of the energy of the average snapshot, while, during the evolution of the combustion, the variability is low, with the respective percentage falling as low as 2% (see Appendix A). This implies that the variability of the captured images and, consequently, the spatially integrated signal curves during the main part of the combustion, where the trends are observed and discussed, is low. More specifically, when the spatially integrated signal curves are presented, error bars are included for each curve at the point of the maximum values of the signal. The error bars represent the standard deviation

of each curve at the aforementioned point and so that the reader can form an own opinion as to the significance of observed trends

2.4.1 Flame's natural luminosity high speed imaging

The optical filters used for natural luminosity imaging were two GG395 high transmissivity filters (Edmund optics) to cut out the ultraviolet wavelengths. The transmissivity of the optical filter is presented below.

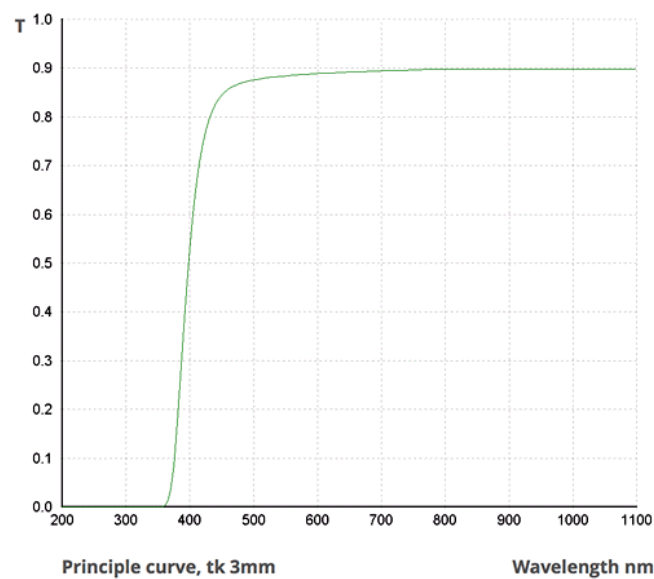


Figure 2.17 Transmissivity curve for GG395 optical filter.

The natural luminosity signal can be qualitatively correlated with soot volume fraction, although care must be taken during the interpretation due to a number of factors that affect the intensity of the signal, mainly the local temperature of the combustion (Mueller & Martin, 2002). As a consequence the peak value of the Spatially Integrated Natural Luminosity (SINL) signal can be indicative of the levels of soot produced during the combustion, while the rate of decay of the SINL signal at the expansion stroke, as well as the level of the SINL signal late at the expansion stroke can be correlated to soot oxidation rate and efficiency, respectively.

2.4.2 Chemical species chemiluminescence high speed imaging

As discussed in Section 1.5.2, the detected signal through an optical filter around a specified wavelength contains not only the chemiluminescence signal emitted by the species at this wavelength, but also background signal, stemming from broadband CO_2^* chemiluminescence and thermal radiation. In theory, if the background signal is subtracted from the detected signal, the remaining is the net chemiluminescence signal.

However, measuring the background signal at a specific wavelength using a single optical filter with a detection window around this wavelength is practically impossible. For this reason, a more complex methodology has to be followed in order to isolate, at least with some approximation, the net chemiluminescence signal and use it qualitatively to describe the combustion behaviour. The methodology followed in the present thesis is based on the one proposed by M. I. Najafabadi *et al.*, 2017, which was the first study to investigate the interference of background signal to chemiluminescence signal of OH^* , CH^* and C_2^* species in PPC operation. However, in the present thesis, the methodology is extended to investigate the spatially resolved chemiluminescence signal in addition to the spatially integrated signal. The methodology in the present thesis combined spectroscopic measurements of the combustion with high speed imaging through narrowband optical filters in order to assess the effect of the background signal in spatially resolved chemiluminescence measurements.

Firstly, spectroscopic measurements were employed to capture the spectral signature of the combustion using an Andor Mechelle ME5000 high band pass echelle spectrograph, coupled to an Andor DH534T-18F-03 ICCD camera. More details about the equipment can be found in Table 2.5. Combustion luminescence is captured through the optical piston, *via* the 45° tilted mirror, whose spectral reflectance can be seen in Figure 2.16. In order to ensure the widest field-of-view possible and capture as much combustion luminosity as possible, a Thorlabs LA4464 UV Fused Silica collimating lens was employed. The collimating lens had a focal length of 60.2 mm and was placed at such a distance from the tilted mirror to maximize the field-of-view. The collimated light was focused into a Thorlabs FT038 optical fibre, which guided the collected light into the spectrograph. The spectrograph was calibrated using a mercury vapour lamp, as shown in Figure 2.18.

Table 2.5 Specifications of the spectrograph and camera

Spectrograph	Andor Mechelle ME5000
Camera	Andor DH534T-18F-03 ICCD
Wavelength range	200 – 975 nm
Wavelength accuracy	Better than ± 0.05 nm
Channel width	1 pixel
Shutter rate	1 Hz
Intensifier size	18 mm
Gate	5 ns
Resolution	1024 x 1024 pixels
Pixel size	13 μ m

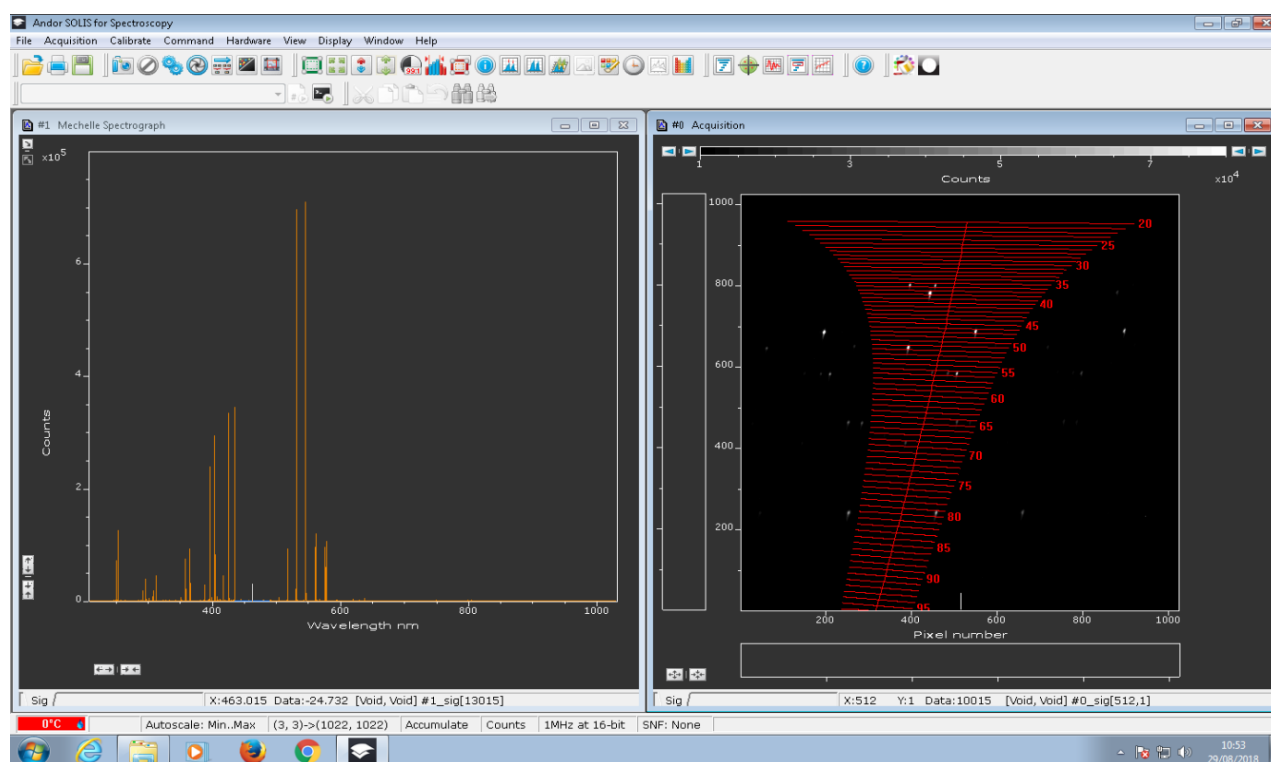


Figure 2.18 Calibration of the spectrograph using the mercury vapour lamp.

For the spectroscopic measurements, the acquisition took place at the CA50 of the combustion, accumulating 15 firing cycles, with a 1:3 skip-firing ratio. The shutter of the camera was kept open for 278 μ s at each cycle, which corresponds to 2 CAD at 1200 rpm engine speed. The obtained spectrum was smoothed out using a 2nd degree Savitzky-Golay filter to reduce the noise. An example of the obtained spectrum can be seen in Figure 2.19. In order to determine the strength of the background chemiluminescence and thermal radiation signal along the spectrum, a polynomial curve was fitted to the data. This polynomial curve represents, approximately, the level of background signal at each wavelength of the spectrum and it will be used later to adjust the level of the background signal captured with the optical filters.

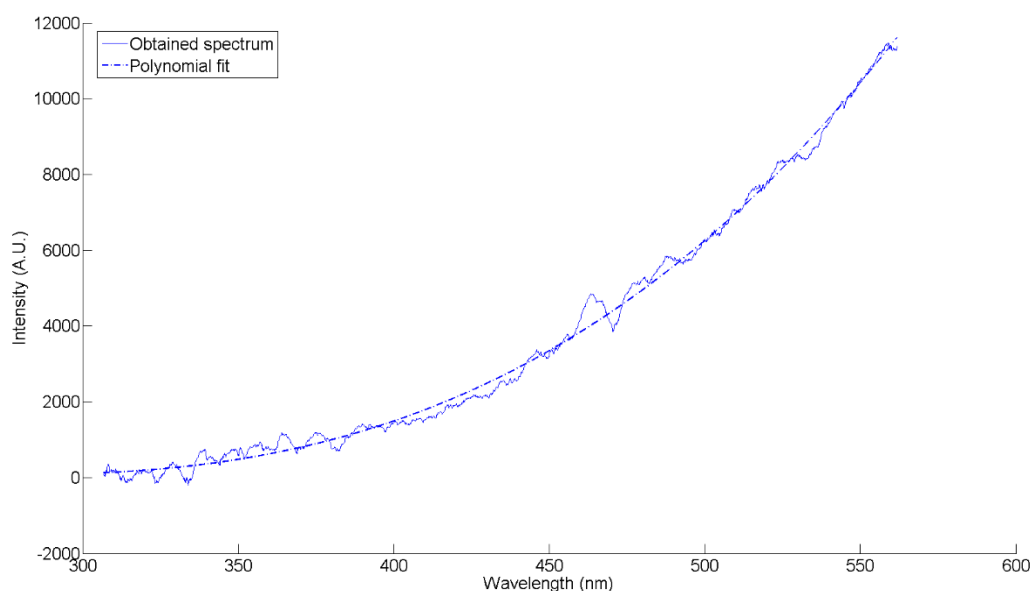


Figure 2.19 Example of obtained combustion spectrum and polynomial fit.

In the present thesis, the assessment of the background signal to the captured chemiluminescence signal was done for two excited radicals; the CH^* and the C_2^* . As discussed in section 1.5.2, the CH^* chemiluminescence depicts the reaction zone of the flame and therefore, throughout this study, CH^* chemiluminescence is considered a proxy of the reactivity of the in-cylinder charge. C_2^* chemiluminescence serves as a proxy of the fuel-rich areas formed during the injection of the fuel inside the cylinder and it is therefore considered indicative of soot formation.

For the assessment on CH^* chemiluminescence three optical filters were used; a 430 nm narrowband filter (Ealing 35-3318, FWHM 10 nm, $T_{\max}=45\%$) capturing the chemiluminescence signal with background interference, and two narrowband filters at 410 nm and 450 nm respectively, capturing the background signal on either side of the emission wavelength (Thorlabs FB410-10 with FWHM 10 nm and $T_{\max}=43\%$ / Thorlabs FB450-10 with FWHM 10 nm and $T_{\max}=51\%$).

For the assessment on C_2^* chemiluminescence three optical filters were used; a 514 nm narrowband filter (Ealing 35-3573, FWHM 10 nm, $T_{\max}=54\%$) capturing the chemiluminescence signal with background interference, and two narrowband filters at 490 nm and 540 nm respectively, capturing the background signal on either side of the emission wavelength (Thorlabs FB490-10 with FWHM 10 nm and $T_{\max}=46\%$ / Thorlabs FB540-10 with FWHM 10 nm and $T_{\max}=56\%$).

The optical filters mentioned above were chosen so that any interference with other chemiluminescent species is avoided. The transmissivity curves for the optical filters are shown in Figure 2.20.

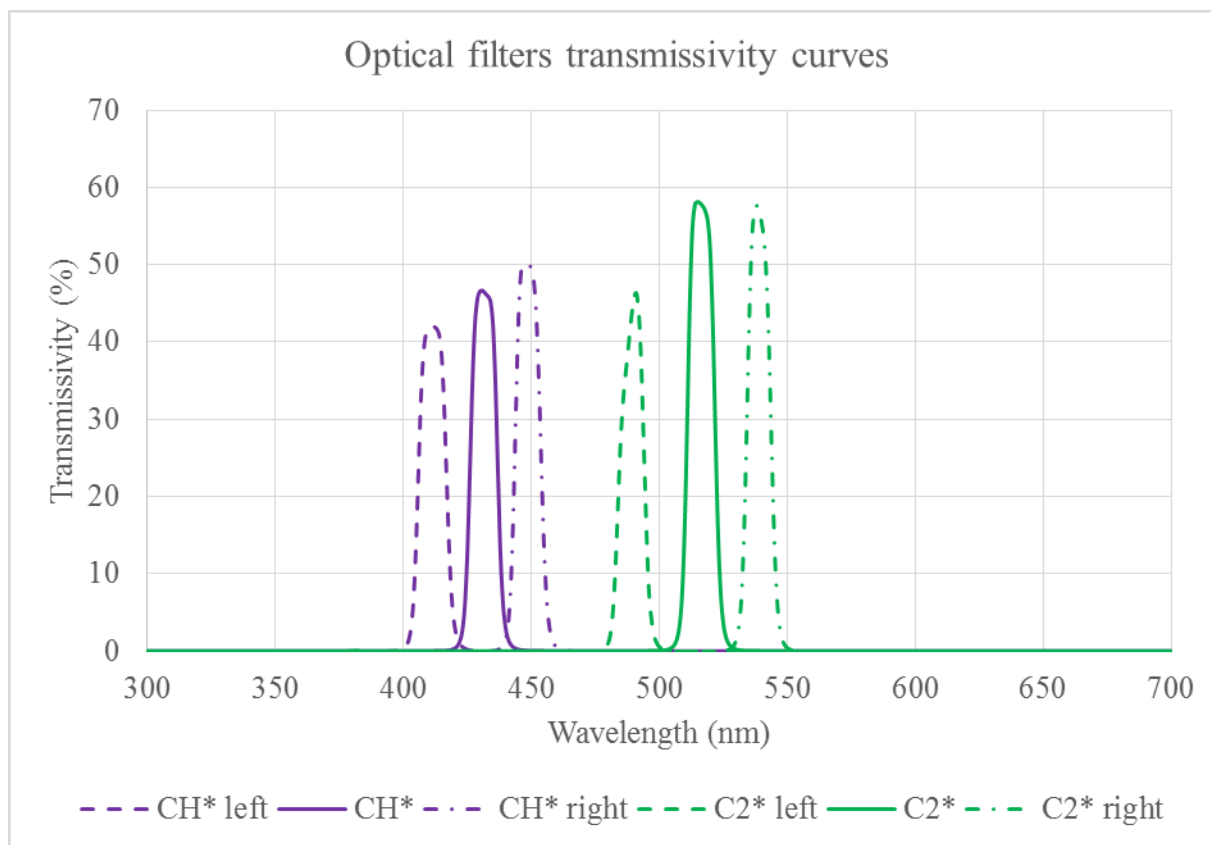


Figure 2.20 Transmissivity curves of the optical filters used for chemiluminescence.

Using the aforementioned optical filters and the high-speed camera, the spatially resolved combustion luminescence was captured at the respective wavelengths. An example of the cycle-averaged images before any adjustment is given in Figure 2.21.

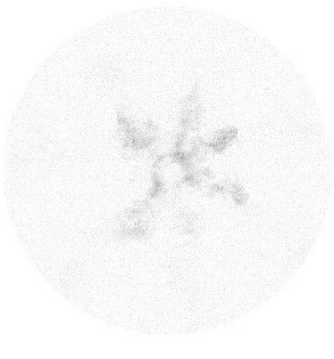
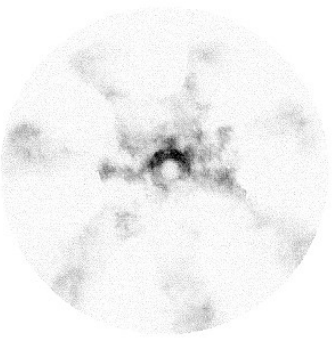
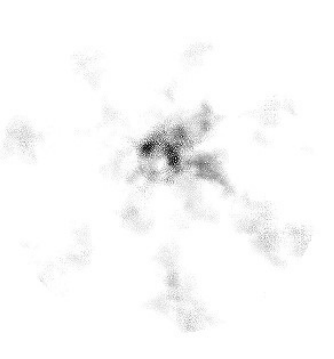
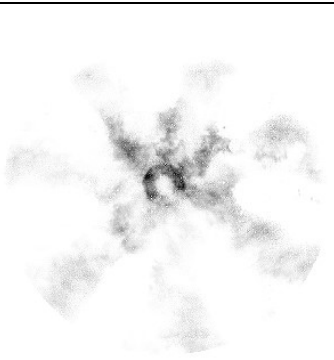
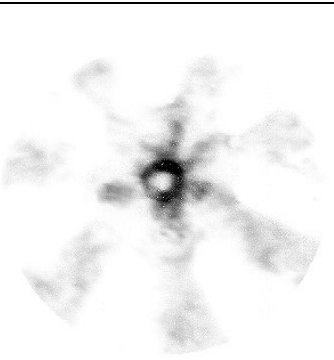
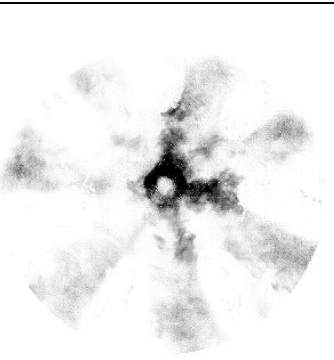
Same CAD	$\lambda=410$ nm	$\lambda=430$ nm	$\lambda=450$ nm
CH [*]			
Same CAD	$\lambda=490$ nm	$\lambda=514$ nm	$\lambda=540$ nm
C ₂ [*]			

Figure 2.21 Example of average images as captured using optical filters in different wavelengths.

In order to calculate the equivalent spatially resolved and spatially integrated background signal intensity at the wavelength of interest (*i.e.* 430 nm for the CH^{*} and 514 nm for the C₂^{*}), the signals on either side had to be adjusted to the specific wavelength, using the polynomial fit from the obtained combustion spectrum. The adjustment also considered the spectral sensitivity of the camera, as shown in Figure 2.15 and the variation in filter transmissivity as shown in Figure 2.20. The calculation of the equivalent spatially resolved background signal intensity for every crank angle degree considered, was done using the following equation:

$$I_{\lambda,adj}(x,y,t) = I_{\lambda}(x,y,t) \cdot \frac{\text{Fitted spectral intensity}(\lambda_0)}{\text{Fitted spectral intensity}(\lambda)} \cdot \frac{\text{Camera Response}(\lambda_0)}{\text{Camera Response}(\lambda)} \cdot \frac{\text{Filter Transmissivity}(\lambda_0)}{\text{Filter Transmissivity}(\lambda)},$$

where x and y refer to each pixel of the captured images, t refers to each crank position, λ_0 refers to the wavelength of interest and λ refers to the wavelength at either side of the wavelength of interest.

Once the background signal intensity was adjusted to the wavelength of interest, it could be subtracted from the total captured signal intensity at this specific wavelength. Figure 2.22 shows an example of the cycle-averaged spatially resolved signal intensity images at the wavelength of interest (green colour), merged with the adjusted cycle-averaged spatially resolved signal intensity images from the wavelengths on each side of the wavelength of interest (blue colour for left, red colour for right). It also presents the spatially resolved corrected signal intensities.

There are two sets of images for each chemiluminescent species from the subtraction of the two different adjusted background signals (left and right of the wavelength of interest). To choose between the two sets, the spatially integrated signal of the corrected image at each crank position is calculated for both sets and the set with the lowest spatially integrated signal overall is chosen. Figure 2.23 and Figure 2.24 show the spatially integrated signals for the CH^* and C_2^* assessment respectively. Both raw and adjusted background signals are included in the graphs, along with the raw and net chemiluminescence signal. Please note that the colour-coding is the same as in Figure 2.22.

The net chemiluminescence signal intensities are calculated using the methodology described above and then used to discuss the observed trends in combustion behaviour. The reader should note that this methodology is only applied to the experimental data of the Volvo D5 optical engine, as the experimental work on the Hydra optical engine preceded the relevant studies on which the methodology was based.

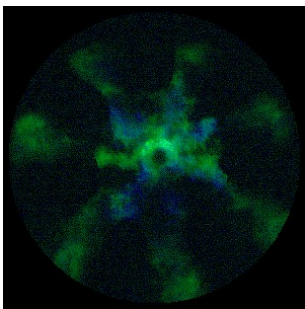
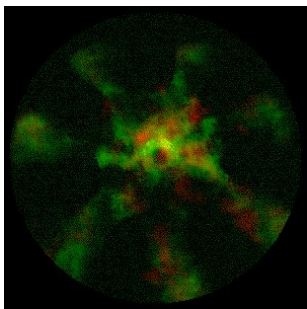
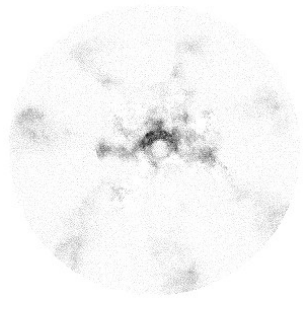
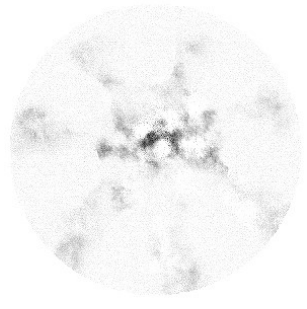
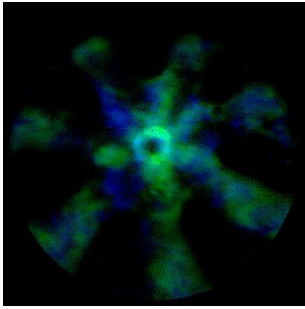
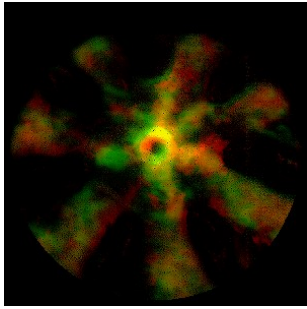
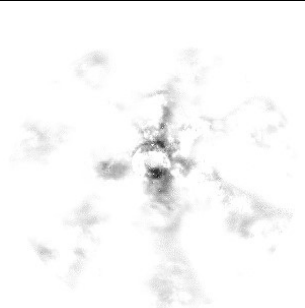
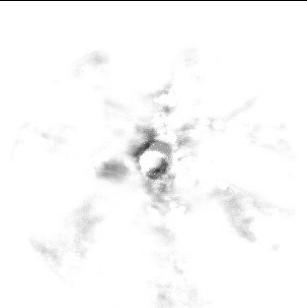
Same CAD		$I_{430nm}(green) vs I_{410nm,adj}(blue)$	$I_{430nm}(green) vs I_{450nm,adj}(red)$
CH*	Merged images		
	Corrected images		
Same CAD		$I_{514nm}(green) vs I_{490nm,adj}(blue)$	$I_{514nm}(green) vs I_{540nm,adj}(red)$
C ₂ *	Merged images		
	Corrected images		

Figure 2.22 Example of merged and corrected spatially resolved chemiluminescence and adjusted background signal intensities.

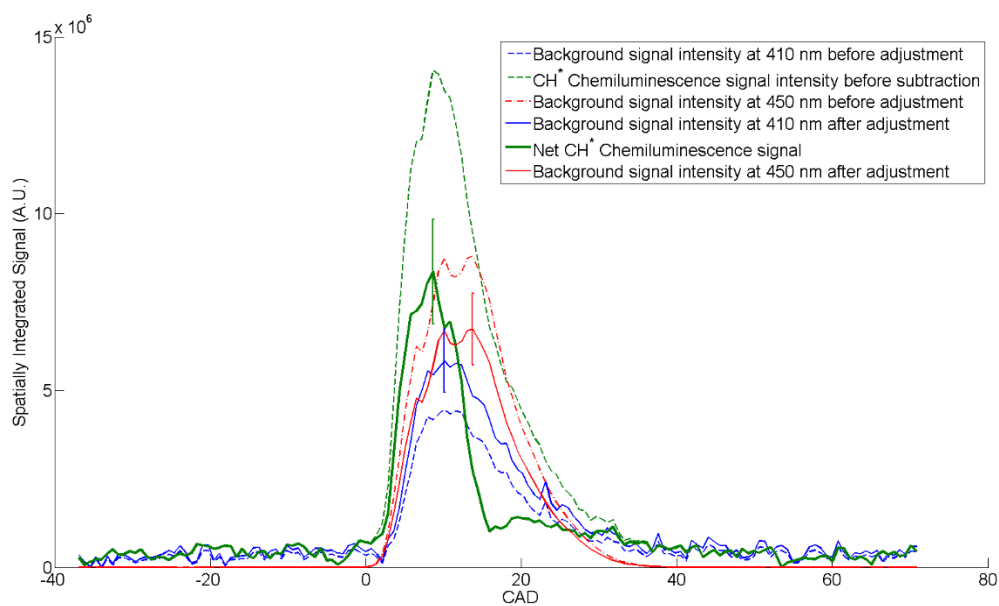


Figure 2.23 Example of the spatially integrated raw and adjusted background signals and the spatially integrated raw and adjusted CH^* chemiluminescence signals.

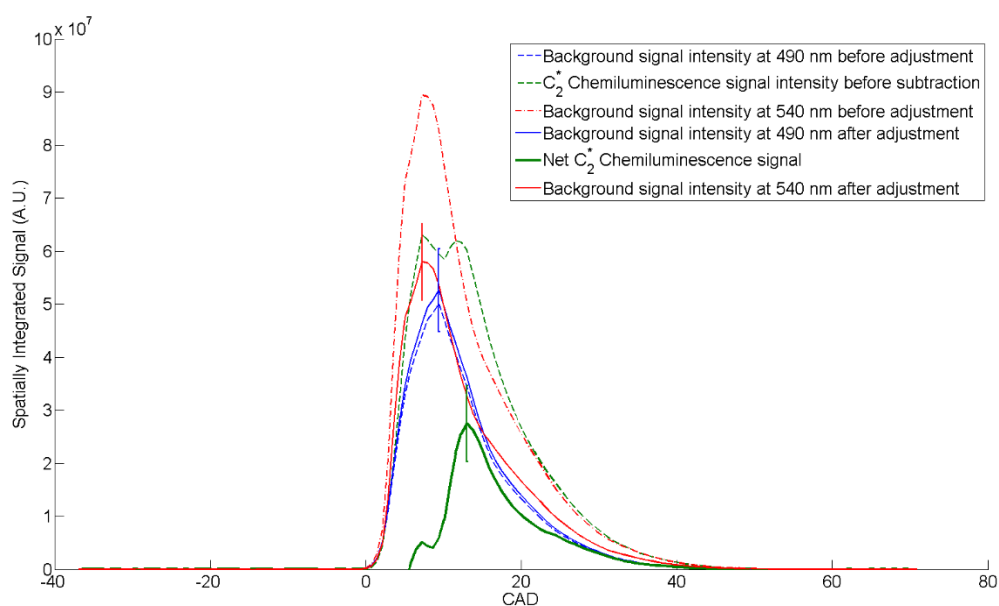


Figure 2.24 Example of the spatially integrated raw and adjusted background signals and the spatially integrated raw and adjusted C_2^* chemiluminescence signals.

2.5 Planar Laser Induced Incandescence

For the qualitative planar Laser Induced Incandescence (pLII) measurements, the laser used for exciting the soot particles was a Quanta-Ray Pro 270-10 Hz Nd:YAG laser, operating at 532 nm wavelength, with 900 mJ of energy *per* pulse, a pulse width of 2 ns and a Gaussian beam profile. Choosing the 532nm instead of the 1064nm wavelength of the laser was purely in the interest of safety, as the 532 nm laser beam is visible, contrary to the 1064 nm laser beam which is in the infrared and, thus, invisible. The laser beam was guided into the combustion chamber using appropriate mirrors compatible with this specific wavelength and transformed into a focused laser sheet, approximately 2 mm thick and 60 mm wide, using a series of cylindrical lenses from Thorlabs with focal lengths of -50 mm, 150 mm and 1000 mm. The laser sheet passed through two opposite located side windows of the combustion annulus, thereby crossing the bore and terminated at a beam dump. A LaVision FlowMaster 3S CCD camera, fitted with a LaVision VC-IRO intensifier and a 105 mm (1:1.8) UV Nikkor lens was used to capture the pLII signal through a Samrock FF01-405/10-25 filter that had a centre wavelength of 405 nm with guaranteed minimum bandwidth of 10 nm, which filtered out most of the chemiluminescence and other fluorescence signals. The spectral response of the camera and the intensifier is shown in Figure 2.25 and Figure 2.26 respectively. The camera pointed at the 45° tilted mirror and captured the incandescent soot particles through the optical piston. The intensifier delay time, relatively to the triggering signal, was 160 ns, the gate duration was 50 ns and the intensifier gain was 70%. The delay time setting was chosen so that the intensifier was energised immediately after the laser pulse had extinguished. The camera shutter opened at the same time with the intensifier, and the exposure time was 1ms. Figure 2.27 shows a schematic of the pLII setup, while Figure 2.28 shows a schematic of the synchronization scheme used in the pLII setup.

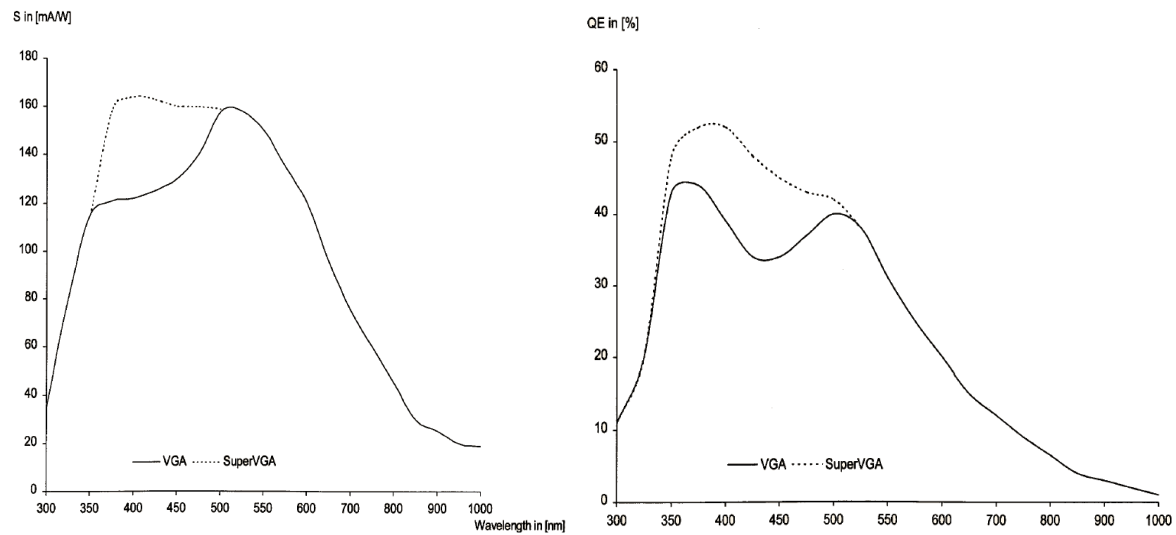


Figure 2.25 Sensitivity (left) and Quantum Efficiency (right) of the LaVision FlowMaster 3S camera (VGA curve).

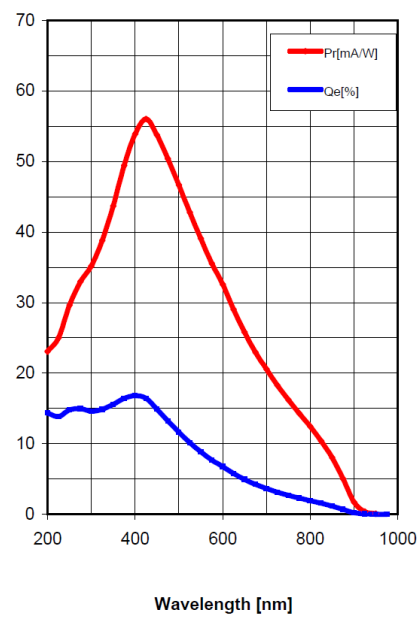


Figure 2.26 Spectral response and quantum efficiency of the LaVision VC-IRO intensifier.

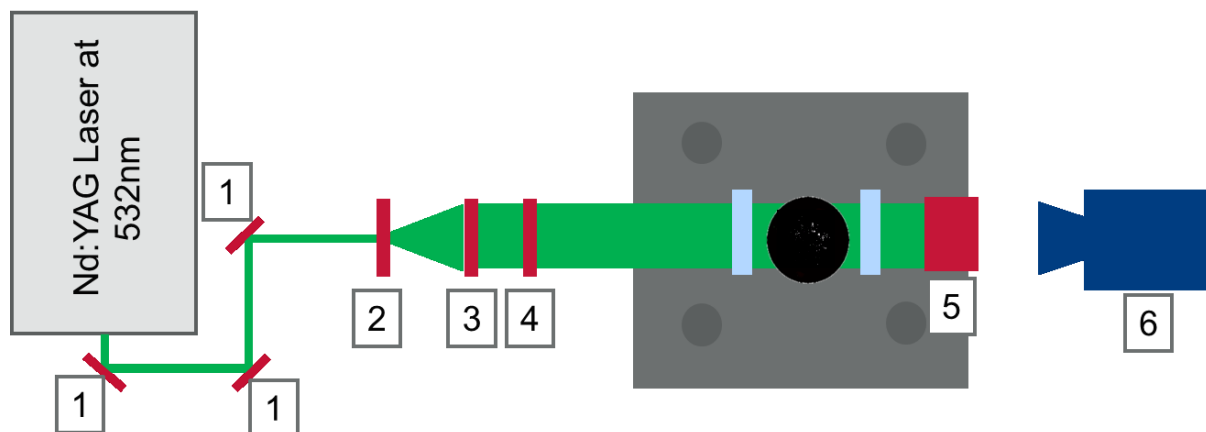


Figure 2.27 Schematic of the pLII setup. Key: 1. Mirrors, 2. $F=-50\text{mm}$ cylindrical lens to expand the laser beam, 3. $F=150\text{mm}$ cylindrical lens to collimate the expanded laser beam, 4. $F=1000\text{mm}$ cylindrical lens to focus the laser beam into a thin sheet, 5. Beam dump, 6. LaVision FlowMaster 3S camera fitted with an intensifier (Hong, 2015).

