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Unraveling advanced compression ignition combustion using optical diagnostics

Ron Zegers
Unraveling advanced compression ignition combustion using optical diagnostics

PROEFSCHRIFT

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Ronald Peter Christiaan Zegers

geboren te Heerlen
Dit proefschrift is goedgekeurd door de promotor:

prof.dr. L.P.H. de Goey

Copromotoren:
dr.ir. C.C.M. Luijten
en
dr. N.J. Dam

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Summary

Despite the expected upsurge of hybrid and electric cars in the coming decades, internal combustion will remain the main power supply for (long-distance) transport. Buses, trucks, ships and airplanes will still rely on combustion engines. Nevertheless, emission legislation is becoming more stringent and the oil price continues to rise. Consequently, there still exists a serious interest in new developments that may improve combustion efficiency and fuel flexibility, and reduce emissions; both fundamental and applied research in this field is thriving.

Recently, a lot of research has focused on advanced compression ignition combustion strategies, like premixed charge compression ignition (PCCI). These strategies aim at combining the advantages of gasoline and diesel combustion within the same engine. As a result, engines will be more efficient and have extremely low exhaust emissions enabling to meet emission legislations without after treatment.

To gain understanding of the physical processes occurring in the ‘black box’ of an engine, engines can be made optically accessible, thus allowing a look inside the engine during the combustion process. In these so-called "optical engines" various optical diagnostics can be used to visualize in-cylinder processes, and to supply data on e.g. flow fields and temperature. As a next step, empirically measured data can be used to optimize numerical calculations, which, in turn, hopefully will improve to such extent that they can be used to replace testing of real manufactured prototypes.

One of the main topics in the area of internal combustion research is to identify and explain phenomena around the onset of ignition, including the injection of fuel itself. In the short time period between the start of fuel injection and ignition (typically a few milliseconds), all events of interest take place. Large cycle-to-cycle variations in real engines require fast diagnostic techniques to resolve these short relevant time spans in single combustion cycles. Therefore, fast optical diagnostic techniques are of major importance.

In this thesis, novel optical diagnostics and improved combustion strategies for compression ignition internal combustion engines are presented. Each chapter focuses on a specific optical measurement technique, which can be used to characterize the different aspects of compression ignition combustion in detail.

Velocities

To investigate mixing of fuel and air, the in-cylinder gas-flow velocities were measured using particle image velocimetry (PIV) in chapter 3 and 4. From these velocities, the turbulence intensity was derived, which gives an indication of the degree of mixing of the charge. These measurements have been performed in a 2D plane parallel to and a few mm below the cylinder head. The 2D measurements indicate that the turbulence
is homogeneous and that the swirl center shifts towards the center of the combustion chamber during the compression stroke due to squish motion.

Increasing the recording rate by using a high-speed laser in combination with a high-speed camera enables to investigate the flow evolution in a single cycle, and to record more data before window fouling occurs. To achieve enough contrast between the particles and the environment, the relatively low laser power of the high-speed laser system needs to be compensated by using hollow microspheres as tracers which have a larger size compared to the more generally used oil droplets. The spatial resolution of the presented in-cylinder velocity fields in chapter 4 is shown to be lower for the high-speed measurements than for the low-speed measurements presented in chapter 3. High-speed PIV is therefore not a substitute for high-resolution low-speed PIV, but an additional measurement method to track fast changes. The high-speed PIV results during and after injection results show a sudden change of air motion at the start of injection as a result of air entrainment at the core of the spray. Furthermore, as expected, spray injection causes a considerable increase in the cycle-to-cycle fluctuations of the flow pattern, the more so for longer injection durations. In the case of multiple injections in a single engine cycle the air-entrainment during the first injection is consistent, and fluctuations between consecutive cycles are small. When fuel vapor from previous injections re-enters the investigated plane via impingement on the cylinder wall or the top of the piston, the flow structure changes drastically and loses coherence, compensating the inward motion due to air entrainment by subsequent injections. The spray induced flow can evolve into various structures, which might influence the actual mixing of fuel and air, causing differences in local fuel/air ratios between different injection strategies and therefore change the ignition delay in PCCI combustion.

The high-speed PIV data has been subjected to proper orthogonal decomposition analysis (POD). The results confirm the observed changes in flow structures during and after injection. POD might be a good tool for comparison of experimental PIV data with future CFD results. However, in highly unsteady flow, as observed during injection, a good representation of the instantaneous velocity field using only a few POD modes is not possible and therefore comparison between PIV and CFD data needs careful selection of the amount of modes.

Phase-invariant POD was found not to be an appropriate method to represent highly fluctuating flows during and after fuel injections. Because of the lack of resemblance between flow fields of consecutive crank angles, the use of separate basis functions for each CAD is more appropriate; alternatively one could use separate sets of base functions before and during injection.
Temperature gradients

Temperature gradients occurring before, during and after fuel injection were measured using a 2-color laser induced fluorescence (LIF) technique with toluene as a fuel tracer (chapter 5). The toluene fluorescence signal was recorded simultaneously in two disjunct wavelength bands by a two-camera setup. After calibration, the LIF signal ratio is a proxy for the local temperature. Good agreement has been found between our new calibration curves and previously presented results by other researchers. A detailed measurement procedure is presented to minimize measurement inaccuracies and to improve precision. N-heptane was used as the base fuel and 10% of toluene was added as tracer. The toluene LIF method is capable of measuring temperatures up to 700 K; above that the signal becomes too weak. The precision of the spray temperature measurements is 4% of the temperature and the spatial resolution 1.3 mm. The fluorescence signal was also used as a fuel tracer to investigate the fuel distribution in the optical engine. The technique was found to be very sensitive to disturbances by fluorescence of the base fuel in the wavelength range of interest. However after calibration, comparison of the temperature gradients inside the spray with Large-Eddy Simulation (LES) shows similar results.

Combustion visualization

Controlling ignition delay is the key to successfully enable partially premixed combustion in diesel engines. Chapter 6 presents experimental results of partially premixed combustion in an optically accessible engine, using primary reference fuels in combination with artificial exhaust gas recirculation. By varying the fuel composition and oxygen concentration, the ignition delay was adjusted.

OH chemiluminescence is a useful tool to measure flame lift-off of burning fuel sprays. A similar approach has been presented using a high-speed spectral measurement setup to measure the position of the flame after injection when running in partially premixed mode. In general, increased ignition delay results in a longer lift-off length of quasi-steady flames. When combustion starts after fuel injection is completed, the longer ignition delay results in flame fronts closer to the injector due to "reflection" of the fuel against the piston wall.

The mixing of fuel and air during the ignition delay period defines the local equivalence ratio. To investigate the influence of the ignition delay on the gas volume involved in combustion and the corresponding local equivalence ratio, the position of the flame is determined using high-speed visualization of OH-chemiluminescence. This enables a cycle resolved analysis of the location of OH formation, i.e. the flame position. A clear correlation was observed between ignition delay and flame location, proving that a longer ignition delay increases mixing. Emission measurements using fast-response analyzers of CO, HC and NOx confirmed the decrease in local equivalence ratio as a function of increasing ignition delay.
Furthermore, multiple injection strategies were investigated, applying pilot as well as post injections, in combination with a main injection at constant load. From these results it is concluded that both pilot and post injections result in an increase of unburned hydrocarbon and CO emission and a slight decrease of nitric oxide emissions.
“In time when dinosaurs walked the earth
When the land was swamp and caves where home
In an age when prize possession was fire
To search for landscapes men would roam”

Iron Maiden - Quest for fire

Introduction
Controlled fire has been in use since 400,000 years [10]. However, controlling combustion is still a challenge. The dependence of human society on combustion applications is tremendous: think of central heating boilers, electricity production and turbines for airplanes. Still, the knowledge of many combustion phenomena is empirical rather than fundamental.

Combustion research and the forthcoming knowledge got a boost since 1938 with the first edition of "Combustion, Flames and Explosions of Gases" by Lewis and van Elbe [55]. Another hallmark in the understanding of pollutant formation is the discovery of nitric oxide (NO) formation via the thermal NO mechanism by Zel’dovich (1939). To control combustion within engines is even more complex and is in development since the first commercial single-cylinder atmospheric two-stroke internal combustion engine was demonstrated by Lenoir in 1859. In the 150 years since Lenoir’s invention, engine efficiency increased tremendously, from about 4% in 1860, via the introduction of the 4 stroke otto (1876) and diesel (1892) engines in with an efficiency of 25%, to a record of almost 60% in 2011 for heavy-duty diesel engines, although only at specific operating points and fuels [64].

After the invention of the combustion engine by Otto and Diesel the principle has not changed, although the main components evolved into sophisticated and expensive equipment. Diesel injectors are currently capable of injecting fuel up to 2500 bar and turbo’s can tremendously boost inlet pressures. Combined with improved knowledge on lubrication and materials, internal combustion engines have become efficient power suppliers.

Internal combustion engines are currently the major source of power for vehicles and will remain so in the near future. While hybrid and electric cars will be increasingly common in the coming decades, combustion engines (including turbine engines) will still be the main power supply for (long-distance) transport purposes by means of buses, trucks, ships and airplanes. Simultaneously emission legislation is becoming more and more stringent every few years and oil prices will keep rising. Therefore, there is a serious need for highly efficient and clean combustion engines.

Fundamental research on heavy duty engines increased tremendously in 1992 when the first EURO emission legislation came into place by forcing manufacturers to restrict their engine-out emissions. The amount of particulate matter decreased 97% since then to 0.01 grams per kWh in the EURO VI norm in 2013 and NO\textsubscript{x} is decreased by 95% to 0.4 gram per kWh. Consequently, combustion research is still a field of big importance and recent developments in combustion engines show a new perspective on the use of engines in terms of green and clean propulsion.

Consequently diesel engine powered vehicles nowadays require a combination of operating strategies. On one hand the vehicles need advanced combustion systems to reduce engine-out emissions. On the other hand, reliable and cost-effective after-
treatment systems are needed to clean the remaining emissions. For best performance the engine and the after-treatment system should be combined in a proper way.

The current state of technology enables manufacturers of road vehicles to effectively optimize the tradeoff between fuel efficiency and the costs of after-treatment systems. Advanced combustion systems using exhaust gas recirculation (EGR), increased injection pressure, improved piston-bowl geometries and improved in-cylinder flows have resulted in a substantial reduction of engine-out emissions [17]. Nevertheless, exhaust gas emissions legislation can only be met by applying relatively expensive after-treatment systems. DeNOx techniques like selective catalyst reduction (SCR) and NO\textsubscript{x} adsorbers are combined with Diesel oxidation catalyst (DOC) and Diesel particulate filters (DPF). These after-treatment systems are well developed these days, but the DeNOx techniques still need expensive rare earth metals (such as barium, rhodium and platinum) or an additional tank for the reagent urea, which is better known as Adblue.

Ideally, engine-out emissions are close to the most stringent legislation to circumvent the need for exhaust gas after-treatment in as many engine operating points as possible. Therefore, current combustion research focuses on understanding fuel sprays for conventional diesel combustion, and on finding optimal combustion strategies beyond the main two, being the spark ignition (Otto) engine and the compression ignition (Diesel) engine.

Diesel engines suffer from a NO\textsubscript{x} and soot tradeoff: by changing operating conditions to decrease one, the other emission is increased. Solutions to avoid the NO\textsubscript{x} and soot tradeoff are being developed since the end of the 20\textsuperscript{th} century and started by mimicking gasoline combustion in a Diesel engine in a process called homogeneous charge compression ignition (HCCI) [116]. By premixing the highly diluted charge, traditional diffusion flames are avoided. An overview of the current status and more detail on advanced compression ignition combustion strategies is reviewed in chapter 2.

Knowledge gaps

Using advanced compression ignition combustion strategies, the engine cannot be operated easily in every operating point without decrease in efficiency or increase of engine-out emissions. Hydrocarbon and carbon monoxide emissions may increase rapidly because of variations in ignition delay and the occurrence of extremely lean regions. Therefore, knowledge of the connection between operating strategies and mixture formation and air-utilization is needed [17]. To investigate the thermal and concentration stratification occurring when applying different fuel-injection strategies, there is a need for measurement methods to measure these distributions with sufficient
1.1 Thesis outline

In this thesis combustion related phenomena are investigated by literally looking into the engine through a few glass windows. Using optical diagnostic techniques, phenomena like air entrainment and temperature stratification after injection can be visualized and boundary conditions measured. In this thesis the in-cylinder velocities, temperatures, combustion characteristics and emissions are measured and visualized using three different diagnostic techniques which will be covered in the following chapters.

1.1 Thesis outline

In chapter 2 an introduction is given to recent developments in advanced compression ignition engines, including premixed compression ignition combustion (PCCI). The main factor to influence combustion behavior of advanced compression ignition strategies is the amount of mixing of fuel and air prior to combustion, which leads to both concentration and temperature stratification.

To influence the combustion behavior, local equivalence ratios and temperatures need to be adjusted, so detailed knowledge on the mixing process is very important. Therefore the flow field development during the compression stroke is measured in an optically accessible engine. For these measurements slow-speed, phase-locked particle image velocimetry (PIV) is used and flow analysis is conducted, as described in chapter 3.

During compression, fuel is injected and the influence of multiple injections on the in-cylinder flow behavior is investigated by using time-resolved PIV (TR-PIV) in chapter 4. Hereby the increasing velocities and corresponding fluctuations due to fuel injection are measured. Proper orthogonal decomposition (POD) analysis is used in this chapter to analyse the coherent structures present during and after fuel injection.

During injection, the temperature distribution in fuel sprays and the mixing is measured using planar laser-induced fluorescence (PLIF) of toluene in chapter 5. This will give more insight in heating of the fuel spray due to air entrainment and the typical temperature distribution prior to combustion. The influence of injections on the fuel distribution is also investigated in this chapter.

Finally in chapter 6 we investigate the influence of several (multiple) injection PCCI strategies on combustion. Combustion is also investigated by looking at chemiluminescence and hydroxyl fluorescence combined with exhaust emission measurements. The various injection strategies also include the use of different primary reference fuels. The thesis concludes with a general discussion in chapter 7 where also an outlook is given on optical engine research.
“My intellect’s the power
With diesel power”

The prodigy - Diesel power

Combustion strategies
2.1 Introduction

The two main combustion strategies used in internal combustion engines today are the compression-ignition diesel engine and the spark-ignition Otto engine. Both have specific advantages and drawbacks. Diesel engines are more efficient than gasoline engines, due to the higher expansion ratio and the lack of throttle losses. Combined with a higher energy density of the fuel a diesel engine has a lower fuel consumption. An Otto engine, on the other hand, produces much smaller amounts of particulate matter (PM), because of fuel-air premixing. The emission formation can be explained by means of figure 2.1. In this schematic overview, the spark ignited mixture shows a flame front (dark) which produces hot combustion products which in turn result in high levels of nitric oxides ($\text{NO}_x$) due to the high local temperature. For the compression ignited diesel sprays the black region around the spray depicts the thermal $\text{NO}_x$ production region and the blue region depicts the (fuel-rich) soot formation region.

![Figure 2.1 Schematic representation of combustion strategies in IC engines. The left image represents a gasoline spark ignition engine, where a flame front travels from the ignition point induced by the spark outward. The compression ignition picture depicts the injection of diesel sprays which ignites during the injection. The HCCI image shows a homogeneous charge which ignites due to compression. The PCCI picture is similar to the HCCI picture, but with stratified regions.](image)

A conceptual model of a conventional diesel spray in figure 2.2 [18] explains the production of $\text{NO}_x$ and soot in more detail by displaying the specific areas in a burning diesel spray. From the injector onward, a liquid spray entraining air exists, which forms the rich fuel/air mixture that will start to burn and expand until there is a fully developed diesel spray. In the spray there is a large area with high soot concentration, which is oxidized at the periphery with a thermal NO production zone.

A Premixed Charge Compression Ignition (PCCI) engine combines the Otto and diesel combustion principles. In this regime the engine operates unthrottled and under auto-ignition conditions (diesel) but has a premixed or homogeneous charge (Otto). The load of the engine is controlled through the equivalence ratio ($\phi$). When the charge is near-homogeneous as is the case in homogenous charge compression ignition (HCCI), ignition and combustion occur almost simultaneously throughout the whole volume. Because of the absence of a flame front, local temperatures are low and therefore
2.2. Advanced compression-ignition combustion

The PCCI strategy mentioned in section 2.1 is (just) one of the advanced compression-ignition strategies being developed to produce only minor amounts of NO$_x$ and PM with an efficiency comparable to conventional diesel engines. The charge is diluted by operating lean or using high amounts of EGR and also (partially) pre-mixed.

These approaches towards clean diesel combustion share the same conceptual behavior, and have been called HCCI, PCCI, modulated kinetics (MK), uniform bulky combustion system (UNIBUS)[32], premixed lean diesel combustion (PREDIC)[74], narrow angle direct injection (NADI)[108] to inject fuel earlier using a narrow angle injector to promote premixing and avoid wall-wetting, partially premixed combustion (PPC)[76], Modulated kinetics (MK)[41, 43], reactivity controlled compression-ignition (RCCI)[99] using a dual fuel strategy and “premixed enough” combustion (using gasoline)[39]. All these approaches can be classified under low temperature combustion (LTC) regimes, although some researchers define LTC more specifically as shown in figure 2.3.

All advanced compression ignition strategies can be categorized in three classes based on injection strategy [4], which will be discussed in more detail in the next subsections: port fuel injection (PFI), early direct injection (DI) and late direct injection. However there are also studies investigating multiple injection strategies, which connect the three classes [11, 23].

\[\text{Figure 2.2 Conceptual schematic of conventional diesel combustion after Dec [18].}\]  

\[\text{NO}_x \text{ formation, which typically scales exponentially with } T^{-1}, \text{ can be orders of magnitude lower. The absence of fuel rich zones also avoids the formation of particulate matter (soot).}\]
2.2. Advanced compression-ignition combustion

2.2.1 Port fuel injection

In the case of PFI, the fuel is injected into the intake manifold. When using fuel with a high vapor pressure, like gasoline, propane or ethanol, this is an easy way to achieve a homogeneous mixture resulting in HCCI, which is dominated by chemical kinetics [30].

Drawbacks of HCCI combustion are:

- The combustion of the homogeneous mixture results in a rapid heat release, high cylinder pressure rise rates and peak pressures [31] causing noise and rapid wear of engine hardware [110].
- Unstable combustion in many operating points [110].
- Low combustion efficiency at low load [110].
- High unburned hydrocarbons (HC) and carbon monoxide (CO) emissions [30 110], due to the fuel adhering to the cylinder liner [40].
- Deteriorating indicated mean effective pressure (IMEP) and decreasing power output due to excessively advanced ignition [40 85].
- Difficult or no control of start of combustion (SOC) when the entire amount of fuel is injected at once [30 40].

All these drawbacks have led to a shift in research activities over the last decade towards strategies using direct injection.
2.2.2 Early Direct injection

The definition of early DI is rather arbitrary, but can be loosely defined as an injection near the onset of the second half of the compression stroke (90 crank angle degrees before TDC), ensuring enough premixing time between the end of injection and start of ignition. The PREDIC concept can be classified as early DI. Early injection can, however, result in spray impingement on the cylinder liner due to low gas densities early in the compression stroke [31]. Drawbacks of this wall wetting are high CO and HC emissions and the risk of diluting the oil when liquid fuel is transferred to the sump via the piston rings. In light-duty engines, these problems can be very severe due to their small dimensions and higher rotational speeds compared to heavy-duty engines.

Controlling the auto-ignition is very difficult for early DI. Together with high pressure rise rates, this approach is limited to low loads.

2.2.3 Late Direct injection

Late direct injection is the advanced injection strategy which approaches the conventional diesel injection timing closely. PCI and MK concepts are using this strategy. The MK concept injects the fuel even after the conventional injection timing. In order to achieve a premixed mixture, mixing should be rapid and the ignition delay should be large enough to separate the injection and the combustion. The desired ignition delay can be achieved using several approaches, but EGR is the most used. In this approach the combustion can still be controlled by injection timing.