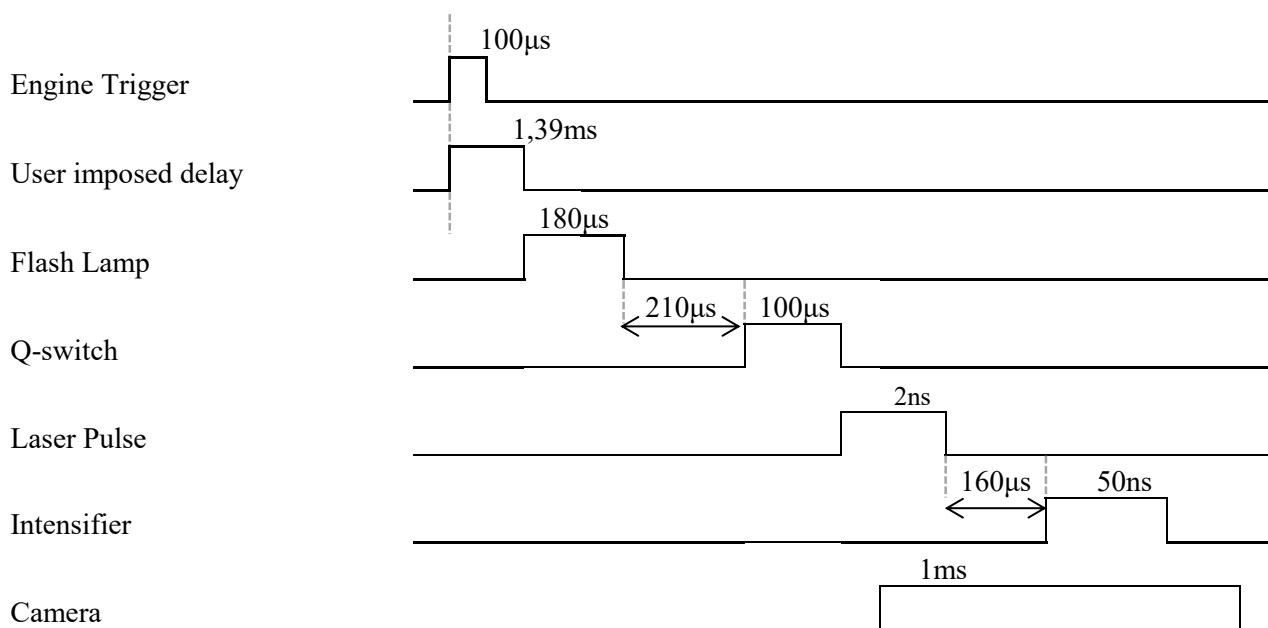


Figure 2.28 Schematic of the synchronization scheme used in the pLII setup.

For the experiments on the Ricardo Hydra engine, due to the size and the positioning of the optical windows, the laser sheet had a width of approximately 30 mm, while pLII measurements of the non-luminous soot after the end of the combustion event were obtained at 31 CAD and 50 CAD aTDC, just above the piston surface. To remove the intensifier noise a threshold of 4% of the maximum pixel intensity is adopted and thus, any pixel with intensity lower than 4% of the maximum pixel intensity is considered of zero intensity (see Figure xxx).

The final images presented are a result of intensity thresholding as described above and averaging over 100 consecutive cycles.

The Volvo D5 facilitated wider optical access into the combustion chamber due to its unique design, as explained in section 2.2. As a result, the laser sheet entering the combustion chamber had a width of approximately 60mm. There were three combinations of piston and laser position tested; laser at 31CAD aTDC with piston at 34CAD aTDC (L31P34), laser at 31CAD aTDC with piston at 54CAD aTDC (L31P54) and laser at 45 CAD aTDC with piston at 54CAD aTDC (L45P54). The final images presented are a result of intensity thresholding to remove noise caused by the intensifier and averaging over 50 cycles with a skip-firing ratio of 4. The reason behind choosing a smaller number of cycles with skip-firing was the higher sensitivity of the optical parts of the Volvo D5 engine, compared to the Ricardo Hydra engine.

To calculate the laser fluence for the LII application, the energy of the laser beam was measured just before the collimating lens and was found to be approximately 700 mJ/pulse, due to energy losses at the laser path. The area of the laser sheet cross section was approximately 1.2 cm^2 and so the laser fluence was calculated at around 0.6 J/cm^2 which is well within the plateau area of the laser fluence-LII signal dependency for a Gaussian sheet laser profile (Bengtsson and Alden, 1995).

Finally, it is important to note here that the time-gated planar LII technique in high pressure environments, applied throughout this study is inherently biased towards detecting larger size soot particles. This due to the very fast decay of the emitted LII signal from smaller soot particles and due to the dominance of the large particles' contribution to the collected signal because of their longer and stronger incandescence (Cenker *et al.* , 2015). This bias is pronounced at the chosen detection wavelength where the incandescence signal is dominated by black-body radiation of particles with temperatures above 4000K (Quay *et al.* , 1994), which are mainly larger particles with higher heat capacity than the smaller particles.

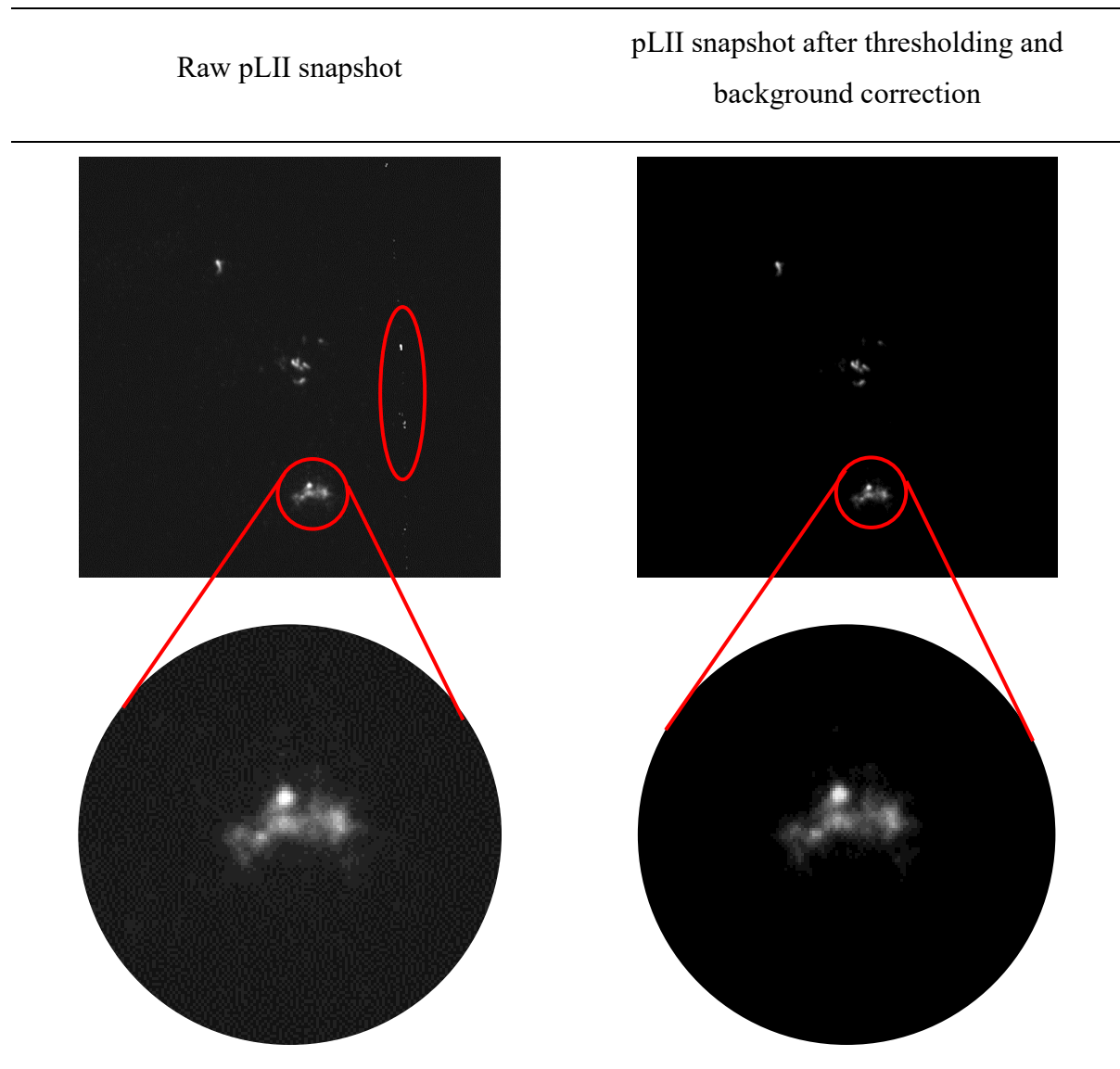


Figure 2.29 Example of thresholding in a pLII snapshot.

2.6 Chapter Summary

The physical setup of both the Ricardo Hydra optical engine and the Volvo D5 optical engine was described, along with the auxiliary instrumentation, the data acquisition process and the controlling systems. Details of the engine specifications and the fuel delivery system were cited. The methodology and the physical setup of the experimental techniques used throughout the study were elaborated.

3 Experimental work on the Ricardo Hydra Optical Engine

3.1 Introduction

The present chapter covers the experimental work on the Ricardo Hydra Optical Engine. The work comprises three different experimental campaigns, which investigate three major engine operation strategies, commonly employed by engine manufacturers. The analysis identifies combustion patterns and characteristics and describes how these are affected when different operation strategies are employed. The investigation focuses on soot emissions formation and their evolution throughout the combustion event. Most importantly, the employed operation strategies are evaluated in regard to their efficiency in reducing engine-out soot emissions.

The first experimental campaign focuses on the effect of EGR on combustion. In this regard, three different CO₂ mass flow rates (up to 7.5% of the initial mass flow) were added in the air inlet stream in order to simulate the thermal capacity and dilution effects of EGR. The engine operated at the low speed (1200 RPM) low load (1.6 bar IMEP) regime. Findings are discussed based on optical measurements, namely line-of-sight high speed imaging of C₂^{*} and CH^{*} chemiluminescence and *via* pressure trace and apparent heat transfer rate (AHTR) analyses.

The second experimental campaign investigates the effect of the injection dwell time (*i.e.* the time between two subsequent injection events) and injection pressure on soot reduction potential. The conducted experiments consider two different dwell times for three different injection pressures. The engine operated under a double injection scheme comprising one pilot and one main injection, under low load and low engine speed (*ca.* 2.8 bar IMEP at 1200 RPM) conditions. The conducted experiments considered two different dwell times (2 CAD and 4 CAD, corresponding to 278 μ s and 556 μ s at 1200 RPM respectively) for three different injection pressures (300, 500 and 800 bar). The fuel quantity of the pilot injection was kept constant at 1 mg/cycle, whereas the fuel quantity of the main injection was adjusted to maintain

the same IMEP value among all cases considered. Findings were analysed *via* means of pressure trace and apparent heat transfer rate (AHTR) analyses, as well as a series of optical diagnostics techniques, namely high-speed flame natural luminosity and chemiluminescence imaging and planar Laser Induced Incandescence (pLII) measurements at 31 and 50 CAD aTDC.

The third experimental campaign employs the aforementioned diagnostics for the assessment of the effects of various post-injection schemes on soot reduction potential. The engine operated under a multiple injection scheme of two pilot and one main injections, typical of a partially premixed combustion (PPC) mode, at the lower end of the load and engine speed range (*ca.* 2.0 bar IMEP at 1200 RPM). Experiments considering the influence of the post-injection fuel amount and the post-injection timing within the expansion stroke, under a constant total fuel mass *per* cycle, have been conducted. Findings were analysed *via* means of pressure trace and AHTR analyses, as well as a series of optical diagnostic techniques, namely high-speed line-of-sight flame natural luminosity imaging, CH* and C₂* line-of-sight chemiluminescence, as well as planar Laser Induced Incandescence (pLII) measurements at 31 and 50 CAD aTDC.

3.2 EGR Campaign

3.2.1 Introduction

Stringent vehicular exhaust emission norms, along with increased demand for high performance and comfort levels, challenge engine manufacturers nowadays (Dec, 2009; Reitz, 2013; Taylor, 2008). The quest for substantially improved thermal efficiency and reduced exhaust emissions simultaneously, has led to the introduction of innovative technological concepts and the use of alternative fuels (Epping *et al.* , 2002; Saxena and Bedoya, 2013; Stanglmaier and Roberts, 1999; Yao *et al.* , 2009). Control schemes, based on selection of multiple injection strategies, high-pressure common-rail systems and increased exhaust gas recirculation, can lead to a modification of conventional diesel combustion, towards operation that potentially exhibits characteristics of a partially premixed combustion mode (G. T. Kalghatgi, 2015; Lu *et al.* , 2011). Among the commonly considered novel operation modes, Partially Premixed Combustion (PPC) constitutes an attractive alternative, since the partially

premixed mixtures allow auto-ignition at low and high loads with high efficiency and lower NO_x and particulate matter (PM). However, relatively high emissions of hydrocarbons (UHC) and carbon monoxide (CO) may be present as well as undesired increased noise levels. The overall partial premixed combustion (PPC) mode can be possibly, effectively controlled through the use of a variety of fuel types and the exploitation of combustion phasing control strategies (Bao *et al.* , 2014; Eastwood *et al.* , 2007). Controlling the combustion phasing in PPC entails tuning the auto-ignition kinetics, as it is affected by the charge composition (global equivalence ratio and local fuel stratification at the start of combustion) and reactants temperature and pressure histories. Consequently, research has focused on describing theoretically, numerically and experimentally the governing phenomena occurring in engines operated under such conditions and overcoming its inherent weaknesses (Benajes *et al.* , 2008; Dec and Kelly-Zion, 2000; O'Connor and Musculus, 2013a; Pickett, 2005).

EGR strategies have already been implemented on conventional diesel engines aiming to reduce combustion temperature and NO_x emissions by diluting the inlet charge and increasing its heat capacity (*e.g.* Ladommatos *et al.* , 1996). Depending on the load, high EGR rates have been found to reduce NO_x but to increase soot (Soulopoulos *et al.* , 2015; Tree and Svensson, 2007). The application of EGR induces a major issue related to the fact that the start of combustion is not controlled in a straightforward way as in the no-EGR case. Complex injection schemes and a combination of high and low pressure increased EGR levels, have been employed to prolong the overall ignition delay and to enhance premixing, thus resulting in lower smoke levels. Multiple injection strategies, depending on the adopted injection schedule (Chartier *et al.* , 2011; Desantes *et al.* , 2007), could potentially introduce an appropriately stratified charge, both in terms of stoichiometry and temperature that eventually, NO_x formation would be avoided due to lower temperatures, and soot formation would be avoided as fuel rich pocket formation is diminished.

Previous results on a light-duty metal engine at Imperial College London suggest that, at low load and low speed, the effect of EGR on (particularly NO_x) emissions is eminent (see Figure 3.1). However, the effect of the application of EGR on the flame structure and its regimes, that would eventually

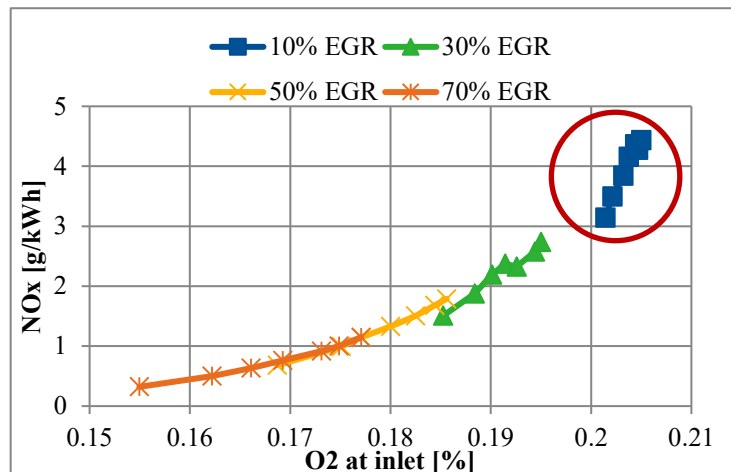


Figure 3.1 The effect of EGR on NO_x emissions in a Ford Puma I4 2.2L research engine. Even low levels of charge dilution can affect NO_x formation significantly (Private communication with N. Soulopoulos).

control emission formation, cannot be easily understood solely on the basis of global observations. Hence a campaign *via* the application of

optical techniques under the aforementioned set of operating conditions could enlighten the observed behaviour. Low levels of CO_2 are gradually added to create a cautiously controlled environment and perform a parametric study. The effects of dilution levels (up to 7.5% mass flow rate CO_2 addition) on the inlet O_2 concentration reduction are shown in Table 3.1. The engine operated with constant fuel *per cycle* under IMEP levels at the order of 1.6 bar at 1200 RPM and typical pilot/main injection schedule at 250 bar rail pressure.

The work presented in the current section focuses on the alteration of flame characteristics when the aforementioned operation strategies are applied. This is achieved *via* in-cylinder pressure analysis, high-speed imaging of ignition and combustion through CH^* and C_2^* line-of-sight in-cylinder chemiluminescence. Each of the above species is a proxy of a fundamental combustion property; CH^* to the heat release zone and the flame front, while C_2^* is indicative of fuel-rich areas. Apart from engine durability concerns, the campaign deals with low levels of IMEP because the investigation of PPC-like conditions in the lower end of the load range, is particularly important, since this load regime is currently the most plausible regime for applying such an operation strategy while retaining the use of conventional diesel fuel with existing equipment of passenger cars (Bakker *et al.*, 2014). Moreover, the investigation of low level EGR rates is in line with the findings of earlier work on a similar metal engine largely sharing the same geometry with the optical one utilized here, where the combination of low load and low EGR had resulted in significant reduction in both NO_x and smoke exhaust levels. Additionally, the small intervals in the performed simulated EGR sweeps (of the order of 2-

2.5%), can be considered as indicative amounts of the relative EGR quantity change from cylinder to cylinder or from cycle to cycle.

3.2.2 Methodology

The optical engine operated at 1200 RPM under a constant load of 1.6 bar IMEP. The air intake temperature was set at 60°C to compensate for the extensive heat losses of the optical engine, while the injection pressure was 250 bar for all cases considered. The injection schedule consisted of one pilot injection with 1.68 mg of fuel at -14.6 CAD aTDC and the main injection with 6.59 mg of fuel at -5.6 CAD aTDC.

According to Heywood (Heywood, 1988), regardless of the EGR composition, if the heat capacity of the diluent is kept constant, the dilution will have the same effect on NO_x reduction, as seen in Figure 3.2. Therefore, the CO₂ addition could have dilution/physical, thermal and chemical effects similar to EGR. Its dilution effect is primarily the change of the inlet oxygen mass fraction; here shown in Table 3.1. Thermal effects correlate with the higher, compared to air, CO₂ specific heat capacity at temperatures typical of combustion, although the opposite is valid at low temperatures, when, for example, fuel injection is realized. Finally, chemical effects are associated to CO₂ decomposition or its role in shifting kinetic pathways. The combined manifestation of the above is in turn reflected on the in-cylinder pressure trace and the respective rate of heat release.

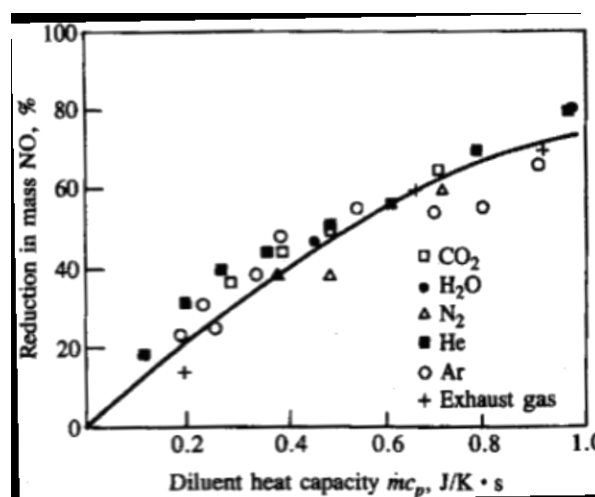


Figure 3.2 Effect of heat capacity of various diluents on NO mass reduction. (Heywood 1988)

Table 3.1 Cases studied.

Case	EGR _{rate} ^[def. equivalent] (%)	CO ₂ mass flow rate addition	Total mass flow rate increase	O ₂ mass fraction decrease
A	0	0	0	0
B	2	2.5%	0.3%	0.6%
C	4	4.9%	1.5%	1.1%
D	6.5	7.2%	3.5%	1.6%

3.2.3 Results and discussion

In the present section, analysis is performed by means of in-cylinder pressure trace, as well as means of C₂^{*} and CH^{*} high-speed chemiluminescence.

Figure 3.3 presents the in-cylinder pressure traces and Cumulative Apparent Heat Transfer (CAHT) for cases A to D. Table 3.2 shows the thermodynamic details of the combustion. It is apparent that CO₂ addition affects the overall ignition delay and reduces the maximum pressure recorded but, solely judging from in-cylinder analysis, one could argue that this is only to a limited extent (contrary to findings from optical measurements –see below) due to low levels of added CO₂. In-cylinder pressure analysis reveals that the combustion event initializes at approximately 5 CAD aTDC, CA50 is achieved at approximately 10 CAD aTDC while combustion is generally prolonged. As expected, CA50 is retarded as the CO₂ levels are increased. Moreover, the maximum cylinder pressure decreases in terms of absolute value and appears later in the combustion cycle. This scales with the reduced maximum pressure rise rate, which could possibly point towards reduced combustion noise. At the highest CO₂ dilution, *i.e.* Case D, the peak pressure is 5% lower, implying also a larger associated drop in the peak in-cylinder gas temperature.

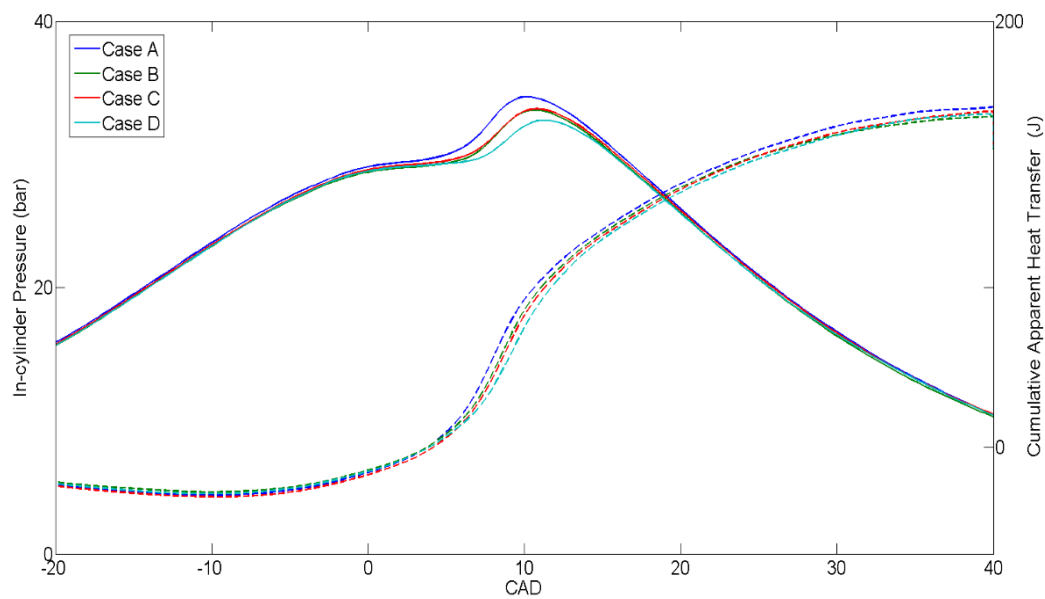


Figure 3.3 Evolution of in-cylinder pressure (left y-axis) and calculated Cumulative Apparent Heat Transfer (right y-axis) for all cases considered.

Table 3.2 In-cylinder pressure analysis results.

Case	IMEP (bar)	CA50 (CAD aTDC)	$P_{\max}$ (bar)	$P_{\max}$ Location (CAD aTDC)	$PRR_{\max}$ (bar/CAD)
A	1.6	9.7	34.3	5.1	1.4
B	1.6	10.0	33.4	5.3	1.3
C	1.6	10.2	33.4	5.4	1.2
D	1.6	10.8	32.6	5.6	1.0

Figure 3.4 presents the evolution of the spatially integrated C_2^* chemiluminescence signal, as a proxy for the intensity of the fuel-rich areas of the combustion, while Figure 3.5 shows the spatially resolved signal at the location of the maximum integrated signal for all cases considered. Although there is no clear trend regarding the intensity of the fuel rich areas in Figure 3.4, it is clear that as the CO_2 dilution increases, the fuel rich areas appear earlier in terms of CAD. In the same figure we can observe that Case C exhibits the lowest signal amongst all cases and this can be confirmed by Figure 3.5 as well. It should be noted here that,

even for the case with the most advanced combustion timing, *i.e.* Case D (see Table 3.2), the start of combustion precedes the end of injection at all times, allowing more time for the air and fuel to mix. It is observed in Figure 3.5 that the fuel-rich areas are more uniformly distributed with the increasing CO_2 , especially in Case C, which assists in avoiding dense fuel-rich regions. This remark, in combination with the low levels of C_2^* signal late at the combustion cycle for Case C may suggest a decrease in soot formation inside the cylinder, as soot originates from the fuel-rich areas, and, as a consequence, a decrease in engine-out soot emissions. However, an indication about soot oxidation is needed to further substantiate this argument.

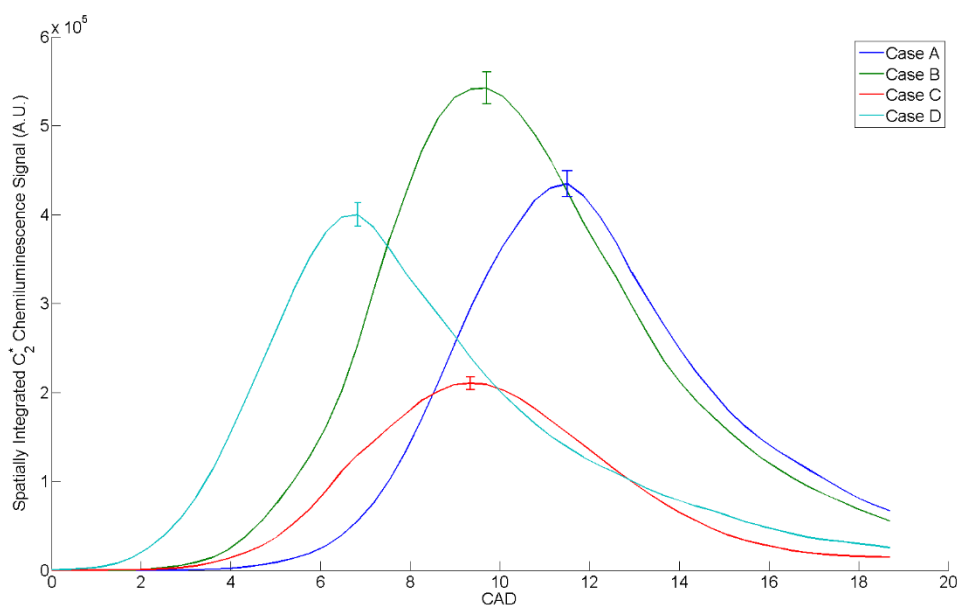


Figure 3.4 Evolution of the spatially integrated C_2^* chemiluminescence signal for all cases considered.

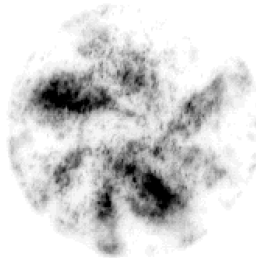
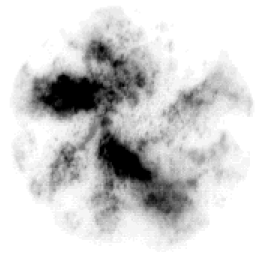
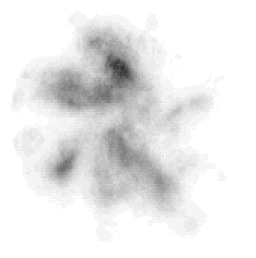
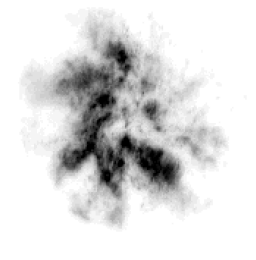
Case / Max signal Location	C_2^* spatially resolved chemiluminescence signal
A / 11.5 CAD aTDC	
B / 9.3 CAD aTDC	
C / 9.3 CAD aTDC	
D / 6.8 CAD aTDC	

Figure 3.5 Maximum spatially resolved C_2^* chemiluminescence signal for all cases considered.

Mixture charge reactivity is correlated to the rate of heat release and therefore to the intensity of CH^* signal (even in arbitrary units) as a typical proxy of the flame front marker. Here, the use of line-of-sight CH^* chemiluminescence measurements exhibits the flame zone characteristics, capturing in detail the main combustion event. Figure 3.6 presents the evolution of the spatially integrated CH^* chemiluminescence signal, while Figure 3.7 shows the spatially resolved signal at the location of the maximum integrated signal for all cases considered. The case without dilution exhibits denser and more intense areas of CH^* concentration, as compared to the zones consequent on increasing CO_2 addition. Moreover, the flame front gradually loses its distinctive pattern, which correlates with the notional axis of the fuel spray jets, and spreads across the cylinder reflecting the swirling flow, and becoming spatially homogenized. Finally, by comparing the CH^* integrated signal among the considered cases, one may argue that both reaction initialization and peak occurrence are retarded, and this is confirmed by pressure trace analysis. An unexpected observation behavior is that, although an increase of the CO_2 addition between cases B and C does not significantly affect the CH^* measurements, while Case D exhibits the lowest levels of recorded signal.

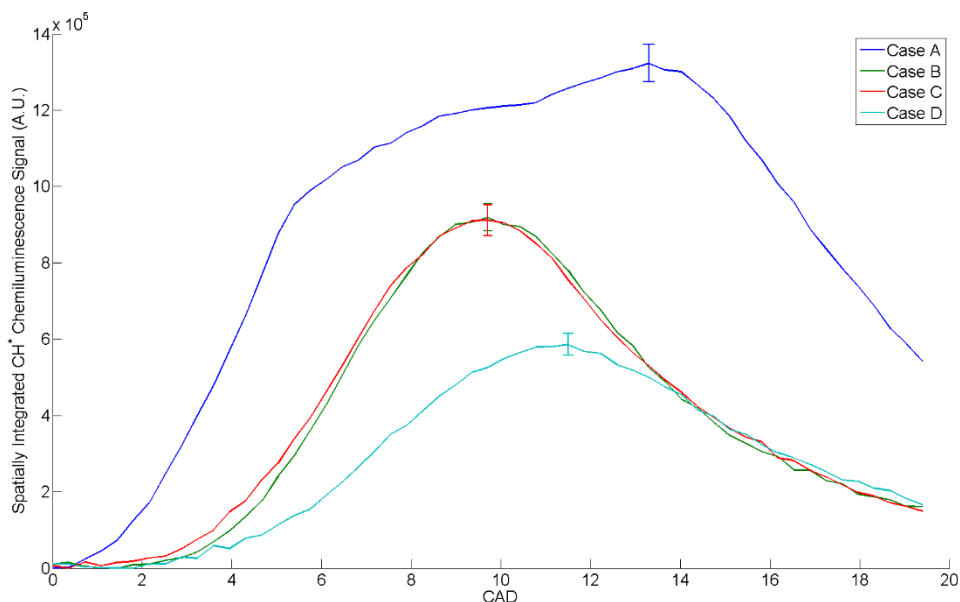


Figure 3.6 Evolution of the spatially integrated CH^* chemiluminescence signal for all cases considered.

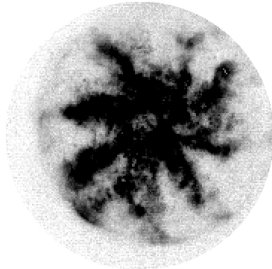
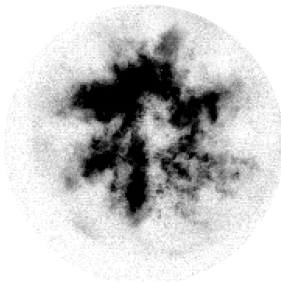
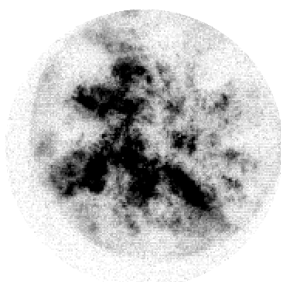
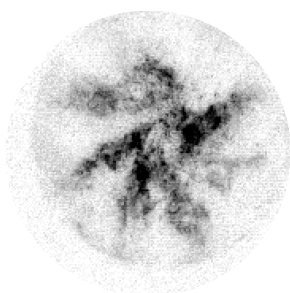
Case / Max signal Location	CH* spatially resolved chemiluminescence signal
A / 13.3 CAD aTDC	
B / 9.4 CAD aTDC	
C / 9.7 CAD aTDC	
D / 11.5 CAD aTDC	

Figure 3.7 Maximum spatially resolved CH* chemiluminescence signal for all cases considered.

The intake air dilution entrained substantial changes in the flame structure, despite the small amounts of CO₂ that were introduced. When the CO₂ content of the charge increased, the fuel rich areas inside the cylinder, as depicted by the C₂^{*} chemiluminescence, became visible earlier in terms of CAD and their intensity dropped, indicating weak soot formation. This could be attributed to the lower in-cylinder temperature, as a result of the charge dilution. The weak fuel-rich areas suggest that the dilution of the charge with CO₂ may lead to a reduction of engine-out soot emissions, due to the altered environment during injection and the retarded combustion initiation which facilitates fuel vapor and air mixing. However, the weaker reactivity of the charge with the increasing addition of CO₂, combined with the lower oxygen availability, do not advocate for the latter argument, as soot oxidation potential is reduced. If we would like to have a more definitive picture on engine-out soot emissions, we would need additional measurements to depict soot oxidation levels and non-luminous soot distribution inside the cylinder.

3.2.4 Conclusions

The strong influence of incorporating an EGR stream at low load and speed on a production engine on both NO_x and soot exhaust levels (Soulopoulos *et al.*, 2015) has sparked the interest for visualizing its effect on combustion process in an optical single cylinder engine, with an identical cylinder head and similar engine geometry. The added CO₂ levels corresponded to cases of low real composition EGR, which was previously shown to reduce significantly the exhaust emissions at the chosen inlet conditions, load and speed (Soulopoulos *et al.*, 2015). In particular, the effects of dilution levels (up to 7.5% mass flow rate CO₂ addition), were evaluated towards the direction of achieving a partially premixed combustion (PPC) operation mode under low loads of *ca.* 1.6 bar IMEP at 1200 RPM on a Hydra optical engine. Flame characteristics were captured by means of high speed imaging of the C₂^{*} and CH^{*} chemiluminescence.

In brief, the major findings of the work have as follows;

- The increase of CO₂ addition results in lower peak pressures, lower noise (PRR) and in an overall delay of the combustion process (as indicated by CA50 values), while also influences the spatial characteristics of reaction zones (CH* images) and differentiates the extent of fuel-rich areas (C₂* measurements).
- Fuel-rich areas were distributed over larger areas of the bore with increasing dilution and the formation of ultra-rich regions was avoided, as indicated by comparably lower C₂* integrated signal intensity. Considering that the total fuel mass *per* cycle is kept constant, C₂* imaging eventually suggests that higher CO₂ levels enhance the premixed areas of the charge, due to longer ignition delay.
- Areas of heat release are found to be less intense and more confined at higher dilution levels. Moreover, CO₂ addition naturally results in less total heat release levels (as indicated through the total signal of CH*). The latter is also confirmed by the measured apparent rates of heat transfer.
- There can be no definitive argument on the engine-out soot emissions, as to have one, additional measurements to depict soot oxidation levels and non-luminous soot distribution inside the cylinder are necessary.

3.3 Dwell Time Campaign

3.3.1 Introduction

Multiple injection patterns can be employed under a PPC operation mode, since the lower in-cylinder temperature during the pre-injection events does not lead to auto-ignition of the pilot fuel, and thus the desired fuel-air mixture stratification can be achieved. The injection strategy is also a means of controlling the heat release profile and combustion phasing of the cycle (*e.g.* Benajes *et al.* 2008; Anselmi *et al.* 2010; Bao *et al.* 2014). Furthermore, modern fuel injection technologies allow the implementation of higher injection pressures and close-coupled multiple injection schemes, which can accelerate liquid atomization and enhance the mixing process, partly due to the increased turbulent mixing and partly due to the increased strain rates in the gaseous phase, which may delay ignition. Therefore, a variation in the dwell time (spacing) between consecutive injection events during the same cycle, as well as in the injection pressure, can help towards achieving PPC operation mode (Hong, 2015).

In the literature, splitting a single fuel injection into two or more subsequent injection events has been identified as an effective means for reducing engine particulate emissions. Early experiments in single cylinder metal engines (*e.g.* Nehmer and Reitz 1994) suggest that for a double injection scheme *per* cycle, there is an optimal pilot fuel quantity for reducing the engine-out particulate emissions and that a shorter dwell time could lead to lower Brake Specific Fuel Consumption (BSFC). However, no apparent correlation between the dwell time and particulate emissions was found. Following those early experiments, and in an effort to understand the governing mechanism behind the reduction of particulates when using multiple injections, numerical methods were employed (*e.g.* Patterson *et al.* 1994; Han *et al.* 1996). The computations revealed that during a single injection event, fuel rich regions form at the spray tip, which are the main source of soot formation. During a multiple injection event, however, those fuel rich areas are not present, the mixture becomes leaner and more stratified and soot formation weakens.

In an effort to study experimentally the physics behind the effect of multiple injection events on combustion, various experiments have been conducted using constant volume chambers (*e.g.* Amagai *et al.* 1999; Zhang *et al.* 2001; Zhang and Nishida 2004). The aforementioned experiments reveal that a well-tuned double injection, in terms of dwell time, could shorten the

ignition delay, by altering the fuel evaporation rate inside the chamber. They also showed that the amount of premixed combustion could be controlled by the dwell time and the amount of injected fuel in the first injection, as long as it is below 25 percent of the total fuel injected, *per* cycle.

Furthermore, experiments conducted in either “metal” or optical engines outlined a number of effects associated with double injection schemes (*e.g.* Koyanagi *et al.* 1999; Lee and Reitz 2003; Ehleskog *et al.* 2007; Busch *et al.* 2015). One of the most important effects of pilot injection, and the reason why it is commonly exploited in production engines, is the combustion noise suppression caused by the shorter ignition delay and the lower heat release peak, due to the premixed combustion mode. Many experimental studies have also focused on the effect of double injection schemes on emission reduction (Gill *et al.* , 2005; Liu & Reitz, 2005; Ogawa and Miyamoto, 2000; Vanegas *et al.* , 2008; Yang and Chung, 2013). Chen S. used a 4-cylinder engine to study the effect of multiple injections on NO_x and particulate emissions (Chen, 2000). The results showed that for light duty diesel engines, the use of a small amount of pilot fuel injection (less than 1.0 mg/ stroke) leads to simultaneous NO_x and soot emission reduction, with the appropriate combination of EGR rates and injection timings. Beatrice *et al.* used one metal and one optical engine, equipped with the same cylinder head, to study various multiple injection schemes (Beatrice *et al.* , 2001). They concluded that the interaction between pilot and main injection fuel jets is highly complex, requiring further investigation. However, they mention that the optimum dwell angle between the pilot and the main injection events appears to be a strong controlling factor for emission reduction. Further investigation of this topic on an optical engine by Beatrice *et al.* suggests that the main controlling factor of air-fuel mixing with the use of a double injection scheme is the swirl motion of the charge inside the cylinder (Beatrice *et al.* , 2002). When a double injection scheme is utilized, the swirling gas flow removes the combustion products from the region of first pulse during the dwell time before the second pulse is injected. However, if the dwell is too long, the products of the first pulse can be transferred into the region of the second pulse. Therefore, proper tuning of the dwell time is necessary for soot emission control. Subsequent studies (Ehleskog and Ochoterena, 2008; Yao *et al.* , 2009) agree on the potential of soot reduction with the use of double injection schemes, because of improved air utilization that promotes soot oxidation. However, they observed no clear relationship between dwell time and net soot production.

From the previous work which was presented above, it becomes evident that there is incomplete knowledge on the mechanism behind the close-coupled pilot injections, the way they interact with the main injections and their effect on the combustion evolution and pollutant formation. The work presented in this chapter attempts to assess the effects of injection pressure and dwell time between two subsequent injections on combustion characteristics, as well as soot production and its evolution inside the bore of a single cylinder optical diesel engine. The assessment is attempted *via* means of in-cylinder pressure trace analysis, high-speed imaging of flame natural luminosity and chemiluminescence, as well as planar Laser Induced Incandescence (pLII).

3.3.2 Methodology

Table 3.3 shows the operating conditions used for the experiments. The optical engine operated at a constant speed of 1200 RPM and the engine load was kept constant at approximately 2.8 bar IMEP. All the investigated cases feature a double injection scheme, consisting of a pilot and a main injection event, with the changing variables being the injection pressure and the dwell angle between the two subsequent injection events. Three different injection pressures, namely 300, 500 and 800 bar, as well as two dwell angles, 2 and 4 CAD (corresponding to 278 μ s and 556 μ s at 1200 RPM respectively) were investigated, resulting in a total of 6 cases. The timing of the main injection was kept constant at 2 CAD bTDC, whereas the timing of the pilot injection was set to either 6 or 8 CAD bTDC. The fuel quantity of the pilot injection was kept constant at 1mg/cycle and the fuel quantity of the main injection was adjusted in each case to maintain the same engine load levels. Figure 3.8 shows, indicatively, the current of the fuel injector for two cases, with different dwell time between the injections, at 300 bar injection pressure.

Table 3.3 Injection schedule details for all cases considered. Engine speed at 1200 RPM and engine load at 2.8 bar IMEP in all cases.

Case no.	Injection Pressure (bar)	Pilot Injection Timing//Quantity (CAD aTDC //mg)	Main Injection Timing//Quantity (CAD aTDC //mg)
A	300	-8//1	-2//17
B	300	-6//1	-2//17.6
C	500	-8//1	-2//15.2
D	500	-6//1	-2//17.2
E	800	-8//1	-2//15
F	800	-6//1	-2//17

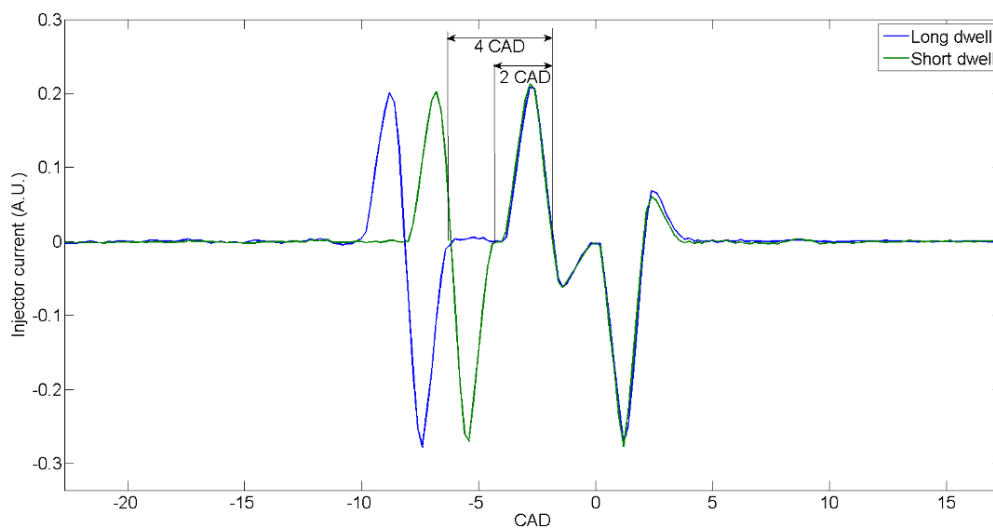


Figure 3.8 Definition of dwell angle between two subsequent injection events within the same engine cycle, based on the current of the injector. Both long dwell and short dwell cases are shown.

3.3.3 Results and discussion

Figure 3.9 shows the evolution of the in-cylinder pressure whereas Table 3.4 shows some thermodynamic characteristics of the combustion for all the cases considered. When the injection pressure increases, the peak in-cylinder pressure increases as well, from *ca.* 40 bar for Case A to 50 bar for Case F, as Figure 3.9 shows and, as a result, the maximum pressure rise rate (PRR) increases, as shown in Table 3.4. This rapid increase in pressure will lead to higher combustion noise. Moreover, higher injection pressure leads overall to advanced combustion in terms of timing, as indicated by CA50, which is the crank location where the 50% of the total heat release has occurred. An advanced combustion event seems to favour combustion efficiency, because, for the cases where the CA50 values are between 6 and 8 CAD, namely Cases C-F, less fuel *per* cycle is required to maintain the same levels of engine load for both dwell angles, as seen in Table 3.4.

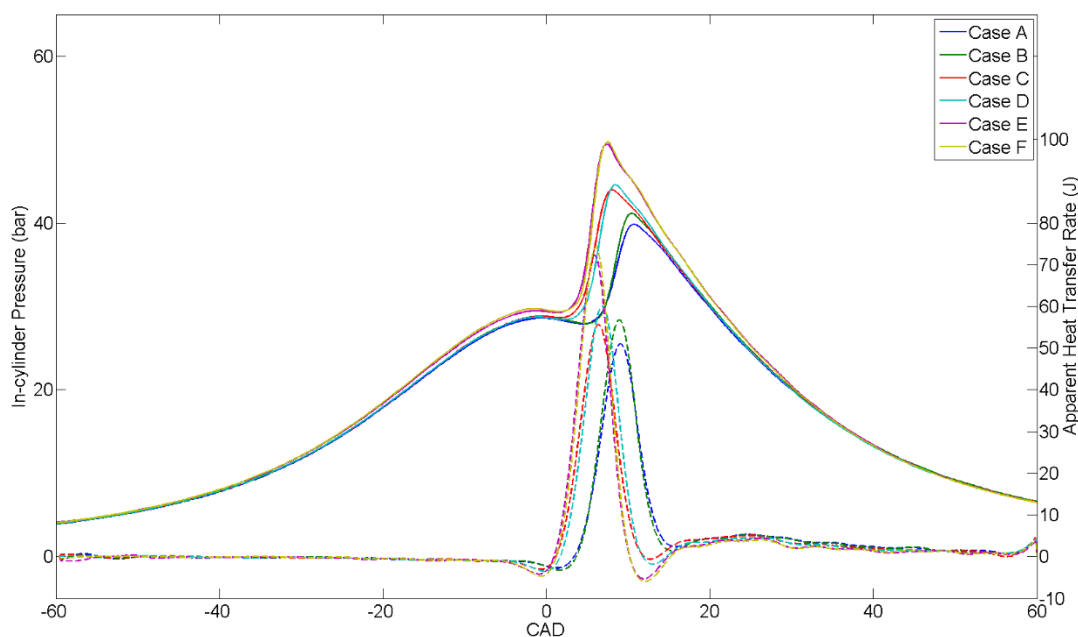


Figure 3.9 Measured in-cylinder pressure as a function of crank angle degree for all cases considered.

Table 3.4 Global combustion characteristics for all cases considered in the present work.

Case	IMEP (bar)	Max Pressure (bar)	Max PRR (bar/CAD)	CA50 (CAD)	CA90-CA10 (CAD)
A	2.8	39.9	4.1	10.8	29.8
B	2.8	41.2	5.0	10.4	28.8
C	2.8	44.0	5.4	7.6	28.2
D	2.8	44.7	6.0	7.9	28.7
E	2.8	49.6	8.2	6.4	25.9
F	2.8	49.8	8.5	6.5	25.3

The same effects as increasing the injection pressure, but less pronounced, can be observed when shortening the dwell time between the two injections. Maximum in-cylinder pressure and maximum pressure rise rate slightly increase, while combustion timing advances and combustion duration becomes shorter, especially for the 300 bar injection pressure cases, *i.e.* Cases A and B. It is important to note here that with a shorter dwell time, more fuel *per cycle* is required in order to achieve the same levels of engine load, thus deteriorating the combustion efficiency.

The Cumulative Apparent Heat Transfer (CAHT) presented in Figure 3.10 can provide a better insight into the combustion process, by revealing the different phases of combustion. When the fuel is injected, the jet formed at the nozzle exit breaks up and fuel atomizes. Then, because of the high temperature environment, it vaporizes and, because of the swirl, it mixes with the air inside the cylinder, to form a combustible mixture. This process of atomization and vaporization absorbs energy from the in-cylinder environment, which explains the initial drop in the CAHT graph (Figure 3) for all cases. When the injection pressure increases this drop becomes more evident, indicating better atomization of the fuel injected, leading, eventually, to better mixing of the in-cylinder charge. The drop is slightly pronounced when the dwell time is shorter, however, the underlying cause here is different, as it will be illustrated later with the analysis of the optical measurements.

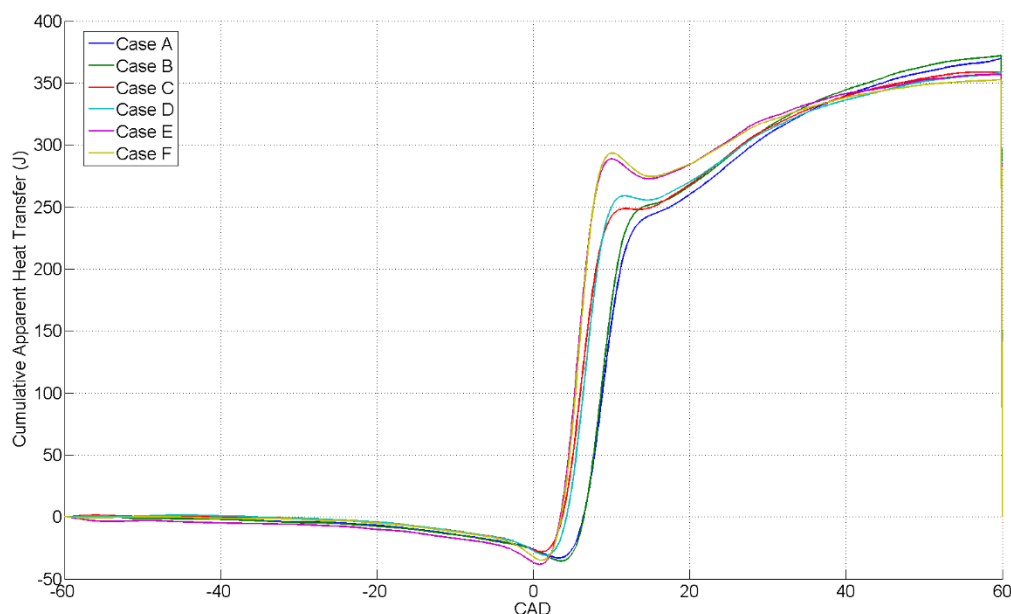


Figure 3.10 Cumulative apparent heat transfer rate as a function of crank angle for all cases considered.

The first part of the CAHT graph, after the initial drop, corresponds to the premixed phase of the combustion process and is characterized by high gradients of apparent heat transfer. With the increase of the injection pressure, or with shorter dwell time, the premixed phase becomes more rapid and reaches higher values of heat transfer in the CAHT graph, explaining the higher in-cylinder pressure and PRR values observed. Moreover, when the premixed phase of the combustion is more intense, more energy is released from the fuel earlier in terms of crank angle and so the whole combustion process advances, as indicated by the smaller values of the CA50. The diffusion part of the combustion, *i.e.* the second part of the CAHT graph characterized by smaller gradients of apparent heat transfer compared to the premixed phase, is shorter when the premixed phase is more intense, leading to overall shorter combustion duration, as described earlier.

In the following paragraphs, the analysis of the optical measurements takes place. The optical diagnostics used are, namely, high speed imaging of the flame natural luminosity and C_2^* chemiluminescence, as well as pLII for non-luminous soot detection at 31 and 50 CAD aTDC. The analysis is performed on two axes: the effect of dwell time and the effect of injection pressure.

i. Dwell time

For the analysis of the effect of dwell time, we are focusing on the 300 bar injection pressure cases, *i.e.* Case A and B. That is because the effect is more pronounced for lower injection pressures and a large portion of the combustion for these cases takes place at the inner part of the bore, where we have optical access, thus providing a better understanding on the combustion process.

Figure 3.11 shows the spatially integrated C_2^* chemiluminescence signal for all six cases considered. The recorded signal for Case B, *i.e.* the case with the short dwell, is lower throughout the expansion stroke, compared to Case A. For Case A the signal peaks at 16 CAD, whereas, for Case B the signal peaks earlier, at 14.4 CAD, and the peak value for Case B is down by 60% when compared to the peak value recorded for Case A. Later at the expansion stroke, this divergence between the recorded values for the two cases becomes larger, as at 31 CAD aTDC the recorded signal for Case B is 86% lower and at 50 CAD aTDC 96% lower. In order to get a better understanding on the spatial distribution of the fuel rich areas inside the cylinder, as depicted by the C_2^* chemiluminescence, we present the recorded images at the crank angle where peak integrated signal was observed for both cases. Figure 3.12 and Figure 3.13 show the average recorded image of C_2^* chemiluminescence at 16 CAD for Case A and 14.4 CAD for Case B respectively. The longer dwell between the pilot and the main injection leads to more intense fuel rich areas, which occupy a larger portion of the cylinder area, as seen in Figure 3.12. On the other hand, Figure 3.13 shows that with the shorter dwell, these fuel rich areas become less intense and limited towards the inner part of the cylinder. It is also important to note here that for the long dwell case the fuel rich areas coincide with the plumes formed by the fuel jets during the injection, whereas for the short dwell case they appear more homogenized, losing their distinctive pattern. From the previous analysis, we can infer that, as soot originates from the fuel rich areas of the combustion (M. I. Najafabadi *et al.* , 2017), for the short dwell case the generation of soot inside the cylinder is reduced and becomes more spatially homogenized, compared to the long dwell case.

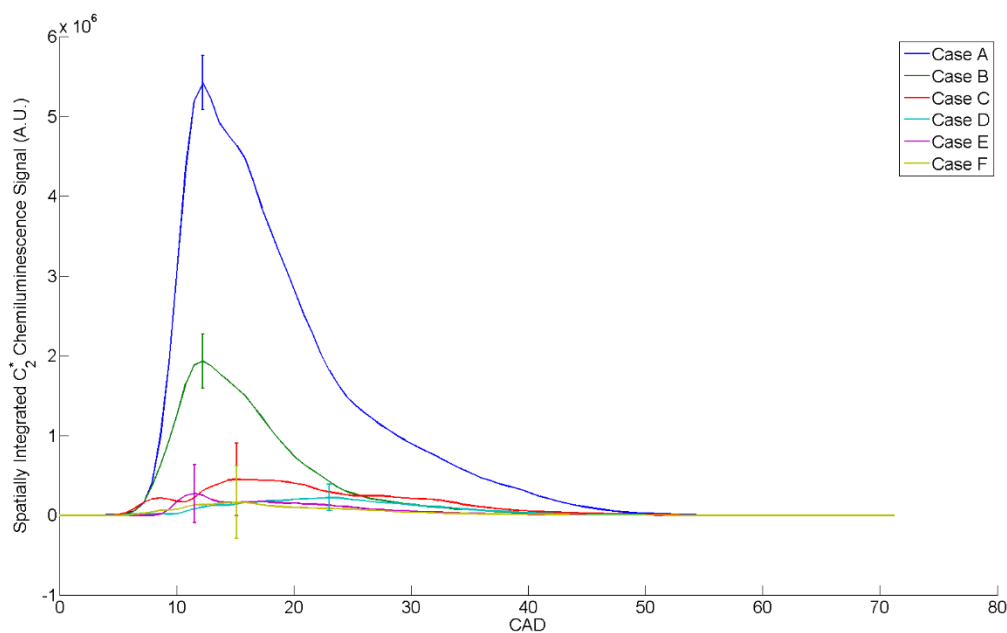


Figure 3.11 Spatially integrated C_2^* chemiluminescence signal as a function of crank angle for all cases considered.

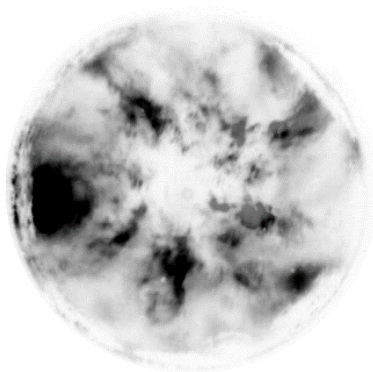


Figure 3.12 Average spatially resolved C_2^* chemiluminescence signal at 12.2 CAD aTDC for Case A.

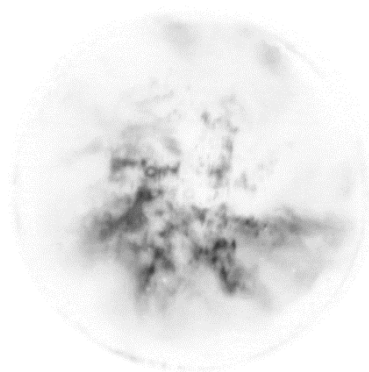


Figure 3.13 Average spatially resolved C_2^* chemiluminescence signal at 12.2 CAD aTDC for Case B.

Figure 3.14 shows the spatially integrated natural flame luminosity signal for all six cases studied. Comparing Cases A and B again, we notice that the soot oxidation rate of the short dwell case is higher than the long dwell case, as inferred by the steeper negative slope of the respective curve in Figure 3.14. Additionally, the short dwell case (Case B) overall exhibits

lower luminous soot concentration than the long dwell case (Case A) late at the combustion cycle (after 30 CAD aTDC), thus featuring a higher soot reduction potential.

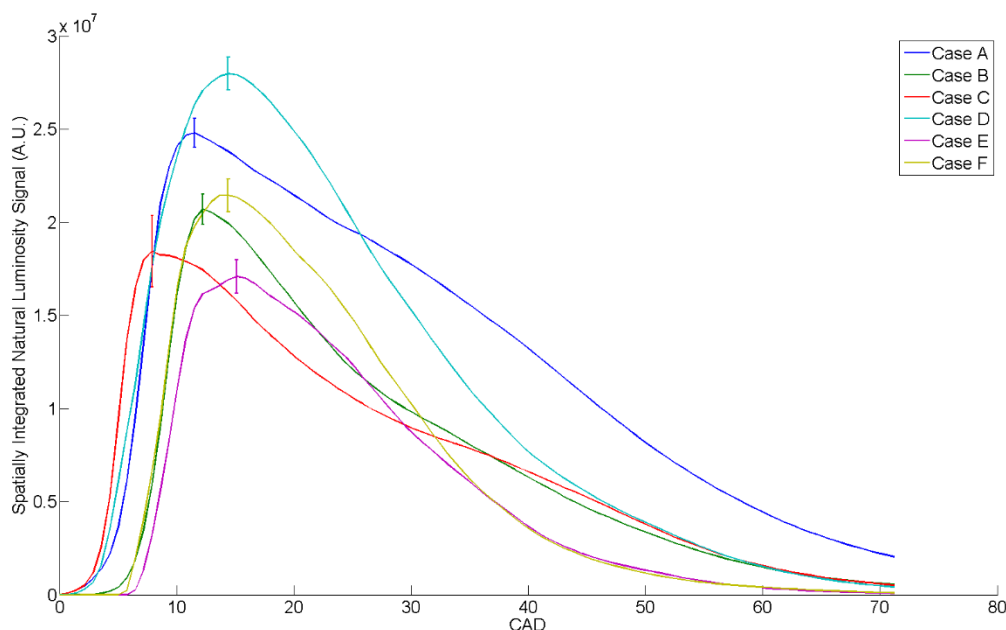


Figure 3.14 Spatially integrated natural luminosity signal as a function of crank angle for all cases considered.

So far, we have discussed the two opposing forces that affect the soot distribution inside the cylinder: soot generation inside the fuel rich areas of the charge and soot oxidation as indicated by natural luminosity signal characteristics. To get a better view on engine-out soot emissions, we need to observe the pLII measurements.

Figure 3.15 shows a juxtaposition of chemiluminescence measurements and pLII measurements for two different crank angles, namely 31 and 50 CAD aTDC for Cases A and B. The chemiluminescence measurements are superimposed using different colours for each species: white for the natural luminosity of the flame, green for the C_2^* and red for the CH^* . Figure 3.16 shows the spatially integrated pLII signal for both crank angles and both cases.

First, it is important to note in Figure 3.15 that there is a strong correlation between the areas where soot is captured by the pLII and the fuel rich areas of the cylinder charge (green colour). This further supports the argument made about the soot generation from the fuel rich pockets. Interestingly, as seen in Figure 3.16, for the long dwell case, the recorded pLII signal is higher

at 50 CAD than at 31 CAD, implying that soot generation still surpasses soot oxidation, because of the intensified fuel rich zones. For the short dwell case, the recorded pLII signal at 50 CAD is lower than at 31 CAD, as soot oxidation dominates over soot generation. Therefore, since soot chemistry freezes later at the expansion stroke, the short dwell case will exhibit lower engine-out soot emissions.

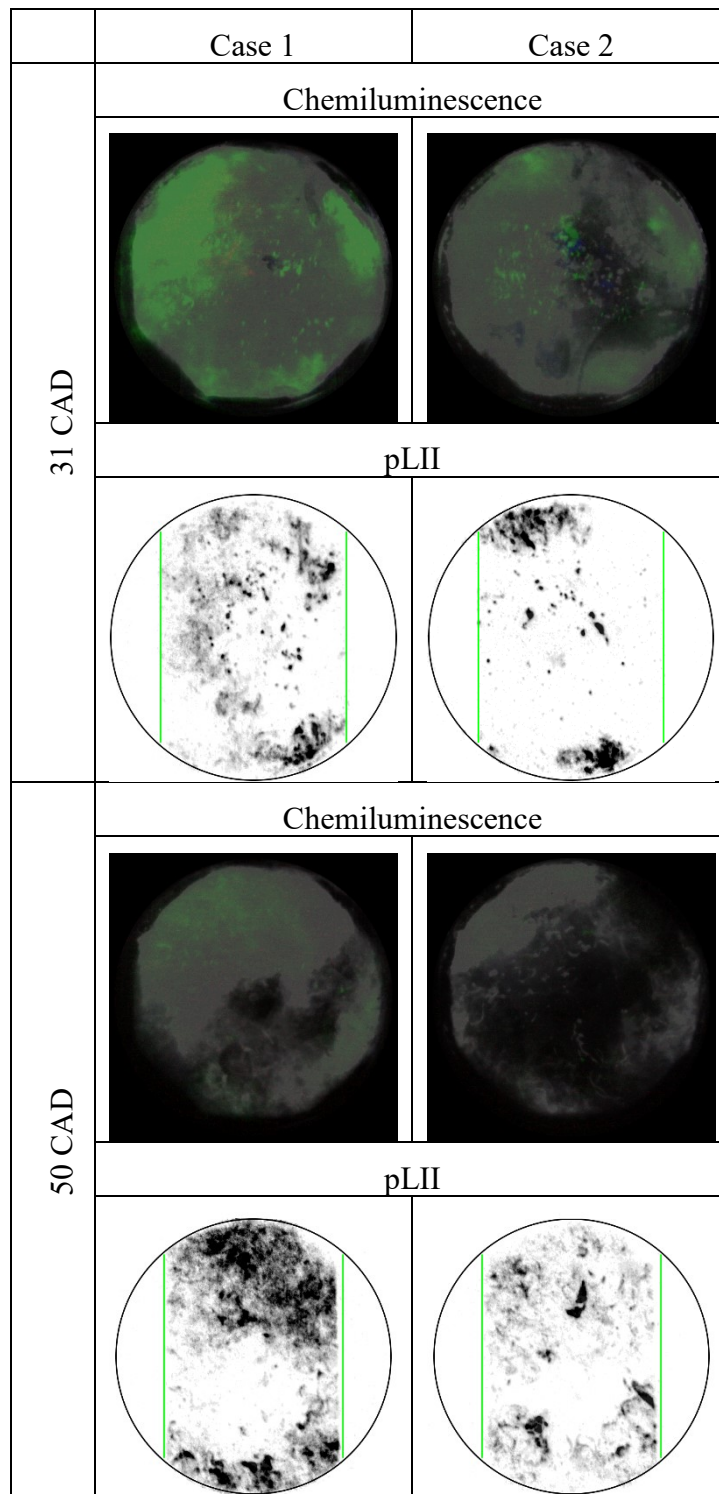


Figure 3.15 Rows 1 and 3: Superimposed average images of flame spectroscopic measurements with natural luminosity (white), CH^* chemiluminescence (red) and C_2^* chemiluminescence (green); Rows 2 and 4: The corresponding planar LII imaging. The black circle represent the optical window contour, whereas the green lines represent the laser sheet contour.

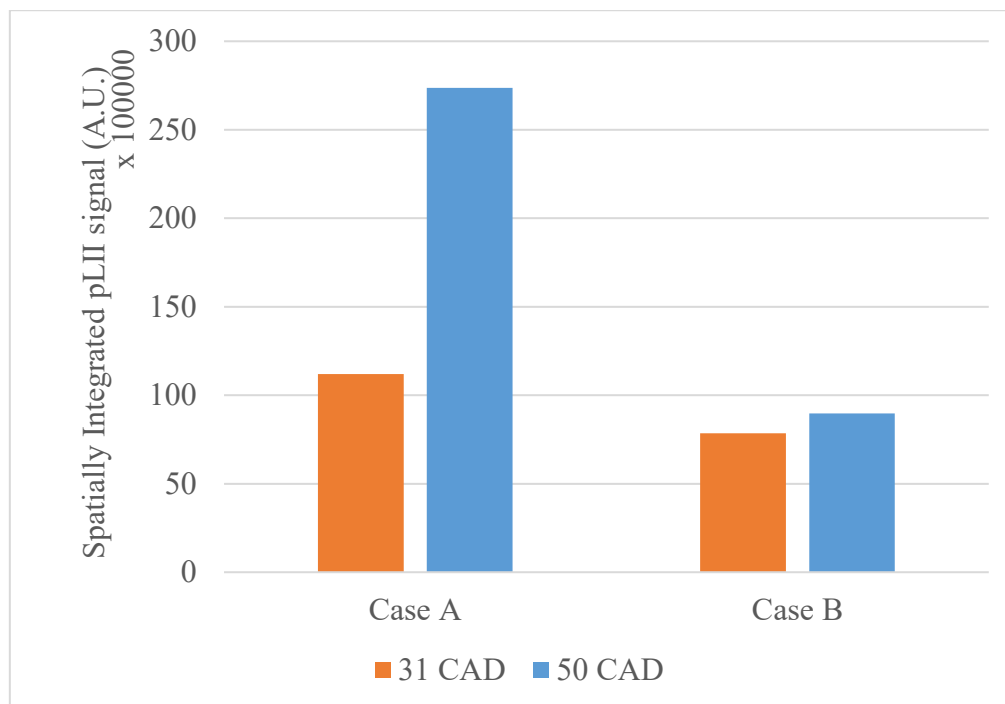


Figure 3.16 Spatially integrated pLII signal for Cases A and B at 31 and 50 CAD aTDC.

At this point, we have already discussed the effect of dwell time on combustion behaviour and the implications on soot emissions; however, we have not yet referred to the cause of this different behaviour.

Figure 3.17 shows a sequence of instantaneous natural luminosity images for Cases A and B at selected crank angles. The first visible auto-ignition spot for Case A can be observed at 0.7 CAD aTDC, whereas for Case B appears later, at 5.8 CAD aTDC. As the first visible auto-ignition spot indicates the onset of the diffusion phase of the combustion, we can deduce that combustion in Case A enters the diffusion phase earlier and while the main injection is still in progress, as seen at the image corresponding to 2.2 CAD aTDC. On the contrary, combustion in Case B enters the diffusion phase at 5.8 CAD aTDC, very close to the end of the main injection event. This delay in the onset of the diffusion phase allows for better stratification of the in-cylinder charge, by giving the fuel of the main injection more time to mix with the rest of the charge, leading, as seen earlier, to enhanced premixed combustion phase and less intense, more homogeneous fuel-rich zones.