2.2.4 Multiple injection strategies

Using pilot or early-pilot injections in combination with a close to conventional injection timing around TDC might have the best possibilities in overcoming the drawbacks of the separate injection strategies. By adjusting the timing and amount of fuel in each injection, the combustion behavior can be optimized for every engine operating point. This requires closed loop control of the injection events. UNIBUS is an example of a multiple injection strategy.

2.3 Parameters influencing LTC combustion

In the previous section the four main injection strategies were explained, arranged according to their injection timing. However, there are more parameters influencing low temperature combustion and these will be addressed in this section. Some of these parameters are a result of the injection strategy.
2.3 Parameters influencing LTC combustion

2.3.1 Stratification

If a mixture would be completely homogeneous, without temperature and concentration gradients, the reaction duration would be extremely short. Even in an engine operated with a HCCI strategy, thermal stratification exists, due to cooling at the piston, cylinder wall and cylinder head [22]. HCCI combustion is very sensitive to temperature changes [97], and therefore post-ignition and cycle-to-cycle variations have a large influence.

Lower intake manifold temperature is preferable for HCCI operation, reducing maximum cylinder pressure and pressure rise rate, and delays the rate of heat release peak [47]. By adjusting the thermal stratification in an engine, the potential of extending the high-load operating limit is very large [22, 97]. The thermal width of the charge, to avoid excessive pressure rise rates, depends on the combustion phasing and fueling rate. The thermal stratification reduces the pressure rise rate (PRR) even more when retarding the combustion phasing due to volume expansion [17].

Higher fuel inhomogeneity lengthens combustion, however its influence on combustion duration is less significant compared to the effect of temperature inhomogeneity for a given fuel/air ratio [75]. Appropriate in-cylinder inhomogeneity has the ability to reduce pressure rise rate, maximum pressure and thus knocking intensity [93]. The local equivalence ratio $\phi$, defined as the stoichiometric air to fuel ratio divided by the actual air to fuel ratio, should also be above a certain flammability threshold to avoid very fuel lean regions (overleaning), for example near the injector between the end of injection and start of combustion leading to unburned HC emissions [73]. Implementing and creating partial fuel stratification needs to be controlled accurately, because too much stratification can quickly lead to unacceptable $\text{NO}_x$ emissions [96]. The interaction of the engine flow and the fuel injection can be high in the case of direct injection. Swirl makes turbulence more homogeneous throughout the cylinder and enhances mixing of air and fuel to give a premixed mixture in a very short time [84]. The turbulence intensity depends heavily on the engine speed and the turbulent mixing rate can also be increased using higher injection pressures.

2.3.2 Exhaust gas recycling

Tuning the amount of exhaust gas recycling (EGR) is the most commonly used technique to adjust temperature and ignition timing. The components of EGR are carbon dioxide, water, nitrogen, oxygen, carbon monoxide, particulate matter, unburned hydrocarbons, $\text{NO}_x$ and combustion reaction intermediates. Carbon dioxide and water have relatively high heat capacities. EGR gas increases the inlet temperature when mixed with fresh charge, decreases the oxygen concentration and increases the heat capacity of the mixture, reducing the temperature at the end of compression (thermal effect). Due to the lower temperature and oxygen concentration, the oxidation rate is lowered, resulting in higher soot emissions. Extremely high EGR levels, however, lower
2.3. Parameters influencing LTC combustion

Combustion temperatures below those required to pyrolyse fuel and form precursors of soot [37]. Cooled EGR is more effective in reducing NO\textsubscript{x} compared to uncooled EGR, except at low speed and light load modes [94].

Another possibility is trapping the exhaust gas in the cylinder by opening the inlet valve before top dead center (TDC) using variable valve actuation. When produced NO\textsubscript{x} (a few ppm) is transferred to the incoming charge via EGR or residuals, the auto-ignition can be advanced in HCCI, resulting in knock via rapidly advanced combustion phasing [88].

2.3.3 Fuel composition

Fuel composition has little influence on the reaction rate and reaction duration once reaction has been initiated, however it does define the auto-ignition temperature (and thus timing). Fuel composition mainly affects the low temperature reaction, which in turn affects the main reaction start. The ignition delay (ID) between start of injection (SOI) and start of combustion (SOC) consists of physical ID and chemical ID. Main fuel parameters influencing the physical ID are the density, heating value and latent heat of vaporization. The chemical ID is influenced by the auto ignition and distillation properties [38]. An increase of density, volatilization difficulties and higher latent heat of vaporization (due to decreasing air/fuel blend temperature) all result in a delayed combustion start/ignition. Common ways to classify fuels is according to the ease of auto-ignition via the Cetane number (CN), or via the resistance to auto ignition via the Octane number (ON). A high cetane number represents less resistance to auto-ignition were the straight chain paraffin iso-cetane defines CN=100. A high octane number represents the resistance against auto-ignition were the branched chain paraffin iso-octane defines ON=100. Gasoline has a high octane number and therefore has little or no low temperature reaction and combustion initiation in the order of 950 K. Diesel fuels show significant low temperature reaction and have initiation temperatures in the range of 750 K.

2.3.4 Compression ratio and load

Higher loads in premixed charge strategies increase the very strong tendency to advance the main reaction (due to increased engine temperatures), resulting in unacceptable heat release rates and rates of pressure rise [116][117]. Where a specific strategy might seem favorable at low loads, the results might change drastically at higher loads, especially for homogeneous mixtures.
2.4 Controlling mixing

Controlling the mixing and in-cylinder pressure is crucial to adapt the combustion strategy to the desired load. Therefore the equivalence and temperature distribution need to be adjusted as functions of the operating point. There are several ways to influence mixing:

1) Extend the mixing time by choosing the fuel composition, blends or combinations [59], advancing injection timings or using EGR to dilute and decrease temperatures.
2) Increase mixing by designing the piston bowl and injection nozzle geometry, increasing swirl by designing the inlet manifolds and increasing fuel pressure to increase vaporization.

To investigate if mixing strategies are useful, appropriate diagnostics are needed. Measurements of in-cylinder velocity fields can act as a "guiding quantity" of mixing, whereas measurements of temperature fields can visualize the result of mixing on the thermal stratification. Combined with flame localization, in-cylinder pressure and exhaust emission measurements differences in mixing strategies can be carefully examined.

This study is therefore largely aimed at a thorough investigation of such diagnostics since the in-cylinder measurement of flow and combustion parameters are still far from being well-established.
Flows in engines using PIV

This chapter is an integral copy of the article which has been presented at the European Combustion Meeting, Vienna 2009. Minor changes have been made to improve the quality. It is published as:

3.1 Introduction

Premixed charge compression ignition (PCCI) is one of the most promising combustion strategies for internal combustion engines in the future, since PCCI combustion is able to realize very low soot and nitric oxide emissions. PCCI is a low temperature combustion (LTC) strategy, which combines the efficiency of a diesel and the low particulate emission of an Otto engine. Although premixed, the air-fuel mixture is not completely homogeneous, as in a homogeneous charge compression ignition (HCCI) engine. By premixing the charge, combustion in both HCCI and PCCI is dominated by chemical kinetics [30] instead of fuel/air mixing. The temperature and concentration gradients present in the charge when using the PCCI combustion strategy (charge stratification) slow down combustion because the mixture will not ignite everywhere at once. The lower rate of heat release extends the operating range from low load to medium or even full load. Temperature stratification has the highest influence on the rate of heat release and pressure rise rate and extends the high-load operating limit [97]. To achieve low emissions of NO\textsubscript{x} and particulate matter (PM) the combustion should be decoupled from the injection of the fuel to avoid a spray that burns predominantly in diffusion mode. This decoupling makes the combustion process difficult to control. When allowing a certain degree of stratification in the cylinder, control over the combustion process by injection timing is partly restored.

![Figure 3.1 Engine cycle with pressure peak at top dead center (TDC). PCCI concepts classification by injection timing range. 1=port fuel injection, before bottom dead center (BDC), 2= early DI, 3=conventional diesel combustion, 4=late DI.](image)

The classification of PCCI combustion concepts is given in figure 3.1. In this figure the pressure inside the engine is plotted as a function of time and different classes of PCCI injection timing are shown. Port fuel injection is only applied for gasoline fueled PCCI engines and takes place before bottom dead center (BDC). For early direct injection (DI) PCCI combustion, collision of injected fuel spray against the cylinder liner, so-called wall-wetting, is one of the main hurdles to overcome. A possible solution to achieve a good evaporation is to raise inlet temperatures by recycling uncooled exhaust gas (EGR) [9]. However recycling cooled exhaust gas is more effective in reducing NO\textsubscript{x} compared to uncooled EGR, except for low speed and light load modes.
Late DI demands a long ignition delay to separate the injection and combustion events. EGR is the most widely used approach to achieve long ignition delays. The high heat capacity of the exhaust gas lowers the temperature at the end of compression and therefore also the combustion temperature. To achieve a premixed charge, rapid mixing is needed in the time given by the ignition delay. However, this strategy comes at a fuel penalty caused by the relatively late combustion timing.

The aim of this project is to investigate the influence of charge stratification on PCCI combustion to reduce the heat release rate and extend the operating range. This will be investigated by measuring in-cylinder velocities, concentrations and temperatures. Since the swirl inside an engine influences the turbulence resulting in a more homogenous distribution, the first step is to investigate the flow fields present in the engine. From these flow fields, the turbulence level and amount of swirl inside the engine can be calculated. The swirl develops due to non-perpendicular impingements of the intake jets at the opposite cylinder wall [105] and from the mutual interaction of these jets [15].

### 3.2 Materials and Methods

The test setup consists of a single cylinder optically accessible heavy duty engine, based on a DAF MX engine, driven by an electrical motor. A front-view of the setup is shown in figure 3.2. The piston is elongated and the upper part of the liner and piston bottom are both made of sapphire. Via a mirror, positioned under 45 degrees, optical access to the combustion chamber is obtained. The hydraulic cylinder can be lowered, allowing easy access to the combustion chamber. The engine specifications in Table 3.1 show fixed valve timings and variable compression ratios (CR). The compression ratio can be changed quickly between measurements by positioning the cylinder head on a different height, thereby changing the volume at TDC.

For PIV measurements a 10 Hz Nd:YAG Continuum Surelite laser is used with a pulse energy of 140 mJ. The camera is a Kodak Megaplus ES 1.0 CCD with 1008 x 1018 pixels. The particles used for seeding are produced using laskin nozzles and have a diameter of 1 µm. The oil type is a silicon oil with a viscosity of 100 cSt. Pre- and post-processing of the PIV images is done using the commercially available software program PIVview (Pivtec). The velocity analysis is done using the commercial software program Tecplot.

**Pre-processing** In addition to the background subtraction, a dynamic threshold is applied with a lower threshold of 5% and an upper threshold of 95% together with a low-pass filter (anti-alias) with a filter size of 0.5 pixel.
3.2. Materials and Methods

Figure 3.2 Front-view of the optical accessible one cylinder engine.

Table 3.1 Engine specifications, top dead center (TDC) is defined as 360 CAD.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>130 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>156 mm</td>
</tr>
<tr>
<td>Connecting rod</td>
<td>270 mm</td>
</tr>
<tr>
<td>Displacement volume</td>
<td>2.07 liter</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9.5 - 14</td>
</tr>
<tr>
<td>IVO</td>
<td>715 CAD</td>
</tr>
<tr>
<td>IVC</td>
<td>190 CAD</td>
</tr>
<tr>
<td>EVO</td>
<td>500 CAD</td>
</tr>
<tr>
<td>EVC</td>
<td>10 CAD</td>
</tr>
<tr>
<td>Piston bowl/crown</td>
<td>Flat bathtub</td>
</tr>
<tr>
<td>Piston bowl diameter</td>
<td>90 mm</td>
</tr>
<tr>
<td>Piston bowl depth</td>
<td>20 mm</td>
</tr>
</tbody>
</table>

IVO = inlet valve opening, IVC = inlet valve closing,
EVO = exhaust valve opening, EVC = exhaust valve closing.

Interrogation parameters  Multi-grid interrogation is used starting with 128 times 128 pixels and ending at 32 times 32 pixels, combined with a window overlap of 50%.
For the correlation peak detection a least squares Gaussian sub-pixel peak fit is used.

**Post-processing values** A normalized median filter is used, this validation filter has a threshold value of 2.2. The maximum displacement is set to three times the average piston speed divided by the time between two images. Also a signal-to-noise ratio filter is applied, with a minimum value of 1.5 [87, 100]. After processing, the amount of rejected vectors is around 3% for measurements in the horizontal plane. The eliminated spurious vectors are replaced by vectors acquired by bi-linear interpolation. When neighboring vectors are also marked as outliers, a Gaussian-weighted interpolation scheme is used [83].

### 3.3 Results

We will present the low-speed PIV measurements executed using the 10 Hz laser. In figure 3.3 the instantaneous, ensemble average and Reynolds decomposed velocity field can be seen, measured at 15 CAD before TDC with an engine speed of 1000 rpm. The ensemble averaged velocities in x and y direction are calculated from N=65 image pairs using:

\[
U_{x,y} = \frac{1}{N} \sum_{i=1}^{65} U_{x,y}(i),
\]  

(3.1)

The timing between two consecutive images is 20 µs, to achieve a large enough pixel displacement, but a low number of “out of plane” vectors.

#### 3.3.1 Reynolds decomposed velocity fields

The Reynolds decomposed velocity, \(u(\theta, i)\), is the difference between the instantaneous velocity and the ensemble averaged velocity given by:

\[
U(\theta, i) = U(\theta) + u(\theta, i),
\]  

(3.2)

where \(U(\theta, i)\) is the instantaneous velocity at crank angle \(\theta\) and cycle \(i\). The ensemble mean velocity over several cycles is given by \(U(\theta)\).

#### 3.3.2 Turbulence intensity

The turbulence intensity is defined as the root-mean-square value of the difference between the length of the instantaneous velocities, \(U\), and the length of the ensemble
3.3. Results

![Velocity field images](image)

Figure 3.3 Instantaneous (top), ensemble averaged (middle) and Reynolds decomposed (bottom) velocity field, measured at 15 CAD BTDC, 1000 rpm, CR=9.6 and 5 mm below cylinder head.

Averaged velocities, $\overline{U}$.

$$u' = \sqrt{\frac{1}{N} \sum_{i=1}^{N} \left( \left( \sqrt{U_x(i)^2 + U_y(i)^2} \right) - \left( \sqrt{\overline{U_x}^2 + \overline{U_y}^2} \right) \right)^2} \quad (3.3)$$

The number of image pairs per crank angle investigated here is $N=65$. In figure 3.4 the turbulence intensity levels are plotted for two compression ratios, together with the velocity field. The turbulence intensity is homogeneous throughout the field, except for the area around the swirl center. The median of the turbulence intensity throughout a field as a function of crank angle and compression ratio can be seen in figure 3.5. There is no difference in turbulence intensity for different compression ratios, whereas the intensity decreases toward TDC for all compression ratios. This decrease in turbulence intensity can also be seen in figure 3.6. In this figure the mean velocities and turbulence intensities for three different engine speeds are plotted against crank angle. A lower piston speed around TDC is the reason for lower mean velocities and thus the decay in turbulence intensity towards TDC.
3.3. Results

Flows in engines using PIV

Figure 3.4 Turbulence intensity [m/s] at 20 CAD BTDC, 1000 rpm and 5 mm below cylinder head.

Figure 3.5 Median of turbulence intensity (left) and turbulent kinetic energy (right) at different compression ratios at 1000 rpm, 5 mm below cylinder head.

3.3.3 Turbulent kinetic energy

The turbulent kinetic energy (TKE) [m$^2$/s$^2$] is given by equation 3.4:

$$TKE = \frac{1}{N} \sum_{i=1}^{N} \frac{1}{2} \left( (U_x(i) - \bar{U}_x)^2 + (U_y(i) - \bar{U}_y)^2 \right)$$  \hspace{1cm} (3.4)

For a compression ratio of 9.6, at 1000 rpm and 20 CAD BTDC, the TKE is shown in figure 3.7 superimposed on the corresponding ensemble averaged velocity field. At higher compression ratios, the maximum value of the TKE is higher, where the median of the TKE is almost the same for all compression ratios, see figure 3.5. This denotes a less homogeneous distribution of the turbulence throughout the volume for higher CR values.
3.3. Results

![Figure 3.6](image1.png)

**Figure 3.6** Median of the mean velocity and turbulence intensity [m/s] as function of crank angle for three different engine speeds at a compression ratio of 9.6.

![Figure 3.7](image2.png)

(a) CR=9.6
(b) CR=13.9

**Figure 3.7** Turbulent kinetic energy [m$^2$/s$^2$] at 20 CAD BTDC, 1000 rpm and 5 mm below cylinder head.

### 3.3.4 Vorticity

The production of turbulent kinetic energy is driven by vorticity and strain rate [56]. The out of plane component of the vorticity is given by the following equation:

$$\omega_z = \frac{\partial U_y}{\partial x} - \frac{\partial U_x}{\partial y}.$$  \hspace{1cm} (3.5)

In figure 3.8 the vorticity is plotted for four different compression ratios as a function of crank angle. In this figure a rise in vorticity can be seen as a function of crank angle, together with a slightly higher vorticity at higher compression ratios.
3.3. Results

Flows in engines using PIV

Figure 3.8 Median of the ensemble averaged vorticity at different compression ratios at 1000 rpm, 5 mm below cylinder head.

3.3.5 Swirl center position

The position of the swirl center in the cylinder for different engine speeds and crank angles is given in figure 3.9. The swirl center shifts towards the center of the cylinder during the cycle. This can probably be attributed to the squish present as a result of the upward motion of the piston. When raising the compression ratio, the distance between the piston and cylinder head becomes smaller and will probably enhance the squish, but this results only in a minor influence on the position of the swirl center.

Figure 3.9 Swirl center position from 300 CAD (60 CAD bTDC) to 385 CAD (25 CAD aTDC), CR=9.6.
3.4 Conclusions and discussion

In-cylinder velocity measurements in an optical accessible engine have been performed using particle image velocimetry. We focussed on the in-cylinder velocities, aimed at a future detailed study on the effect of charge stratification in PCCI combustion. The position of the swirl center is investigated as a function of crank angle and shows a shift towards the middle of the combustion chamber, probably due to squish. Turbulence intensities are calculated and show a homogenous distribution throughout the field, except for the region around the swirl center (compression ratio of 9.6). For higher compression ratios, both the turbulence intensity and turbulent kinetic energy distribution become less homogeneous. Using particle image velocimetry it is possible to get quantitative information of the velocity field inside the cylinder. Importantly, the optical engine with a flat piston bottom does not represent a production type. Due to this, the in-cylinder flow will be different from the production type diesel engine, which has a toroidal shaped piston bottom.
“Hold that second
Hold the time
Hold that picture in your mind
Hold the smoke
Hold the fire
Hold all that that you desire”

The Black Keys - Keep me

High speed PIV
during fuel injection

This chapter is an integral copy of the article that is published in Experiments in Fluids. Minor changes have been made to improve the quality. It is published as:

4.1 Introduction

Due to stringent emission legislation, there is widespread interest in alternative combustion strategies for internal combustion engines. One of these new strategies is the combination of the Otto and Diesel principles, the so-called premixed charge compression ignition (PCCI) concept. This strategy involves a certain amount of premixing of fuel and air inside the engine prior to ignition. The exact amount of pre-mixing is crucial for combustion behavior and emission formation. Premixing occurs in the ignition delay (ID) period, which is defined as the time between start of injection (SOI) and the start of combustion (SOC). When a diesel engine is operated in PCCI mode, the ignition delay is (much) larger as compared to conventional diesel combustion, and larger than ten crank angle degrees (CAD) as shown by Leermakers et al. [54]. Currently, injection pressures in heavy-duty diesel engines of up to 2500 bar, combined with a long ignition delay, improve the mixing of fuel and air prior to combustion. The influence of the global air flow in the cylinder on the mixing process is of particular interest. Due to the high momentum of the spray, direct injection of a diesel fuel is expected to have a stronger influence on the mixing process than the swirl motion that may or may not be present before injection.