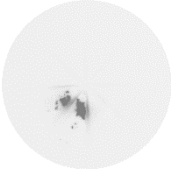
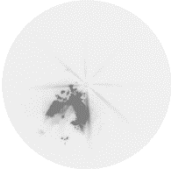
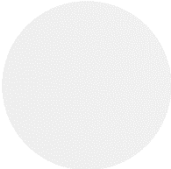
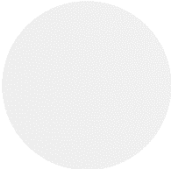
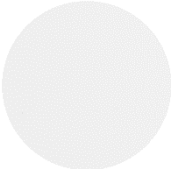
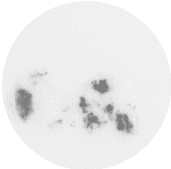
	CAD aTDC			
	0.7	2.2	5.8	7.9
Case A				
Case B				

Figure 3.17 Instantaneous Natural Luminosity images at selected CAD for Cases A and B.

The reason for this delay of visible auto-ignition in Case B over Case A can be attributed to the injection timings. In the short dwell case (Case B) the plume of fuel vapour from the pilot injection has little time available to rotate (due to the swirling flow) away from the axis of the jet. When the main fuel is injected, the pilot fuel cools down and the visible, hot auto-ignition is delayed. On the contrary, for the long dwell case (Case A) the plume of the pilot fuel has more time available to rotate away from the jet axis and to heat up, so its capability to auto-ignite remains somewhat unaffected by the main fuel injection.

The point raised in the previous paragraph emphasizes the essence of the PPC mode. Better pre-mixing and charge stratification can be achieved, not only by using advanced injection timings, but also by placing the individual injection events properly.

ii. Injection pressure

The injection pressure has significant effect on the thermodynamics of the combustion, by enhancing atomization and vaporization of the fuel, thus favouring the mixing of the in-cylinder charge and the premixed phase of the combustion. In the present section, we investigate the effect of injection pressure using optical diagnostics.

In Figure 3.11 we can see that the total C_2^* decreases drastically with the increase of the injection pressure. Therefore, the fuel rich areas inside the cylinder diminish and soot generation is weak. This behaviour can be attributed to the enhanced mixing caused by the increase of injection pressure; however, we should note that limited optical access can affect the measurements. It is interesting to observe the peak in the C_2^* signal for the high-pressure cases with the long dwell, *i.e.* Cases C and E. The peaks appear at around 15 CAD aTDC for both cases and they indicate the presence of fuel rich areas within the inner part of the cylinder, close to the injector tip. Figure 3.18 and Figure 3.19 show, indicatively, the instantaneous spatially resolved C_2^* signal at 15 CAD aTDC for Cases E and F (800 bar injection pressure) respectively. The short dwell case (Case F) exhibits lower intensity in the fuel rich areas formed close to the injector tip than the long dwell case (Case E), indicating that the short dwell limits the generation of soot in the area around the injector tip, an area that is the main source of soot at high injection pressures.

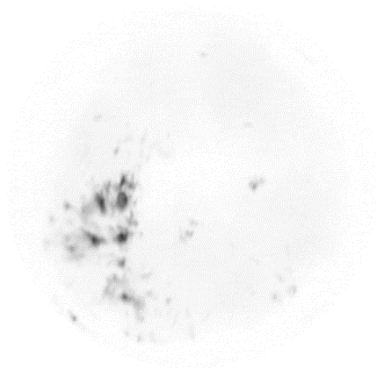


Figure 3.18 Instantaneous spatially resolved C_2^* signal at 11.5 CAD aTDC for Case E.

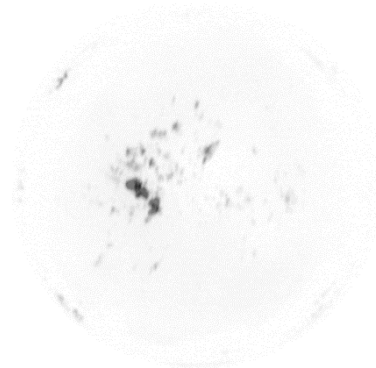


Figure 3.19 Instantaneous spatially resolved C_2^* signal at 15 CAD aTDC for Case F.

As far as the natural luminosity of the flame is concerned, in Figure 3.14 it is evident that the trend that existed at 300 bar injection pressure is reversed for the cases featuring higher injection pressures. The SINL signal is higher for the short dwell cases (Cases D and F) when compared to the respective long dwell cases (Cases C and E).

It is interesting to note that for the 800 bar injection pressure cases, *i.e.* Cases E and F, the natural luminosity of the flame, whose presence indicates the diffusion phase of the combustion, appears instantly at around 10 CAD aTDC. In Figure 3.20 and Figure 3.21, which show the instantaneous spatially resolved natural luminosity signal at 11 CAD aTDC for Cases E and F respectively, the natural luminosity signal forms a “cloud”, which indicates high homogeneity of the charge and suggests that volumetric ignition and combustion takes place, rather than flame propagation. There is an exception of some local spots of high natural luminosity signal, which occur as a result of the local fuel rich spots mentioned earlier. This instant appearance of the natural luminosity signal explains what can be observed in Figure 3.10, for the high-pressure cases, as a drop in the CAHT curve, indicating a dissociation between the premixed and the diffusion phase of the combustion.

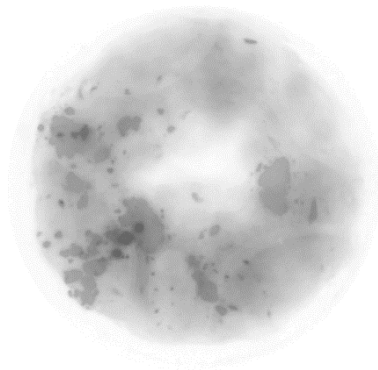


Figure 3.20 Average spatially resolved Natural Luminosity signal at 11 CAD aTDC for Case E.

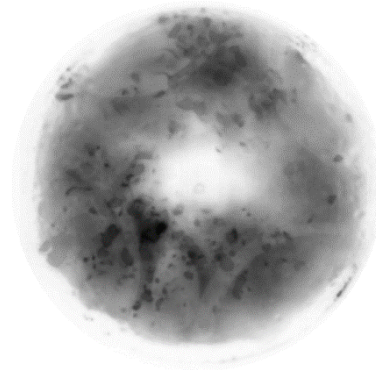
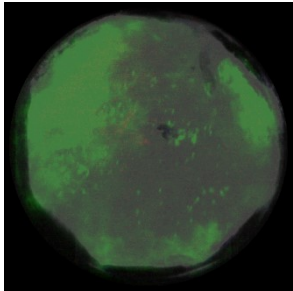
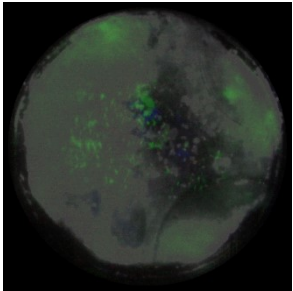
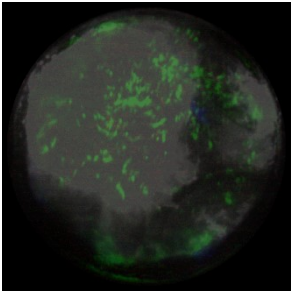
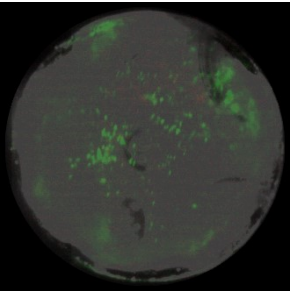
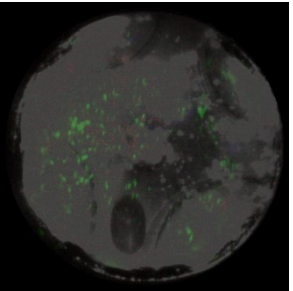
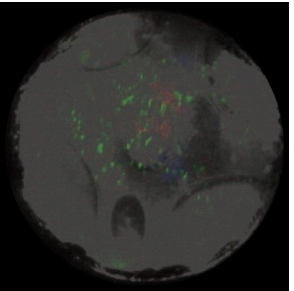
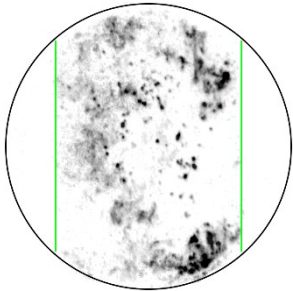
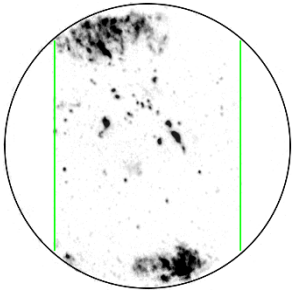
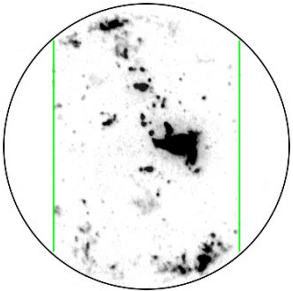
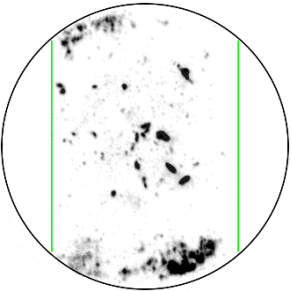
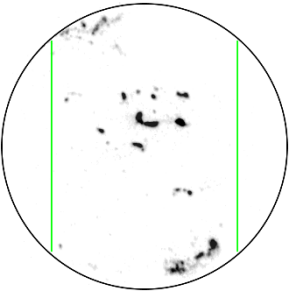
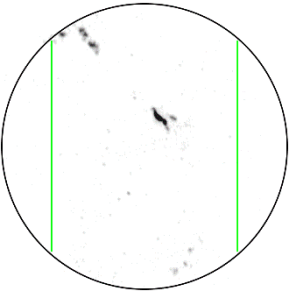


Figure 3.21 Average spatially resolved Natural Luminosity signal at 11 CAD aTDC for Case F.

Figure 3.22 shows a juxtaposition of chemiluminescence measurements and pLII measurements for two different crank angles, namely 31 and 50 CAD for Cases A to F. The chemiluminescence measurements are superimposed using different colours for each species: white for the natural luminosity of the flame, green for the C_2^* and red for the CH^* . Figure 3.23 the spatially integrated pLII signal for both crank angles and all cases considered.

In Figure 3.23, the trend between the injection pressure and the soot formation is clear. High injection pressure leads to low levels of soot, late at the expansion stroke and, arguably, at the exhaust. This is a result of weak soot formation, combined with intense soot oxidation, as demonstrated earlier. In Figure 3.22, it becomes evident that, for the 800 bar injection pressure cases, *i.e.* Cases E and F, the distribution of soot at 31 CAD aTDC is mainly attributed to the fuel rich spots close to the injector tip, as it has been previously discussed. At 50 CAD aTDC soot particles can only be observed at the lip of the bowl and the integrated pLII signal intensity is lower than it was at 31 CAD aTDC. We should note here that the 500 bar cases, *i.e.* Cases C and D, seem to be the intermediate step between the low pressure and the high pressure cases, when it comes to soot distribution and signal intensity.

	Case A	Case B	Case C	Case D	Case E	Case F
31 CAD	Chemiluminescence		Chemiluminescence		Chemiluminescence	
						
	pLII		pLII		pLII	
						

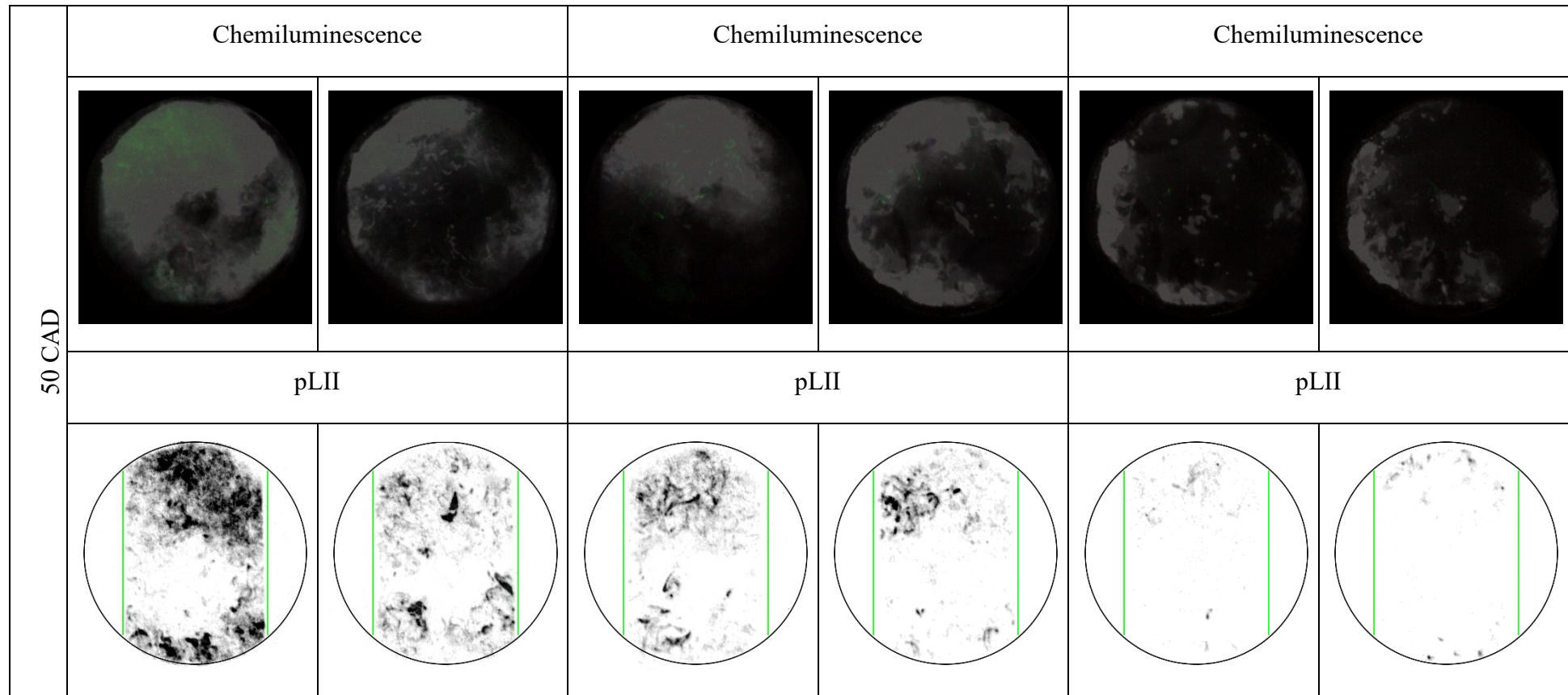


Figure 3.22 Rows 1 and 3: Superimposed average images of flame spectroscopic measurements with natural luminosity (white), CH^* chemiluminescence (red) and C_2^* chemiluminescence (green); Rows 2 and 4: The corresponding planar LII imaging measurements at 31 and 50 CAD aTDC respectively. The white continuous lines represent the optical window contour, whereas the green dashed lines represent the laser sheet contour.

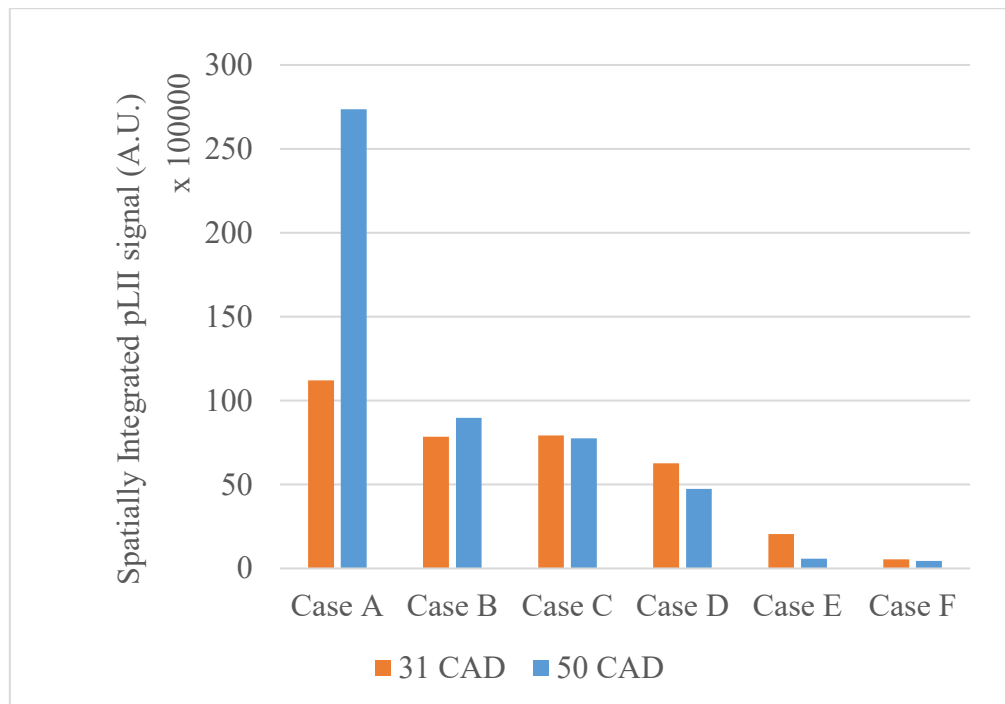


Figure 3.23 Spatially integrated pLII signal for all cases considered at 31 and 50 CAD aTDC.

3.3.4 Conclusions

The effect of the injection dwell time (*i.e.* the time between two subsequent injection events) and injection pressure on soot reduction potential was experimentally investigated in an optical, single cylinder, light duty diesel engine, operated in PPC mode under low load (2.8 bar IMEP) and low speed (1200 RPM) conditions. Results are deduced *via* a combination of in-cylinder pressure trace analysis, spectroscopic chemiluminescence measurements and LII signal at 31 and 50 CAD aTDC.

In brief, results indicate:

- The competition between soot generation and soot oxidation is the determinant factor for the engine-out soot emissions. As soot is being generated in the fuel rich areas of the cylinder, as depicted by the C_2^* chemiluminescence, better mixing inside the cylinder creates less fuel rich areas, thus, weak soot generation.
- A short dwell time causes longer delay for the first hot spot at the fuel injected during the pilot injection to appear, presumably because of the cooling induced by the fuel injected during the main injection. A longer delay provides the charge with more time

to mix, thus enhancing the mixing process and the premixed phase of the combustion, while suppressing the diffusion phase.

- An increased injection pressure enhances the atomization and vaporization of the fuel, which, in combination with the in-cylinder flow, enhances the pre-mixing of the charge. In this case, too, the premixed phase of the combustion is pronounced, while the diffusion phase is suppressed.
- In both cases, then, soot formation weakens, while, as observed in the flame natural luminosity signal evolution, soot oxidation remains at the same order of magnitude. This leads to a reduction in the non-luminous soot levels, detected using the pLII technique, late at the expansion stroke. Consequently, the case with the high injection pressure and the short dwell exhibits the lowest levels of detected soot and, presumably, the lowest engine out soot emissions amongst all cases considered.

To conclude, the present work shows that better premixing and charge stratification can be achieved, not only by using advanced injection timings, but also by placing the individual injection events properly.

3.4 Post-injection Campaign

3.4.1 Introduction

In the previous section we investigated a pilot-main injection scheme the dwell time between the two injections as an effective means of reducing engine-out soot emissions in a compression ignition engine. Other commonly employed injection schemes incorporate both pilot and post injections to further reduce engine-out soot emissions.

O'Connor and Musculus provide a succinct review on the application of post injections for soot emission reduction in diesel engines (O'Connor and Musculus, 2013b). Post-injection of a carefully chosen fuel quantity at appropriate timing, enhances charge mixing creating thereby favourable conditions to enhance soot oxidation *via* an increase in the local heat release. The dwell between main and post injection also directly influences the number and mass of soot particles and a relative optimization can potentially lead to a promotion of the soot oxidation rates favouring nanostructures. A late post injection assists the processes of the exhaust gas after-treatment systems, in particular promoting regeneration in DPFs (*e.g.* Mohan *et al.* 2013), and generates hydrocarbons for an active lean NO_x catalyst (*e.g.* Yamamoto *et al.* 2006).

Generally, optimization of the post-injection scheme can influence the relative strength of the accumulation and the nucleation modes, resulting in significant changes in the size and the number of the soot particles (Desantes *et al.* , 2007, 2011; Li *et al.* , 2015; Xu *et al.* , 2014). In LTC engines, the beneficial effects of post-injections on soot reduction, stemming from the variation of the post injection quantity and timing, are quite sensitive to engine speed and load conditions, EGR levels, and engine operation strategy (*e.g.* Chen 2000; Chartier *et al.* 2011; Frapolli *et al.* 2014). Moreover, the effect on other emissions, *e.g.* UHC and CO, should also be appropriately addressed, especially under operation conditions that promote their formation, such as PCCI conditions (*e.g.* Park and Bae 2013; Martin *et al.* 2016).

Soot-formation processes in mixing-controlled diesel flames have been well characterized, but relatively little is known about the amount and distribution of soot in the absence of appreciable heat release during the expansion and exhaust strokes. Further, although the beneficial effect of post injection on soot reduction stems both from late cycle mixing and the associated temperature increase, the dominant mechanisms can vary significantly depending on the engine

operating conditions and it is hard to establish clear guidelines for the application of such schemes at a practical level. The latter suggests that the development of methodologies able to illuminate soot oxidation processes, rather than just quantifying engine out soot emissions, might be helpful. Laser Induced Incandescence (LII) is based on the incandescence (or thermal radiation) signal induced by a laser excitation; soot particles are heated-up by absorbing laser light, leading to a strong non-linear increase in thermal radiation with temperature (H. A. Michelsen *et al.*, 2015). Soot volume fraction is approximately proportional to both the integrated signal and peak signal level and particle sizes may be deduced from the decay rate of the LII signal (*e.g.* Melton 1984; Bladh *et al.* 2008). The technique has been applied for the understanding of soot fundamentals in flames, *e.g.* Schulz *et al.* 2006; Tropea *et al.* 2007; Desgroux *et al.* 2013, as well as in combustion vessels, *e.g.* Idicheria and Pickett 2005; Vogel *et al.* 2013. LII has also been used to study combustion in engines operated under conditions pertinent to the LTC and PPC modes. Crank angle resolved LII measurements have been utilized to study the effect of in-cylinder soot oxidation *via* the recording of soot particle sizes in an optical high speed diesel engine operated under relative early injection timings (Menkiel, 2012). The in-cylinder soot volume fraction derived from time resolved LII data revealed that, for different fuel injection timings, a relatively higher amount of soot was produced for shorter ignition delays compared to longer ignition delays. This can be correlated with the effect that diffusion-burn phase has on soot because, for shorter ignition delay, the fuel has less time available for mixing, leading to prolonged diffusion-burn phase and, consequently, to higher soot generation. Further, LII images of soot distribution at various crank angle positions have been combined with natural luminosity images in order to assess in-cylinder soot production as a function of ambient O₂ mole fraction in a heavy-duty diesel (Aronsson *et al.*, 2010).

The aim of the work presented in this section is to assess the potential of post-injection strategies (fuel mass quantity and dwell time from main injection) to suppress in-cylinder soot levels and develop phenomenological understanding of the associated observations. A combination of in-cylinder pressure analysis, high speed imaging of time resolved line-of-sight CH* and C₂* in-cylinder chemiluminescence measurements, and qualitative planar Laser Induced Incandescence (pLII) measurements after the end of the combustion process, evaluated the effect of each post-injection strategy on the flame evolution patterns, the heat release and global combustion characteristics and soot reduction potential.

3.4.2 Methodology

Table shows the operating conditions used for the experiments. The optical engine operated at a constant speed of 1200 RPM and the inlet temperature was kept constant at 60°C. Four different post-injection schemes were examined with both post-fuel amount and injection timing being independent experimental parameters. In particular, the post-injection fuel amount varied between 7% (1 mg) and 15% (2 mg) of the total amount of fuel *per* cycle, while three injection timings were considered at 5, 10 and 15 CAD aTDC. The details of all post-injection schemes are shown in Table 1. In each experiment, a multiple pilot injection scheme was retained (1 mg or 7% of the total fuel amount for each pre-injection), while the main injection event occurred at the same time (-5.4 CAD aTDC) and its fuel amount was adjusted in order to have a total of 15 mg of fuel injected in every cycle (including the post injection). The overall conditions have been chosen to represent a PPC-like operation. A “Base Case” without any post injection is also considered for comparison.

Table 3.5 Injection schedule details for all cases considered. Engine speed: 1200 RPM in all cases.

Case	Pre-injections (CAD)	Pre-injections (mg/cycle)	Main injection (CAD)	Main injection (mg/cycle)	Post-injection (CAD)	Post-injection (mg/cycle)
Base Case	-14.6/-11.2	1/1	-5.6	13	-	-
A	-14.6/-11.2	1/1	-5.6	12	5	1
B	-14.6/-11.2	1/1	-5.6	11	5	2
C	-14.6/-11.2	1/1	-5.6	11	10	2
D	-14.6/-11.2	1/1	-5.6	11	15	2

3.4.3 Results and Discussion

The recorded pressure traces and the corresponding apparent heat transfer rates (AHTRs) are presented in Figure 3.24, while Table 3.6 summarizes the global combustion characteristics for all cases considered in the present work. Figure 3.25 presents the Cumulative Apparent Heat Transfer (CAHT) rate profiles for all cases. Note that the trends in the final values of CAHT match the trends in the IMEP values, see also Table 3.6.

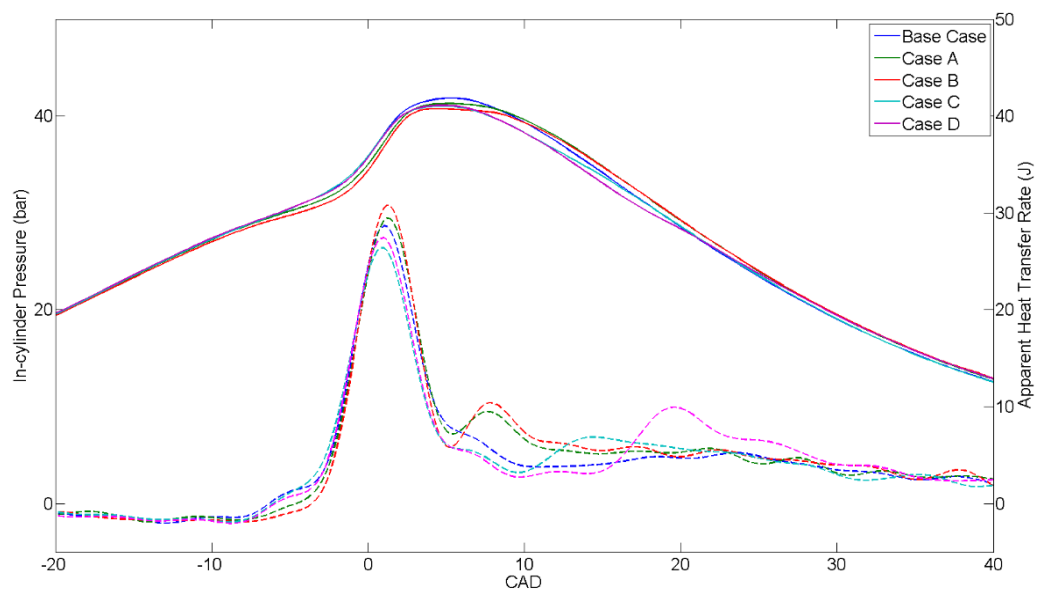


Figure 3.24 Measured in-cylinder pressure (solid lines; left y-axis) and calculated apparent heat transfer rate profiles (AHTR; dashed lines; right y-axis) as a function of crank angle degrees for all cases considered.

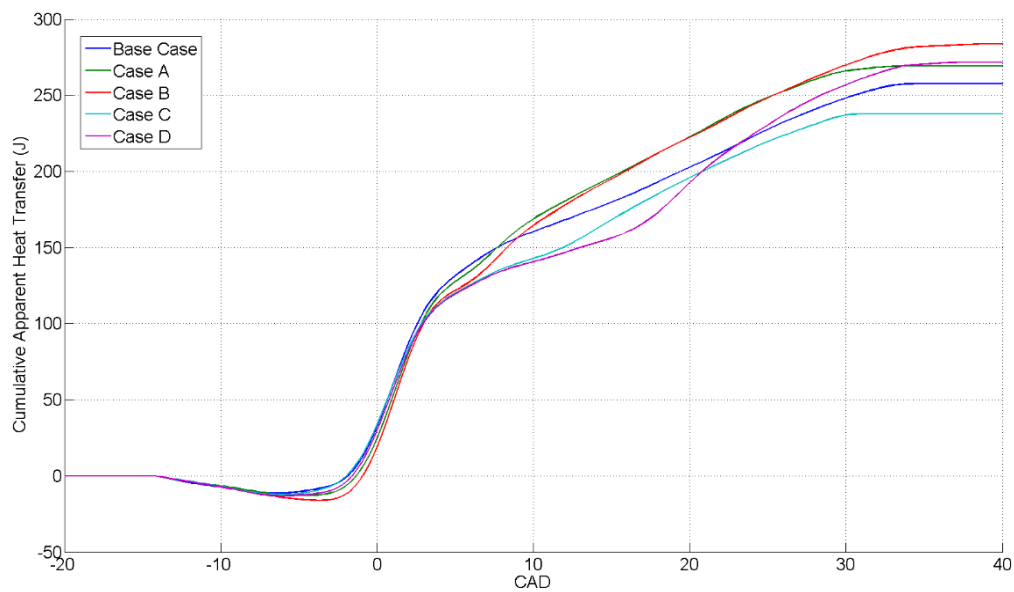


Figure 3.25 Cumulative Apparent Heat Transfer as a function of crank angle for all cases considered.

Table 3.6 Global combustion characteristics for the cases considered in the present work.

Case	IMEP (bar)	CA10 (CAD aTDC)	CA50 (CAD aTDC)	CA90 (CAD aTDC)	CA90- CA10 (CAD)	Max Pressure (bar)	Max Pressure Rise Rate (bar/CAD)
Base Case	2.0	-0.2	4.8	26.0	26.2	41.9	2.4
A	2.1	0.2	6.4	24.9	22.8	41.3	2.4
B	2.1	0.5	7.6	26.8	24.7	40.8	2.5
C	1.9	-0.3	5.9	24.3	24.5	41.0	2.2
D	2.0	0.0	8.6	27.5	27.4	41.1	2.3

The results indicate that there are no significant changes in the general shape of the pressure curves for all conducted experiments. When post injection schemes were employed, the peak in-cylinder pressure was generally decreased, but peak measured values lie within 2% of the base case value. The corresponding AHTR curves in Figure 3.24 evidently reveal a second peak in the respective profiles, indicating the combustion of the post-injected fuel. Further analysis shows that when the late injection of fuel was close to the main combustion phase, *i.e.* Cases A and B, the cumulative apparent heat transfer was higher, while the opposite was true for Cases C and D. The additional heat associated with the post-injection event (locally between 10 and 25 CAD aTDC) slightly modified the charge's pressure and temperature. Although there were no temperature measurements available, estimations based on the equation of state suggests that Cases A, B and D result in *ca.* 40 K higher temperatures at 30 CAD aTDC, while Case C was close to the Base Case. Note that this temperature difference is higher (approximately 60-70 K) at 80 CAD aTDC, where actual in-cylinder T is estimated to be around 900 K at the given piston position. The combustion phasing, as indicated by CA50, *i.e.* the crank angle location where the 50% of the total heat release rate has occurred, was increased in comparison to the Base Case for all the considered cases. The increase of the post injection fuel amount (and hence the consequent decrease in the amount of the main fuel pulse, so as to maintain a constant carbon content *per cycle*) led generally (the exemption being the latest post injection Case D) to reduced combustion duration, as indicated by the values of the difference CA90-CA10 (see Table 3.6). The resulting IMEP is mainly affected by post injections close to the main combustion event, *i.e.* Cases A, B (apparently more so in Case B, where more fuel is injected) while the IMEP obtained from schemes with post injections at 10 or 15 CAD aTDC, *i.e.* Cases C, D, remained relatively unchanged in comparison to the Base Case. On the other hand, as the post injection was retarded relative to the main pulse, the maximum pressure rise rate decreased; a fact that will, arguably, reduce combustion noise.

Figure 3.3.3.3 presents the integrated flame natural luminosity signal for all cases which shows that combustion activity can be either prolonged or shortened when post injection is employed (cf. CA90 and combustion duration values in Table 3.6). For all cases considered, the natural luminosity signal peaks shortly after the moment of the post injection. This indicates that the combustion of fuel from the post-injection enters directly into the diffusion phase, as there is no time available for premixing in a high temperature and pressure environment. Both the dwell time between the main and the post injection, as well as the amount of the post-injection fuel

influence the recorded natural luminosity signals. When a short dwell is employed (Cases A and B), peak luminosity location is approximately 5 CAD retarded as compared to the location of the peak in the Base Case, essentially indicating that the combustion imposed by the post-injection is expressed as an extension of the main combustion event. In the cases of a longer dwell (Cases C and D), two distinct peaks can be observed, suggesting a second, discrete combustion event due to the post injection.

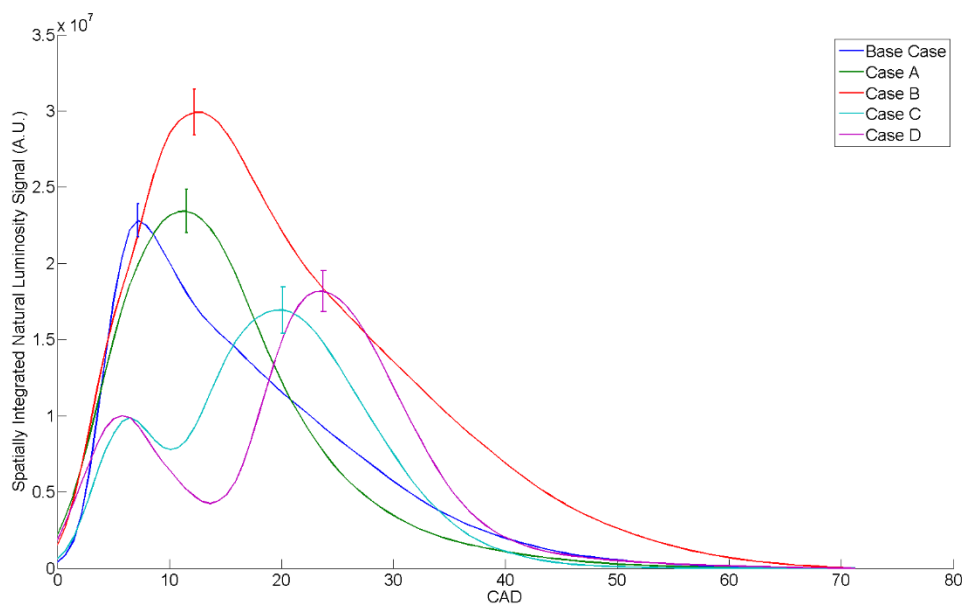


Figure 3.26 Spatially Integrated Natural Luminosity signal as a function of crank angle for all cases considered.

Closer inspection of the average images at the location where the peak Natural Luminosity signal occurs in Figure 3.27 suggests that, for the Base Case, the flames of the main injection are closer to the engine walls, however, due to the limited optical access of the equipment (see Section 2), there is incomplete information about the main combustion event. In Cases A and B, the associated to the post injection event flames, produce luminosity in a larger portion of the optically accessible area, resulting thus to a higher signal. The above can be also corroborated by the fact that in the Base Case, the main injection involves higher fuel mass, acquiring thus higher kinetic energy and resulting in a longer penetration length, leading eventually to wall impingement.

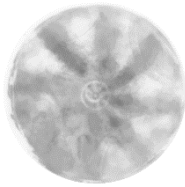
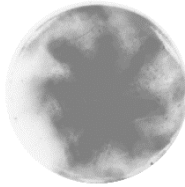
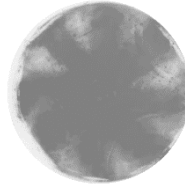
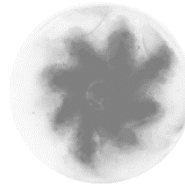
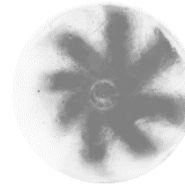
Base Case 10.0 CAD aTDC	Case A 13.7 CAD aTDC	Case B 14.4 CAD aTDC	Case C 22.3 CAD aTDC	Case D 25.9 CAD aTDC
				

Figure 3.27 Average maximum spatially resolved Natural Luminosity signal for all cases considered.

When the amount of the post injection fuel is doubled (Cases A $\rightarrow$ B), peak natural luminosity increases and the corresponding signal remains at higher levels during the expansion stroke which indicates more intense soot oxidation throughout the combustion event following the post-injection. As a consequence, the luminosity signal in Case B is detected later in the cycle, signifying that detectable soot oxidation takes place for longer. Note that CA90 of Case B is also retarded as compared to CA90 of Case A (Table 3.6). On the other hand, the longer dwell Cases C and D feature a lower than the Base Case first local peak (at around 8-10 CAD aTDC) and reach the maximum natural luminosity signal when the combustion of the post injected fuel takes place. Finally, although both Cases C and D feature the same post injection amount, the higher, late (20-30 CAD aTDC) local peak in Case D comes in agreement with the values of the associated cumulative energy release reported in Table 3.7 (see also discussion later).

Table 3.7 Total apparent heat transfer from the end of the main injection until the end of the overall combustion process for all cases investigated (see also text).

Case	Cumulative energy after main injection (after 3 CAD aTDC) (J)
Base Case	162.1
A	188.3
B	194.8
C	165.5
D	182.5

Figure 3.28 and Figure 3.29 present the spatially integrated CH^* and C_2^* chemiluminescence signals, respectively. Processes which are detected as apparent heat transfer are strongly linked to the overall mixture charge reactivity and, therefore, to the intensity of CH^* signal, as the latter is a typical flame front marker. Any post-injection strategy prolongs charge reactivity (higher CH^* signal after 10 CAD aTDC), because a certain amount of fuel is injected later in the cycle, as seen in Figure 3.28. Note the relative higher noise-to-signal ratio in the long dwell Cases C and D, mainly attributed to the overall lower intensity recorded (all images have been acquired with the same resolution of signal intensity). Charge reactivity is not noticeably affected when the post injection amount increases (cf. Cases A and B respectively) at least on the evidence that the peak CH^* signal is not significantly affected, in agreement with Figure 3.24, where the local AHTR values are almost the same.

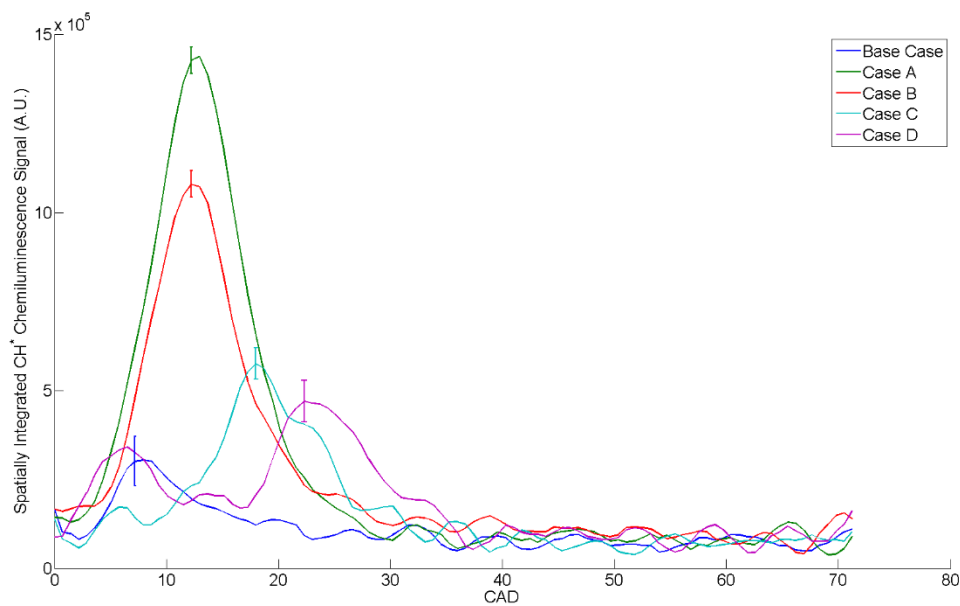


Figure 3.28 Spatially Integrated CH^* chemiluminescence signal as a function of crank angle for all cases considered.

The CH^* chemiluminescence signal can be used to discuss the delay period between fuel injection and combustion. For Cases A and B, the CH^* peak occurs slightly later than the peak of the Base Case (approx. 15 vs. 12 CAD aTDC respectively) indicating the extended mixing and reaction period induced by the post injection (cf. also C_2^* signal peaks in Figure 3.3.3.6). The effect of post injection fuel mass (2 mg in Case B, 1 mg in Case A) does not seem to affect the location of the peak signal and only results in a slightly lower peak value. The long dwell Cases C and D feature significantly lower overall peak integrated CH^* signals. It is interesting to note that the peak of CH^* signals in the short dwell post-injection cases occurs *ca.* 10 CAD after the post injection timing, whereas in the long dwell cases after *ca.* 8 CAD of the post injection, indicating the effect of the in-cylinder thermodynamic conditions at the moment of post-injection. CH^* signals peak at the same location for Case C and for Case D as well, and generally follow the local peak in the apparent heat transfer rates (AHTR), in Figure 3.24. The long dwell Cases C and D featuring late post-injections at 10 and 15 CAD aTDC respectively, result in a secondary heat release/oxidation zone, in agreement with the observations made earlier with respect to CH^* chemiluminescent species.

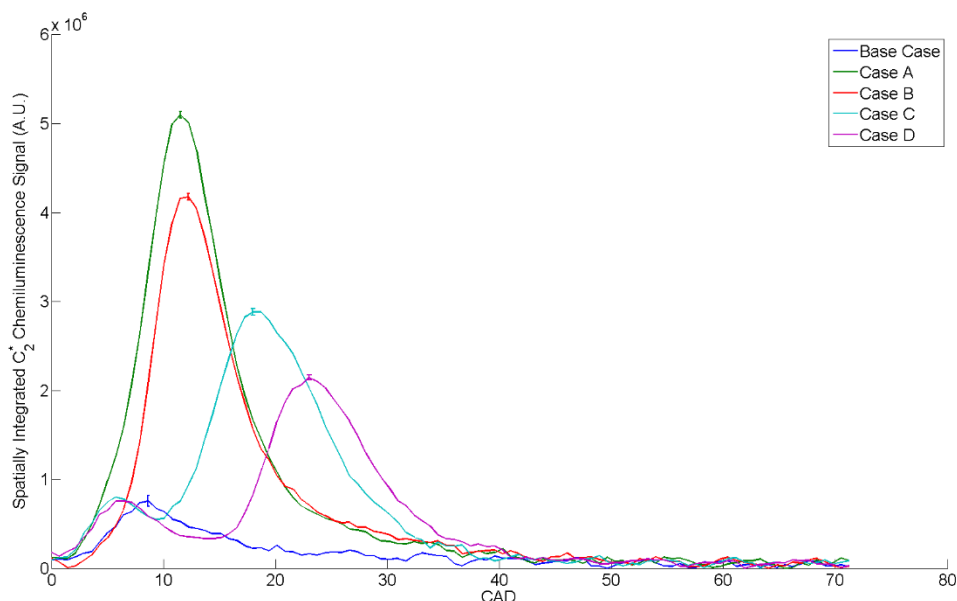


Figure 3.29 Spatially Integrated C_2^* chemiluminescence signal as a function of crank angle for all cases considered.

Measurements of C_2^* chemiluminescence signal in Figure 3.29 provide an insight on the soot formation in the fuel rich areas and thus, the effect of the late injection on exhaust stream and post-injection mixing. Following the discussion made in the natural luminosity section, the diffusion mode of combustion for the post-injected fuel leads to formation of intense fuel rich areas and high levels of soot formation. The onset of the C_2^* signals for Cases A-D is recorded earlier than the respective signal of the Base Case. For Cases A and B (post-injection close to TDC), there is a virtual overlapping of reacting premixed fuel rich areas between the post injection and the main combustion event until 20 CAD aTDC. The peak value of Case B is lower than that of Case A, indicating that a larger amount of post-injection fuel leads to lower soot formation. In Cases C and D, the recorded C_2^* signal peaks between 20 and 30 CAD aTDC in the associated with the post injection combustion event, leading to higher relative soot formation levels, late at the combustion cycle. Note that the relative C_2^* peak location in Cases C and D agrees with the respective post injection timing. Despite the same fuel mass is injected during the post-injection, Case C features a higher peak in the C_2^* signal as compared to Case D, indicating that a post-injection under a hotter environment (5 CAD advanced post-injection) enhances the local formation of reactive fuel rich pockets.

Overall, the combined analysis of optical measurements and pressure traces can be beneficial for delineating the effect of the post injection strategy on in-cylinder combustion evolution. Indications based on pressure trace analysis can be further explained and discussed on the evidence of optical information inside the cylinder. The short post-injection dwell cases (Cases A and B) feature overlapping natural luminosity and chemiluminescence signals between the combustion associated with the main injection and the post-injection, contrary to the long dwell cases (Cases C and D), where the fuel from the post-injection appears to burn in a separate combustion event, late at the combustion cycle. This overlapping observed for the short dwell cases appears to enhance the reactivity of the charge and the intensity of the heat release zones. Additionally, although the fuel rich areas seem to be more intense, indicating higher levels of soot formation associated with the post-injection, they appear earlier in the combustion cycle, allowing more time for the generated soot to oxidize. This behaviour is promising when it comes to engine out soot emissions, however, additional measurements of soot distribution inside the cylinder are necessary to corroborate this argument.

Figure 3.30 shows the spatially integrated recorded pLII signal intensity at 31 and 50 CAD aTDC for all cases considered. It is interesting to note that for the Base Case the total captured signal at 50 CAD aTDC is greater than the one at 31 CAD aTDC. This may either be attributed to the effects of the limited optical access (the laser sheet is visible in the pictures of Figure 3.3.3.8) and/or to the redistribution (*e.g.* recirculation) of the charge, and hence of the soot also, towards the centre of the bore, due to the reverse-squish flow induced by the piston movement. It may also be partly attributed to the fact that the soot oxidation ‘freezes’ as the in-cylinder volume expands. The opposite behaviour is captured for the cases featuring post injections. The greater reduction is recorded for Cases C and D although, for these cases, the pLII signal and, presumably, the analogous soot concentration, is substantially higher in comparison to the Base Case and Cases A and B as well. The pLII signal for Case B is the lowest of all cases and, in fact, at 50 CAD aTDC is even less than the Base Case, exhibiting the overall lowest recorded pLII signal intensity. This is to be contrasted with Case A, for which the post injection is realized at the same timing as Case B but with less injected fuel mass. This additional fuel, during the post injection, presumably creates conditions favourable for soot oxidation *via* an increase in the bulk gas temperature and enhanced mixing (see earlier discussion on the chemiluminescence signals). In the context of engine operation, this may prove particularly

beneficial as the cylinder charge has equal carbon content for all cases considered, while Case B exhibits the higher IMEP value.

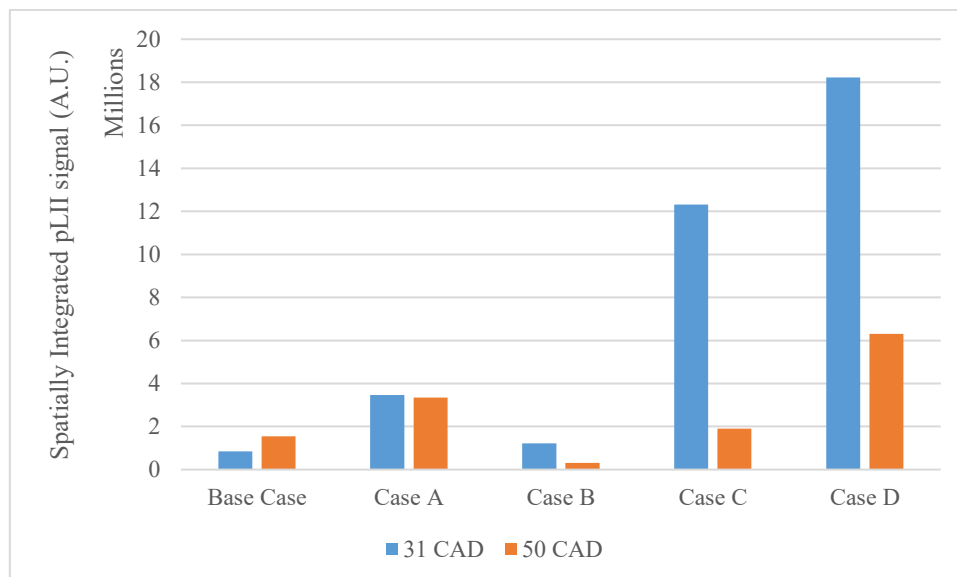
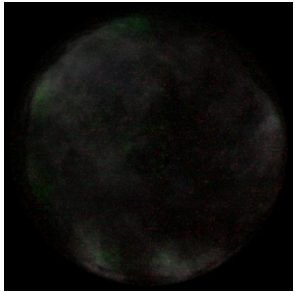
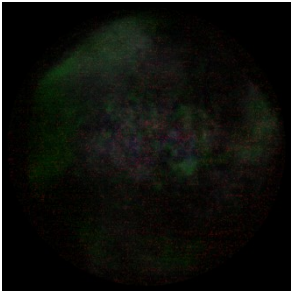
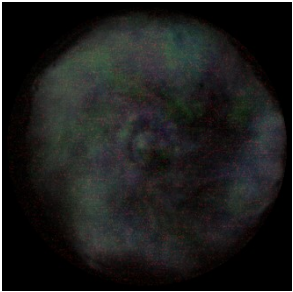
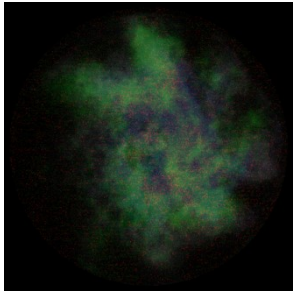
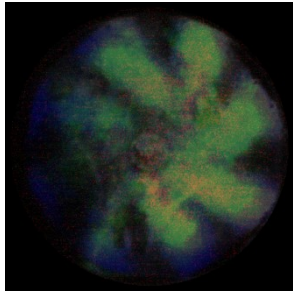
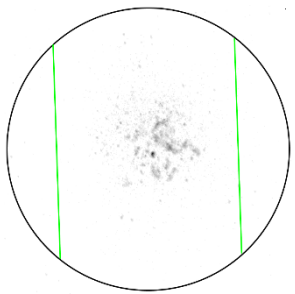
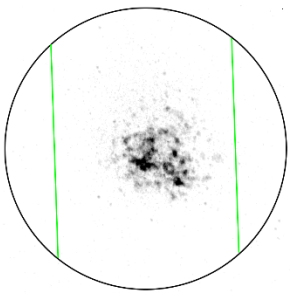
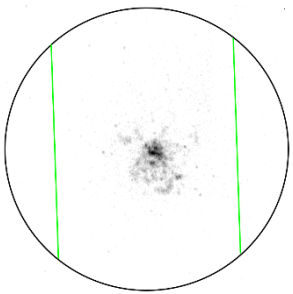
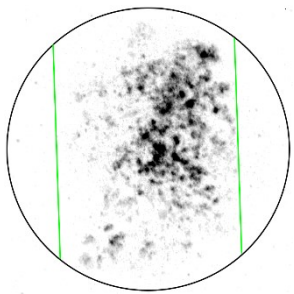
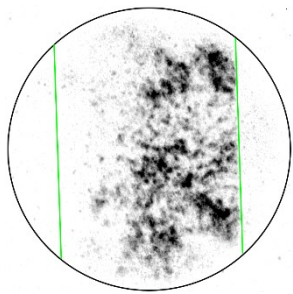


Figure 3.30 Total LII signal for 31 and 50 CAD aTDC for all cases considered.

The effect of the post injection schemes on in-cylinder soot emissions is further discussed *via* assessing the spatial distribution in flame luminosity in terms of broadband visible, CH^* and C_2^* chemiluminescence, against planar pLII measurements at 31 and 50 CAD aTDC. Figure 3.31 presents superimposed and ensemble averaged images of flame spectroscopic measurements at 31 CAD aTDC (upper two rows) and 50 CAD aTDC (lower two rows), and the corresponding pLII imaging measurements for the same piston position. The images exhibit the following colour coding; visible flame luminosity is depicted in White, and the chemiluminescent intensity signal of CH^* and C_2^* are shown as red and green respectively. According to the current understanding of the soot oxidation mechanisms (*e.g.* [11, 12]), soot precursors as well as soot particle oxidation by OH radicals can be of equal importance to the direct oxidation *via* O_2 . Indeed, in the present dataset, natural luminosity signal is substantially intensified, indicating that the post-injections contribute to the creation of conditions conducive to soot oxidation. This is in agreement with the expectation that the fuel in the post-injection serves as a ‘trigger’ for the enhancement of local mixing towards the oxidation of any remaining carbonaceous products from the main combustion. In particular, Case B shows at 31 CAD aTDC a more homogenized distribution of C_2^* species as well as excited CH^* radicals all

over the chamber, indicating enhanced premixing and, as a result, improved soot oxidation, as compared to Case A. This is not the case for Cases C and D, where post injection flame “fingers” are still observable in both chemiluminescence and pLII measurements. Moreover, the topography of the chemiluminescence measurements and, in particular, the poor overlapping of CH^* and C_2^* radicals implies that a significant portion of the detected soot will not be further oxidized for Cases C and D.

	Base Case	Case A	Case B	Case C	Case D
31 CAD	Chemiluminescence				
					
	pLII				
					

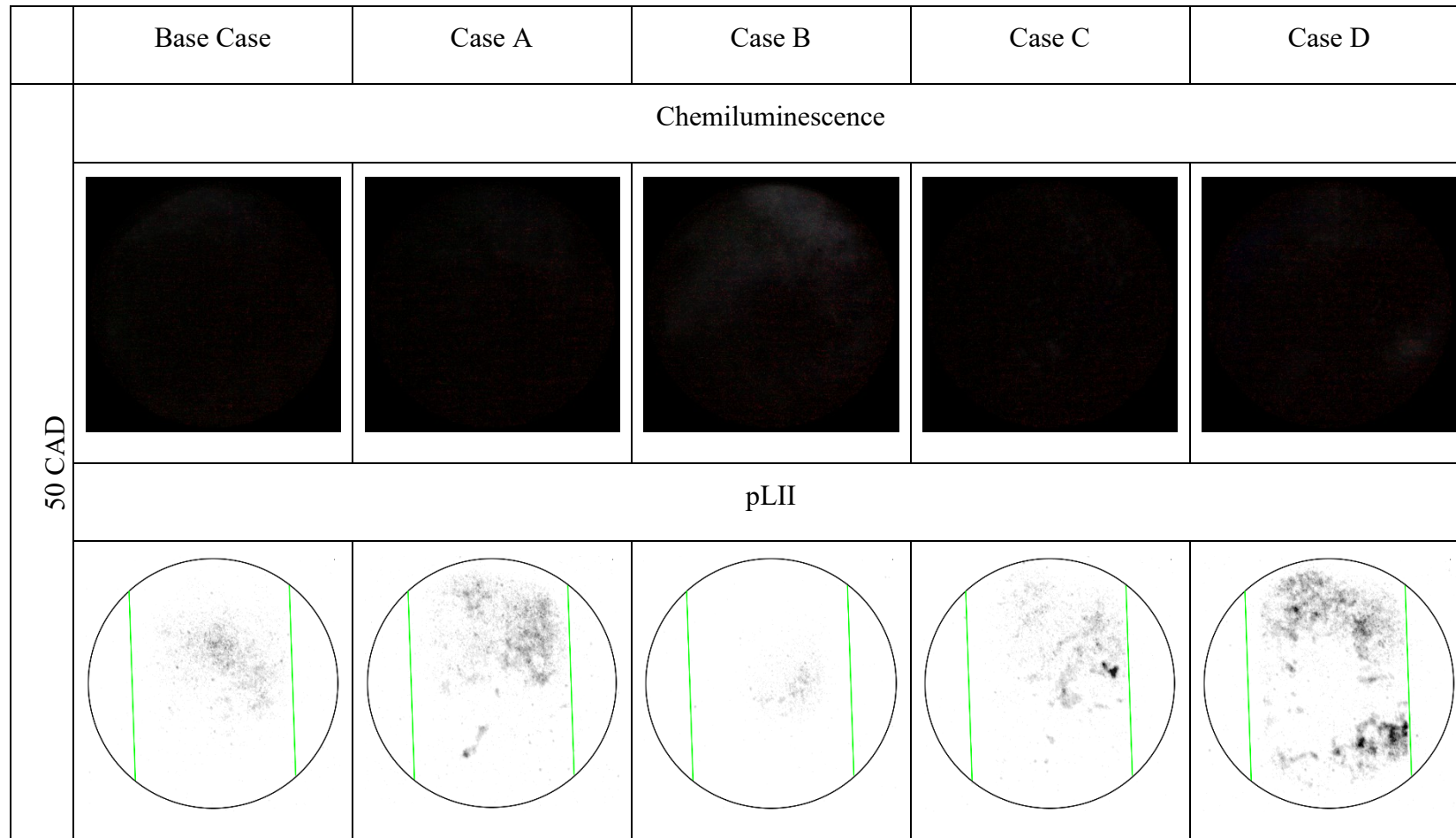


Figure 3.31 Superimposed average images of flame spectroscopic measurements with natural luminosity (white), CH^* chemiluminescence (red) and C_2^* chemiluminescence (green) and the corresponding planar LII imaging measurements at 31 (upper two rows) and 50 CAD aTDC (lower two rows). The black circle represent the optical window contour, whereas the green lines represent the laser sheet contour.

From the preceded analysis we can infer that both the post-injection timing and fuel amount affect the characteristics of the combustion, the flame patterns and the soot distribution inside the cylinder. It appears that an injection scheme featuring a close-coupled post injection with a fuel amount that constitutes around 13% of the total fuel amount injected *per cycle* (Case B) serves as the most beneficial strategy for engine-out soot reduction. In order to understand the reason why this strategy is the most beneficial, let us take a closer look on the interaction between the post-injection fuel jets and the main injection combustion products.

Figure 3.32 and Figure 3.33 show the effect of the post-injection timing on the aforementioned interaction for Cases B and C respectively. The red oval-shaped line indicates the area where the hot gases from the main injection combustion are located, the blue one indicates the area where unburnt oxygen is more likely to be located and the green one indicates the fuel injected during the post-injection. The post-injection fuel in Case B is likely to find more unburnt oxygen than Case C and thus release more heat to maintain the in-cylinder temperature above the soot oxidation temperature threshold. The enhanced heat release is also associated with the higher temperature of the charge when the early post-injection is employed. This process enhances soot oxidation, as seen in the natural luminosity measurements (see Figure 3.26 and Figure 3.27).

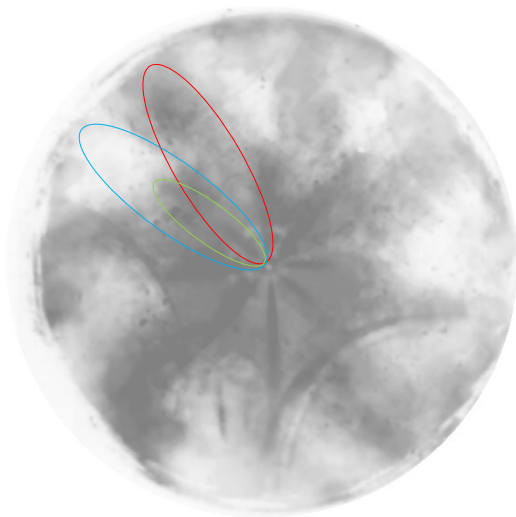


Figure 3.32 Illustration of the effect of post-injection timing on soot oxidation in Case B.

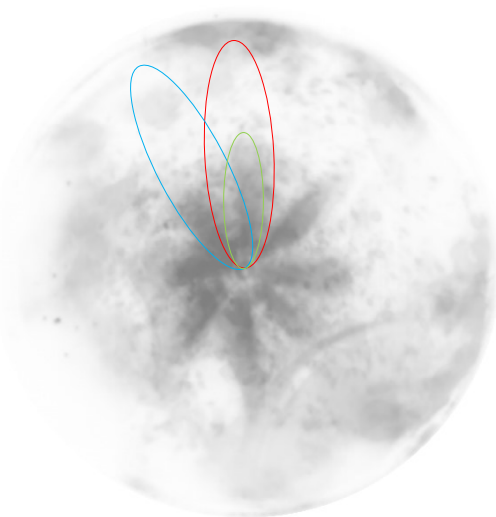


Figure 3.33 Illustration of the effect of post-injection timing on soot oxidation in Case C.

Figure 3.34 and Figure 3.35 show the effect of the post-injection fuel quantity on the aforementioned interaction for Cases A and B respectively. The blue oval-shaped line indicates the area where unburnt oxygen is more likely to be located, while green one indicates the area covered by the flame associated with the post-injection. In Case B, which features a larger amount of fuel during the post-injection than Case A, the flame associated with the post-injection covers a larger amount of the area where unburnt oxygen is likely to be available, thus taking advantage of a larger portion of the available oxygen. This can be attributed to the longer penetration length of the fuel, resulting from the higher kinetic energy of the larger post-injection fuel amount. The larger area of the flame is translated to higher levels of heat release from the post-injection and subsequent increase in the in-cylinder temperature early in the combustion cycle, facilitating soot oxidation for a longer period of time. Additionally, the higher kinetic energy of the post-injection in Case B is likely to induce more turbulence in the area where the oxygen is available inside the cylinder, thus facilitating the oxidation of the already formed soot.

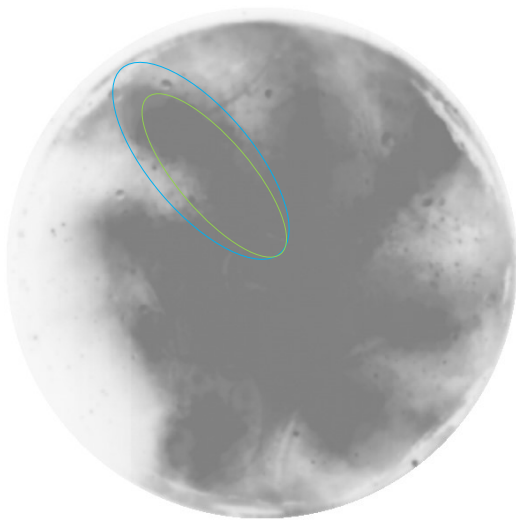


Figure 3.34 Illustration of the effect of post-injection fuel amount on soot oxidation in Case A

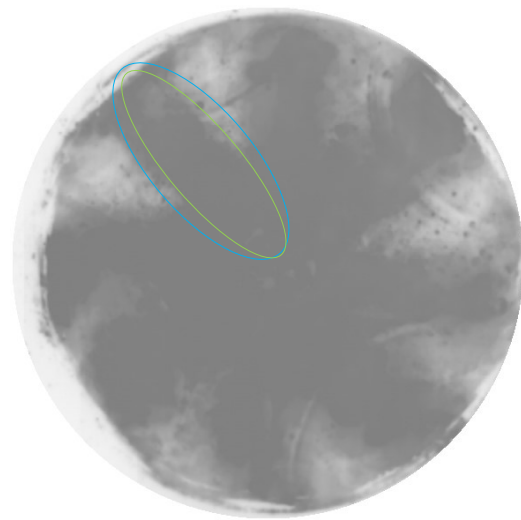


Figure 3.35 Illustration of the effect of post-injection fuel amount on soot oxidation in Case B

3.4.4 Conclusions

The combination of pre- and post-injection events was experimentally investigated in an optical single-cylinder low duty diesel engine operated in a PPC mode under low load, low speed (*ca.* 2.0 bar IMEP, 1200 RPM) conditions. The work focused on the influence of the amount of post injected fuel and timing on flame patterns and the respective implications on exhaust soot emissions. Results are deduced *via* a combination of in-cylinder pressure trace analysis, spectroscopic chemiluminescence measurements and LII signals at selected CAD. In particular, the diverse combustion patterns are visualized by means of high speed imaging of flame natural luminosity and CH^* and C_2^* chemiluminescence, while planar LII experiments at 31 and 50 CAD aTDC indicate the soot concentration at the given piston position. As a result, a magnitude of evidence regarding soot oxidation rates, overall reactivity, premixed fuel rich areas and oxidation zones is captured.

In brief, results indicate:

- The heat release associated with the combustion of the post-injection appears as a second peak in the AHTR curve. This peak appears shortly after the realization of the post-injection, which indicates that the combustion of fuel from the post-injection enters directly into the diffusion phase, as there is no time available for premixing in a high temperature and pressure environment.
- Flame intensity is strongly affected when post injection is employed close to the main injection event, whereas a retarded post-injection event leads to a virtually second flame front formation. This implies that a soot oxidation is more intense when a close-coupled post injection is employed.
- The reaction zones, which are linked to the heat release associated with the post-injection are enhanced and prolonged when a post injection is employed. A close-coupled post-injection exhibits more intense reaction zones and this is related to the higher bulk temperature of the charge when an early post-injection is realized.
- The fuel rich areas, which is where soot is being formed within the cylinder, appear to be more intense, due to the diffusion phase of the post-injection combustion. The most intense and widely distributed fuel rich areas are formed when a close-coupled post-injection is employed. This implies that soot generation during the post-injection is higher in this case.

- According to the pLII measurements, the post-injection strategy which is the most effective in reducing engine-out soot emissions is the one that features a close coupled post-injection with a fuel amount that accounts for the 13% of the total fuel amount *per* cycle (Case B). It appears that although soot formation is high for this strategy, soot oxidation is also enhanced and the competition between them leads to a reduction of soot levels inside the cylinder.
- The reason behind the effectiveness of this strategy lies in its ability to make use of the remaining oxygen inside the cylinder in order to promote soot oxidation for a longer period in the combustion cycle. Both the timing and the quantity of the post-injection facilitate the exploitation of the available oxygen, the former by placing the fuel jets between the product gases of the main combustion event and the latter by increasing the penetration length of the fuel jets.

Overall, results suggest that a careful employment of the appropriate fuel amount close to the main combustion phasing can reduce engine-out soot emission for low load, low speed engine operation. This could be accompanied by slightly increased IMEP levels at the same specific fuel consumption (same carbon content is injected *per* cycle) at the expense of engine noise, as the maximum pressure rise rate is slightly increased.

3.5 Chapter summary

The experimental work conducted on the Ricardo Hydra optical engine was presented. The work consisted of three separate experimental campaigns, focusing on combustion characterization and in-cylinder soot evolution during the transition from conventional CI operation to PPC operation, when intake air dilution, close-coupled pilot-main injection schemes and post injection schemes were employed. The engine operated at the low speed, low load regime while the analysis was performed *via* means of in-cylinder pressure trace, high speed natural flame luminosity and species chemiluminescence, as well as soot Laser Induced Incandescence.

4 Experimental work on the Volvo D5 Optical Engine

4.1 Introduction

The present chapter covers the experimental work on the Volvo D5 Optical Engine. The Volvo D5 optical engine featured a lower compression ratio than the Ricardo Hydra optical engine and thus, the ignition delay of the in-cylinder charge could be longer, especially at low engine speeds, allowing for more extensive pre-mixing of the injected fuel with the surrounding air. The enhanced premixing promotes PPC operation. The experimental work presented in this chapter focuses on the effect of injection timing on combustion stratification and soot formation on the Volvo D5 optical engine at operating conditions that promote PPC operation. More specifically, the engine operated at the low speed, low load regime, under three single injection schemes. Findings were analysed *via* means of pressure trace and apparent rate of heat transfer analyses, as well as a series of optical diagnostics techniques, namely high speed flame natural luminosity imaging, CH^* and C_2^* line-of-sight chemiluminescence, as well as planar Laser Induced Incandescence (pLII). The pLII measurements were conducted in three different combinations of laser sheet position and piston position.

4.2 Fuel Stratification Campaign

4.2.1 Introduction

Since the development of the first compression ignition internal combustion engine by Rudolf Diesel in 1892, significant effort has been put into research and development in an effort to achieve higher thermal efficiency and, consequently, better fuel economy. In the later years, environmental concerns have been raised regarding the air quality globally, as well as locally. The greenhouse effect, which is caused by greenhouse gases, such as carbon dioxide (CO_2) and methane (CH_4), affects the climate globally and is responsible for the abnormal raise in the global temperature. Locally, air quality is affected by toxic gases such as carbon monoxide (CO) and nitrous oxides (NO_x), unburnt hydrocarbons (UHC), as well as particulate matter (PM), which is characterized by their small diameter (varying from 10 nm to 10 μm). All the aforementioned pollutants can be found to a greater or lesser extent in the exhaust products of a compression ignition engine. The increasing number of vehicles powered by compression ignition engines and the deterioration of the air quality, especially in the vicinity of large cities, has led to strict local emissions regulations. In order to meet these regulations and, at the same time, satisfy customer needs, engine developers had to come up with novel and technologically advanced combustion strategies.