A novel method to investigate pre-mixing during the ignition delay period in PCCI engines is to visualize in-cylinder flows using high-speed particle image velocimetry (HS-PIV). In some cases the recording speed is fast enough to resolve a large part of the time-scales present and can therefore be called time-resolved PIV (TR-PIV). During compression Cosadia et al. [16] showed time-resolved results with a recording speed of 1 kHz. PIV has already proven its applicability for visualization of in-cylinder flows in spark- and compression-ignition engines.

Combustion occurring in spark ignition (SI) engines is influenced by the in-cylinder flow, mostly the tumble motion, and is also very prone to cycle-to-cycle variations. Therefore, PIV measurements in spark ignition engines have focused on understanding the influence of cycle-to-cycle variations on combustion behavior and the occurrence of misfires. Recently Müller et al. [71], Peterson and Sick [82], and Krishna and Mallikarjuna [50] showed the applicability of (HS-)PIV near the spark plug region in SI engines. PIV measurements in a light-duty diesel engine with a typical production piston with re-entrant shape and valve cutout geometries were presented by Petersen and Miles [81] and Yu et al. [113]. The optical distortion observed through the curved surfaces of the fused silica piston bowl was successfully corrected for. Qualitative comparison of PIV with large eddy simulation (LES) data has been presented by et al. [113]. Fuel injection phenomena in a gasoline direct injection (GDI) engine was investigated using PIV by Ekenberg et al. [27] by injecting compressed air through the gasoline injector.

Diesel sprays in compression-ignition engines are influenced by the main swirl, which deflects the sprays away from the central axis. This deflection was investigated experimentally and numerically using computational fluid dynamics (CFD) models of spray
4.2 Methods

4.2.1 Experimental setup

The experimental setup consists of an optically accessible engine and a high-speed PIV setup, described in more detail in the next two subsections.

Optically accessible engine

The test setup consists of a one-cylinder optically accessible heavy-duty Diesel engine, based on a Ricardo Proteus block and equipped with a DAF MX cylinder head. The
engine is driven by an electrical motor. A cross-section of the engine setup is shown in figure 4.1. The piston is elongated, and the upper part of the liner and the piston bottom are both made of sapphire. Via an oval aluminium coated mirror (Molenaar Optics), positioned under 45 degrees, optical access to the combustion chamber is obtained. The hydraulic cylinder can be lowered, allowing easy access to the combustion chamber for cleaning. For details of the engine we refer to the work of Doosje [21]. The engine compression ratio is 13.9, all other specifications are summarized in Table 3.1.

The engine is equipped with a common rail injector of Delphi Diesel Systems, which can be used up to 2500 bar injection pressure with up to 5 injections per cycle. Details are presented in Table 4.1. To avoid massive light scattering by the injected liquid fuel, n-heptane is chosen instead of diesel. Because of faster vaporization, n-heptane fuel sprays show only a relatively short liquid core.

Table 4.1 Injector specifications.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holes</td>
<td>7</td>
</tr>
<tr>
<td>Hole size</td>
<td>195 µm</td>
</tr>
<tr>
<td>Flow (nominal)</td>
<td>1.7 l/min</td>
</tr>
<tr>
<td>Spray cone angle $\phi$</td>
<td>143 degrees</td>
</tr>
</tbody>
</table>

Nominal flow is measured by Delphi Diesel systems at 100 bar.

High-speed PIV setup

The HS-PIV setup consists of an EdgeWave IS8II-DE double-cavity high-speed Nd:YAG laser and a Vision Research Phantom V7.1 high-speed CMOS camera, triggered by a
4.2. Methods

High-speed PIV during fuel injection

Laser sheet

Cylinder head

TDC

Φ = 143°

Figure 4.2 Schematic representation of the time-resolved PIV setup during injection via the injector with a spray angle of 143 degrees. The dotted lines represent the position of the upper rim of the piston at specific CA during the compression stroke. The black solid line represents top dead center (360 CAD, TDC).

LaVision high-speed controller and DaVis software. The laser has a repetition rate up to 10 kHz and a pulse energy of 7 mJ at 3 kHz. The high-speed camera has a frame rate of 4.8 kHz at full resolution of 800 x 600 pixels. For HS-PIV experiments the frame straddling technique is used, where one cavity is fired at the end of a certain frame and the other cavity at the beginning of the next frame. This technique reduces the maximum effective frame rate for PIV to 2.4 kHz at full resolution. The laser sheet has a thickness of approximately 1 mm in the engine and the delay between the two laser pulses is 50 µs.

When performing HS-PIV measurements of relatively large areas, the high-speed laser intensity is too low to achieve enough scattering using regular seeding particles, such as silicon oil droplets of 1 µm. Therefore relatively large dry-expanded hollow polymer microspheres (Expancel 920 DE40 D30) are used as seeding material. The diameter of the microspheres is \( d_p = 40 \, \mu m \) and their density is approximately \( \rho_p = 30 \, kg/m^3 \). The microspheres are seeded using a home-built cyclone which is operated by applying a pressure difference between the intake manifold and the cyclone inlet of 0.2 bar. They are easily seeded into the intake air, and do not have any negative influence on the engine behavior, but cleaning of the intake manifold, piston rings, cylinder head an piston is necessary after a few measurements. A similar kind of approach using microspheres can be found in the papers by Towers and Towers [104], Ekenberg et al. [27] and Nordgren et al.[77].

The density ratio between the microspheres and the air at TDC with an intake pressure of 2 bar is \( s = 1.07 \), reducing the difference between the air and particle velocity. The flow fluctuations which can be followed by the particle are determined by using the approach published by Melling [67].
4.2. Methods

The relaxation time (response) of the particles at TDC (800 K) is calculated using 
\[ \tau_p = \frac{d_p^2 \rho_p}{18 \mu} \] 
with a dynamic viscosity of \( \mu = 187 \cdot 10^{-6} \) kg/m s. This results in 
\[ \tau_p = 1.4 \cdot 10^{-5} \text{ s}. \] 
An approximation of the characteristic resolved flow scale near the spray is given by 
\[ \tau_f = \frac{\Delta}{V_{\text{spray}}} \] 
where the spatial resolution \( \Delta = 10^{-2} \) m and a characteristic spray velocity 
\( V_{\text{spray}} = 100 \) m/s resulting in \( \tau_f = 10^{-4} \) s. The resulting Stokes number is 
\[ S_k = \frac{\tau_p}{\tau_f} = 0.14, \] 
providing acceptable flow tracing accuracy [67].

All measurements were carried out at 600 rpm with a resolution of one CAD, resulting in a measurement frequency of 3.6 kHz and a camera frequency of 7.2 kHz. During the compression this recording speed is appropriate to resolve flow time scales of interest. However, during fuel injection, time scales are much shorter and therefore only the large scale fluctuations are captured. This recording frequency limits the camera resolution to \( 512 \times 512 \) pixels. When recording the whole field of view, the resolution is 0.16 mm per pixel. The pulse separation is determined to be 50 \( \mu \)s to get an appropriate displacement for in-cylinder flows of around 10 m/s. On average the particle image size is 2 pixels, corresponding to 0.32 mm. The lowest displacement visible (sub-pixel accuracy) of 0.1 pixels corresponds to 0.32 m/s given the 50 \( \mu \)s pulse separation.

The recorded plane is positioned only two millimeter below the cylinder head to enable the lasershare to enter the combustion chamber during the whole compression stroke. A detailed representation of the piston position at several crank angle positions is shown in figure 4.2. Obviously, the desire to record data during the passage of the piston through TDC limits the probe volume to the uppermost part of the combustion chamber. The probe volume is also limited by the size of the optical part of the piston bowl resulting in a field of view with a diameter of 8 cm. In particular no direct information is obtained from the flow fields within the piston bowl, or velocities in the vertical plane. Using a completely transparent bowl, for instance as used by Cosadia et al. [16], Chartier et al. [11], and Petersen and Miles [81], might further improve the knowledge on the influence of spray injections on the flow fields and therefore the mixing.

**Pre-processing parameters**

Due to the large scattering efficiency of the microspheres, only minor pre-processing is needed. We normalize the particle intensity by applying a min-max filter with a scale length of 6 pixels. This increases the contrast between the particles and the background. A mask is applied to exclude the edges of the field of view (FOV). After pre-processing, the valve seats can still be seen in the image due to overexposure. The scattering of the fuel spray is reduced by the pre-processing, thereby lowering the error in cross-correlating.
4.2. Methods

Interrogation parameters

The chosen interrogation parameters represent a trade-off between relatively low velocities prior to injection and the much higher velocities during and after injection. During fuel injection, velocities increase tremendously, and combined with a 50 µs inter frame time, a relatively coarse grid is needed. The multipass/multigrid scheme used is explained by Raffel et al. [86]. We use sizes of 128 pixels and 64 pixels, with 75% overlap to increase the spatial-sampling frequency, resulting in an overall spatial resolution of 10.4 mm with a spatial grid spacing of 2.6 mm between each vector. This spatial resolution is two times lower than in the time-resolved PIV study by Cosa-dia et al. [16]. Using the post-processing values as defined in the next paragraph, maximally 28% of the 850 calculated vectors were rejected when injecting multiple times. This can be attributed to high out-of-plane velocities. Using a grid size of 32 pixels during interrogation results in a slightly higher amount of rejected vectors than with a 64 pixels grid size. The combination of a bad laser profile and multiple injections resulted in maximally 35% outliers. To achieve the highest accuracy, spatial resolution has been sacrificed.

Post-processing parameters

Post-processing is applied after each interrogation pass of the multipass/multigrid scheme and after the last pass. A correlation peak height ratio of 1.5 is used to reject spurious vectors and the vector field is median-filtered by removing vectors which are two times larger than the rms value of their neighbors and only re-inserted when the value was smaller than 3 times the rms value. The resulting velocity field is smoothed using a $3 \times 3$ filter.

4.2.2 Experimental datasets

The experiments are split in two datasets: without (1) and with (2) fuel injection.

Dataset 1

The first dataset is used to investigate the statistics of cycle-to-cycle fluctuations without fuel injection. This dataset contains three measurement series of 10 CAD duration (320 to 329 CAD) and a total of 310 cycles. The intake pressure was kept at 100 kPa.
4.2. Methods

**Dataset 2**

The second dataset comprises 11 cases of 140 CAD duration (260 to 339 CAD) and 9 cycles each, with injection of n-heptane in pure nitrogen to avoid combustion. The intake pressure was raised to 200 kPa to reduce the liquid spray volume and therefore overexposure of the camera.

An overview of all cases with varying injection pressures, timings and durations is given in table 4.2. From this dataset, 8 cases are combined to calculate phase-averaged velocity fields of 72 cycles up to the start of injection (260 to 330 CAD). The n-heptane injections are split into a maximum of 5 injections during a cycle. Various injection pressures, timings and durations are investigated. The injection durations range from 1 ms up to 6 ms (3×2 ms) to represent the full engine load range. As the injection durations are kept constant, the injection amount increases with increasing injection pressure. Varying common rail pressure as a result of previous injections is not taken into account.

The low number of cycles prohibits the calculation of a statistically converged phase averaged velocity field, even though flow analysis can still be performed.

### Table 4.2

<table>
<thead>
<tr>
<th>Case</th>
<th>Pressure (bar)</th>
<th>310 CAD</th>
<th>320 CAD</th>
<th>330 CAD</th>
<th>340 CAD</th>
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<tr>
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</tr>
<tr>
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<td>2</td>
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<td>-</td>
<td>3</td>
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</tr>
<tr>
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<td>1</td>
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<td>1</td>
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<td>1</td>
</tr>
</tbody>
</table>

4.2.3 Analysis Methods

As mentioned in the introduction, (HS-)PIV results can be used for qualitative comparison with CFD results as shown by Yu et al.[113]. To be able to make this comparison quantitative, proper orthogonal decomposition (POD) is a promising tool to
4.2. Methods

High-speed PIV during fuel injection compare LES and PIV data as shown by Meyer et al. [68]. The POD technique and its implementation are briefly introduced below.

**Proper orthogonal decomposition**

POD is a mathematical technique used to obtain low-dimensional approximate descriptions of high dimensional processes, or to extract main modes from experimental data, see Chatterjee [12]. The basis functions used are orthonormal and ordered such that a combination of the first few functions gives the best possible reconstruction. For POD on velocity fields, the basis functions are themselves velocity fields, and most of the structure in the measured velocity field can be captured by a linear combination of only a few basis functions. In this case we use the snapshot approach, developed by Sirovich [95], which is more convenient when the number of collected samples is smaller than the space discretization, according to Bizon et al. [7]. The snapshot approach has been applied in engine flow research to extract coherent structures from a turbulent flow in an unbiased way by Druault et al. [24] and Roudnitzky et al. [89], and also for decomposition of soot luminosity during combustion by Bizon et al. [6]. POD has been applied to LES data of a piston-cylinder assembly by Liu and Haworth et al. [58]. The POD, in this case an eigenvalue decomposition, can be performed in a few steps which are explained below, following the strategy of Meyer et al. [68].

**Eigenvalue Decomposition**

At every crank angle position we record \( N \) independent velocity fields during successive engine cycles, the so-called snapshots \( (U^n) \). Each snapshot consists of \( m \) 2D velocity vectors. Phase-dependent POD, as the name suggests, calculates the POD modes for every specific CAD. The components \( u \) and \( v \) in \( x \) and \( y \) direction of each velocity vector \( (u,v) \) for each measured velocity field are stored in a velocity matrix, \( U \), in which each column represents one measured velocity field taken at the same crank angle, hence

\[
U = [U^1 U^2 ... U^N] = \begin{pmatrix}
  u_1^1 & u_1^2 & \cdots & u_1^N \\
  \vdots & \vdots & \ddots & \vdots \\
  u_m^1 & u_m^2 & \cdots & u_m^N \\
  v_1^1 & v_1^2 & \cdots & v_1^N \\
  \vdots & \vdots & \ddots & \vdots \\
  v_m^1 & v_m^2 & \cdots & v_m^N 
\end{pmatrix}.
\] (4.1)

Using the velocity matrix, the \( N \times N \) space correlation matrix, or autocovariance matrix, is defined as:

\[
R = U^T U,
\] (4.2)
for which the eigenvalue problem can be written as

\[ RA = \Lambda A. \]  

(4.3)

The eigenvalues \( \lambda \) from the eigenvalue array \( \Lambda \) are then sorted in decreasing order:

\[ \lambda_1 > \lambda_2 > \ldots > \lambda_N. \]

The eigenvectors \( (A_i) \) are sorted in the same order as the eigenvalues and stored as a matrix used to define the POD mode matrix \( (\Phi) \) which is normalized:

\[ \Phi = \frac{AU}{\|AU\|}. \]  

(4.4)

The POD coefficients \( (a) \) for a specific snapshot are determined by projecting the velocity field of the snapshot onto the POD modes \( \Phi \):

\[ a^n_i = \Phi^i U^n. \]  

(4.5)

Using the POD coefficients and the POD modes, a snapshot \( (n) \) can be reconstructed (subscript \( r \)) using:

\[ U^n_r = \sum_{i=1}^{i_{max}} a^n_i \Phi^i. \]  

(4.6)

When all POD mode contributions are included \( (i_{max} = N) \) the snapshot is fully reconstructed.

For phase-invariant POD, all snapshots are stored in one single matrix \( U \), the resulting number of columns is the product of the number of cycles and the number of visualized CAD positions. The subsequent eigenvalue decomposition procedure is the same as for phase dependent POD. The number of POD modes resulting from this approach again equals the number of columns of \( U \).

Coordinate definition

In almost all flow research in engines conducted with PIV, the coordinate system is cartesian. However, when investigating the injection of a fuel spray in the swirl plane of a combustion engine, in which there is rotational symmetry in both flow and geometry, transforming the velocity information onto a polar grid, as done by Petersen and Miles [81], can be expected to give more insight. By representing the data in radial and azimuthal directions, the influence of a radial injection of fuel on the dominantly azimuthal swirl can be investigated in more detail. The center point of the polar grid is positioned in the center of the combustion chamber, directly below the injector tip. An offset in swirl center position is not taken into account in this investigation, although the swirl center is not necessarily positioned in the center, see Zegers et al. [115].
4.3 Results

Fluctuation definition

To investigate fluctuations, the turbulent kinetic energy (TKE) and its dissipation rate are often calculated from the 2D datasets, as shown by Saarenrinne and Piirto [91], Müller et al. [71] and Druault et al.[24]. By calculating the specific kinetic energy, the in-plane influence of the fuel injection on the flow is investigated as explained by [1]. However, when the fluctuations are of the same order of magnitude as the mean velocity, it is more convenient to define the fluctuations \( u' \) as the Reynolds-decomposed part of the velocity. In this approach the ensemble averaged velocity field \( \overline{U} \) is subtracted from the instantaneous velocity fields \( U \), that is

\[
u' = \frac{1}{m^2} \sum_{i,j=1}^{m} \frac{1}{N} \sum_{k=1}^{N} |U(i,j,k) - \overline{U}(i,j)|, \tag{4.7}
\]

where \( i \) and \( j \) are the \( r \)- and \( \theta \)-components respectively, and \( k \) is the number of cycles. The results represent only two components of the velocity. The third component, in the vertical direction \( z \), is not measured in this setup.

4.3 Results

The seeding material of hollow microspheres shows good laser scattering due to their large size, which can be seen in the PIV images in figures 4.3a and 4.3b. We observe no significant change of the microsphere size during the compression stroke. The heating effect and the increasing pressure seem to roughly cancel out, resulting in...
High-speed PIV during fuel injection

4.3. Results

Microspheres of almost constant size. The injection of fuel does not seem to affect the microspheres through the whole compression and work strokes of the engine. The region of bright scattering by the liquid fuel (figure 4.3b) is longer than would be expected on the basis of the geometrical overlap of the 1 mm thick light sheet and the spray (see figure 4.2). This is probably due to multiple scattering. Inhomogeneities in the light sheet, partly due to obstruction by the fuel sprays, result in low signal levels in the lower left part of the images, resulting in low correlations in those areas and relatively many rejected vectors.

4.3.1 Dataset 1: No injection

A typical instantaneous velocity field (snapshot) is shown in figure 4.4a. Averaging 310 velocity fields, all recorded at 329 CAD, results in the ensemble-averaged velocity field of figure 4.4b. Note that the swirl center is slightly positioned off the central

Figure 4.4 Typical velocity fields in [m/s] at 329 CAD (dataset 1).
axis of the cylinder. This behavior was also observed previously in chapter 3 for the same piston geometry. Reynolds decomposition of the snapshot in figure 4.4a results in the fluctuation field presented by figure 4.4c. The kinetic energy contained in this residual velocity field is still 25% of the energy present in the ensemble-averaged velocity field of figure 4.4b. This procedure was repeated for the complete dataset, and by spatially averaging the velocities and their corresponding fluctuations in radial and azimuthal direction the evolution as function of crank angle degree can be determined as depicted in figure 4.5. From this figure it can be seen that the azimuthal and radial velocity components can be considered to be relatively stable, only showing small deviations over the investigated crank angle range. The radial component is negative, which means the flow is directed towards the center, due to squish motion. The azimuthal velocity component outweighs the radial component in absolute value, which is obvious when considering the swirl motion present in the instantaneous velocity fields. The fluctuations in azimuthal and radial direction are comparable in magnitude.

**Phase-dependent and phase-invariant POD analysis**

Using phase-dependent POD, the dataset of 310 snapshots is used to investigate the dominant flow structures, which contain the highest kinetic energy. When using the first mode to reconstruct an instantaneous velocity field, only the most dominant structures are present. An example is shown in figure 4.6a for the same instantaneous velocity field as given in figure 4.4a. This reconstruction contains 83% of the energy present in the original velocity field, and is similar to the ensemble averaged velocity field. This is confirmed by plotting the difference between the two in figure 4.6b. By scaling the vectors, the swirling motion is still noticeable; the velocities are however in the order of 0.2 m/s and represent only 0.2% of the energy from the ensemble-averaged field. This is because the average velocity is not subtracted from the velocity.
4.3. Results

(a) Reconstructed snapshot of an instantaneous flow field at 329 CAD (figure 4.4a) using only the first mode. The reconstruction contains 83% of the energy of the original snapshot.

(b) Residue when subtracting the mean velocity field from the reconstructed snapshot using only the first mode of the flow field at 329 CAD. The residue contains only 0.2% of the energy in the ensemble-averaged velocity field.

Figure 4.6 Reconstructed velocity field [m/s] and the corresponding residue. Note the difference in velocity scales.

The dominance of the first mode is clearly displayed in figures 4.7a and 4.7b where the POD coefficients and the cumulative energy are displayed for the first 20 modes. When applying phase-invariant POD using the complete crank angle range (3100 snapshots), the resulting cumulative energy plot overlaps the line in figure 4.7b exactly and is therefore not shown. From this we conclude that, for the case with 310 snapshots without fuel injection, there is no need to apply the phase-invariant POD to represent the instantaneous velocity fields as the energy represented by the same modes is equal for phase-dependent and phase-invariant POD.

4.3.2 Dataset 2: Fuel injections

Fuel has been injected at an increased inlet pressure of 2 bar. Eight cases without injection up to 330 CAD (Table 4.2: cases 1, 2, 4 to 8 and 10) were combined resulting in 72 cycles for crank angles up to 330 CAD. The corresponding spatially averaged velocities and fluctuations are depicted in figure 4.8. This figure clearly shows increasing azimuthal and radial velocities, whereas the fluctuations are relatively stable. The increase of in-plane velocities towards top dead center has been shown before in chapter 3. The increase of the radial velocity towards the center (negative
4.3. Results

High-speed PIV during fuel injection

Figure 4.7 First 20 phase dependent POD modes of the instantaneous flow field at 329 CAD as displayed in figure 4.4a.

Figure 4.8 Spatially averaged velocities and fluctuations in r and θ direction of 72 cycles as function of CA at 2 bar inlet pressure. The information from figure 4.5 (at 1 bar) has been plotted as well (320–329 CAD). No fuel is injected.

direction) can be attributed to the increase of squish motion between the piston and the cylinder head. The increase in azimuthal velocity depicts the increase in swirl motion which can be ascribed to the conservation of angular momentum in a smaller volume due to compression. Comparing the crank angle range (320-329 CAD) of figure 4.5 with the same range in figure 4.8 it appears that velocities and fluctuations are similar. We conclude that the increased density, due to the increase in inlet pressure for dataset 2, does not have an important influence on the velocities.

**Typical flow fields during injection**

A typical PIV image during injection is shown in figure 4.3b. Averaged flow fields for a double injection at 330 and 340 CAD (case 8) are shown in figure 4.9. Start of the first actuation is at 330 CAD, the injection starts around 331 CAD and ends 2
ms later around 337 CAD. The second injection starts around 341 CAD. During the initial phase, the flow of the injection (figure 4.9a) shows a distinct motion towards the center of the cylinder, due to air entrainment near the foot of the spray. Note that none of the velocity fields show the outward motion of the spray. This motion is not captured because the illuminated spray is almost completely removed from the recording during pre-processing. Vectors are displayed at positions of the fuel spray because the large 64 px interrogation windows use the surrounding displacement to calculate the displacement at those positions, for the same reason the outward velocity of the regions near the spray are not captured. The largest bias errors therefore occur at the spray positions and it’s near surrounding as shown in figure 4.3b. Moreover, the combination of the 50 µsec pulse separation and the 1 mm thin laser sheet biases our measurements to structures with axial velocities $V_z < 20 \text{ m/s}$. The speed of the fuel spray is in the order of 200 m/s and will induce a large out-of-plane motion.

After 7 CAD of injection, the flow behavior has changed into a less coherent motion, as shown in figure 4.9b. The air entrainment from below the spray (indicated by the green arrow in figure 4.10a), leads to a positive outward motion. Combined with a significant amount of induced out-of-plane motion, the resulting velocity field recorded has very low averaged velocities between 337 and 340 CAD, when the second injection starts.

The inward motion due to air entrainment during the second injection at 345 CAD (figure 4.9c) is strongly influenced by other flow structures present, possibly by the re-entrance of the first injection via impingement on the piston and cylinder, as illustrated by the pink arrow in figure 4.10a or by the entrainment from below.

The influence of injection timing and injection pressure on the flow field is addressed in more detail in the next subsections.

**Influences of injection timing**

The piston position as function of CAD is illustrated in figure 4.10. Upon injection at crank angle degrees between 350 and 370 the fuel will be directed into the piston bowl as visualized in figure 4.10b. For all other crank angles, the fuel does not impinge on the piston bowl wall and enters the squish region above the bowl ring (figure 4.10a). The position of the fuel impingement appears crucial for the flow development inside the cylinder.

The influence of the injections on the flow field has been investigated by averaging the spatially averaged velocity fields for each crank angle degree in $r$ and $\theta$ direction separately. The same has been done for the corresponding fluctuations. Several cases have been combined in figures 4.11 - 4.14.

Injection timing was varied from 330 CAD to 350 CAD with constant injection pressure (cases 6, 7, 8 and 10); the results for the spatially averaged velocities and their
4.3. Results

High-speed PIV during fuel injection

Figure 4.9 Ensemble averaged velocity fields [m/s] for case 8 with a 2 ms double fuel injection at 330 and 340 CAD and 2500 bar injection pressure.

Figure 4.10 Schematic representation at two different CAD of the fuel injection and the corresponding major fuel flow patterns (pink), squish motion (orange) and entrainment motion (green).
4.3. Results

Figure 4.11 Spatially averaged velocity and corresponding fluctuations in $r$ and $\theta$ direction of cases 7 and 8. Fuel was injected at 330 and 340 CAD respectively 340 and 350 CAD with 2500 bar injection pressure and injection durations of 2 ms.

fluctuations are shown in figures 4.11 and 4.12. Within each figure the curves are similar, but shifted in time following the injection timing, showing smaller velocities and fluctuations for the cases with a retarded injection timing. These differences can be explained by the influence of the different piston position and a corresponding change in flow behavior, where the piston position at later injection timings significantly dampens the radial motion.

Pronounced peaks towards negative radial velocity are observed for each injection, indicative for air entrainment (figures 4.11a and 4.12a). After each injection, the radial velocity values switch from negative to positive, which can be attributed to the entrainment wave as observed by Musculus and Kattke [72].

For later injection timings, the average radial velocity during injection is close to
4.3. Results

High-speed PIV during fuel injection

Figure 4.12 Spatially averaged velocity and corresponding fluctuations in $r$ and $\theta$ direction of cases 6 and 10. Fuel was injected at 340 respectively 350 CAD with 2500 bar injection pressure and injection durations of 3 ms.

- At the beginning of each injection, during air entrainment, radial fluctuations are relatively small, rising steeply near the end of the injection as depicted in figures 4.11c and 4.12c. The average azimuthal velocities drop sharply only at the end of the (first) injection, and simultaneously its fluctuations rise (figure 4.11d).

Influences of injection pressure

Injection pressure was increased for the same injection timings from 2000 to 2500 bar (cases 3 and 11, figure 4.13), and from 1500 to 2500 bar (cases 1, 4 and 5, figure 4.14). An increase in injection pressure does not seem to lead to increased velocities
(figure 4.13), the fluctuations do not correlate very well with injection pressure.

Figure 4.13 Spatially averaged velocity and corresponding fluctuations in $r$ and $\theta$ direction of cases 3 and 11. Fuel was injected at 310, 320, 330, 340 and 350 CAD, with injection durations of 1 ms.

Figure 4.14 does show variations in the integrated flow field with changing injection pressure, but does not show a clear trend. This is not to be expected from momentum conservation and might be related to different injector behavior when fuel pressure is changed and a different mass flow. Further research should be performed to investigate how increasing injection pressure, and therefore increasing fuel momentum, influences the flow fields inside the piston bowl.

Phase-dependent POD analysis

Phase-dependent POD analysis is performed on a typical cycle from case 8. By doing so, there are only 9 modes. The POD coefficients for the most dominant modes
4.3. Results

High-speed PIV during fuel injection

Figure 4.14 Spatially averaged velocity and corresponding fluctuations in \( r \) and \( \theta \) direction for cases 1, 4 and 5. Fuel was injected at 330, 340 and 350 CAD with injection durations of 1 ms. 

are shown in figure 4.15a for the whole measurement span (260 to 400 CAD) and for a subrange (330 to 360 CAD) in figure 4.15b. In these figures the dominance of the first mode is obvious before and during the first injection, but not during the second injection (340 to 348 CAD) when the first five modes are almost equally contributing to the instantaneous velocity fields. This indicates that, upon injection, the flow changes into an incoherent flow with unrelated structures in each snapshot. Immediately after the second injection, around 350 CAD, the POD coefficient of the first mode increases rapidly again, indicating a re-establishment of coherent flow. Averaging the POD coefficients over the 9 available cycles reduces the noise-like structures and results in figure 4.16. The drop in magnitude of the dominant POD coefficient during injection corresponds with the decrease in radial and azimuthal
velocities around 340 CAD as shown in figure 4.11.

Figure 4.15 Phase-dependent POD analysis for a typical cycle from case 8. Fuel was injected at 330 and 340 CAD with durations of 2 ms.

In figure 4.17 the normalized eigenvalues (λ) are shown for the accumulation of the first five modes of case 8. The dip in the eigenvalue of the first modes after 340 CAD supports the observed lack of coherent structures in figure 4.16b. For crank angles during the first injection starting at 330 CAD, the first mode contributes about 90% to the total energy, and drops a little bit until the second injection starts at 340 CAD. During and after the second injection, the energy present in the first mode drops to a level of 20% and the first five modes combined only capture 75% of the energy.
4.3. Results

High-speed PIV during fuel injection

Figure 4.17 Normalized eigenvalues ($\lambda$) as function of CAD for the phase-dependent POD analysis for case 8. Fuel was injected at 330 and 340 CAD with durations of 2 ms.

Phase invariant POD analysis

To investigate whether POD modes from different crank angles can be used to represent the measured velocity fields during and after injection, phase invariant POD has also been applied to case 8. There are no phase dependent eigenvalues and eigenvectors any more, only the POD coefficients themselves can be studied to judge the dominance of certain modes. The results of the phase-invariant analysis are shown in figure 4.18. The fluctuations observed in the POD coefficients for one cycle in figure 4.18a are greatly reduced by averaging the POD coefficients for the 9 available cycles as shown in figure 4.18b. To interpret the results, a comparison is made with the phase-dependent POD. Contrary to the phase dependent POD case given in figure 4.15, there is no clear dominance of a single mode, except, surprisingly, at the start of the first and of the second (last) injection. Apparently, these events are sufficiently violent to put a stamp on the global flow pattern even though they dominate the flow for just a few crank angle degrees. The lack of a single dominant mode is confirmed by figure 4.18c, showing that mode 1 contributes around 30% to the total energy and mode 2 in the order of 20%.

The phase-invariant POD analysis can be applied to datasets with a low number of snapshots to increase the amount of data in the analysis. However, using snapshots of the flow fields without injection as a basis for flow fields at crank angle degrees after injection does not seem to be a good strategy for sets of data in which the injection of fuel changes the flow field drastically. Because of the lack of resemblance between flow fields, the use of separate basis functions for each crank angle seems more appropriate. Possibly, a more promising strategy could be to use two sets of base functions, one for the period before injection and one for the period after the start of the first injection.

What can be learned from the performed POD analysis is the lack of coherent struc-
4.4 Conclusions

In this study the applicability of HS-PIV to investigate large in-cylinder flow structures during and after fuel injection has been proven. The hollow microspheres used as seeding material perform very well before, during and after injection.

At crank angles without fuel injection, velocity fluctuations are in the order of 50% of the average absolute velocity. Increasing the inlet pressure and therefore density does not show an influence on the velocities and fluctuations for the invested crank angle range of 320 to 329 CAD. POD analysis shows that a snapshot reconstruction using only the first mode differs only 0.2% in energy from the ensemble averaged velocity field.

Figure 4.18 Phase-invariant POD analysis of case 8. Fuel was injected at 330 and 340 CAD with durations of 2 ms.

tures during an injection. This can be seen from the sudden drop in POD coefficient of mode 1 and the rise of higher modes.

(a) POD coefficients ($a^1_i$) of cycle 1 for modes 1 to 5.

(b) Averaged POD coefficients of 9 cycles ($a_i$) for modes 1 to 5.

(c) Cumulative energy
During injection of fuel, air entrainment is observed, witnessed by a strong increase of the in-plane velocity and the corresponding fluctuations during the injection event. In the case of multiple injections the air-entrainment during the first injection is consistent and the fluctuations between consecutive cycles small. When previous injections re-enter the investigated plane via impingement on the wall or top of the piston, the in-plane structures change drastically into less coherent structures, compensating the inward motion due to air entrainment by subsequent injections. The spray induced flow can evolve in different structures, which might influence the actual mixing of fuel and air, causing differences in fuel and air ratios between different injection strategies and therefore change the combustion delay present in PCCI combustion.

The applied POD analysis supports the observed changes in flow structures during and after injection and might be a good tool for comparison with future CFD results. For CAD with high fluctuations, as observed during injection, a good representation of the instantaneous velocity field using only a few POD modes is not possible.

Phase-invariant POD is not an appropriate method to represent highly fluctuating flows during and after fuel injections. Because of the lack of resemblance between crank angle degrees, the use of separate basis functions for each CAD is more appropriate; alternatively one could use separate sets of base functions before and during injection.

Restricting the investigated crank angle range and therefore enabling the recording of more velocity fields during the ignition delay period increases the amount of snapshots for the POD analysis and increase measurement accuracy.
“Fuel is pumping engines, burning hard, loose and clean”

Metallica - Fuel

5

Temperature measurements using Toluene PLIF

This chapter is an integral copy of the article which has been submitted to Applied Physics B: Lasers and Optics. Minor changes have been made to improve the quality.


* Both authors contributed equally to this work. R.P.C. Zegers is responsible for the engine experiments and M. Yu for the experiments conducted in the high pressure cell. The analysis of the experiments has been done by both researchers. The LES simulations have been performed by C. Bekdemir.
5.1 Introduction

Non-intrusive accurate 2D temperature measurements under high temperature engine conditions have been a challenge for the last decade. Mass-averaged temperature in engines can be derived quite accurately from the pressure trace. However, temperature stratification resulting from injection of fuel into a cylinder filled with hot air cannot yet be measured accurately. Recent research suggests that toluene laser-induced fluorescence (LIF) is an appropriate measurement technique for measuring gas and vapor temperatures [20, 92, 103]. Toluene has a few advantages over other tracers: it has a high fluorescence quantum yield, a relatively large red shift as function of temperature and a lower toxicity than e.g. benzene [69, 92]. The change in fluorescence as a function of oxygen concentration can also be used to determine the oxygen distribution in vapors by using two tracers which react differently on oxygen quenching [46]. A major disadvantage of toluene LIF is the decrease of its effective fluorescence lifetime as function of temperature. This is a result of fast depopulation of the excited state due to inter-system crossing (ISC) [118]. This decrease in fluorescence lifetime decreases the total amount of fluorescence with increasing temperature [44]. Even with high exitation laser fluence and intensification by the CCD camera, signal levels will decrease to noise levels at high temperatures (>700 K) and therefore decrease the accuracy.

The applicability of the toluene LIF technique in an engine environment was investigated by only a few researchers. Luong et al. measured in-cylinder temperatures without fuel injection [61]. Mannekutla et al. showed the potential of the technique for fuel sprays [65], but results were tentative, probably due to fuel fluorescence or fuel decomposition in air. In our research, toluene LIF is used as a tool to measure temperatures during and after injection of fuel inside an engine and a high-pressure cell. Toluene LIF is used because of the promising outlook of the technique and its relatively modest equipment requirements.

Temperature stratification after spray injection is of major influence on partially premixed combustion (PPC) strategies in diesel engines. In the PPC regime, there is a finite time delay between end of injection and start of combustion, resulting in premixed fuel and air regions with various temperatures and fuel concentrations. Earlier toluene LIF studies did not have the aim to measure temperature stratification after fuel injection.

Cycle-to-cycle variations influence the mixture formation and therefore the time and location region where combustion starts. Controlling the ignition event is a major research topic in the application of PPC combustion strategies in practical combustion engines. For this reason we investigate the mixture properties by visualizing the fuel stratification after injection. To investigate the detailed spatial structure of a single fuel spray, toluene LIF is also applied in a high-pressure cell, which has enhanced optical access compared to the optical engine. Finally, a Large-Eddy Simulation (LES) of the same spray is used to interpret the experimental observations via the
calculated velocity field. In section 5.2 we briefly describe the measurement as well as the numerical setup. The aspects which need to be taken into account when performing quantitative toluene LIF experiments are carefully reviewed in section 5.3. The calibration and error analysis is given in section 5.4. The results are shown in section 5.5 followed by the conclusions and recommendations in section 5.6. Finally an outlook is given in section 5.7.

5.2 Materials

In this research, two setups are used, being an optically accessible one-cylinder engine and a constant volume high-pressure cell (HPC).

5.2.1 Optically accessible engine

The engine setup is the same as presented in section 3.2. A schematic layout of the complete setup is given in Figure 5.1 and a cross-section of the engine LIF setup is shown in Figure 5.2. Specifications can be found in Table 3.1 and the engine operated with a compression ratio of 13.9.

The engine is equipped with both a port fuel injection (PFI) system (used for toluene injection in the intake air) and a common rail injector of Delphi Diesel Systems (used for pure heptane injection or a heptane toluene blend). Toluene is pressurized in a 100 ml vessel using compressed nitrogen at 4 bar and is directly connected to two PFI injectors which are controlled by a MOTEC M8 programmable motor management system. The distance between the PFI injectors and the inlet valve is approximately 50 cm. The details of the common rail diesel injector are presented in Table 4.1.
5.2. Materials

Figure 5.2 Cross-section of the optically accessible engine.

5.2.2 High-pressure cell

The core of the HPC is a cubic combustion chamber produced through spark erosion inside a stainless steel cube. Ports on each side of the combustion chamber can be fitted either with a window or with metal plug. The HPC is equipped with sub-systems for heating, cooling, fuel injection, gas supply, control, data acquisition and optical diagnostics. A frontal view of a cross-section of the HPC is shown in Figure 5.3. The mixing fan and the pressure sensors are not shown, because they are installed on the corners of the HPC. The main features of the HPC are listed in Table 5.1. A more detailed description is given by Baert et al. [3].

The gas supply system is based on four mass flow controllers, which enable filling pure gases one by one to prepare the gas mixture inside the HPC. The filling process is monitored by the gas pressure sensor. A mixing fan is used to enhance the mixing. We use \( \text{C}_2\text{H}_2 \), \( \text{O}_2 \), \( \text{N}_2 \) and \( \text{Ar} \) to form a mixture, which, once burned, produces the same density, \( \text{O}_2 \) mass fraction and heat capacity as those expected at the desired engine condition.

In the experimental procedure, the HPC is first evacuated and then filled with the combustible gas mixture. This is spark-ignited and burns during the so-called pre-combustion phase. After that the residual gas of pre-combustion cools down. When the desired pressure (hence bulk temperature) has been reached, typically after about 1 - 2 seconds, the fuel is injected. The optical diagnostic system is synchronized with the fuel injection. A full experimental cycle takes about 15 minutes.
5.2. Materials

5.2.3 LIF setup

Each of the two toluene LIF setups contain two cameras which are used in two different configurations, as described in the sections below. The specifications of the equipment are similar for both configurations.

General specifications

The components used in the toluene LIF setups are specified in Table 5.2. The bandpass filters both have a diameter of 50 mm to fit the UV lenses as indicated by the red and blue blocks in front of the lenses in figures 5.2 and 5.7. Due to failure of camera 2, three different cameras have been used. These have been named camera 1 to 3. The transmission efficiencies of the used filters, lenses, sapphire windows and the dichroic beam splitter, as well as the reflection efficiencies of the aluminium-coated mirror are shown in Figure 5.4. In addition the change in quantum efficiency of the photocathode of the PI-MAX 3 cameras over the wavelength range of interest is also plotted.