Many of these strategies are based on the concept of Low Temperature Combustion (LTC), which favor the reduction of both NO_x and PM emissions, due to the lower local combustion temperature. One of the first LTC concept to be developed was Homogeneous Charge Compression Ignition (HCCI). During HCCI operation, the combustible mixture is formed outside the combustion chamber and then introduced *via* the inlet valves into the chamber, where it is auto ignited during the compression stroke of the engine. The air-fuel mixture inside the cylinder is homogeneous and lean in terms of equivalence ratio (ϕ) and thus burns at lower temperatures in an environment with excess of oxygen. Therefore, NO_x and PM emissions are reduced due to the lean nature of the combustion and thermal efficiency can go up by as much as 50% compared to conventional CI operation. However, HCCI is not panacea, as there are issues with controlling the timing of the combustion, as well as with the peak in-cylinder pressure and the increased CO and UHC emissions. The auto-ignition of the mixture during HCCI operation originates from several spots inside the cylinder, due to the homogeneity of

the mixture, resulting in steep in-cylinder pressure rise rates and rapid heat release, which, not only contribute to excessive engine wear and lubrication problems, but also create difficulties in controlling the phasing of the combustion. Since the conception of the HCCI operation, many researchers have dealt with the issue of controllability both at the low load and high load operation regimes. The experimental results have led to the conclusion that, in order to have some control over the combustion phasing, one must control the heat release rate inside the cylinder. To achieve that, the mixture inside the cylinder should exhibit some level of “controlled” inhomogeneity in fuel distribution inside the cylinder, rather than having an entirely homogeneous charge (Hwang *et al.* , 2009; Sjöberg and Dec, 2006). This “controlled” inhomogeneity is known as fuel stratification and it is the main feature of another LTC strategy, the Partially Premixed Compression Ignition (PPCI) or Partially Premixed Combustion (PPC). During PPC operation, the fuel is injected into the combustion chamber using a high-pressure fuel injector, just like in conventional CI operation, with the exception that the fuel is injected early enough in terms of CAD, before the piston reaches its top dead centre, but still relatively close to the top dead centre. Therefore, the fuel has more time to mix with the air inside the cylinder compared to conventional CI operation, but not enough time to create a homogeneous charge, like in HCCI operation and thus a partially premixed charge is created. Additionally, as the mixture is injected relatively close to top dead centre, combustion controllability is maintained. One can then identify the PPC operation as the link between conventional CI operation and HCCI operation. The desired level of in-cylinder stratification at the start of combustion (SOC) can be controlled either by the timing of the fuel injection, or by employing multiple injection strategies (Noehre *et al.* , 2006).

One of the first attempts to characterize PPC operation in a heavy-duty compression ignition engine at medium and high loads was this of Noehre *et al.* in 2006, which showed that a combination of low compression ratio, high EGR rate and an injection scheme which facilitates PPC operation close to stoichiometric conditions enables simultaneous NO_x and soot reduction. It was identified that this simultaneous reduction can be traced back to long ignition delay due to early injection, which allows good premixing and low combustion temperatures due to high levels of EGR (Noehre *et al.* , 2006). Chaichan in 2014 observed the same trends in a light-duty direct injection CI engine, operating under PPC. The author noticed that, while NO_x and PM emissions go down, UHC and CO emissions increase when the fuel injection timing is advanced and the EGR rate increases, along with the Brake Specific Fuel Consumption (BSFC)

(Chaichan, 2014). Shen *et al.* in 2015 investigated the transition from HCCI to PPC by altering the fuel injection timing. They pointed out that the transition is possible when the injection timing is retarded and the intake temperature is adjusted accordingly to maintain the combustion phasing. Increased levels of EGR favour NO_x emissions reduction, at the price of increased CO and UHC emissions. When a long dwell double injection strategy was used, the second injection can result in local fuel-rich zones, resulting in high amounts of soot at the exhaust. This situation could be avoided by increasing the boost pressure, thus providing more oxygen to enhance soot oxidation (Shen *et al.* , 2015). Along the same line, Qiu *et al.* in 2016 used a 4-cylinder direct injection CI engine operated at the low speed and both low and high load regimes to investigate the effect of various injection strategies on PPC operation. They pointed out that low load operation is dominated by premixed combustion, while at higher loads, the diffusion phase of the combustion is significantly longer. For both low and high load operations NO_x emissions were considerably improved, when compared to conventional CI operation. When a pilot injection was introduced, NO_x, CO and soot emissions increase, while UHC emissions remain practically unchanged. However, the BSFC is reduced for low load operation with the pilot injection. It should be noted here that the pilot injections employed in this work were not close-coupled to the main injection.

Along with the aforementioned experimental work, numerical studies have also been conducted in an effort to simulate PPC operation and predict performance and emissions for a wide range of operating points. For instance, Mubarak Ali *et al.* in 2017 used the experimental data from a single cylinder AVL optical CI engine to tune their software and investigate the effect of injection timing in the development of fuel spray, mixing and combustion. They reported the same emission trends with the experiments and they highlighted that, for a single injection strategy, a start of injection between -20 and -15 CAD aTDC resulted in the largest amount of optimal mixture pockets at $0.9 < \phi < 1.2$ (Mubarak Ali *et al.* , 2017). Maghbouli *et al.* in 2017 followed the same approach, this time with a Volvo D5 based single cylinder CI optical engine and with PRF70 as fuel. They simulated two cases of low and high air-fuel mixture stratification to find that the first stage ignition of the charge occurred earlier for the low stratification case, while high stratification levels would cause a faster transition from the first stage to the second stage ignition. It was concluded that by generating more stratified mixtures and providing the needed activation energy by engine thermal load, would possibly

allow to have control on timing of the second stage ignition and combustion phasing speed of a PPCI engine (Maghbouli *et al.* , 2017).

Fuel and combustion stratification, as inseparable characteristics of the PPC operation, as well as emissions formation have been extensively studied in optical engines, using optical diagnostics. Kim *et al.* in 2008 employed two-dimensional planar imaging and one-dimensional spectrally resolved line-imaging of Laser Induced Fluorescence from CO and UHC to identify the source of these pollutants, in a light-duty CI engine operating in a PPC mode. The collected data suggested that CO emissions mainly originate from the squish region, while, in most cases, UHC emissions stemmed from both the squish volume and the piston crevices. The authors noticed that increased charge dilution tended to increase CO and UHC emissions. Tanov *et al.* in 2014 focused their study on the effect of early, multiple injection strategies and Negative Valve Overlapping (NVO) on the degree of stratification inside the cylinder of a light-duty optical engine operating at the low speed, low load (almost idle) regime, with PRF55 as a fuel. They employed OH^* chemiluminescence to characterize the stratification of the combustion, while the NVO operation ensured that the charge was diluted from the residual gases inside the cylinder. Initial results showed that the combustion with triple injection was more homogenous compared to single and double injection. More detailed analysis showed that in the triple injection case, the first injection created homogeneous charge, the second stratified the charge and created zones with high reactivity and the third one burned these zones and extended the reaction to leaner regions. Most importantly, they claimed that the highest stratification level appeared in the single injection case. The degree of stratification was decreasing when the injection time was advanced. This led to a long ignition delay and hence to better premixing (Tanov *et al.* , 2014). Along the same line, Najafabadi *et al.* in 2016 employed OH^* chemiluminescence and spray shadowgraphy to investigate the ignition sensitivity and the combustion stratification of single injection strategies with SOI that varied from -100 to -12.5 CAD aTDC. The analysis revealed that combustion stratification is low and almost independent of SOI for early injections, while there was a remarkable reverse correlation between combustion timing and stratification level for the late injections (-45 to -12.5 CAD aTDC). The authors also pointed out that injection pressure could adversely affect combustion stratification. However, the effect of injection pressure on the combustion stratification was negligible compared to the effect of injection timing (Najafabadi *et al.* , 2016).

Another useful technique for the characterization of the stratification inside the cylinder is Particle Image Velocimetry (PIV). The velocity field inside the cylinder can provide information on the level of turbulence during the fuel injection, which will determine, to a large extent, the mixing efficiency and the resulting fuel stratification, as well as the turbulence during the combustion, which is linked to the combustion stratification. Tanov *et al.* in 2015 employed the PIV technique to study the effects of multiple injections on the generation of in-cylinder turbulence and the resulting mixing efficiency in a Volvo D5 optical engine running on PRF70 fuel at the PPC region. The results showed that the high amount of spray momentum during the single injection increased the turbulence rapidly, but when the injection took place early in terms of CAD, the induced turbulence did not influence the flow field within the bowl. It is highlighted that for this engine design and low-temperature operating condition, the measurements were indicative of low late-cycle turbulence performance (Tanov *et al.* , 2015). Using the same engine setup, Najafabadi *et al.* in 2017 studied the effect two different 5-hole and 7-hole injectors have on the levels of turbulence inside the cylinder and the fuels stratification which stems from the induced mixing. They discovered that the vortex inside the piston bowl, driven by the spray, persisted during the ignition delay period, thus enhancing the in-cylinder mixing and that this vortex was stronger the closer the injection event was relative to TDC. They concluded that the 7-hole injector led to sharper heat release rates, better mixing and higher combustion efficiency compared to the 5-hole injector.

Finally, it is important to note that, although soot forming mechanisms during PPC engine operation have been studied numerically (Zhao *et al.* , 2013, 2014), there is not sufficient experimental work on soot formation characterization during PPC operation. The following experimental work focuses on fuel and combustion stratification and their effect on combustion characteristics and, more importantly, soot distribution inside the cylinder. The engine setup is identical to the one used by Najafabadi *et al.* 2016 and the conditions investigated are similar because they are representative of the PPC operation. Fuel stratification is manipulated by changing the fuel injection timing. The combustion stratification is studied using high speed imaging of chemiluminescent species and the soot distribution is studied using natural flame luminosity imaging as well as qualitative planar Laser Induced Incandescence.

4.2.2 Methodology

Table 4.1 shows the operating conditions used for the experiments. The optical engine operated at a constant speed of 1200 RPM and the engine load was kept constant at approximately 3.2 bar IMEP. The cases were chosen to cover a wide range of operating modes of diesel engines, varying from highly premixed compression ignition to more conventional compression ignition operation. The investigated cases feature a single injection scheme, varying in timing, at a constant injection pressure of 500 bar. The intake air was boosted to 1.2 bar using a supercharger and was heated to 50°C in order to compensate for the extensive heat losses of the optical engine.

Table 4.1 Injection schedule details for all cases considered. Engine speed 1200 RPM and engine load 3.2 bar IMEP in all cases.

Case	Intake Pressure (bar)	Injection Pressure (bar)	Main injection Timing (CAD aTDC)	Main duration//quantity ($\mu\text{s}/\text{mg}$)	Global equivalence ratio (ϕ)
A	1.2	500	-8	740/14.2	0.010
B	1.2	500	-15	770/14.8	0.011
C	1.2	500	-30	860/16.5	0.012

4.2.3 Results and discussion

Figure 4.1 shows the evolution of in-cylinder pressure and the respective Apparent Heat Transfer Rate (AHTR), whereas Table 4.2 shows some thermodynamic characteristics of the combustion for all cases considered. From Table 4.1 it is evident that when the injection timing is shifted away from the TDC, the injection duration and, consequently, the fuel consumption is increased in order to maintain the same engine load. This is attributed to the advanced ignition that takes place when advanced injection timing is employed. As seen in Table 4.2, CA10 values, which indicate the timing of the ignition, vary from 3.4 CAD aTDC for the late injection cases to -12 CAD aTDC for the early injection cases. An early injection leads to an

early pressure rise inside the cylinder, before the piston reaches top dead centre, creating a force on the piston surface that opposes to the piston movement, thus decreasing the total work produced during the combustion. To compensate for this work loss, more fuel is necessary and the combustion results in higher maximum in-cylinder pressure values, as seen both in Figure 4.1 and Table 4.2.

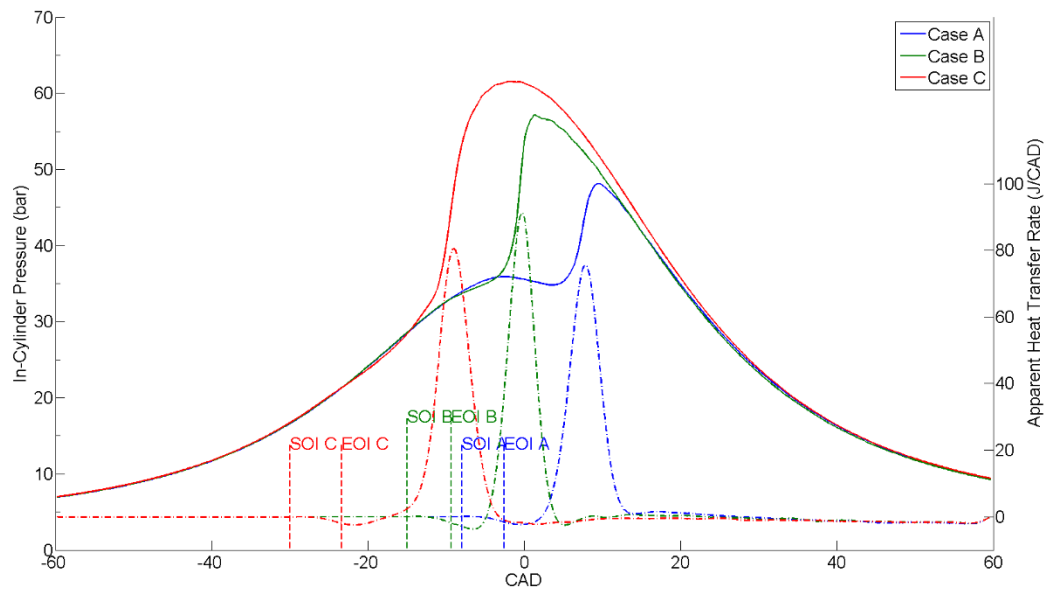


Figure 4.1 Measured in-cylinder pressure (solid lines; left y-axis) and calculated apparent heat transfer rate profiles (AHTR; dashed lines; right y-axis) as a function of crank angle degrees for all cases considered. The colour-coded vertical dashed lines indicate the SOI and EOI for each case.

Table 4.2 Global combustion characteristics for all cases considered in the present work.

Case	IMEP (bar)	CA10 (CAD)	CA50 (CAD)	CA90 (CAD)	Max Pressure (bar)	Max Pressure Location (CAD)	Max Pressure Rise Rate (bar/CAD)
A	3.2	3.4	5.4	8.2	52.3	0.7	10.1
B	3.2	-3.2	-1.4	0.5	59.4	0.0	14.8
C	3.2	-12.0	-9.8	-7.4	61.1	-0.2	8.3

Figure 4.2 shows the Cumulative Apparent Heat Transfer for all cases considered. There are three distinctive parts in each curve, corresponding to three different phases of the combustion. The first part of the curve is the one below the horizontal axis, showing negative values of apparent heat transfer. This part corresponds to the evaporation of the injected fuel, where energy is being absorbed from the surroundings. The cases with early injection timings seem to have larger evaporation duration compared to the cases with late injection timings, which implies that a larger amount of fuel is evaporated and mixed with the surrounding air before ignition. The second part of the curve is the one that features steep gradients of heat transfer and corresponds to the premixed phase of the combustion. This phase is characterized by rapid heat transfer from the fuel to the engine and thus, rapid increase of the in-cylinder pressure and temperature. As seen in Figure 4.2, when an advanced fuel injection is employed (Case C), the premixed phase of the combustion is enhanced. Finally, the last part of the curve features mild gradients of heat transfer and corresponds to the diffusion phase of the combustion. When a late injection scheme is employed, the diffusion phase is present, whereas when the injection event is advanced, the diffusion phase is suppressed and, in some cases, non-existent. From the previous analysis, we can infer that an early injection event provides enhanced evaporation of the fuel and better premixing with the air available inside the cylinder. This, in turn, leads to an enhanced premixed combustion phase and a suppressed diffusion phase. This argument is also supported by Table 4.3, where the ignition delay of the fuel is presented for all cases considered. The case with early fuel injection events (Cases C) exhibits the longest ignition delay, allowing more time for the injected fuel to mix with the air inside the cylinder.

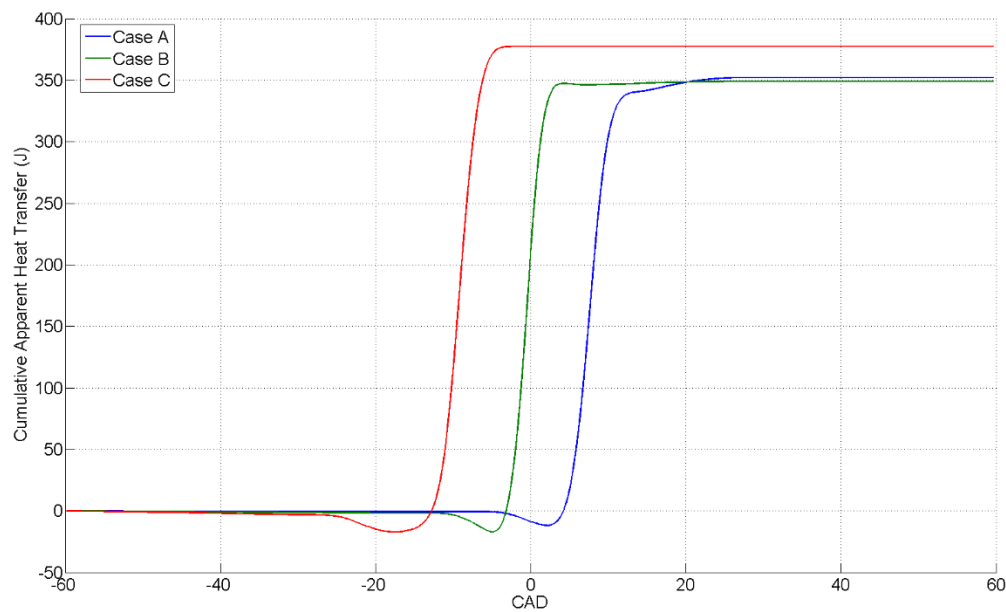


Figure 4.2 Cumulative Apparent Heat Transfer for all cases considered

Table 4.3 Calculated ignition delay for all cases considered.

Case	Start of 1st Injection (CAD)	Start of Combustion (CAD)	Ignition delay (CAD)
A	-8	3.4	11.4
B	-15	-3.2	11.9
C	-30	-12.0	18.0

It is interesting to note here that Case B exhibits the highest PRR values, resulting, presumably, to the highest noise levels, as the ignition takes place slightly before the piston reaches top dead centre. Surprisingly, the cases with the early injection timing, *i.e.* Case C, exhibits low noise levels, even though these cases feature the most premixed combustion amongst all cases, as mentioned above.

Hereafter, the analysis of the optical measurements takes place, investigating the effect of advanced injection timing and the effect of double injection schemes separately. The optical diagnostics used are, namely, high speed imaging of the flame's natural luminosity, CH^* and C_2^* chemiluminescence, as well as pLII for non-luminous soot detection at three different combinations of laser sheet position and piston position.

In order to enhance the credibility of the observed trends in the chemiluminescence measurements, the broadband background signal interference is removed from the chemiluminescence signal using the methodology described in section 2.4.2. For a more detailed presentation of the results of this method on the experimental data of the present chapter, please see Appendix B.

Figure 4.3 shows the evolution of the spatially integrated CH^* chemiluminescence signal with crank position before and after the background interference has been removed, while Figure 4.4 to Figure 4.6 show the maximum cycle-averaged spatially resolved chemiluminescence signal after the background interference has been removed for all the cases considered. First of all, it is interesting to note in Figure 4.3 that the trends observed before the background correction is maintained after the correction as well. When the injection timing advances, the overall reactivity of the mixture weakens, as a result of the enhanced premixing of the charge.

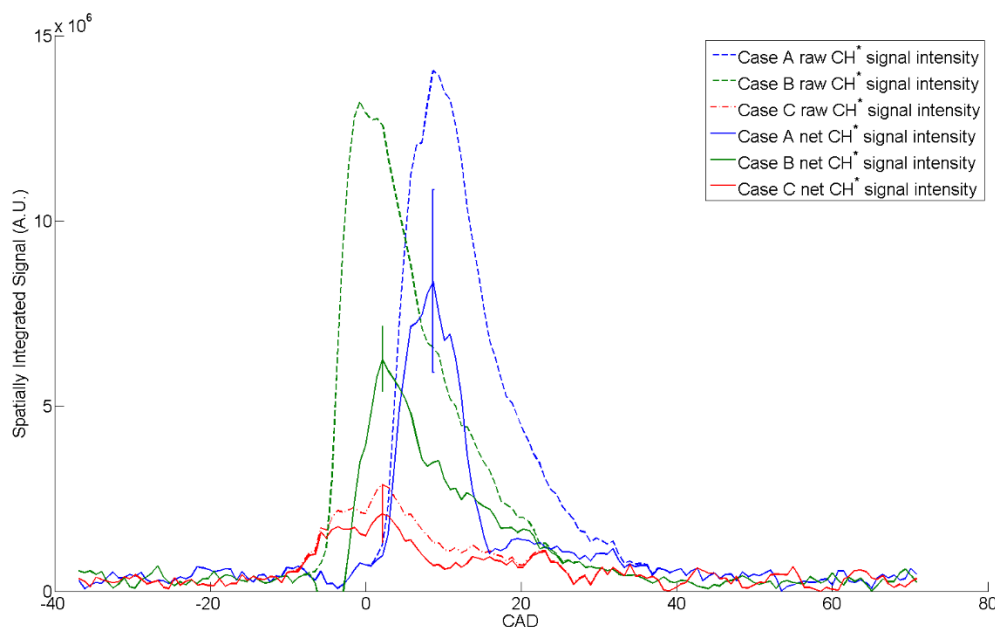


Figure 4.3 Spatially integrated CH^* chemiluminescence signal before and after the background interference correction for all cases considered.

The same trend can be observed when looking at the spatially resolved signal. Case A features the highest levels of CH^* chemiluminescence signal amongst all cases and this signal appears within the path of the fuel jets. In Case B, where the injection timing advances from 8CAD bTDC to 15 CAD bTDC, the reactivity within the path of the fuel jets drops and is limited towards the outer region of the fuel jet, as a result of the slightly longer premixing time available. Finally, Case C exhibits the lowest levels of CH^* signal, stemming from small pockets located mainly at the centre of the combustion chamber.

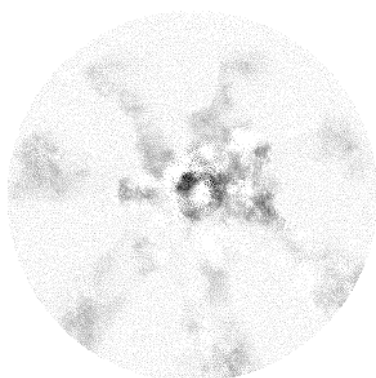


Figure 4.4 Average spatially resolved net CH^* signal at 8.7 CAD aTDC for Case A.

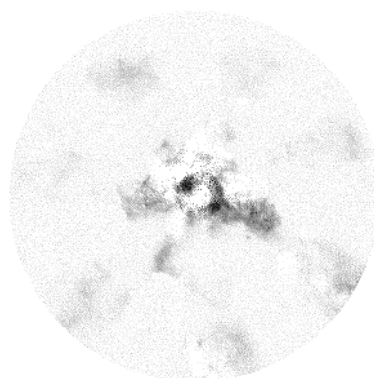


Figure 4.5 Average spatially resolved net CH^* signal at 2.2 CAD aTDC for Case B.

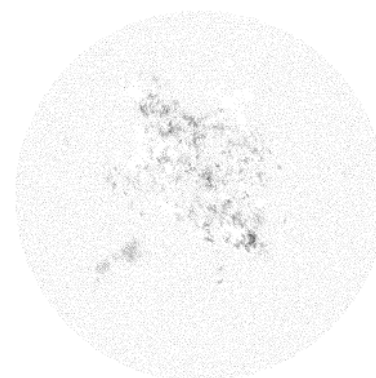


Figure 4.6 Average spatially resolved net CH^* signal at 2.2 CAD aTDC for Case C.

Figure 4.7 shows the evolution of the spatially integrated C_2^* chemiluminescence signal with crank position for the single injection cases. In Figure 4.7 the integrated C_2^* signal appears and peaks earlier, when the injection timing is advanced. Case A and B exhibit similar peak C_2^* signal, and thus rate of soot formation is similar. However, in Case A, soot formation is still ongoing late at the combustion cycle, after 20 CAD aTDC, implying that soot is more likely to survive through the oxidation process. Case C exhibits almost no net C_2^* signal, as a result of the extensive premixing. It becomes clear that the level of fuel and combustion stratification directly and monotonically affects the development of fuel-rich areas inside the cylinder and, subsequently, the formation of soot during the combustion.

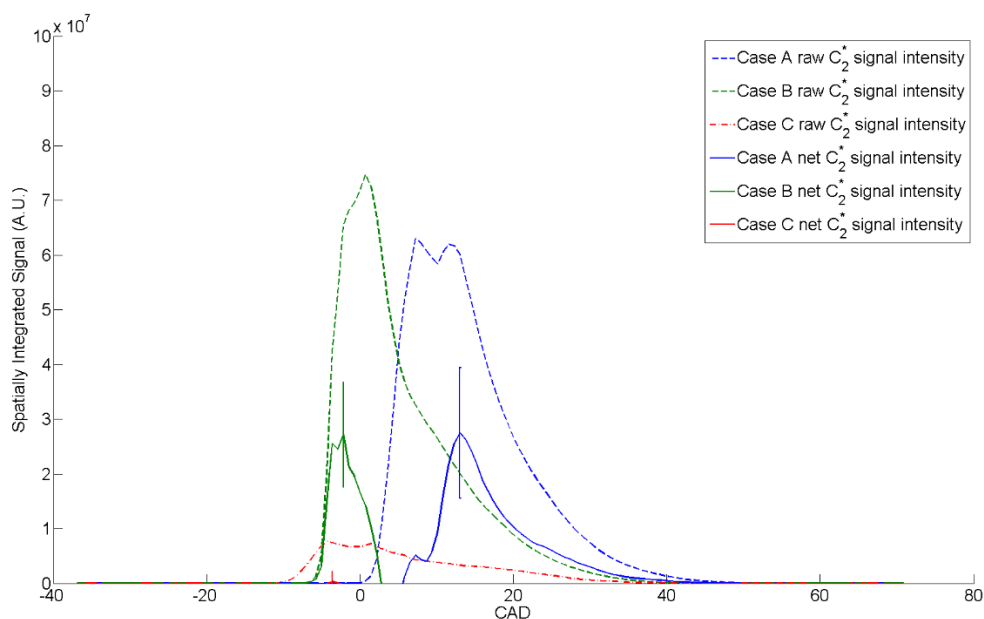


Figure 4.7 Spatially integrated C_2^* chemiluminescence signal before and after the background interference correction for all cases considered.

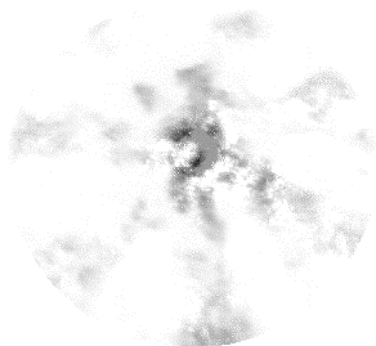


Figure 4.8 Average spatially resolved net CH^* signal at 8.7 CAD aTDC for Case A.

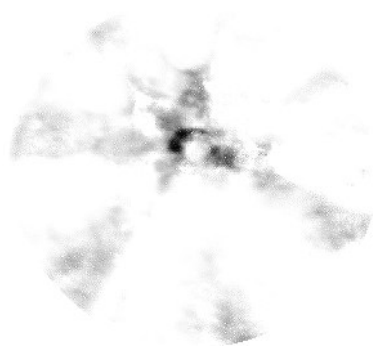


Figure 4.9 Average spatially resolved net CH^* signal at 2.2 CAD aTDC for Case B.



Figure 4.10 Average spatially resolved net CH^* signal at 2.2 CAD aTDC for Case C.

Figure 4.8 to Figure 4.10 show the maximum average spatially resolved C_2^* chemiluminescence signal for Cases A to C respectively. In Case A, the fuel-rich areas appear on the fuel jet paths, implying that the fuel did not have sufficient time to mix with the surrounding air before the start of combustion, which is typical for a conventional CI operation. In Case B, the distinctive fuel jet pattern is still distinguishable, however the fuel-rich areas do

not appear consistently on all 7 fuel jets. Finally, in Case C, the intensity of the fuel-rich areas is largely suppressed, stemming only from local spots around the injector tip.

Figure 4.11 shows the evolution of the spatially integrated natural luminosity signal with crank position, for all cases considered. When the injection timing advances, the overall spatially integrated signal drops, indicating lower combustion stratification and lower soot volume fraction inside the combustion chamber. This behaviour is pronounced for Case C, where longer premixing appears to be beneficial for in-cylinder soot formation, as observed in the C_2^* chemiluminescence measurements, too. In Figure 4.12 to Figure 4.14, it becomes evident that, as the injection timing advances, the natural luminosity signal on the fuel jet paths weakens and is limited towards the centre of the combustion chamber. Especially in Case C, the signal appears in small pockets towards the centre of the combustion chamber, which is similar to what was observed for the CH^* and C_2^* signal distribution.

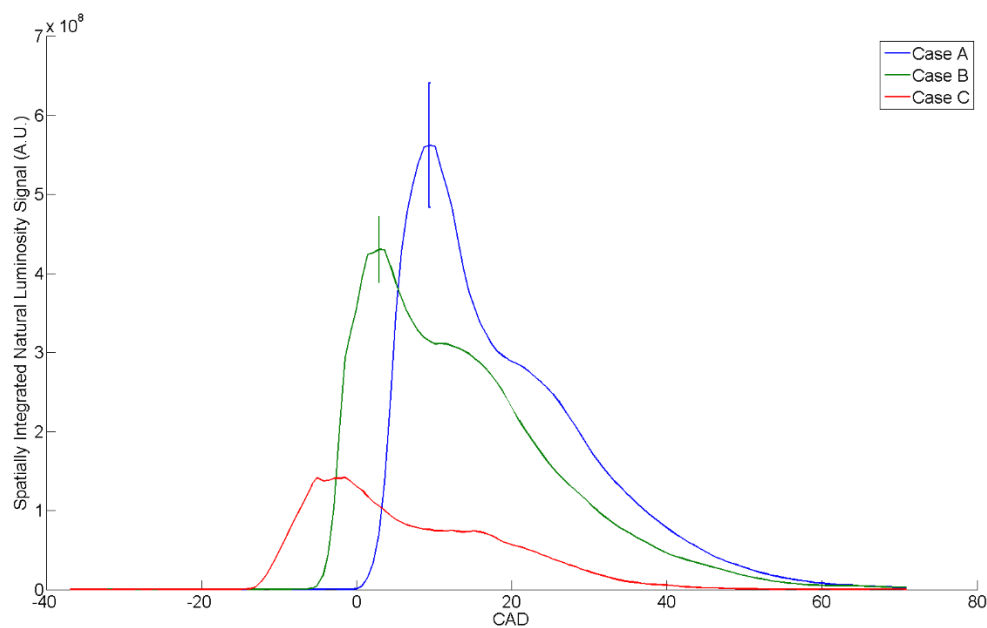


Figure 4.11 Spatially integrated natural luminosity signal as a function of crank angle for all cases considered.

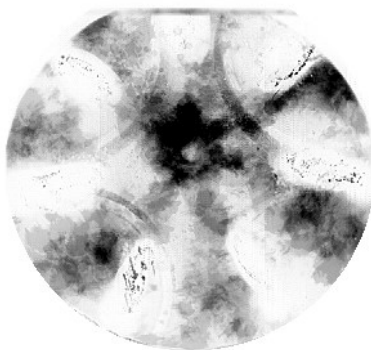


Figure 4.12 Average spatially resolved natural luminosity signal at 9.4 CAD aTDC for Case A.

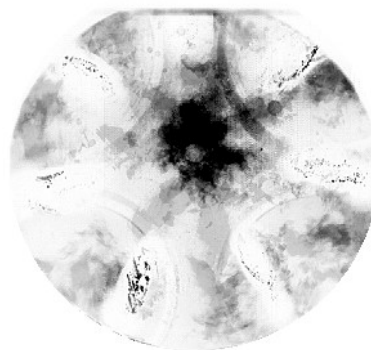


Figure 4.13 Average spatially resolved natural luminosity signal at 2.9 CAD aTDC for Case B.

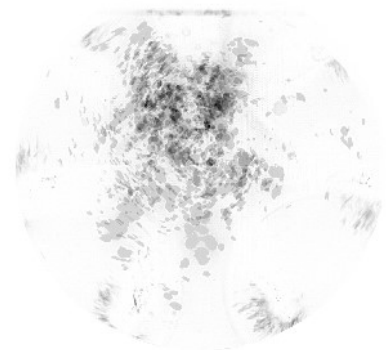


Figure 4.14 Average spatially resolved natural luminosity signal at 1.4 CAD bTDC for Case C.

Figure 4.15 shows the spatially integrated pLII signal for all cases considered, for three combinations of laser and piston positions as well as the total recorded signal when the piston was at 54 CAD aTDC. Figure 4.16 shows the respective spatially resolved pLII signal. For the sake of brevity, we use the notations “L” and “P” for the laser and piston positions respectively. Therefore, the combinations are L31P34, L31P54 and L45P54. Starting with L31P34, the spatially integrated pLII signal decreases when the injection timing advances, as seen in Figure 4.15, and this is in agreement with the trends observed for the fuel-rich areas and the natural luminosity signal. The spatially resolved pLII signal for the L31P34 combination, seen in the first row of Figure 4.16, reveals the areas inside the cylinder where soot is detected. For Case A, these locations are in agreement with the locations where fuel rich areas are detected using C_2^* chemiluminescence, as seen previously in Figure 4.8. The observation above enhances the argument that soot is generated in the fuel-rich areas formed inside the cylinder. The pLII signal at P34 for Case C is the lowest amongst the single injection cases, because the combustion is highly premixed, and the fuel rich areas are scarce. The total recorded pLII signal at P54 is lower compared to P34 for all the cases considered, presumably due to the ongoing soot oxidation and the volumetric expansion. It is interesting to note that at P54, the recorded pLII signal for Case C is virtually null. The overall low pLII signal is not surprising, given that the investigated cases, except for Case A, feature good premixing of the charge and the engine is operating at a low speed and low load regime.