Illumination

In the HPC the laser sheet is aligned with the centerline of the fuel spray. In the optical engine this is impossible, therefore the laser sheet enters the combustion chamber 5 mm below the cylinder head. The laser sheet focuses and is about 1.0 mm thick in the center of the cylinder. Therefore, only a part of the fuel sprays is illuminated.
5.2. Materials

Table 5.1 Specifications of the HPC

<table>
<thead>
<tr>
<th>Basic parameters</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Shape of the chamber</td>
<td>108×108×108 mm³</td>
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<tr>
<td>Window size</td>
<td>50 mm thick, 100 mm diameter</td>
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<tr>
<td>Maximum pressure</td>
<td>300 bar (sapphire windows)</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Heating and cooling system</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall temperature</td>
<td>293 K - 473 K</td>
</tr>
<tr>
<td>Fuel temperature</td>
<td>313 K - 423 K</td>
</tr>
<tr>
<td>Injector holder temperature</td>
<td>333 K</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fuel supply system</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Injector</td>
<td>Delphi single-hole nozzle</td>
</tr>
<tr>
<td>Injector coating</td>
<td>TiN coated</td>
</tr>
<tr>
<td>Nozzle hole size</td>
<td>150 μm</td>
</tr>
<tr>
<td>$C_d$</td>
<td>0.86 (at 800 bar)</td>
</tr>
<tr>
<td>$C_m$</td>
<td>0.77 (at 800 bar)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Sensors</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas pressure</td>
<td>Druck PMP 4070 (0-70 bar)</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>Kistler 4067A5000</td>
</tr>
<tr>
<td>Pre-combustion pressure</td>
<td>Kistler 6041 AU20 (0-250 bar)</td>
</tr>
</tbody>
</table>

Table 5.2 The components of the toluene LIF setups

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laser</td>
<td>Lambda Physik, Compex 350T</td>
</tr>
<tr>
<td>Laser wavelength</td>
<td>248 nm</td>
</tr>
<tr>
<td>Toluene</td>
<td>Sigma Aldrich HPLC grade 99.6% pure</td>
</tr>
<tr>
<td>Dichroic beam splitter</td>
<td>310 nm long pass</td>
</tr>
<tr>
<td>Bandpass filter 1</td>
<td>320 nm FWHM 43.8 nm</td>
</tr>
<tr>
<td>Bandpass filter 2</td>
<td>280 nm FWHM 22 nm</td>
</tr>
<tr>
<td>Lenses</td>
<td>Halle OUC 2.5, f = 100 mm, F# = 2</td>
</tr>
<tr>
<td>ICCD Camera 1</td>
<td>Princeton instruments PI-MAX 3</td>
</tr>
<tr>
<td>ICCD Camera 2</td>
<td>Princeton instruments PI-MAX 3</td>
</tr>
<tr>
<td>ICCD Camera 3</td>
<td>Princeton instruments PI-MAX 2</td>
</tr>
<tr>
<td>ICCD photocathode coating</td>
<td>unigen II</td>
</tr>
<tr>
<td>Camera resolution</td>
<td>16 bit</td>
</tr>
</tbody>
</table>

directly. Figure 5.5 shows a schematic overview of the laser sheet through the spray in the engine; the fluorescence is collected through the sapphire piston bottom. Obviously, the laser sheet directly illuminates only a cross section of the spray. In the
5.2. Materials

Temperature measurements using Toluene PLIF

Figure 5.4 Transmission efficiencies, reflection efficiencies and photocathode quantum efficiency of the optical detection line. All values are based on manufacturer specifications.

actually measured images, the apparent cross sections are larger due to scattering of the laser light and therefore resulting in the visualization of the complete spray.

Figure 5.5 Spray image schematic in the optical engine, with the laser sheet in blue and the spray in orange. The cone angle of the injector is $143^\circ$.

Camera configuration

The toluene LIF thermometry technique requires that two images be taken from the same event in two different wavelength ranges. The wavelength ranges are indicated by the two boxes in the LIF spectrum in Figure 5.6. Requiring these two images can be done either by one single camera or by two separate ones. The former has the advantage of minimized cost, but goes at the expense of spatial resolution and somewhat complicated measures for image separation (see, e.g., Ref. [51]). The latter is more flexible, and is the method chosen here. In both cases, the individual images have to be mapped onto each other, and corrected for pixel-to-pixel variations in sensitivity.

When two cameras are used in combination with sheet illumination, an additional
choice concerns their location. Ideally, the camera’s look at the same scene via the same optical path, as in Figure 5.7a. Under the assumption that distortions in optical path are negligible, an alternative is to look from opposite directions \[60\ 103\](Figure 5.7b), which removes the need for a beam splitter. We have tested both approaches, using a dichroic (rather than a broad-band) beam splitter to split the fluorescence immediately into the desired wavelength ranges.

![Normalized toluene LIF spectrum at three different temperatures](image)

**Figure 5.6** Normalized toluene LIF spectrum at three different temperatures. The two boxes indicate the wavelength regions of the two cameras.

![Schematic view of both camera configurations in the HPC setup](image)

**Figure 5.7** Schematic view of both camera configurations in the HPC setup. a) one side setup b) opposite side setup

For the high-pressure cell setup, the ICCD cameras can be positioned in both configurations. The distance of the cameras is kept the same in both configurations, because the same lenses with fixed focal length are used. For the optical engine setup, the one-side camera configuration is the only option.
5.3 Aspects of quantitative measurements

The dichroic beam splitter is highly sensitive for the light incident angle. The transmission curve of the beam splitter crosses the transmission curve of the 320 nm filter (Figure 5.4). Since the beam splitter is close to the HPC, the incident angle deviates 8 degrees from one side of the image to the other side, resulting in a shift in transmission efficiency of the beam splitter. This results in different intensity ratios across the region of interest. The correction for this effect will be explained in section 5.5.2.

5.2.4 Numerical setup

The LES code used in this study is the fully compressible AVBP solver co-developed by IFP Energies Nouvelles and CERFACS for structured and unstructured meshes [70]. The second order centered Lax-Wendroff convective scheme [52] is combined with an explicit time advancement. Unresolved subgrid-scale turbulence is modeled by a Smagorinsky model [98] with constant coefficient. The dispersed liquid phase is resolved using a mesoscopic Eulerian formalism [29]. The region close to the injector, dominated by high volumetric ratios of liquid fuel and breakup effects, is not included in the simulations. The DIturBC injector model [66] is used to bridge that region by imposing physical flow conditions at a distance approximately 10 nozzle diameters downstream of the injector outflow plane. Its principle is to impose boundary conditions for the two phase flow at a distance downstream of the injector nozzle, in regions where the flow has been dispersed by breakup phenomena and can be addressed by a diluted mesoscopic Eulerian formalism.

The LES has been run on a computational mesh which represents a closed combustion vessel with slightly larger dimensions than the HPC (112³ versus 108³ mm³). It consists of a tetrahedral mesh with local refinement. The mesh size near the nozzle exit is 80 µm and gradually increases to 800 µm towards the other end of the domain, yielding approximately 0.7 million nodes and 4.3 million cells. All boundaries, except for the injection plane, are taken as adiabatic walls which have negligible influence on the spray internal structure. This mesh has been chosen on the basis of spray formation results presented in [5].

5.3 Aspects of quantitative measurements

To perform accurate toluene LIF measurements, special attention has to be paid to several aspects concerning the equipment and measurement procedure. These aspects will be systematically described and discussed in the following subsections. The data processing steps are given in section 5.8.
5.3. Aspects of quantitative measurements

5.3.1 Fuel

The base fuel and tracer should not contain any species which fluoresce in the wavelength range of interest upon excitation at 248 nm. Therefore a high purity grade (high performance liquid chromatography, HPLC) of toluene and n-heptane base fuel are used.

Disintegration or thermal decomposition of the tracer in an oxygen-free environment limits the time span in which experiments can be performed at high temperatures. For the time spans under investigation in engines, this is not an issue. The toluene concentration needs to be adjusted to achieve optimal signal levels in spray vapor and gas phase. The laser beam is attenuated at high tracer concentrations and this decreases the fluorescence signal along the laser path. Note that fluorescence trapping is not expected to play any significant role in our experiment because of the red-shift of the toluene fluorescence upon excitation [92].

Figure 5.8 shows an example of the injection of a pure n-heptane spray (without toluene) in the HPC into residual pre-combustion gas at about 600 K, recorded with the opposite camera configuration. The residual gas intensity yields about 150 counts more than the dark current level. In the 280 nm range the spray is barely visible. However, in the 320 nm wavelength band, the spray increases the signal with 150 counts above the residual gas intensity, which originates from fluorescent minor impurities in the fuel. Previous research also mentions background signal from impurities, but neglected it because the signal levels were below 10% of the toluene signal [103]. This increased background level needs to be corrected for or taken into account when evaluating the results.

![Raw images of pure n-heptane injection](image-url)

**Figure 5.8** Raw images of pure n-heptane injection (320 nm (a) and 280 nm (b)) in the HPC into residual pre-combustion gas at about 600 K. The fuel is injected from the top right corner. The increased signal in the center of the 320 nm image originates from fluorescent minor impurities in the fuel. The top left corner of the 320 nm signal shows increased signal due to a minor light leak.
5.3.2 Laser

The laser energy may vary from shot-to-shot, the normalized fluorescent spectrum however does not change with laser power. The magnitude of the laser fluence \( [\text{mJ/cm}^2] \) also affects the fluorescence yield. At low fluence the fluorescence yield rises linearly with laser pulse energy, but at higher fluence saturation sets in and the fluorescence yield levels off. Eventually, under conditions of full saturation the fluorescence yield is independent of laser pulse energy. Although it is generally advised to stay away from saturation [103], there are no indications that the LIF ratio method will not work any more when the fluorescence yield is saturated.

To delimit the linear regime, we record the LIF signal of toluene vapor at a fixed crank angle degree and inlet temperature in the optical engine and calculate the ratio of the signals from the two cameras. The laser fluence is varied from approximately 50 to 250 mJ/cm\(^2\) with constant toluene concentration and camera gain. The measured average pixel intensities are shown in Figure 5.9a as function of laser fluence. When looking at the calculated 320/280 ratio (see figure 5.9b), it can be concluded that increasing the laser fluence above 150 mJ/cm\(^2\) does not influence the ratio. Remarkably, the ratio does change for a laser fluence below 150 mJ/cm\(^2\). This is explained by the low overall signal levels on the 320 nm camera. Other researchers mention a laser fluence of about 50 mJ/cm\(^2\) as the upper limit for linear fluorescence response [103], which contradicts with our findings. Currently, the results published on the assumed ratio change as function of fluorescence saturation are inconclusive. Therefore all experiments presented below are conducted with moderate laser power levels around 150 mJ/cm\(^2\).

![Figure 5.9 LIF intensities and ratios as a function of laser fluence obtained in the optically accessible engine at 300 CAD, with an inlet air temperature of 50 °C. The 320/280 ratios represent a temperature span of approximately 460 ± 15 K.](image-url)
5.3.3 Camera

The quantum efficiency of the photocathode material used is dependent on the incoming wavelength. In the wavelength region around 300 nm the quantum efficiency of the unigen II coating is in the order of 10%. When the intensity signal on the camera drops, the accuracy decreases due to a lower signal-to-noise ratio. Therefore a $2\times2$ pixel binning approach is used, resulting in a decreased spatial resolution of $512\times512$ pixels.

ICCD cameras only respond linearly to the incoming amount of photons for a certain intensity range. Also multi-channel-plate (MCP) saturation needs to be avoided [57]. Due to the low photocathode quantum efficiency for the wavelength range under investigation, the MCP does not saturate in the performed experiments. To check the linearity range, the camera intensity as function of the incoming light from a white diffusive screen is measured and the light intensity is adjusted by means of the camera lens diaphragm. The observed maximum intensity of the linear range is different for all cameras used and is $1.2\times10^4$ counts for camera 1, $2.5\times10^4$ counts for camera 2 and $6.0\times10^4$ counts for camera 3 (on a full dynamic scale of $6.5\times10^4$). It should be mentioned that this test was performed with a white light source and the MCP might now saturate earlier than for the UV wavelengths.

5.3.4 Optical effects

Fouling of optics and windows needs to be avoided at all times to limit the probability of a wavelength dependence in the transmission. The transmission and reflection characteristics of the used dichroic beam splitter change as function of the angle of the incoming light. The specified transmission characteristics as function of wavelength are only valid for $45^\circ$. To avoid any transmission change over the field of view (FOV) the dichroic beam splitter needs to be positioned as far as possible from the measurement plane to decrease the angle of the incoming light, otherwise a correction procedure is needed as explained in section 5.5.2.

5.3.5 Collisional quenching

Previous research shows a large influence of oxygen on the fluorescent signal by collisional quenching, which changes as function of temperature [44, 45]. To achieve the highest possible signal and to avoid oxidation of the fuel, all experiments have been conducted without oxygen.
5.3.6 Background subtraction

In all experiments the recorded fluorescent images also contain background consisting of several contributions. Part of the background concerns hardware offset (fixed) and dark current (depends on settings). However, the background in the HPC also contains fluorescence from pre-combustion residual gas, and during spray injection also fluorescence of the fuel, as shown in Figure 5.8. To correct for the background fluorescence level of pre-combustion residual gas, the image recorded 0.2 seconds (5 Hz recording speed) prior to the spray image is used as background and subtracted pixel by pixel from the spray measurement. The influence of the decreasing temperature between the background recording and the spray, which is on the order of 10K, is neglected. The shot-to-shot fluctuations of laser power introduce fluctuations in the background signal, however the additional error is less than 1% of the total signal. The background signal from the base fuel is difficult to remove because the temperature response of the fluorescence signal is unknown. This signal increases the intensity of the 320 nm image with 2%-7%, depending on the operating conditions.

5.3.7 Spatial alignment

Spatial alignment is performed by recording a grid image with both cameras. By using a linear conformal transformation, distortions due to translation, rotation and scaling are accounted for. After transformation, the grid lines in both images match each other with an accuracy corresponding to the 2 pixel thickness of the grid lines.

5.3.8 Flat-fielding

A flat-field is an image acquired from a homogeneous light source to determine pixel-to-pixel sensitivity differences. The flat-fielding method corrects for sensitivity variations across the photocathode, phosphor screen and CCD chip. When the flat-field is determined with a lens mounted on the camera, lens aberrations and vignetting are also accounted for [57, 111]. Accurate flat-fields for both cameras are the key to achieve accurate image ratios and minimize the systematic error. Flat-field images were recorded with defocused camera lenses and a diffusive screen illuminated by a white light source. In order to account for possible wavelength dependence in the flat-field images, a UV light source in the spectral range of the toluene fluorescence would have been preferred. Figure 5.10 displays the flat field image of camera 1 and shows a difference in sensitivity between pixels of approximately a factor of two. The vignetting effect in the corners is also clearly seen. In the opposite camera configuration, the co-flat-fielding method without bandpass filters is used to quantify the sensitivity difference between the two cameras. By using the toluene vapor fluorescence, the camera sensitivities are compared for the UV wavelength bands used for the actual measurement. The pixel-to-pixel sensitivity variations in the region of interest
5.4. Calibration

Figure 5.10 Image of a uniformly lit plane recorded with defocused camera lens; camera 1 with Halle lens.

are quite severe and vary by approximately a factor of 2.

5.3.9 Evaluation of spatial alignment and flat field correction

The purpose of spatial alignment and flat field correction is to make sure the two cameras have the same response to the same fluorescent signal, so as not to introduce errors in the ratio calculation. The intensity correlation of an image pair obtained with opposite and one-side camera configurations are shown in figures 5.11a and 5.11b. For the one-side configuration, a 30/70 beam splitter is used instead of the dichroic beam splitter to get nominally the same signal on both cameras. Both the one-side and opposite side configuration display large differences in sensitivity and distribution. Adjusting for differences in spatial alignment has only a minor influence on the distribution because the residual misalignment is smaller than the 12 px averaging (figures 5.11c and 5.11d). After applying the co-flat-field correction obtained from 10 recordings, the pixel-to-pixel intensity is almost identical on the two cameras (figures 5.11e and 5.11f). The standard deviation of the pixel ratio is approximately 3% for the opposite and 2% for the one-side camera configuration. The difference can be ascribed to the difference in optical paths in the opposite-side camera configuration.

5.4 Calibration

Calibrating the LIF ratio vs. temperature relation is a tedious but crucial task. The most direct calibration procedure, at least in principle, would be to determine this ratio for toluene at a known range of temperatures, using exactly the same setup as would be used in the actual experiments. To avoid the need to perform such a calibration each time the optical setup is changed, or for cases in which the ideal
Figure 5.11 Signal intensity correlation for an image pair obtained in one-side (left column) and opposite configuration (right column). (a&b) Original; (c&d) After spatial alignment; (e&f) After co-flat-field correction. The one-side images are obtained by using a 30-70% beam splitter. The used images are $12 \times 12$ px neighbor average filtered.
calibration procedure is not practically feasible, we here use an alternative approach. This approach is based on the notion that the emitted spectrum is characteristic for toluene (independent of the environment, but that the measured spectrum will be different because of all optics in between the emission and the detector. If we start out with accurate emitted spectra over a range of temperatures, and know the optical characteristics of all elements in the detection path, then the spectrum that would be measured at any temperature in a particular setup can be reconstructed, and the LIF ratio calculated. This procedure will be detailed below. We use temperature-dependent spectra recorded at IFPEN as a basis [20], and in the end compare LIF ratios with global temperatures in our optical engine.

5.4.1 Optical path corrections

The correction for the optical path consists of two steps: (1) the IFPEN base data were transformed into emission spectra by correcting for the wavelength-dependence in the IFPEN spectrometer by Tea et al. [103]; (2) the resulting emission spectra are transformed into spectra that we would measure in our setups. From these spectra and the filter characteristics the LIF ratios for our setups are calculated as a function of temperature. The optical path corrections of step (2) include the wavelength-dependent transmission/reflection coefficients of camera lenses, the sapphire piston window, mirrors (incl. the dichroic one in the case of the one-side setup), and the photocathode of the ICCD cameras used; the data as provided by the manufacturer were given in Figure 5.4. As an example, the emission spectra reconstructed from the IFPEN data (purple) are shown in Fig. 5.12, together with the spectra arriving at the ICCD cameras after reflection or transmission by the dichroic mirror (setup of Figure 5.7a). Obviously, the transition region in the transmission/reflection characteristics of the dichroic mirror significantly reduces spectral intensity in the 300 nm region.

For this calibration approach, different overall camera sensitivities (“overall” meaning apart from flat field corrections that have been discussed above) have to be taken into account explicitly. We determine the relative sensitivity in a straightforward way, by recording images from the same object under constant illumination. This factor varies with gate time, camera gain and recording speed, and should be determined separately for each measurement condition.

5.4.2 Engine model

A multi-zone engine model, which incorporates a Woschni correlation to estimate the heat loss to the cylinder walls [25], is used to calculate the bulk in-cylinder temperature. Our model does not account for blow-by losses via the piston rings, and the bulk in-cylinder temperature at the moment of intake valve closing (IVC in Table 3.1) is treated as a variable, ranging between 30-70 ºC. The LIF ratio is
5.4. Calibration

Temperature measurements using Toluene PLIF

determined from the complete FOV without areas near the wall and edges of the laser sheet. Figure 5.13 shows the LIF ratio vs. temperature plots derived in this way for various IVC temperatures, and compares them to the similar data derived from the original IFPEN curves and corrected for the differences in the optical setup used in both experiments, as described in the previous section. Clearly, the curves are very similar, but the IFPEN data cross several of our curves for different IVC temperatures. This may be due to differences in heat loss approximations and blow-by.

5.4.3 Error analysis

The error analysis has been split into sections on the precision and accuracy of the technique, respectively.

**Precision**

The precision or random error is determined by the deviation in background signal ($\sigma_{bg}$) originating from laser power fluctuations and the noise introduced by the camera intensifiers ($\sigma_i$). $\sigma_{bg}$ is approximately 1% of the signal (about 20 counts). And $\sigma_i$ is
5.4. Calibration

Figure 5.13 Temperature calibration lines from the optical engine in Eindhoven and at IFPEN.

...estimated from the standard deviation of the pixel-to-pixel correlation after co-flat-fielding which is 3% in the opposite camera configuration as given in section 5.3.9. The random error of the 320 nm (subscript r for 'red') image is

\[ \sigma_r^2 = \sigma_{i,r}^2 + \sigma_{bg,r}^2. \]  
(5.1)

Similarly, the error of the 280 nm (subscript b for 'blue') image is

\[ \sigma_b^2 = \sigma_{i,b}^2 + \sigma_{bg,b}^2. \]  
(5.2)

The error in the LIF ratio can be determined by using:

\[ \left( \frac{\sigma_R}{R} \right)^2 = \left( \frac{\sigma_r}{I_{r,2}} \right)^2 + \left( \frac{\sigma_b}{I_{b,2}} \right)^2, \]  
(5.3)

where \( \sigma_R \) is defined as the precision of the LIF ratio and is approximately 4%.

Accuracy

The accuracy, defined here as the uncertainty due to systematic error, originates from several sources: unwanted fluorescence from the base fuel, deviations in the flat field matrix and calibration procedure. Unfortunately, the HPLC grade of the n-heptane base fuel primarily emits fluorescence signal in the 320 nm wavelength region upon illumination by 248 nm radiation. The signal intensity contributes 2 - 7% of the total signal depending on the operating point.

The flat field correction at opposite configuration is quite accurate because the toluene LIF signal is used to obtain the co-flat-fielding matrix. The camera response accounted
for in the flat-field matrix is from almost the same wavelength range as in the actual spray measurements.

However, for the one-side configuration, the flat-field approach using the toluene LIF signal itself cannot be used because of the dichroic beam splitter. Therefore, flat-field images are obtained using a white light source.

5.5 Results

The results section is split into two major parts. Part one presents the results in the optically accessible engine, the second part presents the result in the high-pressure cell.

5.5.1 Optically accessible engine results

Residual fuel fluorescence

A basic assumption of the LIF ratio method is that the fluorescence transmitted by the band-pass filters is really due only to the tracer. Thus, it is imperative that the fuel itself does not fluoresce upon 248 nm irradiation. Before each measurement series, we check for fluorescence signal during fuel injection without adding toluene to the system. Initial experiments were performed with spectrometric-grade n-heptane (99.3% pure). This always resulted in clearly recognizable spray images (Figure 5.14), obviously due to fluorescence of trace components in the fuel. The whole fuel line was then flushed 5 times with HPLC-grade n-heptane (99.6% stated purity; Sigma-Aldrich). Figure 5.15 shows the time evolution of the average signal in one of the spray tips (location indicated in Figure 5.14) measured on both cameras, as a function of the amount of fuel flushed through the fuel line (including fuel pump, tubing and injector). After an initial fast decrease, the signal gradually decreases further, but not to zero, especially on the 320 nm camera. After having the system stay idle overnight, the fluorescence was again increased (data points to the right of the vertical dashed line), apparently due to residuals in crevices in the fueling system that leaked back into the fresh fuel.

This residual fluorescence gives rise to LIF ratios that deviate strongly from those expected for toluene fluorescence at the same temperature. Therefore, during actual toluene LIF measurements the signal must be sufficiently high for the residual contribution to be negligible.
5.5. Results

Figure 5.14 Single shot LIF image of a heptane spray in the optical engine. The red square represents the selected area of the spray to determine the intensity decrease as function of the amount of flushed high purity heptane.

Figure 5.15 Intensity as function of pure n-heptane flush. The fueling system was filled with 99.3% pure n-heptane prior to flushing with 99.6% pure n-heptane. Images are recorded at -20 CAD ATDC, (laser power 60 mJ, camera gain=100). The vertical line displays different measurement days.

During fuel injection

Temperature stratification during fuel injection was investigated by adding toluene to the inlet manifold. Together with the nitrogen, the toluene will be entrained by the injected fuel spray, which should allow visualization of the temperature field in and around the sprays. A representative result of the measurements, and a translation of the LIF ratios to a temperature field on the basis of Figure 5.13 is shown in Figure 5.16 (non-reactive conditions). The recorded images are smoothed by $12 \times 12$ neighbor averaging, which also results in a lateral resolution comparable to the thickness of the excitation laser light sheet. Pixel values below 300 counts (after background subtraction) are discarded.

The temperature field, although qualitatively correct, shows a number of artifacts.
Some of these are due to the mechanical construction of the combustion chamber (the valve seats are clearly visible), some others likely relate to multiple scattering or fluorescence trapping by the sprays. The global temperature field is uniform, but shows a LIF ratio-based temperature that is approximately 250 K above the global temperature derived from the pressure trace. The cold fuel spray regions are clearly recognizable.

![Temperature measurements using Toluene PLIF](a) 280 nm (b) 320 nm (c) Temperature

**Figure 5.16** Fuel injection in nitrogen with toluene recorded at 20 CAD before TDC. Injection started 25 CAD before TDC.

*Post injection*

Temperature stratification after injection of a fuel spray can be measured by adding the toluene tracer to the fuel spray, to the intake air or both. To simultaneously measure fuel stratification, the tracer should be added to the fuel. In the PPC regime, fuel is injected prior to combustion. A difficulty for measuring LIF ratios after injection is the decreasing signal due to increasing temperatures and the decrease in toluene concentration due to mixing. Unfortunately, the poor fuel purity, as mentioned before, prohibited measuring the temperature quantitatively in this way. Therefore only the 320 nm part of post-spray measurements for early injection timings (-70 CAD ATDC) is presented in figure 5.17 for 4 recording timings (-60 to -45 CAD ATDC). The first row gives the single shots (Figures 5.17a to 5.17d). The laser enters the image from the top and the focus point is near the middle of the cylinder, therefore not all sprays and clouds are equally illuminated. Scattering of the laser light might result
in larger illuminated areas downstream. The second row gives the average intensities of 10 experiments (Figure 5.17e to 5.17h) and clearly shows the mixing effect after injection. To investigate the cycle-to-cycle variation, the coefficient of variation (CoV) is plotted and shows the highest variations at the edges of the sprays (Figures 5.17i–5.17l). Furthermore the droplets originating from injector dribble after injection are clearly visible in the images.

5.5.2 High-pressure cell results

To investigate the spray temperature field in more detail than is possible in an engine environment, experiments have been conducted in a high-pressure cell. The purpose of the research is to investigate the major characteristic of temperature distribution while spray propagates into hot ambient gas. The experimental conditions are listed in table 5.3.

<table>
<thead>
<tr>
<th>Table 5.3 The experimental conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection pressure: 900 bar</td>
</tr>
<tr>
<td>Injection duration: 2 ms</td>
</tr>
<tr>
<td>$T_{\text{amb}}$: 590 K</td>
</tr>
<tr>
<td>Ambient density: 20 kg/m$^3$</td>
</tr>
</tbody>
</table>

We present two separate data sets. The first one is obtained by a one-side camera configuration and the second by an opposite side arrangement. The laser sheet thickness converges from 1.8 mm to 0.8 mm from left to right in the HPC, and therefore the spatial resolution at the laser entrance side will be lower than at the far end.

Temperature distribution of sprays in the HPC

One-side camera configuration results  The results presented in this section are for the one-side camera configuration. The obtained spatial resolution is 9.28 px/mm. The images are smoothed by $12 \times 12$ neighbor averaging, to represent the same spatial resolution as the average laser thickness in z-direction in the middle of the HPC (1.3 mm). The fuel doped with 10 vol.% toluene is injected into ambient conditions after pre-combustion of $T_{\text{amb}} = 590$ K, for a duration of 2 ms. The laser frequency is 5 Hz, resulting in only one image during injection. As a result, each pair of images originates from separate injections. Pairs of images are acquired at fixed times with a time shift of 0.5 ms between the measurement series.

As stated before, the beam splitter may influence the ratio because the transmission/reflection coefficients are a function of light incident angle. An example of the fluorescent signal recorded about 2 seconds after spray injection is shown in Figure 5.18a as function of the distance from the left side of the field of view. By this
Figure 5.17 Post spray intensities of the 320 nm wavelength part at -60 CAD (1st column), -55 CAD (2nd column), -50 CAD (3rd column) and -45 CAD ATDC (4th column) with injection at -70 CAD ATDC.
5.5. Results

time, the fan has mixed the toluene, heptane and residual gas to a homogeneous concentration and temperature. The mean value and standard deviation of 13 scaled ratio curves is shown in Figure 5.18b. The scaled ratio on the left hand side is higher than that on the right hand side. The same effect is found for measurement sets with different laser powers and toluene concentrations; therefore, the variation of LIF ratio is attributed to angle-of-incidence variations over the beam-splitter.

By using Figure 5.18b as a correction curve, the temperature distributions of the sprays are calculated, as shown in Figure 5.20. The fuel is injected out of the top right corner and the laser propagates from left to right. The rectangle in Figure 5.20 indicates the extent of the laser light sheet. It does not directly illuminate the liquid part of the spray; for our conditions the liquid length of the spray is estimated to be 27 mm using a mixing-limited spray vaporization model [79]. Other areas may be excited indirectly by laser scattering via the walls.

Increased background levels and undesired fluorescence contribute to the systematic errors of the LIF ratio. To account for these systematic errors the temperature calibration curves have been adapted. The edge of the spray is assumed to have the same temperature as the ambient gas. Based on this assumption, the systematic error is corrected by linearly fitting the calibration relation with a correction factor $C$ resulting in a new temperature calibration function of $T_2(R) = CT(R)$. In this case the correction factor is determined to be $C = 0.84$. Bear in mind that this procedure only gives accurate values at the edge of the spray.

When observing the experimental spray development in a two-dimensional cross-section one can conclude that the temperature field coincides with the simplified and generally accepted view of a diverging spray due to air entrainment, which should result in heating of the fuel from the core region outwards and in downstream direc-
5.5. Results

Temperature measurements using Toluene PLIF. However, at the 1.58 and 2.08 ms point, large gaps in the spray pattern might indicate increased air entrainment and therefore increased temperatures in these regions compared to other nearby regions. The experimental results are compared with LES to verify the findings.

Comparison with LES  First, the simulated spray penetration is compared to the measured one. The fuel vapor penetration in the LES is taken at the point in space where the mass fraction of the fuel is $Y_{fuel} = 0.001$, as defined in the first Engine Combustion Network workshop [33]. For the experiments, the vapor edge is clearly visible and determined visually. The spray penetration length of a single LES and one experiment are shown in Figure 5.19. The first measured point in time ($t_0=0.58$ ms) is fixed on the LES penetration. Subsequently, this is taken as a reference for all other measured points. The error bar of the point at 2.08 ms is defined from the minimum and maximum values from the 9 measurements. Figure 5.19 shows a good agreement between the experiments and the LES.

![Figure 5.19](image)

The results of the LES are also given in figure 5.20. Velocity vectors are colored with the corresponding temperature at that point. Furthermore, the contour of the fuel spray ($Y_{fuel} = 0.001$) is indicated with the solid black line. The LES reveal heavy mixing in a pulsating fashion due to spray induced vortices at the edge, resulting in hot spots of ambient air enclosed in the spray core region. The temperature stratification inside the spray seems to persist a little bit longer in time when comparing the experimental results with LES at time point 2.08 and 2.58 ms. The LES predicts temperatures in the order of 500 K for the inner spray areas whereas the experimental results are much closer to the ambient condition of 590 K. Possible reasons might be the averaging effect in the experiment due to the laser thickness. The assumption on which the
correction factor $C$ is defined might also cause this deviation.

Figure 5.20 Comparison between temperature distribution in the HPC and velocity fields obtained with LES. The black rectangle in the top left image represents the laser sheet area. Toluene outside this rectangle is excited by multiple scattering. The velocity vectors are colored according to temperature. The limited field of view of the experimental setup prohibits visualization of the complete spray at 2.58 and 3.58 ms.
Opposite camera configuration results

Due to failure of one of the two PI-MAX 3 cameras we used the PI-MAX 2 camera instead. However, due to poor performance of the PI-MAX 2 camera, the spray measurements showed extremely poor spatial resolution. Therefore, we only provide vapor results in this section. The toluene is added to the HPC in a cup and vaporized by heating the HPC.

The temperature calibration line in figure 5.21 is obtained by using the same approach as in section 5.4. It differs from the one presented in figure 5.13 because of the lack of dichroic beam splitter. The image also presents two data points obtained from separate measurements using evaporated toluene from a little cup in the HPC at two different ambient temperatures. As expected the ratio calibration points match well with the IFPEN data and once again prove the validity of the calibration approach presented in section 5.4.

\[ \text{320nm / 280nm ratio} \]

\[ \begin{align*}
0.4 & \quad 0.6 & \quad 0.8 & \quad 1 & \quad 1.2 & \quad 1.4 & \quad 1.6 & \quad 1.8 & \quad 2 \\
\end{align*} \]

\[ \text{Temperature [K]} \]

Figure 5.21 Temperature calibration line for the opposite camera configuration. The square and triangle represent two measurement points toluene vapor ratios at known ambient temperature.

5.6 Conclusions and recommendations

The approach to measure temperature distributions during and after fuel injection in an optically accessible engine and high-pressure cell using planar toluene LIF has been elaborately explained. Fluorescence intensities below the required threshold preclude investigating the temperature stratification after fuel injection around top dead center. The initial inaccuracies originating from pixel-to-pixel differences have been successfully corrected and the obtained temperature fields demonstrate the applicability of the technique and reveal the detailed temperature distribution in a spray. The random errors are evaluated and a precision of 4% is achieved. Comparison with LES
is satisfactory and shows similar temperature hot spots and gradients as observed in the experimental results.

This research displays the strengths and weaknesses of toluene planar LIF in detail. Therefore this conclusions section is further split into different items to represent each part of the setup/method.

**Cameras** Images should be obtained preferably using identical camera types, to avoid large differences in optical properties. The intensity regimes in which the cameras respond linearly vary tremendously for the cameras used here. It is highly recommended to check the linearity regimes for the wavelength region of interest and stay in between these values at all times [57].

**Laser power** The results presented here are inconclusive for the dependence of the ratio change on toluene saturation levels. Additional research should confirm if any such effect exists.

**Base fuel** Unfortunately, the spray images showed a substantial background signal level on the 320 nm camera. This originates from impurities in either the HPLC grade n-heptane base fuel, or from impurities left in the fueling system from previous experiments. We recommend to acquire the highest purity base fuel available and only use non-lubricated fuel pumps and fuel lines which have been purged thoroughly.

**Optical configuration** An opposite side camera setup seems to be beneficial over a one-side setup because it avoids the dichroic beam splitter requirement, but the precision is lower because of different light paths. Unfortunately, we were not able to measure fuel sprays due to camera failure. In the one-side setup, the dichroic beam splitter is found to have different transmission/reflection coefficients for different incoming angles. This needs to be corrected for. However, its influence can be reduced by using a dichroic beam splitter of which the transmission slope is exactly in between the band-pass filter transmission curves.

**Flat-fielding** The presented results show that co-flat-fielding using the LIF intensity of the toluene vapor prevents the need for separate flat-fields, and avoids incorporation of the wavelength-dependence of flat-field images. For the one-side setup with a dichroic beam splitter, the co-flat-fielding can be performed by replacing the dichroic beam splitter by a regular beam splitter.

**Signal to noise** The technique shows high enough signal levels in dense spray regions at moderate temperatures. For post-injection studies, the toluene percentage in the
base fuel should be adjusted to maximize the signal levels for the crank angle range of interest. Pixels with counts lower than approximately 500 are affected by noise and therefore result in less accurate ratio calculation.

**Cycle-to-cycle variations** Fuel tracer visualizations show that cycle-to-cycle variations are high at the periphery of the spray and can persist well after injection.

### 5.7 Outlook

Toluene LIF is capable of visualizing temperature distributions with high precision. However, due to the low signal levels at high temperatures, small disturbances from the optical path or fuel purity result in systematic errors which reduce the accuracy. Detailed data sets of spectra recorded at accurate temperatures including the optical pathways are still needed for calibration by anybody interested in applying toluene LIF. Furthermore, the details of the unwanted fluorescence of the heptane base fuel requires further research. The possible effect of saturation on toluene LIF ratios needs to be investigated in more detail. By aiming at saturated toluene fluorescence, signal levels could be increased significantly, possibly enabling the use of the method at higher temperature ranges.

### 5.8 Appendix Data processing

The toluene LIF data processing procedure includes removing background, spatial alignment, flat field correction, ratio calculation and converting to temperature distribution. We obtain an image pair, of which $I_r(x, y)$ is the pixel intensity of the “red” part of the spectrum (320 nm) at position $x, y$ and $I_b(x, y)$ is the pixel intensity of the “blue” part of the spectrum (280 nm). In case of one-side configuration, the data correction procedure for both images can be expressed as:

$$I_{r,corr}(x, y) = (I_r(x, y) - B_r(x, y)) \cdot TR(x, y) \cdot F_r(x, y)$$  \hspace{1cm} (5.4)

and

$$I_{b,corr}(x, y) = (I_b(x, y) - B_b(x, y)) \cdot F_b(x, y),$$  \hspace{1cm} (5.5)

where $B(x, y)$ is the background signal, $TR(x, y)$ the spatial transformation matrix and $F(x, y)$ is the flat field matrix. In case of opposite configuration, the data processing procedure for both images can be expressed as:

$$I_{r,corr}(x, y) = (I_r(x, y) - B_r(x, y)) \cdot TR(x, y) \cdot C_r(x, y)$$  \hspace{1cm} (5.6)
and
\[ I_{b,\text{corr}}(x, y) = (I_b(x, y) - B_b(x, y)), \quad (5.7) \]

where the major difference is the co-flat-fielding matrix \( C_r(x, y) \). To only present temperature regions which have reasonable accuracy, pixels with less than 300 counts are neglected because of the low signal-to-noise ratio. The temperature distribution can be obtained with the calibration relation.

\[ T(x, y) = T(R(x, y)) = T\left( \frac{I_{r,2}(x, y)}{I_{b,2}(x, y)} \right) \cdot BS(x, y), \quad (5.8) \]

where \( R \) is the LIF ratio distribution and \( BS(x, y) \) is the correction for beam splitter, which is only used in one-side configuration of the HPC experiments.
Correlation of flame location and ignition delay

This chapter is an integral copy of the article which is accepted for presentation at the SAE Powertrain, Fuel and Lubricants conference. Minor changes have been made to improve the quality. It is published as:

6.1 Introduction

Partially premixed combustion (PPC) strategies rely on a certain amount of ignition delay (ID) to achieve a premixed charge. This delay can be accomplished in several ways, as described extensively by Dec [17]. More specifically, the ignition delay can be increased by choosing a less reactive fuel, lowering the compression ratio, supplying residual exhaust gas or using a multiple injection strategy. In this research partially premixed combustion is investigated in an optically accessible engine, using primary reference fuels (PRF’s) and "artificial" exhaust gas recirculation (EGR). The used PRF’s are blends of n-heptane and iso-octane. The use of PRF’s is inspired by the fact that reduced chemical kinetic mechanisms exist for these fuels, with a limited amount of species and reactions, so that they can be applied readily in computational fluid dynamic (CFD) simulations. However, it must be recognized that the ignition behavior of the applied PRF’s will differ from real diesel and gasoline mixtures for mixing-controlled processes like PPC, due to their significantly higher volatility [28]. For example, the boiling point of n-heptane is 98 °C, which is substantially lower than a typical boiling trajectory like 179-366 °C for diesel [90]. The heat of evaporation, on the other hand, is 316 kJ/kg for n-heptane, which is higher than the value of 250 kJ/kg for diesel. Weber et al. [109] studied the difference between n-heptane and commercial diesel combustion. They concluded that the ID is similar, however with a much higher heat release rate for the n-heptane fuel. This results in a higher mean gas temperature and therefore higher nitric oxide (NOx) and lower CO emissions. The unburned hydrocarbon (UHC) emissions are slightly higher if n-heptane is used as a surrogate fuel, which might originate from its higher volatility as well. A multiple injection strategy increases the control over the amount of premixing. This control is needed to enable PPC combustion strategies and to optimize emission formation across the full load range. Injection of fuel before the main injection provides control over the charge stratification, which can be accomplished by changing the timing and duration of these so-called pilot injections. Early pilots can decrease soot emissions without significant influence on NOx emission because of enhanced pre-mixing [23]. In general the HC emissions increase due to presence of unburned fuel in over-lean regions and fuel trapped in crevices. If early pilots are applied using nozzles with a wide umbrella angle the durability of the cylinder liners, piston rings and oil due to wall-wetting might be affected [23]. Injections after the main injection (so-called post injections) reduce soot emissions by enhanced soot oxidation as described by Park et al. [78] and Bobba et al. [8]. The latter authors also concluded that the optimal post injection timing is approximately 40 CAD after top dead center (TDC) and is independent of the main injection timing (at 9-10 bar IMEP load). Payri et al. [80] suggested that the optimum injected fuel mass in the post-injection pulse is around 15-20 mg/cycle for partial and full load cases, which means about 7-10% of the total mass for full load conditions or 15-20% for partial load modes. Post-injections are also claimed to be effective in the reduction of HC emissions in premixed charge compression ignition that are induced by the over-lean mixtures near the nozzle [11].
By applying post injections, combustion in these regions near the injector is enhanced, leading to lower HC emissions. Still the exact amount and timing of the post injection is of major importance [11]. The high unburned HC concentration around the injector region for a PPC combustion mode with US diesel is confirmed by Kim et al. using laser-induced fluorescence (LIF) techniques [42]. Nevertheless it should be emphasized that the results cited so far are not generic and difficult to compare due to different engine specifications, injectors, fuels, pressures and other engine parameters. In this research the specific objectives are to determine the effect of fuel reactivity, oxygen concentration and equivalence ratio on the combustion characteristics and emissions in relation to ignition delay. Furthermore, the position of the flame front as function of ignition delay is investigated using OH* chemiluminescence. From the flame front determination, an approximate premixed combustion volume is determined, including an estimate of the corresponding local equivalence ratios. First we explain the optically accessible engine setup and the optical diagnostic methods used. The results section is divided in the parts for varying oxygen concentration, fuel reactivity and equivalence ratio followed by the emissions as function of ignition delay. After these generic results, the spectra of the OH* chemiluminescence are presented followed by the analysis of flame location versus ignition delay and versus load sections. The last part of the results section is dedicated to preliminary results of multiple injection strategies.