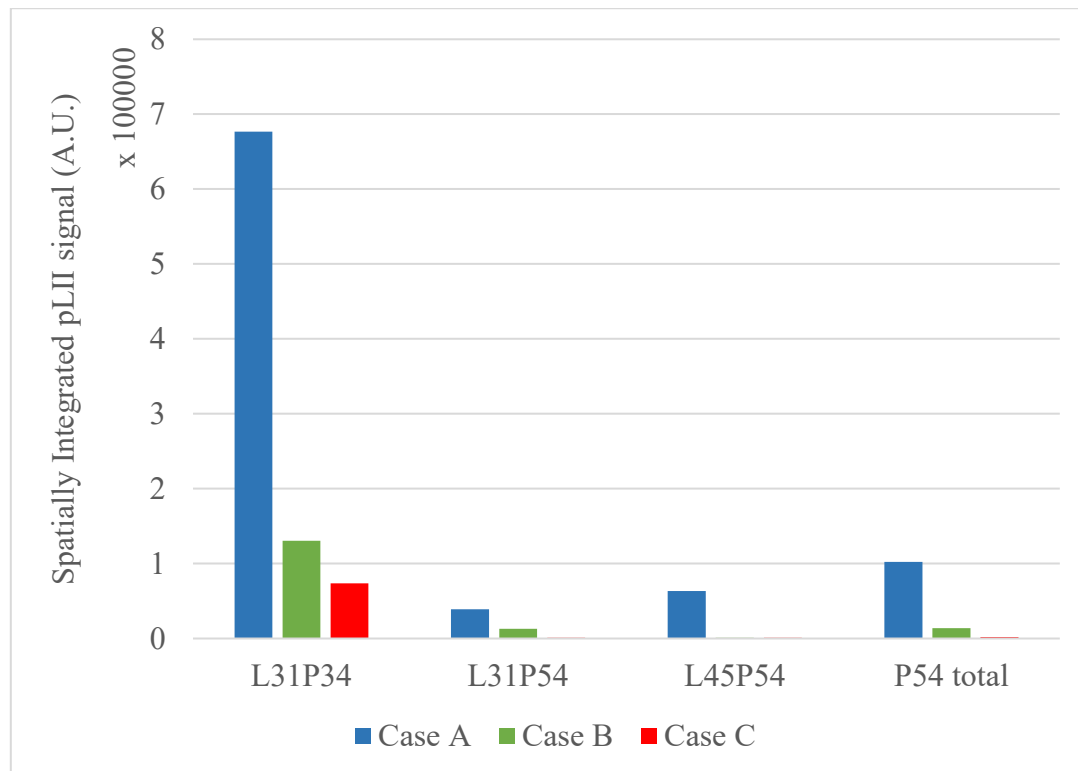


Figure 4.15 Spatially integrated pLII signal for three combinations of laser and piston positions, for the single injection cases, *i.e.* Cases A to C.

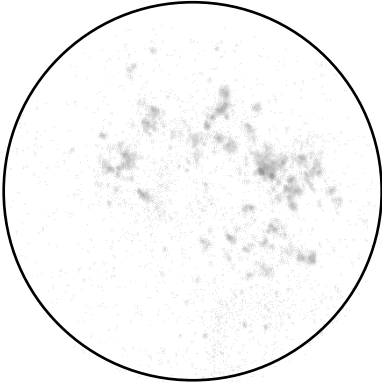
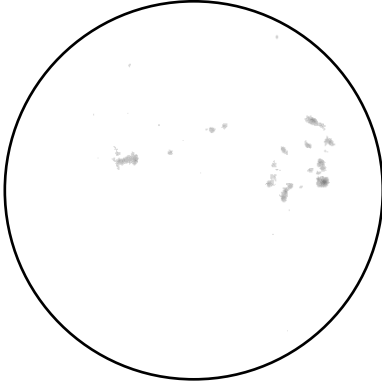
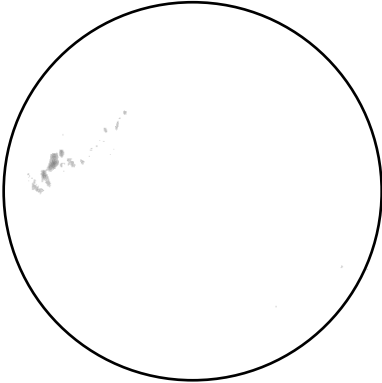
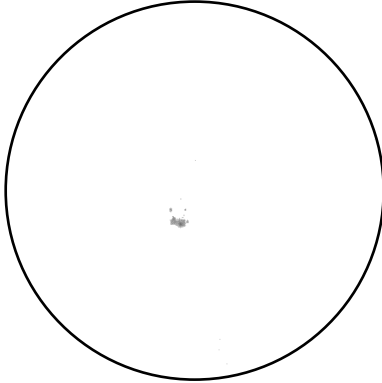
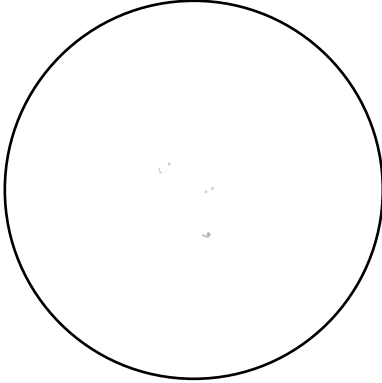
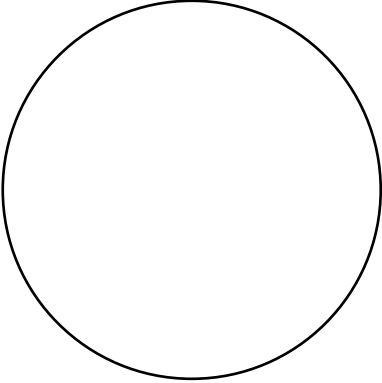
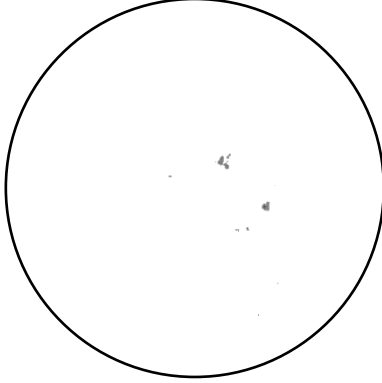
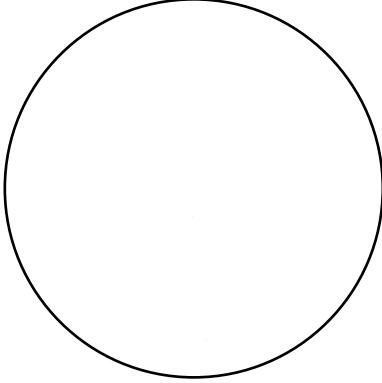
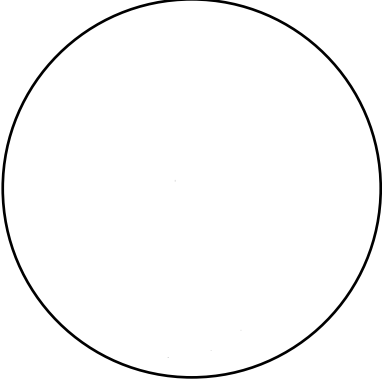
		pLII signal		
Laser Position	Piston Position	Case A	Case B	Case C
31 CAD	34 CAD			
31 CAD	54 CAD			
45 CAD				

Figure 4.16 Spatially resolved ensemble-averaged pLII signal for three combinations of laser and piston positions, for all cases considered. The black circle represent the optical window contour.

4.2.4 Conclusions

The effect of injection timing on combustion stratification and soot formation is investigated on the Volvo D5 optical engine at operating conditions that promote PPC operation. More specifically, the engine operated at the low speed, low load regime (approx. 3 bar IMEP constant at 1200 RPM), using three single injection schemes with a constant injection pressure at 500 bar. The duration of the main injection event was adjusted for all cases, so that the IMEP is kept constant. Findings are analysed *via* means of pressure trace and apparent rate of heat transfer analyses, as well as a series of optical diagnostics techniques, namely high speed flame natural luminosity imaging, CH^* and C_2^* line-of-sight chemiluminescence, as well as planar Laser Induced Incandescence (pLII). The pLII measurements are conducted in three different combinations of laser sheet position and piston position. It is important to note that for the chemiluminescence measurements, the importance of background interference has been assessed and removed.

In brief, results indicate:

- For the CH^* and C_2^* chemiluminescence measurements, the interference of background chemiluminescence and thermal radiation signal is significant. The background interference must be removed to get more credible qualitative observations. The observed chemiluminescence signal trends are largely maintained after the background interference correction.
- When the injection timing advances, the stratification of the combustion becomes weaker, as indicated by the natural luminosity signal, due to the better pre-mixing between the injected fuel and the surrounding air.
- The early injection case (*i.e.* Case C) exhibits the lowest levels of soot late at the combustion cycle and, presumably the lowest engine-out soot emissions. This is because of the efficient pre-mixing, caused by the early injections, which limits the formation of locally fuel-rich areas inside the cylinder. This comes at the price of increased fuel consumption.

4.3 Chapter Summary

The experimental work conducted on the Volvo D5 optical engine was presented. The work focused on PPC operation at the low speed, low load regime, facilitated by advanced single and double injection strategies, varying in injection timing, and leading to different levels of combustion stratification and, subsequently to different combustion patterns and soot evolution. The analysis was performed *via* means of in-cylinder pressure trace, high speed natural flame luminosity and species chemiluminescence, as well as soot Laser Induced Incandescence.

5 Conclusions and Future Work

5.1 Introduction

The main objective of the present thesis was to assess the effect of various engine operating strategies, under low speed and low load operation, when moving from conventional Compression Ignition to Partially Premixed Combustion, on in-cylinder mixing and combustion characteristics as well as to study the governing mechanisms of soot formation and oxidation. The strategies investigated were, namely, intake air dilution (resembling EGR), close-coupled pilot and main injections, post-injections and advanced injection timings. The first three strategies were investigated in a Ricardo Hydra optical engine, which was equipped with a Ford Puma engine head, while the latter strategy was investigated in a Volvo D5 optical engine. The techniques used throughout the thesis were in-cylinder pressure analysis, high speed imaging of the flame's natural luminosity and of chemiluminescent species, as well as Laser Induced Incandescence (LII). To summarize the study, three main topics, based on the research objectives, are discussed below. Firstly, the aspect of enhanced in-cylinder mixing, as a result of the implementation of the operating strategies mentioned above, which leads to switching from conventional CI to PPC operation, is discussed. Secondly, the implication of the operating strategies on combustion behaviour are summarized and, finally, soot formation and oxidation processes are discussed.

5.2 In-cylinder mixing

The fuel/air mixing inside the cylinder is a process governed by time and fluid flow characteristics, such as the swirling flow induced by the air path design, as well as turbulence induced both by the piston movement and the fuel injection. In a direct injection engine, the mixing process takes place from the start of the first fuel injection until the start of combustion and largely affects the subsequent combustion and emissions formation.

Enhanced air-fuel mixing can be achieved either by increasing the available mixing time, or by altering the fluid flow characteristics. Inlet air dilution with a stream of CO_2 , which was investigated in section 3.2, led to longer ignition delay of the charge, as revealed by the in-cylinder pressure analysis and the C_2^* chemiluminescence imaging, due to the higher heat capacity of the triatomic molecules of carbon dioxide and the lower oxygen content of the charge. Enhanced premixing of the in-cylinder charge due to longer ignition delay was also observed for the case of a close-coupled pilot-main injection scheme, as seen in section 3.3. When the dwell time between the pilot and the main injection was halved from $556\ \mu\text{s}$ to $278\ \mu\text{s}$, the ignition delay was prolonged, presumably due to the overlapping of the main injection plume with the pilot injection plume and the induced cooling of the former to the latter, which prevented the hot auto-ignition of the pilot fuel to take place. Advanced injection timings as early as 30 CAD bTDC, which were investigated in section 4.2, resulted in lower combustion stratification, due to enhanced pre-mixing, stemming from longer ignition delay caused by the low thermodynamic potential away from TDC. Finally, higher injection pressures, which were investigated in section 3.3, increase the turbulence of the flow around the fuel jets, causing better atomization of the liquid fuel, faster evaporation and, consequently, more efficient mixing. It is reasonable to conclude that a combination of the aforementioned strategies can be used in order to shift the engine operation towards a PPC mode with the desired level of fuel stratification inside the cylinder before the start of combustion, especially for the low speed regime, where mixing time availability is not an issue.

5.3 Combustion

The mixing process discussed above affects to a large extent the subsequent combustion event. The degree of fuel-air mixing before the start of combustion affects the ratio between the premixed and the diffusion phase of the combustion, which can be derived directly from the calculated CAHT curve. This ratio between premixed and diffusion phase largely affects emission formation during the combustion. One of the main issues with conventional CI operation is that fuel is injected close to, or even after TDC and has little to no time available to mix with the surrounding air. The injected fuel auto-ignites before the end of injection and thus a large amount of fuel burns in a diffusion mode as soon as it is injected. On the contrary, during PPC operation, the enhanced fuel-air mixing leads to a prolonged and more intense premixed phase, while suppressing the diffusion phase of the combustion. It is important to

note here that an enhanced premixed phase features higher peak pressure and pressure rise rate and, presumably, higher noise levels.

Higher injection pressure and short dwell times between the pilot and the main injection in section 3.3 were found to enhance the premixed phase and suppress the diffusion phase of the combustion. The same was observed for the advanced injection timing cases in section 4.2. When an early injection timing was employed, the diffusion phase of the combustion was virtually eliminated, nevertheless at the expense of higher fuel consumption. On the contrary, inlet air dilution with CO₂ addition resulted in weakened premixed phase, lower peak pressures, lower noise (PRR) and in an overall delay of the combustion process. The spatial characteristics of reaction and oxidation zones were influenced, and the extent of fuel-rich areas was limited.

When a post-injection was employed, the AHTR curve exhibited a second peak at the start of the post-injection, indicating that the post-injected fuel entered directly into diffusion burning mode. As a result, flame intensity and fuel-rich areas intensity were enhanced at the time of the post-injection and when the post-injection was close-coupled to the main injection event, the reaction and oxidation zones were enhanced and prolonged, leading to higher in-cylinder bulk temperatures.

5.4 Soot formation and oxidation

Engine-out soot emissions are determined by the outcome of the competition between soot formation and soot oxidation rates. Soot is formed inside the fuel-rich areas of the cylinder, which were depicted with the use of C₂^{*} chemiluminescence, where the local equivalence ratio exceeds the value of 2. On the other hand, soot oxidation takes place at the outer surface of the diffusion flame, where soot particles come in contact with the available oxygen, provided that the appropriate thermodynamic conditions are in place.

The formation of fuel-rich areas can be avoided and thus, soot formation can be weakened, when the air-fuel mixing before the start of combustion is enhanced, as described in section 5.2. PPC operation promotes the conditions for more efficient and prolonged mixing periods, thus limiting, or even eliminating in some cases, the formation of fuel-rich areas inside the cylinder. This was the case when inlet air dilution and injection schemes featuring either early fuel injection timings, or high-pressure close-coupled pilot-main injection events, were employed. On the other hand, soot oxidation after the main combustion event is enhanced when

a close-coupled post-injection is employed. The enhanced soot oxidation is associated with the ability of this strategy to increase the bulk gas temperature, allowing for soot oxidation to take place for a longer period at the expansion stroke, as well as its ability to make better use of the remaining unburnt oxygen inside the cylinder. Both the timing and the quantity of the post-injection facilitate the exploitation of the available oxygen, the former by placing the fuel jets between the product gases of the main combustion event and the latter by increasing the penetration length of the fuel jets.

5.5 Future work

The current research on compression ignition engines focuses on combustion strategies and their combination with novel fuels, in an effort to achieve higher efficiency with lower emissions (*e.g.* Reitz and Duraisamy 2015; Shuai 2015; Kokjohn 2015; Kavuri, Paz, and Kokjohn 2016). The experimental work conducted for the purposes of this thesis, especially in section 4.2, has revealed a need for fuels with longer ignition delay than commercial diesel when operating under early-injection PPC, so as to provide sufficient premixing time between the start of injection and start of combustion, and shift the peak pressure location closer to the TDC, for better efficiency. Low octane gasoline or gasoline blends serve as attractive alternatives to commercial diesel fuel in light-duty compression ignition engines (what is known as Gasoline Compression Ignition or GCI), when operating under PPC mode, as they feature longer ignition delay (Ahmadi & Ghadikolaei, 2014; Cracknell *et al.* , 2014; G. Kalghatgi & Johansson, 2017; G. T. Kalghatgi, 2014).

Moving towards that direction, it would be interesting to test commercial gasoline and other gasoline blends fuels available in the fuel storage of the Mechanical Engineering Department in both the Volvo D5 optical engine and the Ford Puma I4 metal engine, using a combination of optical diagnostics and exhaust gas measurements. It is important to note that GCI under PPC mode can lead to high levels of instantaneous heat transfer and pressure rise rate and thus, combinations of EGR, inlet air boosting and multiple injection schemes should be employed and investigated, in order to control the in-cylinder pressure and the heat transfer efficiently.

Additionally, as far as the Volvo D5 optical engine is concerned, a new, shallower, omega-shaped optical piston could be manufactured and installed to follow the trends of Euro 6 diesel

engines. New optical diagnostics can be applied with little modification to the existing laser setup, such as Formaldehyde LIF and Tracer LIF.

6 References

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Appendix A: Statistical analysis of high speed images

Standard Deviation analysis is a statistical method which gives an estimation of how far away from the mean lies a value of the dataset. Proper Orthogonal Decomposition (POD) is a widespread method of data analysis which is often used to extract “mode shapes” or basis functions from experimental data. In the case of high-speed imaging of fast evolving phenomena, such as combustion in engines, the POD method can be employed to extract information about coherent spatial structures during the evolution of the flame and to determine the cycle-to-cycle variations of the flame structures. For a detailed analysis of the POD method used refer to Bizon *et al.* , 2009, 2010.

For the purposes of the present thesis, both methods were employed mainly to determine the cycle-to-cycle variability of the flame structure, with an example given below. Both the SD and POD analyses used images from 10 consecutive firing cycles. Since the average image at a given crank position was not subtracted by the dataset during the POD analysis, the 1st POD mode coincides with the average and the 2nd POD mode presents the most prevalent spatial fluctuations of the flame structure at a given crank position (the fluctuations with the highest “energy”). When the total energy of the 2nd mode is low, compared to the energy of the 1st mode, then the fluctuations are limited and of low intensity, meaning that the cycle-to-cycle variability is also low. On the contrary, higher energy of the 2nd POD mode implies higher cycle-to-cycle variability.

Figure A 1 presents the average image and the standard deviation image at selected crank positions for Case B in the post-injection campaign (see Section 3.4). Figure A 2 presents the average Standard Deviation and the Coefficient of Variance of the high speed images for each crank position.

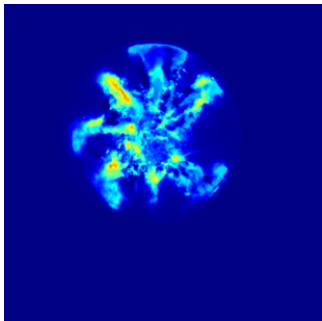
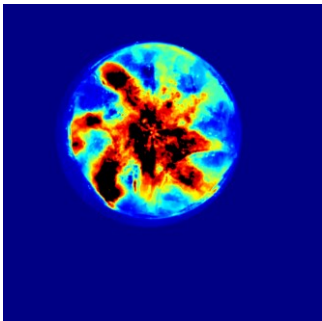
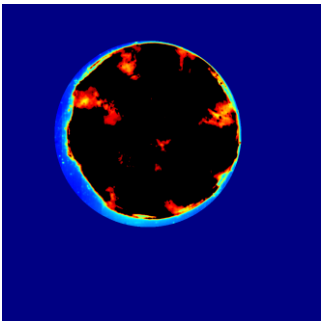
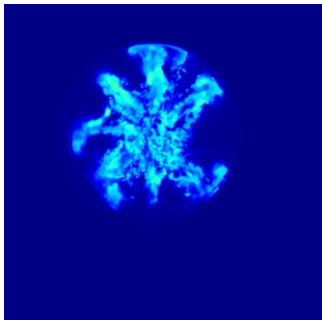
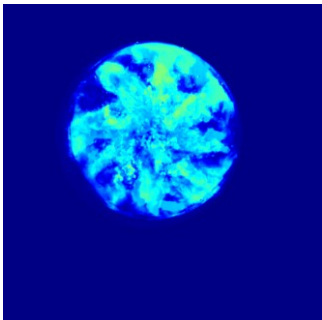
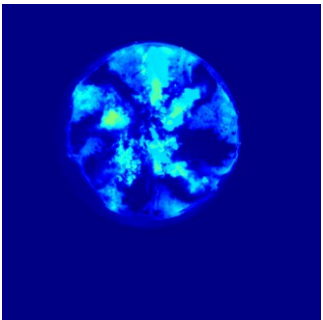
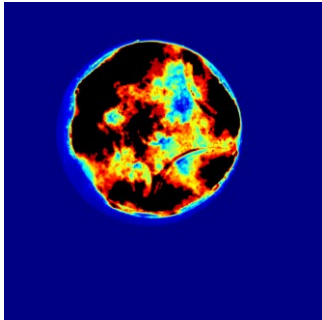
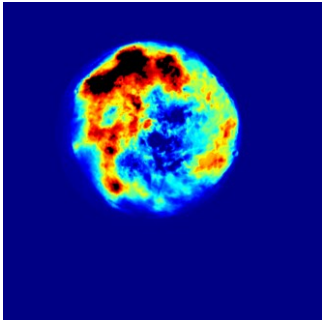
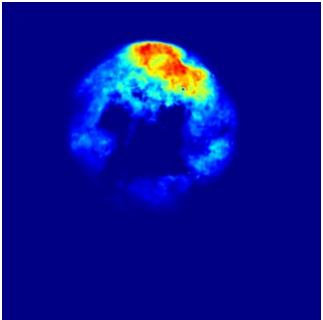
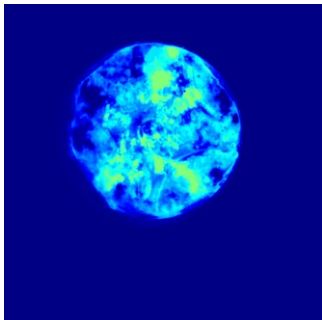
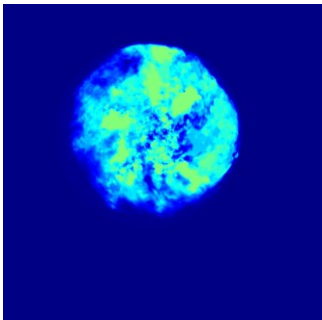
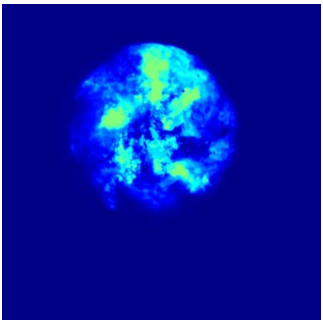
CAD	1.4 CAD	4.3 CAD	10.8 CAD
Average image			
Standard deviation			
CAD	20.1 CAD	32.4 CAD	44.6 CAD
Average image			
Standard deviation			

Figure A 1 Indicative selection of SD images for natural luminosity data from Case B in the Post-injection campaign.

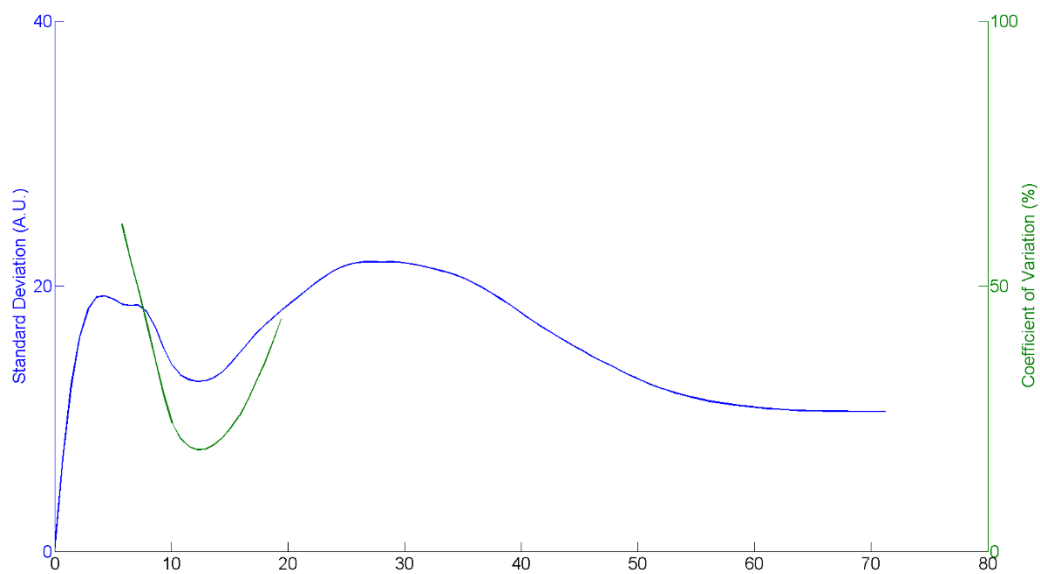


Figure A 2 Average SD and COV of high speed natural luminosity images for each crank position of Case B in the post-injection campaign.

In Figure A 2 it becomes evident that the Coefficient of Variation has its lowest values (approx. 20%) around the CA50, which is the midpoint of the combustion evolution, indicating that combustion variability is minimum around CA50.

Figure A 3 presents the first two POD modes of the spatially resolved natural luminosity signal for the same crank positions of Case B in the post-injection campaign. Let us, for example, focus on the two POD modes at 1.4 CAD to illustrate how the POD method can be used. Looking closely to the 2nd POD mode, it becomes evident that the most prevalent fluctuation close to the start of combustion appears for the 12 o'clock combusting fuel jet. This could indicate instabilities of the jet during the fuel injection, or in-cylinder flow instabilities.

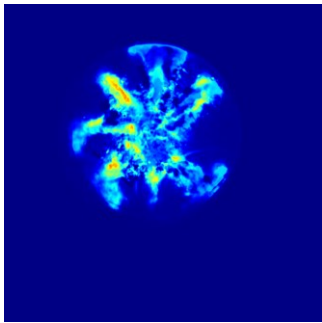
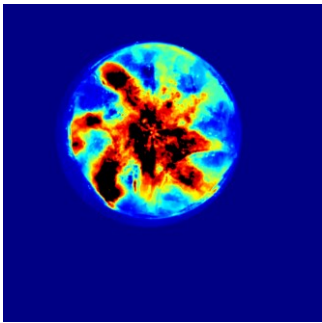
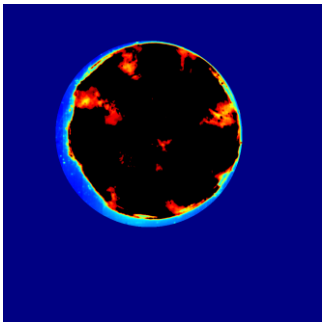
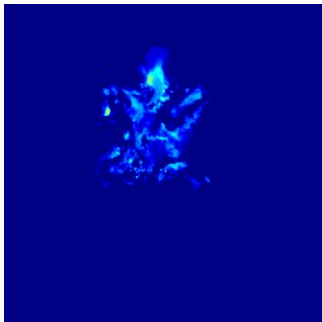
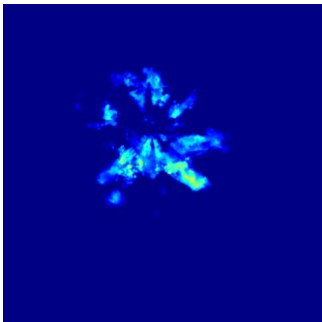
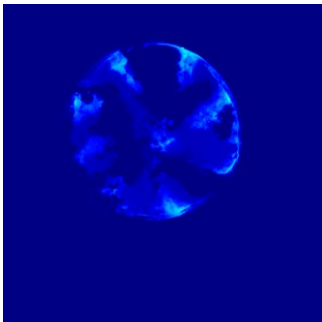
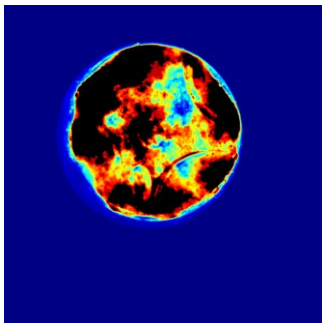
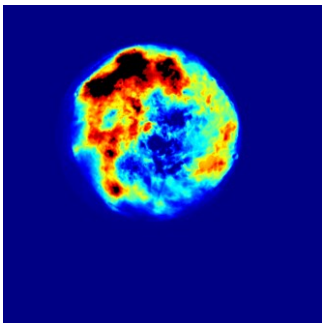
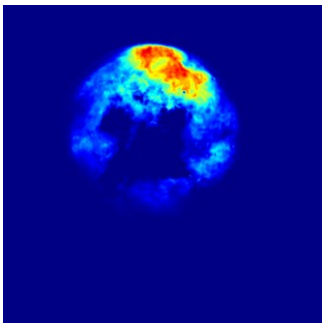
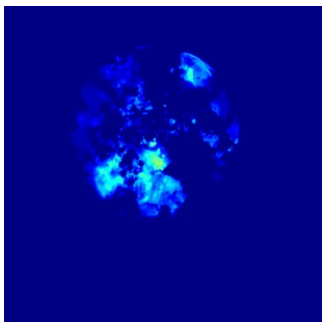
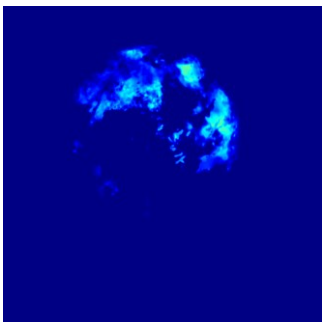
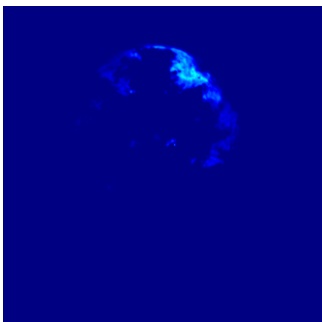
CAD	1.4 CAD	4.3 CAD	10.8 CAD
1 st POD mode (average)			
2 nd POD mode			
CAD	20.1 CAD	32.4 CAD	44.6 CAD
1 st POD mode (average)			
2 nd POD mode			

Figure A 3 Indicative selection of POD images for natural luminosity data from Case B in the Post-injection campaign (Section 3.4).

Figure A 4 presents the energy of the first 8 POD modes for the selected crank positions, while Figure A 5 presents the energy percentage of the 2nd POD mode to the 1st POD mode (average). For the crank angles close to the start and end of combustion (as indicated by the CA10 and CA90 values, see Table 3.6), the energy percentage of the 2nd mode to the 1st mode lies around 35%, indicating significant cycle-to-cycle variability, which can be confirmed by observing the POD images in Figure A 3. However, for crank angles around the midpoint of the combustion, as indicated by CA50, this percentage decreases significantly, reaching values below 2%, indicating very low cycle-to-cycle variability.

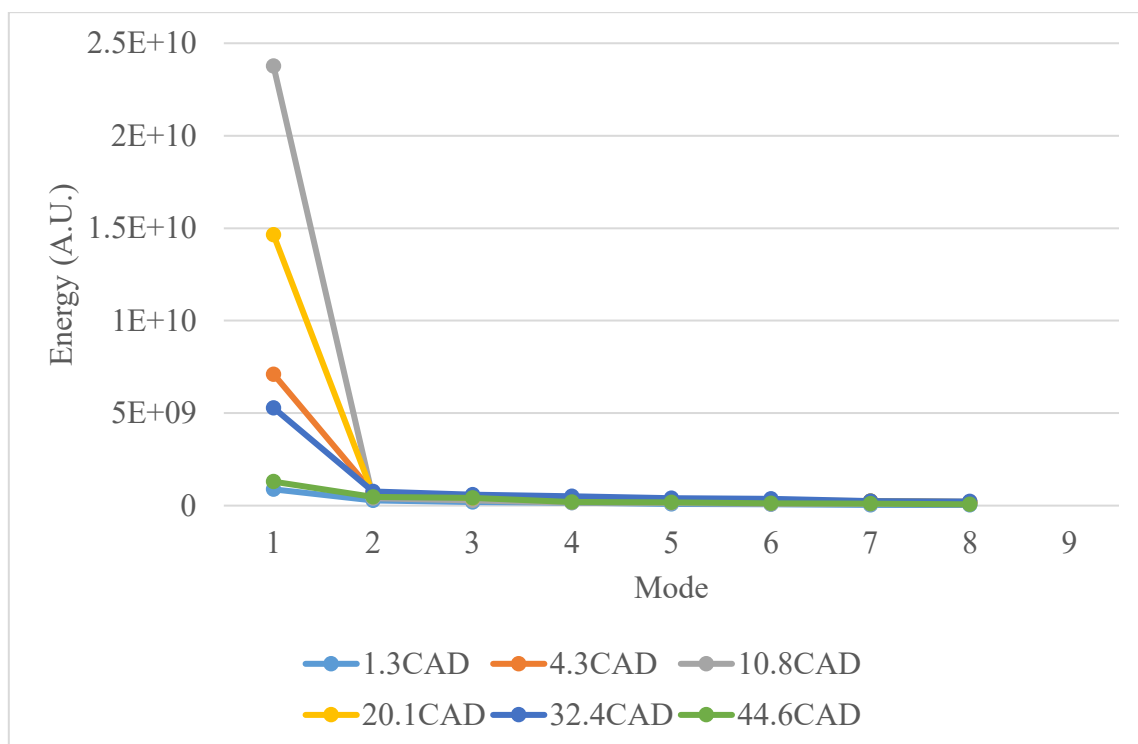


Figure A 4 Energy of the first eight POD modes at the selected crank positions.