6.2 Materials and methods

6.2.1 Optically accessible engine

The single cylinder optically accessible heavy-duty diesel engine used in this study is the same as in previous chapters. A schematic view is presented in Figure 6.1. Artificial exhaust gas is added to the inlet using nitrogen and carbon dioxide from pressurized cylinder packages.

A common rail is added to the fueling system to damp pressure fluctuations. The injection pressure is measured by a sensor between the common rail and the injector. The system is able to inject 5 times per cycle. The Delphi MX injector specifications used are shown in Table 6.1. For SOA later than 35 CAD before TDC the spray is always directed into the piston bowl. Earlier injections might lead to more unburnt HC’s or wall-wetting due to fuel impact onto the piston rim. If fuel is injected near TDC (Figure 6.2), the spray will be pointed approximately 5 mm above the sapphire window position.

The elongated part of the piston is neither cooled nor lubricated; consequently, special PEEK polymer piston and guidance rings are used. Specifications can be found in Table 3.1 and the engine operated with a compression ratio of 15.45.

The combustion parameters are acquired and analyzed using a high-speed acquisition system (Smetec Combi). The used timing definitions are sketched in Figure 6.3.
6.2. Materials and methods

**Correlation of flame location and ignition delay**

**Figure 6.1** Schematic overview of the optically-accessible heavy-duty engine test rig.

**Table 6.1** Injector specifications (Delphi Diesel Systems). Nominal flow is given as measured by Delphi Diesel Systems at 100 bar

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holes</td>
<td>6</td>
</tr>
<tr>
<td>Hole size</td>
<td>194 µm</td>
</tr>
<tr>
<td>Nominal flow</td>
<td>1.4 l/min</td>
</tr>
<tr>
<td>Spray angle</td>
<td>140 degrees</td>
</tr>
</tbody>
</table>

**Figure 6.2** Piston position at 350 CAD.

Ignition delay is determined as the time between start of combustion (SOC) and start of injection (SOI); ID=SOC-SOI. The CA5 point defines the SOC and SOI is defined...
as the start of actuation (SOA) plus a physical opening delay of the injector of 2 CAD (SOI=SOA+2). The mixing period (MP) is defined as the time between the start of combustion and the EOI; MP=SOC-EOI.

Figure 6.3 Schematic overview of the timing definitions. In this drawing the MP is negative.

The pressure oscillations measured by the in-cylinder pressure transducer arising from the sound waves occurring during rapid combustion are consistent with research by Vressner et al. [107] and are zero-phase filtered using a fourth-order Butterworth filter. Due to the optical design of the setup, continuous firing would lead to a too high (thermal) load on the engine and sapphire window. Therefore, a burst firing scheme is used. The injector is only allowed to inject during recording using a relay in between the ECU and the injector. During burst-firing, the thermal conditions change and therefore also the liner and piston temperatures. Consequently, the charge temperature will also increase during the burst and this will affect auto ignition. The number of cycles needed to get a satisfactory steady-state situation was determined earlier to be at least 10. To determine the emissions in such a burst-firing scheme and to investigate cycle-to-cycle fluctuations, fast response exhaust gas analyzers (Horiba MEXA 1400FR) are used. The exhaust gases are sampled close to the exhaust valves and guided by an electrically heated sampling line at 190 °C. The setup allows cycle-resolved emission concentration measurements, but with a consistent delay of 2 cycles at 1200 rpm. The recorded HC and NO values from the fast-response analyzers stabilize within 10 cycles [114]. The CO emissions however do not stabilize completely and seem to fluctuate around a steady mean after the 10th cycle. Due to the burst firing scheme, a smoke meter cannot be used to measure soot in the exhaust. To achieve consistent results for all engine settings, a standard of 15 combusting cycles per acquisition is used throughout the study, of which the last 5 cycles are used for data analysis.

6.2.2 Optical diagnostics

Natural light emission from combustion consists of soot incandescence and chemiluminescence from reaction species. Soot incandescence is the dominant source of luminosity over a broad wavelength range during the combustion of traditional diesel fuel
Correlation of flame location and ignition delay

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and visualizes regions of soot formation and high temperatures during the combustion process. On the other hand, chemiluminescence arises from spontaneous, non-thermal radiation of excited gas-phase molecules that are formed during the combustion process. These excited intermediate species (for example OH*, CH*, C2*, CO2*, HCO*, CH2O*) emit light in a relatively narrow spectral range as they relax to their ground state. The most studied chemiluminescence is the relaxation of electronically excited hydroxyl radicals, OH*, (emission peak near 310 nm) resulting mainly from hydrocarbon atoms reacting with oxygen [13]. The OH* chemiluminescence is well correlated with the rate of heat release and is a marker of premixed combustion [63]. By using a spectrograph the emission spectrum can be monitored to ensure that the signal is not polluted by other intermediate species (HCO) or broad-band radiation of soot [106]. By using band-pass filters, intermediate species formation can be visualized in 2D (line of sight) when using UV sensitive intensified CCD cameras, as presented by Mancaruso et al. for OH*, HCO* and CH* [63]. Figure 6.4 shows a schematic overview of the spectral visualization setup. A UV Nikkon f/4.5 105 mm lens was mounted in front of a spectrograph (Chromex 250is) with a UV-blazed grating of 1200 grooves/mm that focuses the field of view (FOV) of the combustion chamber via a 45 mirror onto the entrance slit. By adjusting the orientation of the grating, the light from the FOV can be dispersed in the desired wavelength range. Between the high-speed camera (HSC) and the spectrograph, a high-speed Intensified Relay Optics unit (IRO) from LaVision is connected to amplify the weak spectral signal and to make the UV range visible for the UV insensitive camera. The CMOS high-speed camera is a Vision Research Phantom V7.1 with a 12 bit dynamic range, operated at a resolution of 512x512 pixels. The width of the spectrograph entrance slit, the gating time and gain of the IRO can be adjusted to control signal intensity. With this setup and selected spectrograph grating a spectral range of 27 nm can be recorded simultaneously. The data acquisition is triggered and synchronized with the crank angle positions by a high-speed controller from LaVision.

Figure 6.4 Spectral visualization set-up.
The field of view is indicated in Figure 6.5a and covers a narrow strip from the center of the combustion chamber to the piston edge in the region of one spray. The direction of the spray deviates by a small angle from the FOV. However, when looking to the light luminosity during combustion the FOV does cover the burning area (Figure 6.5a).

![Figure 6.5 Field of view (FOV) of the spectrograph. Picture through the piston window. a) during post injection (fuel spray visible); b) during combustion (soot cloud luminosity). The swirl is clockwise oriented.](image)

A typical OH* emission recording is shown in Figure 6.6. The horizontal axis represents the distance from the injector (0 mm) to the piston edge (45 mm). The vertical axis represents the wavelength range observed. The spectral information reveals two intensity peaks in the middle of each image. This is the typical OH* emission at 306.5 and 309 nm. The OH*-emission is integrated over a wavelength range of 305-310 nm, the background is subtracted and the curve is filtered to remove noise. The trailing edge of OH*, used to define the flame front position, is defined as the point where the tangent line of the steepest gradient crosses 0, depicted by the vertical lines in Figure 6.7. The thus defined flame position has quite large cycle-to-cycle variations but does not systematically depend on the cycle number.

### 6.2.3 Artificial EGR system

Due to burst-firing, the engine does not produce stable exhaust gases to use for recirculation; therefore EGR needs to be artificially supplied. Simulating EGR with increasing complexity influences the combustion results. However, a simplified EGR composition is sufficient because it allows the oxygen concentration and specific heat ratio to be well matched. This produces acceptable deviations compared to "real" exhaust EGR, as shown by Colban et al. [14]. In this study, we use the exhaust gas equivalent (E-EGR) to define the oxygen mass percentage \( [O_2]_{mass} \) of the intake air.
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Figure 6.6 Typical OH*-emission recording, 4 bar IMEP, n-heptane, CA50=366 CAD, entrance slit=200 řm, IRO gate=100 řs, gain=65.

Figure 6.7 Flame front determination based on trailing edge of OH* emission.

The resulting artificial EGR (A-EGR) dilution percentage is defined as the volumetric dilution of the intake air with a N2/CO2 mixture. The N2/CO2 ratio is fixed ratio at 2.63 corresponding with the simple EGR case of Colban et al.[14]. The artificial dilution is equivalent to a much higher real equivalent exhaust gas recirculation level, since in practice the recirculated exhaust gas still contains a certain level of oxygen depending on the operating point. For reference both volumetric percentages (A-EGR and E-EGR) are indicated in Table 6.2 for a specific operating point at which most experiments have been conducted. The resulting overall equivalence ratio (φ) is also depicted in the table.
Table 6.2 Artificial EGR dilution at 1200 rpm, Pin=1.1 bar, 45 mg n-heptane injection per stroke, 4 bar IMEP.

<table>
<thead>
<tr>
<th>E-EGR:</th>
<th>10 %</th>
<th>20 %</th>
<th>30 %</th>
<th>40 %</th>
<th>50 %</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-EGR</td>
<td>3 %</td>
<td>7 %</td>
<td>11.5 %</td>
<td>19 %</td>
<td>27 %</td>
</tr>
<tr>
<td>$[O_2]_{mass}$</td>
<td>22%</td>
<td>19.5%</td>
<td>18.6%</td>
<td>17.1%</td>
<td>15.4%</td>
</tr>
<tr>
<td>$\phi$</td>
<td>0.28</td>
<td>0.32</td>
<td>0.33</td>
<td>0.36</td>
<td>0.4</td>
</tr>
</tbody>
</table>

6.2.4 Operating parameters

All experiments have been conducted at 1200 rpm, an inlet pressure of Pin=1.1 bar and an intake air temperature of approximately 293 K. The fuel injection pressure is kept at 1000 bar and the duration is adjusted to achieve approximately 4 bar IMEPnet for all cases unless explicitly stated otherwise. The low load is selected in order to prevent damage to the optical setup. The point where 50% of the heat is released (CA50) is kept constant at approximately 6 CAD after TDC by adjusting the start of actuation (SOA). The notations of the used blends and their corresponding cetane numbers (CN) are summarized in Table 6.3.

Table 6.3 PRF compositions in volume% and the corresponding cetane number.

<table>
<thead>
<tr>
<th>Notation</th>
<th>PRF0</th>
<th>PRF25</th>
<th>PRF50</th>
<th>PRF75</th>
<th>PRF100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iso-octane</td>
<td>0 %</td>
<td>25 %</td>
<td>50 %</td>
<td>75 %</td>
<td>100 %</td>
</tr>
<tr>
<td>N-heptane</td>
<td>100 %</td>
<td>75 %</td>
<td>50 %</td>
<td>25 %</td>
<td>0 %</td>
</tr>
<tr>
<td>CN</td>
<td>56</td>
<td>45.5</td>
<td>35</td>
<td>24.5</td>
<td>14</td>
</tr>
</tbody>
</table>

6.3 Results

In this section we first present the influence of changes in oxygen concentration (EGR percentage) and fuel reactivity on combustion timing and emissions. Secondly, we analyze the change in combustion behavior when varying the equivalence ratio by differing the amount of injected fuel, while keeping other parameters fixed. A constant 17% oxygen mass concentration is used to keep the pressure rise rate limited. Thirdly, the resulting engine-out emissions for varying PRF’s and EGR percentages are re-examined as function of ignition delay. Furthermore the flame location as a function of ignition delay is presented. The results section is concluded with results on natural luminosity and the corresponding engine out-emissions when applying multi-injection strategies.
6.3. Results

6.3.1 Varying oxygen concentration

The artificial exhaust gas dilution is varied between 0% and 50% E-EGR with PRF0 as a fuel. The start of actuation is advanced with decreasing oxygen concentration to keep the CA50 point as close as possible to 6 CAD after TDC (aTDC). The resulting pressure traces are shown in Figure 8. These show a decreasing slope for higher EGR, which also follows from the difference between CA5 and CA50 points in Figure 6.9. The maximum pressure rise rate (PRR) decreases with increasing EGR, indicating a slower combustion. The CA90 point, however, increases proportional to the CA50, meaning that the combustion rate between these points is fairly constant with changing EGR%. The advancing SOA leads to a larger ignition delay and corresponding mixing period.

Figure 6.8 Pressure traces for varying EGR percentages, fuel=PRF0.

Figure 6.9 Combustion characteristics for varying EGR percentages, fuel=PRF0.
6.3. Results

Correlation of flame location and ignition delay

The increased ignition delay has a large influence on the engine-out emissions (Figure 6.10). CO and HC values are significantly higher for increased dilution, but NO-emissions decrease due to the lower local combustion temperatures.

![Figure 6.10 Emissions as function of E-EGR %, fuel=PRF0.]

6.3.2 Varying fuel reactivity

The fuel reactivity has been varied by changing the ratio between iso-octane and n-heptane. The oxygen concentration was kept at 23% (mass percentage, so 0% EGR). The pressure traces for changing PRF ratio are shown in Figure 6.11. For increasing iso-octane (volume) percentage, the initial pressure rise originating from the n-heptane part of the fuel is advanced because of the earlier SOA. This two-stage ignition behavior has also been described by Yang et al. [112]. It should be noted that for the PRF75 blend, the injected amount of fuel had to be increased to approximately 56 mg per injection, as compared to approximately 43 mg for the other fuel blends. Due to overleaning, the engine failed to ignite while aiming for 4 bar IMEP. Even at higher loads the first cycles displayed very low efficiencies and had a retarded CA50. This is likely to reduce the heating of the liner and piston during the first cycles of each burst and therefore will have less stable boundary conditions as compared to the PRF0 to PRF50 points. Injecting more fuel led to a relatively higher equivalence ratio, whereby the ignition of the n-heptane raised the temperature enough to combust the iso-octane part of the fuel.

The CA50 points was kept as close as possible to 6 CAD aTDC, in which case also the CA5, CA10 and CA90 points were found to be relatively constant (Figure 6.12). As expected, an increase of ignition delay is observed for less reactive blends. The maximum PRR decreases slightly for the PRF50 and PRF75 point, however no clear trend is observed.

Figure 6.13 shows the corresponding emissions. The resulting extra mixing time leads
6.3. Results

![Figure 6.11](image)

Figure 6.11 Pressure traces for PRF 0, 25, 50 and 75.

![Figure 6.12](image)

Figure 6.12 Combustion characteristics for varying fuel reactivity.

to a similar trend as for the EGR dilution: with increasing iso-octane concentration in the blend, HC and CO emissions rise. On the other hand, NO\textsubscript{x} production does not show a clear trend. Because the emissions are given in gram per gram fuel, the emissions of the PRF75 break the trend of rising CO and HC emission as function of PRF due to the higher amount of fuel injected.

6.3.3 Varying equivalence ratio

To investigate the effect of increasing global equivalence ratio on the emissions and combustion parameters, the injection duration (\(t_{\text{inj}}\)) is increased, which obviously increases the load as well. All tests in this series were performed with PRF0 at a constant SOA of 348 CAD and with a constant oxygen concentration of 17% (40%...
E-EGR. Other parameters are given in Table 6.4.

Table 6.4 Injected fuel mass ($m_{fuel}$) and overall (global) equivalence ratio ($\phi$) as function of injection duration ($\tau_{inj}$) with an oxygen concentration of 17%, fuel=PRF0.

<table>
<thead>
<tr>
<th>$\tau_{inj}$ (ms)</th>
<th>0.9</th>
<th>1.25</th>
<th>1.6</th>
<th>1.95</th>
<th>2.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_{fuel}$ (mg)</td>
<td>45</td>
<td>64</td>
<td>80</td>
<td>103</td>
<td>117</td>
</tr>
<tr>
<td>$\phi$</td>
<td>0.37</td>
<td>0.53</td>
<td>0.66</td>
<td>0.85</td>
<td>0.97</td>
</tr>
</tbody>
</table>

Pressure traces for increasing fuel injection duration are given in Figure 6.14. Obviously, injection of more fuel results in an increased maximum pressure. This stabilizes after an overall equivalence ratio of 0.66, presumably due to the occurrence of locally fuel rich zones resulting in incomplete combustion and elongated combustion duration.

The derived combustion parameters are depicted in Figure 15 as function of overall equivalence ratio. The CA5 and CA10 points are relatively constant and therefore the ignition delay is constant with increasing equivalence ratio. The CA50 and CA90 points are retarded, as is expected when injecting more fuel. For the equivalence ratio of 0.97 the injection duration of 2.2 ms (15.8 CAD) combined with the 2 CAD injector opening delay (SOI-SOA) is too long to achieve a positive mixing period and therefore a proper PPC mode, more a conventional diesel combustion with a huge premixed phase. The slight overlap between injection and combustion results in a conventional diesel mode. The pressure rise rate reaches a maximum for an equivalence ratio around 0.7. Longer injection duration decreases the PRR slightly, most likely because of the vaporization of the fuel due to a lack of a positive mixing period.

For injection durations of 1.95 ms ($\phi$=0.85) and 2.2 ms ($\phi$=0.97), the spray regions will have high local equivalence ratios, resulting in the formation of CO which cannot be oxidized efficiently due to the low temperatures during expansion, as also proven by multi-zone simulations [25]. The existence of fuel rich zones is confirmed by the

Figure 6.13 Emissions for varying fuel reactivity.
exhaust emission of CO, shown in Figure 6.16 together with the lack of HC emissions. The CO emission shows a U-shape curve, indicating the point with maximum combustion efficiency. NO-emissions are about constant.