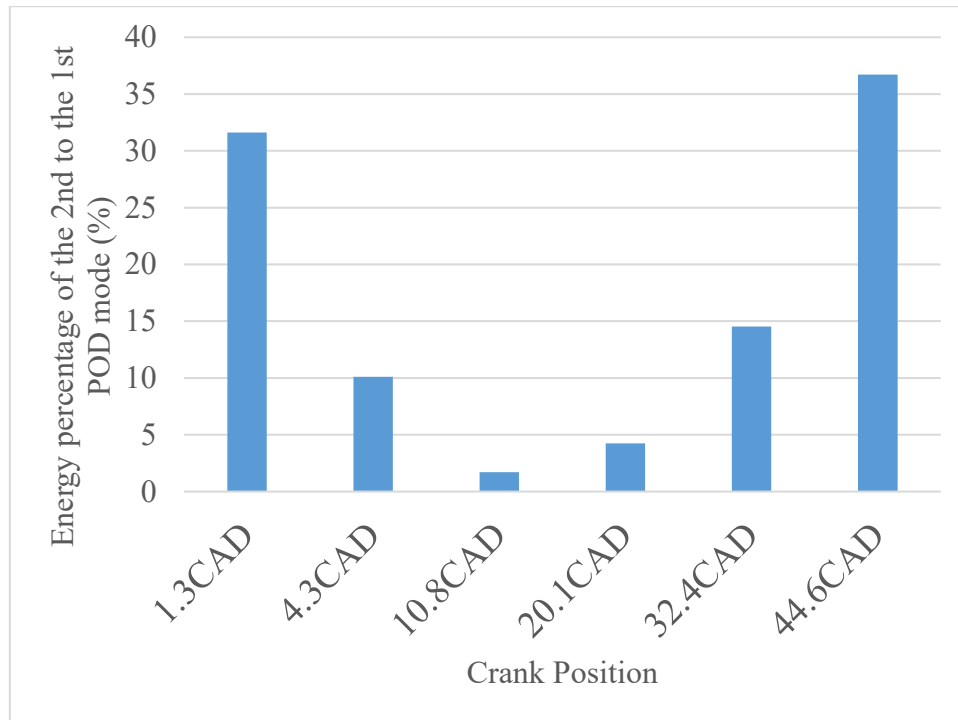


Figure A 5 Energy percentage of the 2nd mode to the 1st mode.

Appendix B: Assessment of chemiluminescence measurements in Chapter 4

The effect of the background interference in the chemiluminescence signal of CH^* and C_2^* was assessed for the measurements conducted on the Volvo D5 optical engine and which were presented in Chapter 4. For the assessment, the methodology described in section 2.4.2 was used. In this appendix the results of this assessment are presented in more detail.

The operating conditions were described in Chapter 4. Here is a quick recap:

- Case A: Single injection of 14.2 mg at 8CAD bTDC and 500 bar with 3.2 bar IMEP
- Case B: Single injection of 14.8 mg at 15CAD bTDC and 500 bar with 3.2 bar IMEP
- Case C: Single injection of 16.5 mg at 30CAD bTDC and 500 bar with 3.2 bar IMEP

Figure B 1 to Figure B 3 presents the spatially integrated signals captured for each of the three experimental cases and for each of the two chemiluminescent species. Each graph contains the raw (dashed line) and adjusted (solid line) signals captured at three wavelengths; one is the wavelength of detection for the chemiluminescent species and the other two are wavelengths on both sides of the chemiluminescence wavelength, where only background signal is detected and there is no interference with any other chemiluminescent species.

Figure B 4 to Figure B 9 present a compilation of raw and adjusted spatially resolved CH^* and C_2^* chemiluminescence signals for all the cases considered.

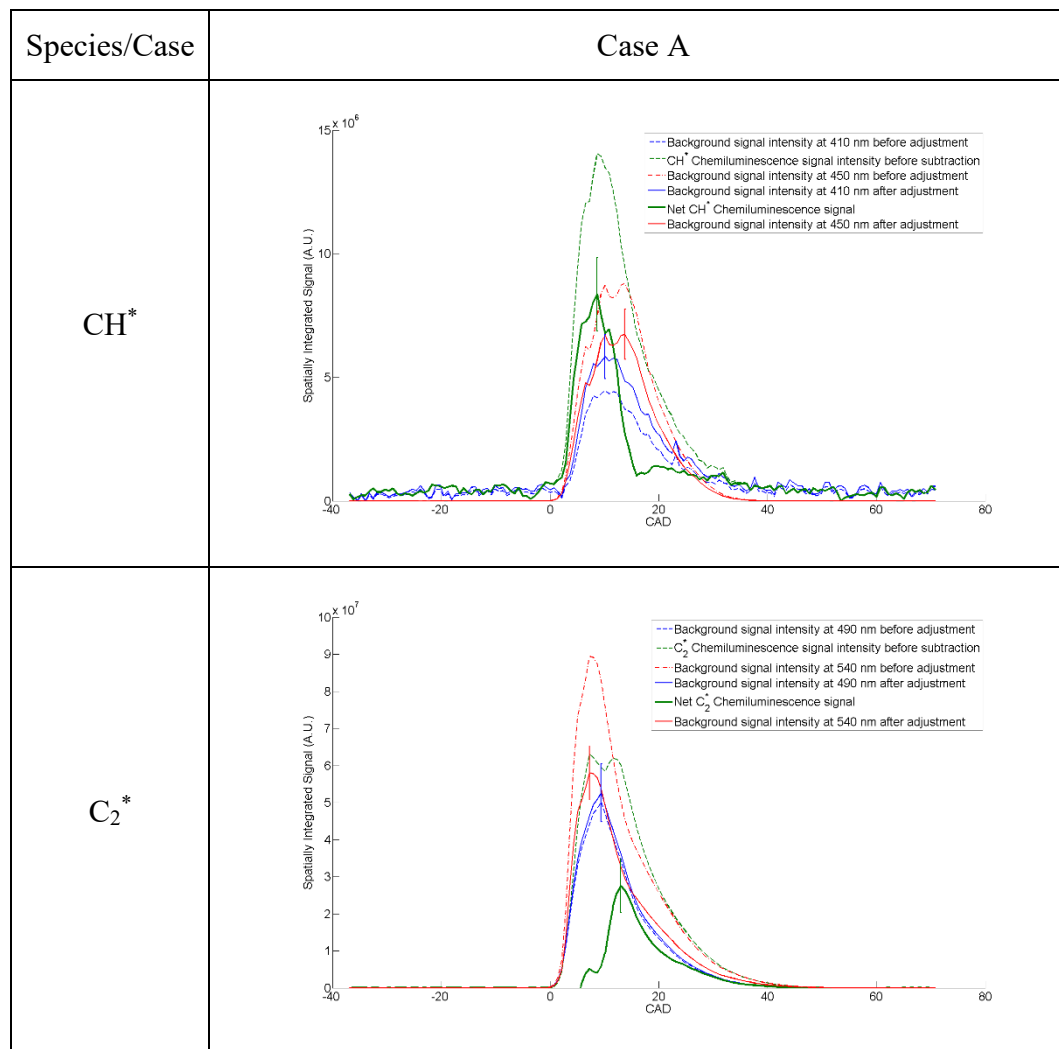


Figure B 1 Assessment of background interference on spatially integrated chemiluminescence measurements for Case A.

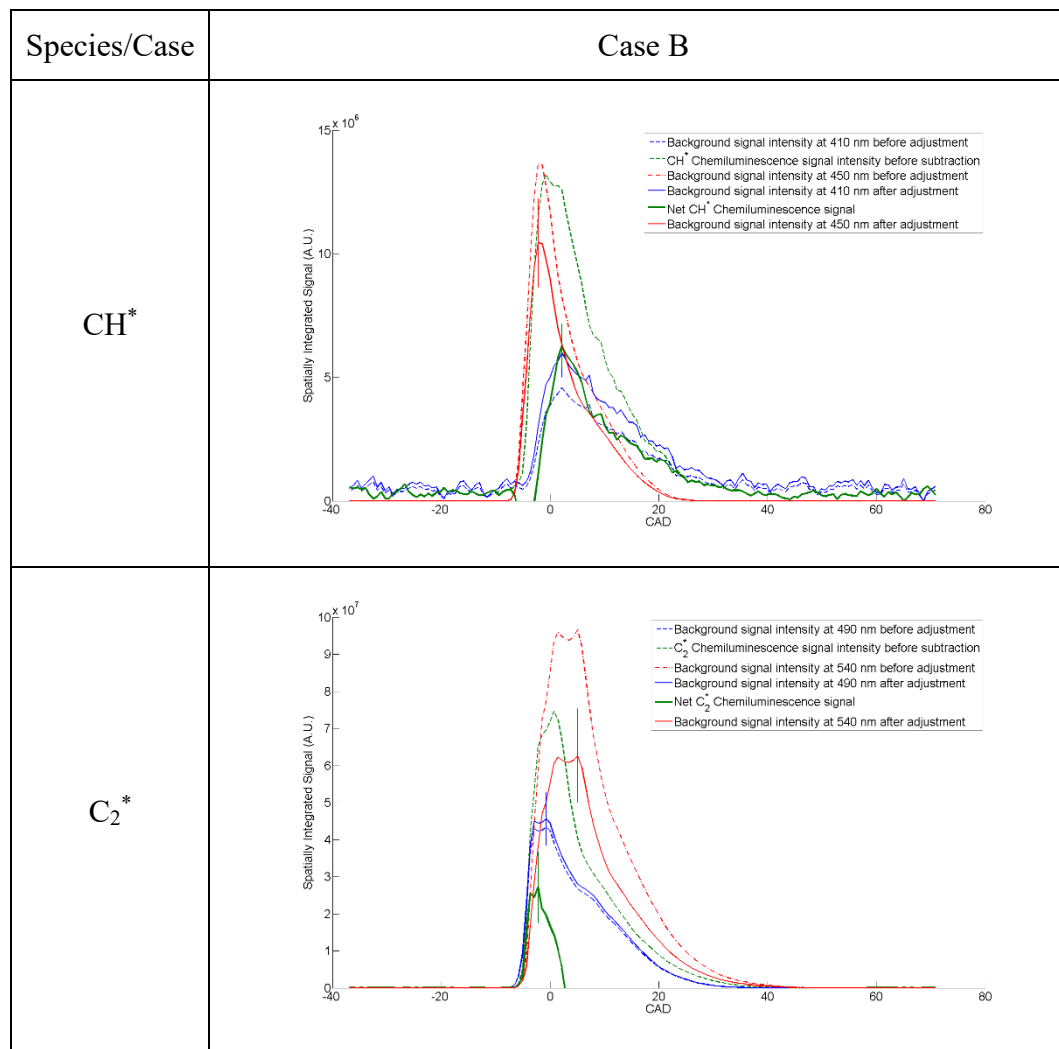


Figure B 2 Assessment of background interference on spatially integrated chemiluminescence measurements for Case B.

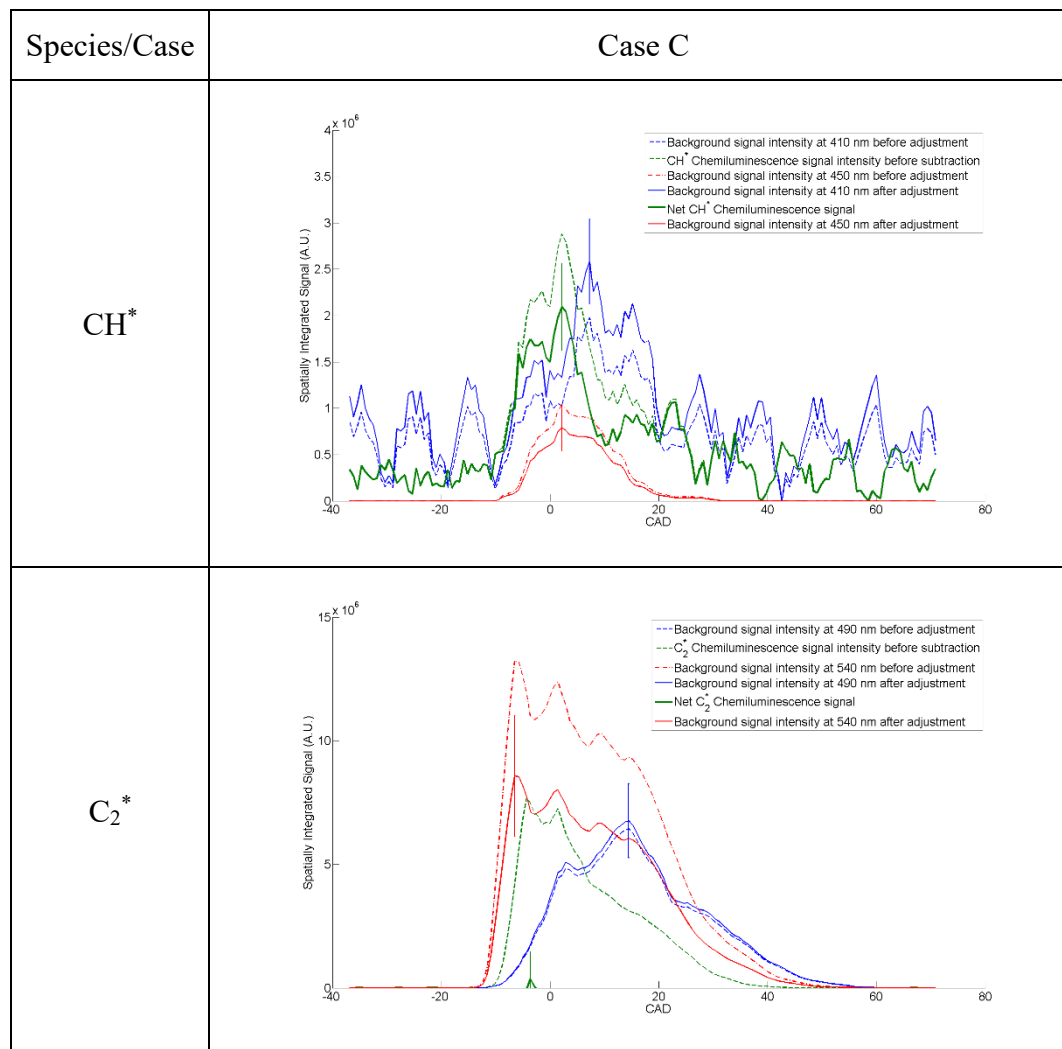


Figure B 3 Assessment of background interference on spatially integrated chemiluminescence measurements for Case C.

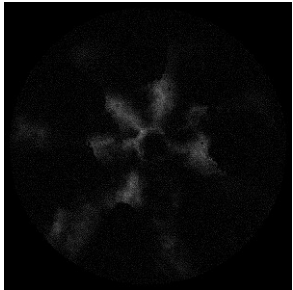
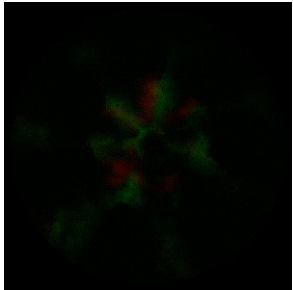
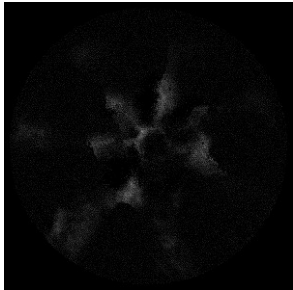
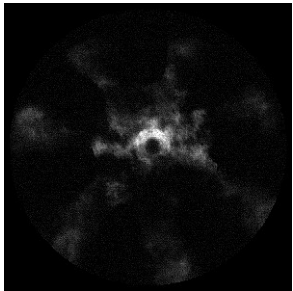
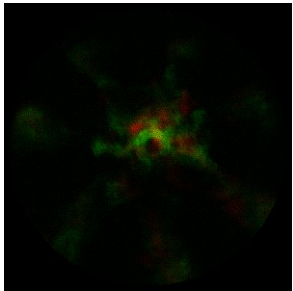
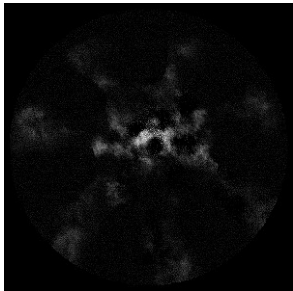
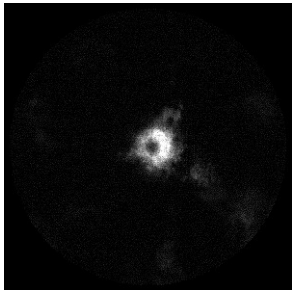
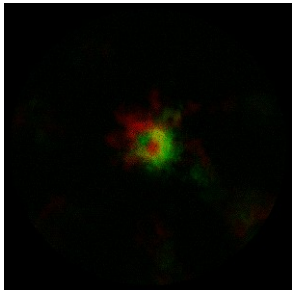
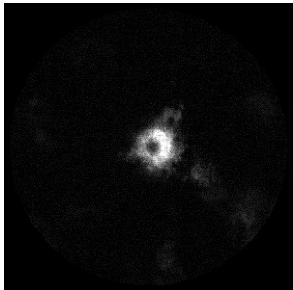
CAD/Spatially resolved signal	CH [*] signal before correction	Fused CH [*] (green) and adjusted background (red) signal	Net CH [*] signal
4.3			
8.0			
15.2			

Figure B 4 Compilation of spatially resolved CH^{*} signal images before and after background correction for selected crank positions of Case A.

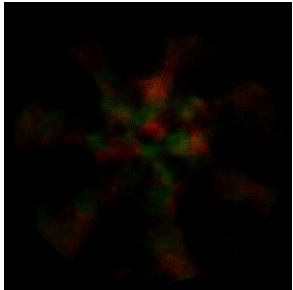
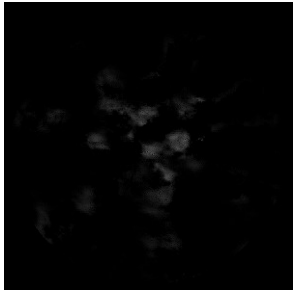
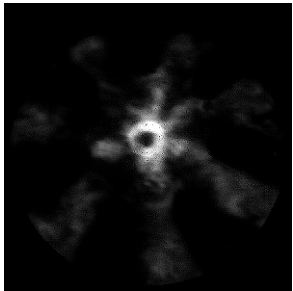
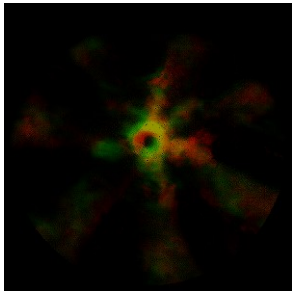
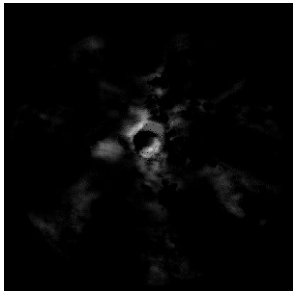
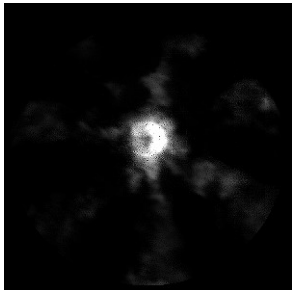
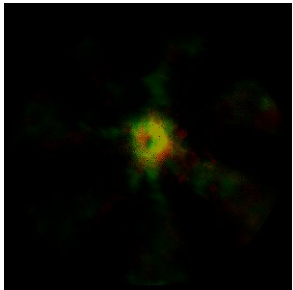
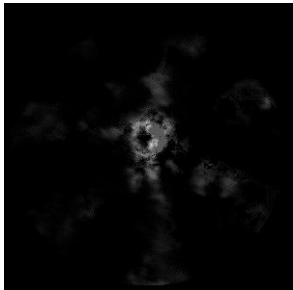
CAD/Spatially resolved signal	C_2^* signal before correction	Fused C_2^* (green) and adjusted background (red) signal	Net C_2^* signal
4.3			
8.0			
15.2			

Figure B 5 Compilation of spatially resolved C_2^* signal images before and after background correction for selected crank positions of Case A.

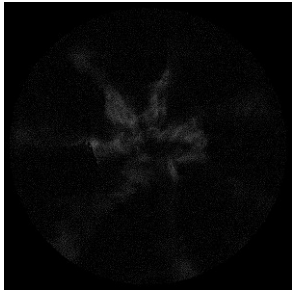
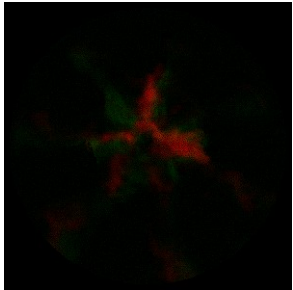
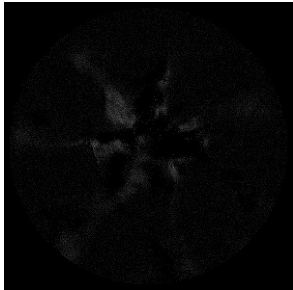
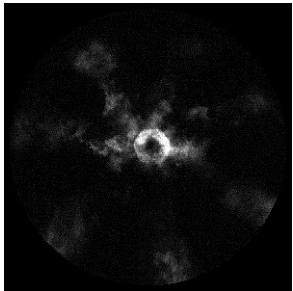
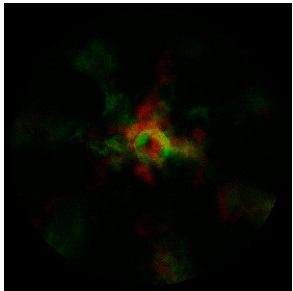
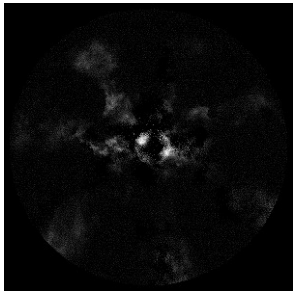
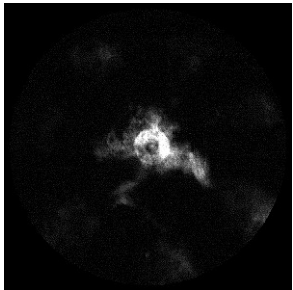
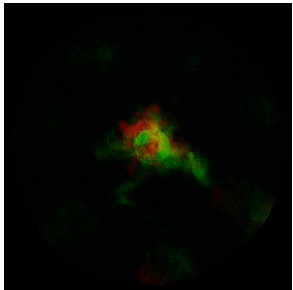
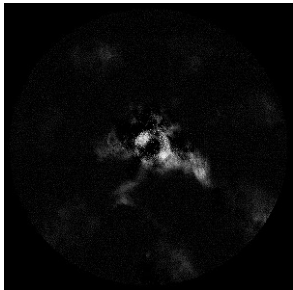
CAD/Spatially resolved signal	CH [*] signal before correction	Fused CH [*] (green) and adjusted background (red) signal	Net CH [*] signal
-3.6			
0.0			
2.9			

Figure B 6 Compilation of spatially resolved CH^{*} signal images before and after background correction for selected crank positions of Case B.

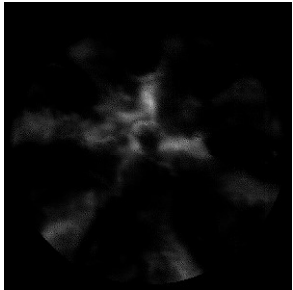
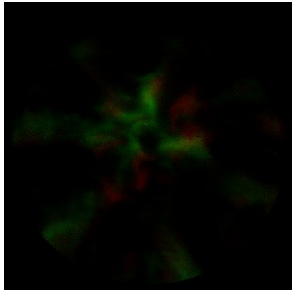
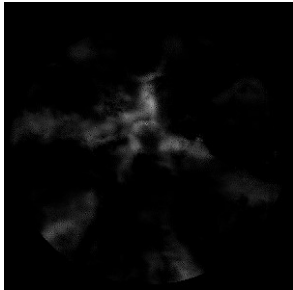
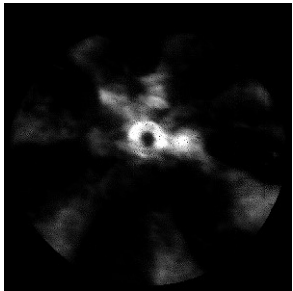
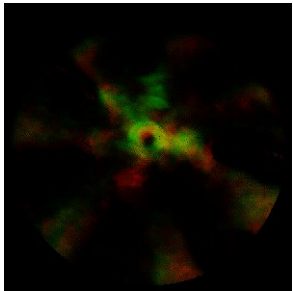
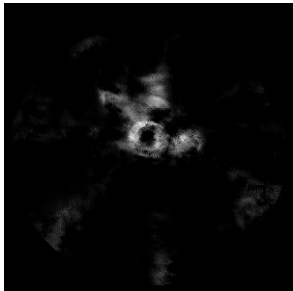
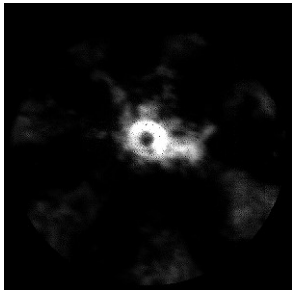
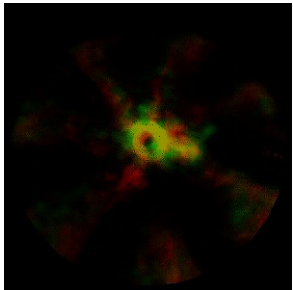
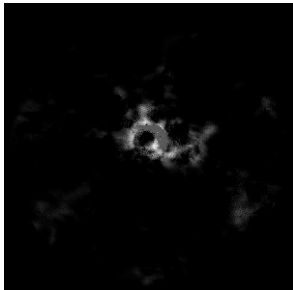
CAD/Spatially resolved signal	C_2^* signal before correction	Fused C_2^* (green) and adjusted background (red) signal	Net C_2^* signal
-3.6			
0.0			
2.9			

Figure B 7 Compilation of spatially resolved C_2^* signal images before and after background correction for selected crank positions of Case B.

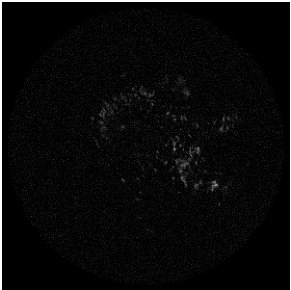
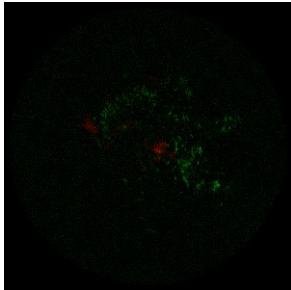
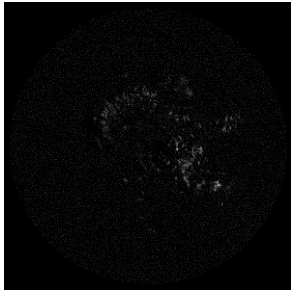
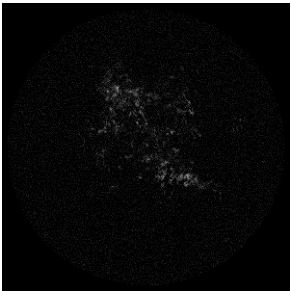
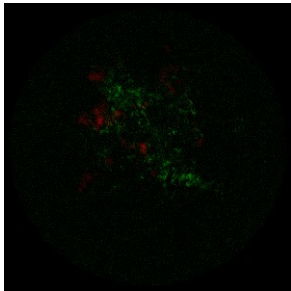
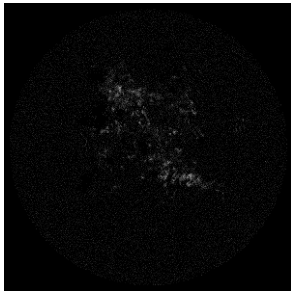
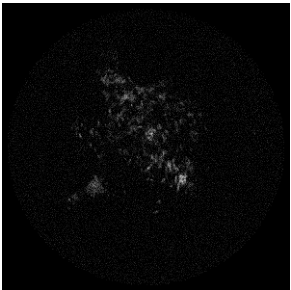
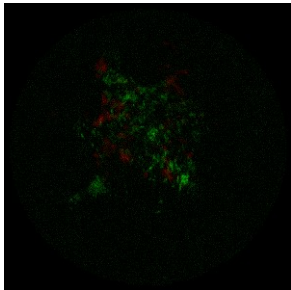
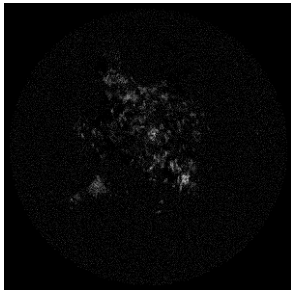
CAD/Spatially resolved signal	CH [*] signal before correction	Fused CH [*] (green) and adjusted background (red) signal	Net CH [*] signal
-4.3			
0.0			
2.9			

Figure B 8 Compilation of spatially resolved CH^{*} signal images before and after background correction for selected crank positions of Case C.

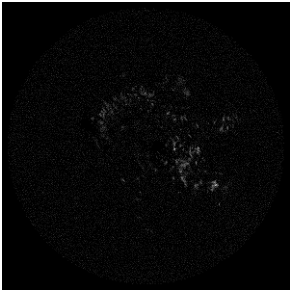
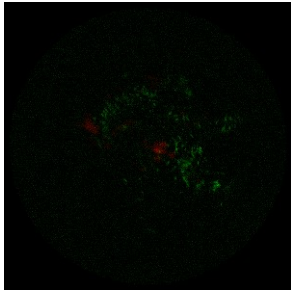
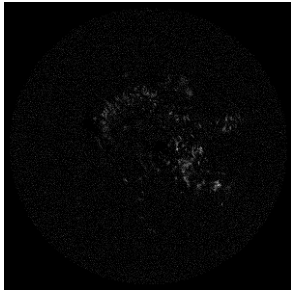
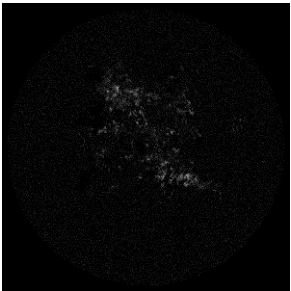
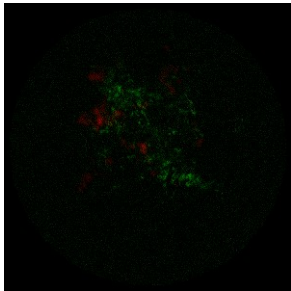
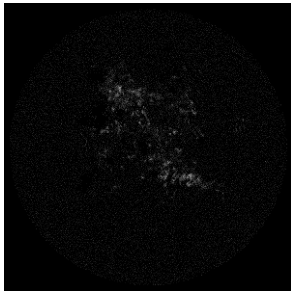
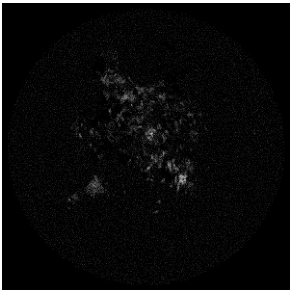
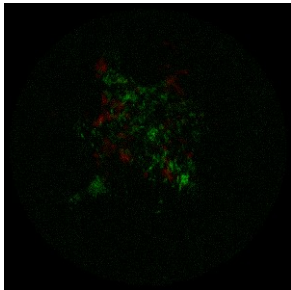
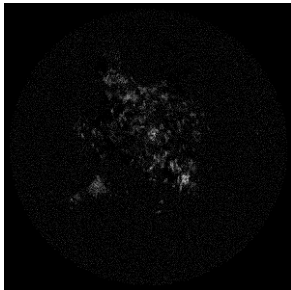
CAD/Spatially resolved signal	C_2^* signal before correction	Fused C_2^* (green) and adjusted background (red) signal	Net C_2^* signal
-4.3			
0.0			
2.9			

Figure B 9 Compilation of spatially resolved C_2^* signal images before and after background correction for selected crank positions of Case C.

Appendix C: Discussion on the outliers of the pLII measurements

The pLII measurements on the Volvo D5 optical engine revealed some interesting outliers which are worth mentioning and discussing in this appendix. More specifically, for the experimental cases featuring a main injection at 15 CAD bTDC, the snapshots shown in Figure C 1 were captured with the laser sheet being placed at the equivalent piston position of 31CAD aTDC and the actual piston position being at 54 CAD aTDC.

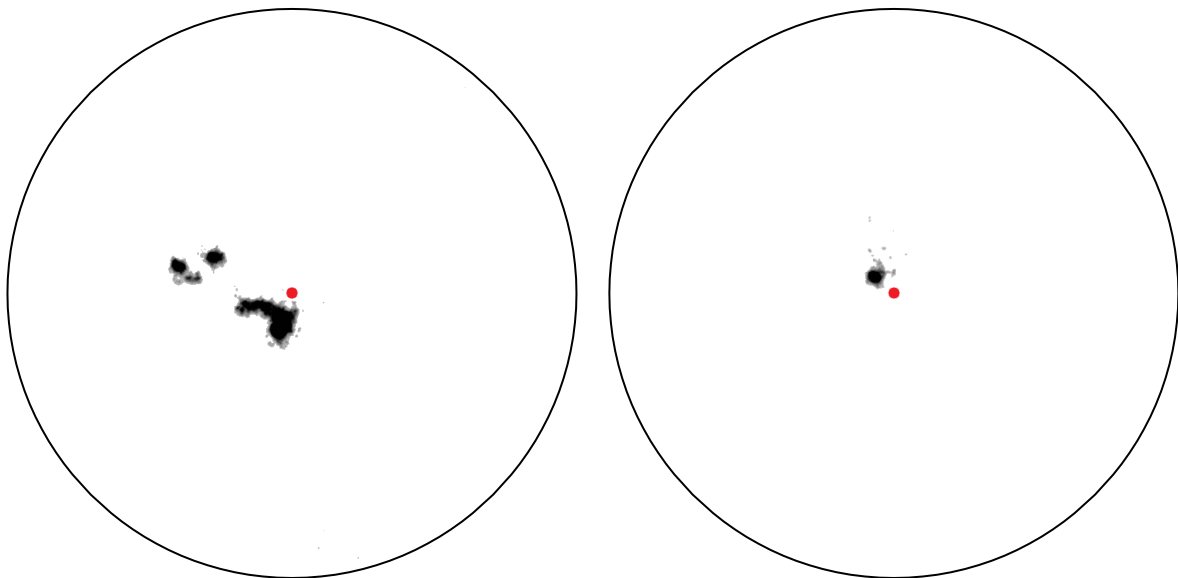


Figure C 1 Outliers of the pLII measurements on the Volvo D5 optical engine, for the cases with main injection at 15 CAD bTDC. The red dot represents the position of the injector tip.

The formations in the figures above exhibit high signal intensity and are located close to the injector tip (red dot) but away from the piston surface. The high signal intensity could indicate that the captured signal stems from reflections of the laser sheet and the proximity of the formations to the injector tip could indicate that they were caused by fuel drip off from the injector.