The HC emission decreases as function of equivalence ratio, indicating that an important cause of HC emissions is over-leaning of certain regions in the cylinder at low equivalence ratios. This shows the importance of dealing with these lean-areas for low loads by, for instance, multiple injection strategies, which will be discussed in a later section. An additional reason for the decrease might be the decreased relative importance of injector dribble on the HC emission in gram/gram fuel, when assuming a constant dribble quantity.
6.3. Results

6.3.4 Emissions versus ignition delay

As shown above, ID has a large influence on emissions. Leermakers et al. [54] showed a strong correlation between ID and emission of unburned hydrocarbons and CO for a full-metal engine. To investigate whether such a correlation exists for this optical engine, all emission results obtained with varying EGR percentage and fuel composition are plotted as function of ignition delay in Figures 6.17 to 6.19. These measurement points were all produced at a load of approximately 4 bar IMEP and at constant inlet pressure. PRF75 results are excluded since these were measured at higher load. The emissions of CO (Figure 6.17) and HC (Figure 6.18) for all measurement point nearly collapse on a trend line with a positive slope as a function of ID. The higher CO and HC emissions are explained by over-leaned regions arising at large ID. A decreasing trend is shown for NO emissions as function of ID (Figure 6.19). However there is not a single trend line on which all data collapse as with CO and HC emissions. In PPC mode longer mixing drastically drops local temperature. EGR adds on top of that a higher specific heat so that even if mixing time is equal differences in EGR will further reduce NO. UHC and especially CO will show an opposite trend (need high T). UHC you will also see crevice effect (assuming in crevices CO is not even formed). Comparing Figures 6.17, 6.18 and 6.19 seem to corroborate this.

6.3.5 OH* chemiluminescence

The increased mixing with increasing ignition delay is visualized by recording OH* chemiluminescence. Figure 6.20 shows the OH*-emission recordings for several CA positions of PRF0 combustion with 0% EGR. In this figure the intensities are averaged over 5 cycles and plotted as function of wavelength. The observed signal has two typical OH*-emission peaks at 306.5 and 309 nm. At 5 CAD aTDC, the first signal is observed, corresponding with CA5, after which the signal increases rapidly at 6 CAD.
6.3. Results

Figure 6.17 CO emission results as function of ID.

Figure 6.18 UHC emissions result as function of ID.

Figure 6.19 NO emission results as function of ID.
From 7 CAD on, most of the fuel has combusted and OH* intensities decrease again. The observed background signal is most likely HCO chemiluminescence [63].

![OH* emission intensity as function of wavelength for various CA, PRF0, EGR0.](image)

**Figure 6.20** OH*-emission intensity as function of wavelength, for various CA, PRF0, EGR0.

To investigate whether the trailing edge position (the inner boundary) of the OH* signal (see Figs. 4 and 5) changes during combustion, its position is plotted in Figure 6.21 for two situations (PRF0EGR0 and PRF25EGR30) and for the relevant CAD range in which significant OH* chemiluminescence is observed. For the latter situation, combustion starts further away from the center of the cylinder after which it progresses to a relatively steady position for crank angle degrees around CA50 which is close to 6 CAD aTDC. The vertical error bars show the standard deviation of 5 measurement cycles and is quite large, especially for the 5 CAD aTDC point of the PRF0 fuel without EGR.

For further investigation, we use the +6 CAD point to determine the flame position.

### 6.3.6 Flame location versus ignition delay

The flame position for (partially) premixed combustion can provide useful information regarding the trends in emissions. In particular, HC emissions might find their origin near the injector due to local over-lean fuel regions. Therefore, the flame location is further investigated as a function of ignition delay.

Figure 22 shows the (quasi steady-state) OH* position as function of the ignition delay for different combinations of fuel and inlet dilution. The error bars depict the standard deviations, which are a result of large cycle-to-cycle variations. The shortest ID is produced by PRF0 and no EGR dilution. The longest ID in the figure results
6.3. Results

**Figure 6.21** Minimal distance of OH* from the injector as function of CAD for PRF0 without EGR (SOA -10 CAD aTDC) and PRF25 with 30% EGR (SOA -19 CAD aTDC).

**Figure 6.22** OH* position as a function of ignition delay.

from PRF50 and 30% EGR. It can be noticed that some points have the same ID with different settings; for example PRF50 with 7% inlet dilution and PRF25 with 18.5% dilution both create an ID of 22 CAD. The observed distance of OH* is qualitatively depicted in Figure 6.23. For larger ignition delays the injected amount of fuel has more time to mix with air towards the middle of the bowl (after rebounding from the piston bowl rim) before SOC. Therefore OH* is first observed closer to the centerline depicted by the dashed-dotted line in Figure 23 which corresponds with the 0 mm point in Figures 6 and 7. To estimate the local equivalence ratio in the burning region, the combusting cloud is approximated by a sphere with a diameter (D) equal to the distance between the piston wall and the flame edge. This is defined as the combustion volume (CV) represented by the circles in Figure 23.
6.3. Results

Correlation of flame location and ignition delay

For the shortest and longest delays, the approximations of the combustion volume and both global and local equivalence ratios are presented in Table 6.5. In this approximation it is assumed that all fuel mass ($m_f$) is trapped in the combustion volume $CV$. The oxygen mass in the combustion volume ($m_{O_2}$) is only a small part of the total oxygen available in the cylinder at TDC.

Table 6.5 Flame diameter ($D$), corresponding combustion volume ($CV$), fuel mass ($mf$), oxygen mass ($m_{O_2}$), global equivalence ratio and local equivalence ratio approximations for the shortest and longest ID.

<table>
<thead>
<tr>
<th></th>
<th>$D$</th>
<th>$CV$</th>
<th>$mf$</th>
<th>$m_{O_2}$</th>
<th>$\phi$</th>
<th>$\phi_{cv}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PRF0EGR0</td>
<td>14.9</td>
<td>10.4</td>
<td>45</td>
<td>42.6</td>
<td>0.27</td>
<td>3.7</td>
</tr>
<tr>
<td>PRF50EGR30</td>
<td>23.1</td>
<td>38.8</td>
<td>45</td>
<td>128</td>
<td>0.31</td>
<td>1.1</td>
</tr>
</tbody>
</table>

While the average equivalence ratio in the cylinder increases with increasing EGR percentage for the same amount of fuel injected, the local equivalence ratio in the combustion volume ($\phi_{cv}$) decreases due to the larger combustion volume as a result of the longer ignition delay. Even when accounting for the air dilution, the $\phi_{cv}$ ratio decreases from 3.7 to 1.1 if combustion is delayed. While this is only a crude approximation, it illustrates how a longer ignition delay leads to lower local equivalence ratios via a larger mixing time/volume. However, as shown in Figure 18, this larger mixing time shows higher HC emissions. Since this, according to the above analysis, cannot result from richer conditions in the combustion volume, it is most probably the result of over-leaning near the middle of the piston bowl which is far more probable if the CV is large.
6.3.7 Flame location versus load

Following the same method as described in the previous section, the increase in combustion volume was also measured as a function of the fuel injection duration (i.e., load variation). The fuel used is PRF0. The OH* position as a function of equivalence ratio is shown in Figure 6.24 for two different crank angle degrees close to the CA50 point. The mixing period as given in Figure 15 is short enough to avoid a lifted diffusion flame at the recorded CA.

![Figure 6.24 OH* position as function of global equivalence ratio for the +7 and +8 CAD aTDC.](image)

The size of the flame region increases with fuel injection duration. This trend can be explained by the increased jet momentum due to longer injection. This supports the jet to penetrate further to the middle (during the ID) after rebounding from the piston bowl rim. By increasing the injection duration from 0.9 to 2.2 ms, the diameter of the flame region (OH*-emission) enlarges from 18 mm to 25 mm. The resulting estimated local equivalence ratios are shown in Table 6.6, showing that even though the combustion volume increases, the local equivalence ratio rises with injection duration. However, the rise is only a factor of 1.42 in local equivalence ratio, while the average equivalence ratio increases with a factor of 2.63 over this range of injection duration. This difference is the result of the larger CV due to the longer injection. While this is only a rough approximation, the emissions confirm the higher $\phi_{cv}$ by showing more CO emissions (see Figure 16) and increased soot formation, observed as soot deposition on the piston by visual inspection.

6.3.8 Multiple injection strategies

In the previous parts it was shown that a longer mixing time, as expressed by ignition delay, leads to higher CO and HC emissions. As a possible explanation for the higher
HC emission, we mentioned the lean mixture near the injector. Chartier et al. [11] managed to reduce the UHC emission, caused by overleaning near the injector, by adding a small post injection of fuel after the main injection. As mentioned before, pilot injections may be used to create a relatively homogenous mixture prior to the main injection and therefore enable the control over the combustion phasing. During pilot and post injection strategies experiments the engine was operated with a lower compression ratio of 13.9, compared to 15.45 for all previous measurements. This is due to constraints of test cell availability. All experiments were performed with PRF0.

### Pilot injections

The different pilot timings (-50 to -20 CAD aTDC) with injection durations of 0.4 ms contained approximately 33% of the fuel injected and are combined with a main injection of which the SOA is adjusted between -6 CAD for the -20 CAD pilot injection and -8 CAD for all other pilot injection timings, to keep the CA50 point as constant as possible. The changing pilot injection timings are compared with only a single main injection with a SOA of -8 CAD, hereafter referred to as "main only". Representative natural luminosity images of a pilot injection (see Figure 6.25) show a great variation in combustion areas across the field of view, which is not repeating in other cycles and can therefore be attributed to cycle-to-cycle variations.

![Figure 6.25](image-url) Representative natural luminosity recordings for a combined pilot injection (-30 CAD) and main injection (-6 CAD).
The natural luminosity at each CAD is averaged for five cycles and plotted in Figure 6.26. It shows a decrease in luminosity for advanced pilot injection timing. The high natural luminosity for the -20 CAD pilot injection timing can be ascribed to an increase in stratification. The higher overall natural light intensity for the pilot injection recordings compared to the main only combustion suggests more soot formation.

![Figure 6.26 Natural luminosity as function of CAD for different pilot injection timings.](image)

The single pressure traces for the last combustion cycles (of each series of 5 that were averaged in Fig. 6.25) are given in Figure 6.27 and show a late start of combustion for the main only point. The pressure traces of the -40 and -50 CAD pilot injection are almost identical. The pilot injections already start to release heat before the main injection occurs, as can be noticed from the higher pressure traces for advanced pilot injection. As a consequence, this results in smaller ignition delays for the main injection. Figure 6.28 shows the CA5, CA10, CA50 and CA90 points which are relatively stable and do not present clear trends with pilot injection timing. The presented ID and MP are determined for the main injection and increase with advancing the pilot injection. This corresponds with a higher PRR due to a more homogenous charge. Due to the stratification at retarded pilot injection timings, combustion starts at a specific point and then gradually ignites other regions. However, with these low average equivalence ratios, it is expected to produce little or no soot after oxidation. This is also confirmed visually by a lack of deposits in the combustion chamber.

**Post injections**

Adding a post injection might decrease the HC emission by "refueling" overlean regions. However it is not likely to reduce soot; it might even lead to higher soot production because the post injection does not burn in premixed-mode, rather showing a conventional diesel flame. The effect of post-injections on emissions was investigated
by keeping the main SOA constant at -10 CAD aTDC and varying the post injection timing from 0 to +40 CAD in increments of 5 CAD. The injection duration was 0.72 ms (67% of the fuel) for the main injection and 0.4 ms (33% of the fuel) for the post injection. Representative natural luminosity images for a combined main and post injection recording is shown in Figure 6.29. The post injection (+15 CAD) enters the hot volume and starts to burn almost immediately (around +20 CAD) with high luminosity.

Figure 6.30 shows the averaged natural luminosity over 5 combustion cycles for changing post injection timings. A large increase in natural luminosity is observed during and after the post injection due to a negative MP for the post injection, in combination with already high temperatures due to combustion of the main injection.
6.3. Results

Figure 6.29 Representative natural luminosity recordings for a combined main (-10 CAD) and post (+15 CAD) injection.

The slope of the intensity increase after the post injection decreases for later post injection, suggesting an increased mixing time due to lower temperatures. Individual pressure traces are shown in Figure 6.31 and show a slight increase in pressure shortly after the post injection starts.

The derived combustion characteristics (Figure 6.32) show some deviation in CA5 and CA10 due to a difference in the location of maximum heat release used to determine these points. This also translates to the same trends for the ID and MP. The CA50 and CA90 points retard for later post injection timings except for the +35 CAD point. Probably the injector behaved differently due to pressure fluctuations in the common rail. The pressure rise rate is stable because it is the largest for the combustion of the main injection which is constant.

The emission results for the combined pilot/post and main injections are shown in Figure 6.33. The emissions at the -10 CAD point are from a main only injection and given as a reference. The figure shows that the addition of a post injection does not improve CO and HC emission. The NO emission decreases with respect to a main only injection, because of lower temperatures due to a decreased amount of fuel in the main injection and lower combustion efficiencies. CO and HC emissions increase for later post injections, implying lower fuel efficiencies. This is due to a less efficient combustion as result of a decrease of temperature. For the pilot injections, the CO and HC emissions are found to increase for more advanced pilot timings. A reason is that the fuel has more time to move into crevices and zones where flame quenching occurs. High HC emissions results in lower temperatures and therefore decreased conversion rates of CO to CO2. This is definitely the case for a pilot SOA before -35 CAD, because the wide-angle injector will spray on top of the piston edge and propagate fuel particles into the crevices. This is also clearly seen by the increase in HC emissions for the pilot injections before -35 CAD. An additional explanation might be the increased injector dribble for a multiple injection strategy and therefore increased HC emissions. NO increases with high combustion efficiency and low HC and CO.
6.4 Conclusions

Combustion characteristics and emissions of PPC combustion were investigated in an optically accessible engine, by changing the fuel reactivity using different PRF blends and the amount of artificial EGR. By combining all emission results, clear trends of CO and UHC emissions as function of ignition delay are observed, comparable with full metal engine tests presented in literature. For NO\textsubscript{x} emission the results indicate no clear trend with ignition delay when the latter is accomplished in different ways (i.e. different trends are found for varying EGR percentage and fuel reactivity). High-speed visualization of OH\textsuperscript{*} was used to determine the edge of the flame location closest to the injector. The inner boundary of the area in which OH\textsuperscript{*} is visible clearly correlates with ignition delay. Using a simple approximation for the size of the combustion volume, local equivalence ratios were derived which were shown to be
helpful in explaining/correlating observed emission results. Finally, natural luminosity measurements of multiple injection strategies show increasing light emission for both pilot and post injection strategies. Emission analysis results of pilot injection strategies as presented in this paper confirm the findings in literature concerning increasing CO and HC emissions for earlier pilot injections. The results of the pilot injection strategy do not correspond with earlier presented results by Chartier et al. [11]. A possible explanation might be the increased injector dribble for a multiple injection strategy and therefore increased HC emissions. Additionally, the NO$_x$ emissions decreased for both pilot and post-injection strategies.

Figure 6.32 Combustion characteristics for a combined main and post injection.

Figure 6.33 Emission results for pilot/post + main vs. main injection.
“This is the end”

The Doors - The end

General discussion
7.1 Optical diagnostics

The main topic of research in internal combustion engines is focused on determining phenomena timed around the start of ignition, including fuel injection. All events of interest then take place in a few milliseconds. Large cycle-to-cycle variations in engines require fast diagnostic techniques to resolve these small relevant time spans. In this thesis, novel optical diagnostics and improved combustion strategies for compression ignition combustion have been presented. Each chapter has focused on one specific measurement technique, which can be used to characterize different aspects of compression ignition combustion. This final chapter discusses the specific choices that were made during this research and indicates opportunities for further improvements.

7.1 Optical diagnostics

7.1.1 Particle image velocimetry

Particle image velocimetry has a proven track record of visualizing in-cylinder flows and quantifying velocity fields [21, 105]. Increasing the recording rate by using a high-speed laser and high-speed camera enables to investigate the flow evolution in a single cycle, and to record more data before engine fouling occurs. To achieve enough contrast between the particles and the environment the relatively low laser power of the high-speed laser system needs to be compensated by a larger tracer size. The spatial resolution of the presented in-cylinder velocity fields in chapter 4 is shown to be lower for the high-speed measurements than for the low-speed measurements presented in chapter 3. High-speed PIV is therefore not a substitute for high resolution low-speed PIV, but an additional measurement method to track fast changes.

7.1.2 Toluene LIF

Measuring temperature gradients using 2-color toluene LIF has been proven to be a powerful method [61, 103]. However, the technique is very sensitive to disturbances by fluorescence of the base fuel in the wavelength range of interest. Good agreement has been found between the calibration curves of current (chapter 5) and previously presented results [20]. To increase the applicability of the method additional research needs to be done to increase the signal levels. Increasing the signal level can be achieved by e.g. increasing the laser fluence and avoiding the use of a dichroic beam-splitter or replacing the beam-splitter with a version which has a higher transmission in the 320 nm range. Furthermore the source of the interfering fluorescence from the base fuel needs to be further investigated.
7.1.3 OH chemiluminescence

OH chemiluminescence is a useful tool to measure flame-lift off of burning fuel sprays [49]. In this thesis a similar approach has been presented using a high-speed spectral measurement setup to measure the position of the flame after injection when running in a partially premixed mode, see chapter 6. In a burning spray increased ignition delay results in a longer flame lift of length [49]. When combustion starts after injection a longer ignition delay results in flame fronts closer to the injector due to “reflection” of the fuel against the piston wall.

7.2 Future perspectives

7.2.1 Optical access

To understand all phenomena in an engine in detail, optical diagnostics will be needed in the future to validate numerical simulation. However, with all optical diagnostics, the potential advantage of not being intrusive raises some design concerns for obtaining optical access.

To measure combustion phenomena, the optical access of an engine needs to be as large as possible. In this thesis all results presented are from a single viewpoint, through the piston. For PIV this results in a restriction to a 2D field without information on the 3rd dimension. The same holds for temperature measurements using LIF, the temperature can only be measured in one plane. To circumvent this, a fully optically accessible piston in combination with a fully optically accessible liner would increase the possibilities in measuring in all directions when the piston is close to TDC, and inside the piston. The drawback would be the increase of fragile parts and the increased difference in thermal properties between the optically accessible engine and a conventional engine. A piston with a geometry close to production type (toroidal shape) will facilitate easy translation of the results in an optical engine to conventional, production type, engine setups. Preferably two identical setups should be operated of which only one is optically accessible to measure the level of conformity.

7.2.2 Engine operating range

Running an optical engine is a delicate job because of the decreased lubrication and cooling possibilities of the piston and liner, resulting in operating points of limited load, pressure rise rates and peak pressures. Therefore they are usually operated in a low load regime with approximately 20 consecutive combustion cycles in a row. For a production type engine the behavior of alternative combustion strategies at high load is as important as the behavior at low load. Future experimental setups need to able to operate at full loads to make sure the phenomena observed at low loads
are consistent at high load as well, when the heat release is much higher. This might require an optical engine design with regular cooling of the cylinder and lubrication of the piston. By using an engine without an elongated piston but equipped with an adapted piston top and additional cylinder liner part, optical access via side windows can be obtained.

7.2.3 Hardware

For quantitative measurements, calibration of all used equipment is of utmost importance. However, knowledge of the limitations of the hardware, such as (ICCD) cameras and lasers is just as important. These limitations need to be covered in more detail prior to the start of the experiments. A closer collaboration with manufacturers and specialists might increase the reliability of the used equipment, and the absolute quantification of measured data.

7.2.4 PPC research

To achieve partially premixed combustion in diesel engines will need a paradigm shift at engine manufacturers and fuel suppliers. Achieving more premixing needs a fuel that is less prone to auto-ignition. As this might need a dedicated new fuel type (e.g., gasoline diesel blend), it is unlikely that this will be implemented in the near future. However, using a dual-fuel technique using a combination of diesel and gasoline might be a solution [36] [53]. As aftertreatment systems are becoming more sophisticated, these maybe even avoid the requirement of low engine out emissions of e.g. HC and CO as long as the thermodynamically properties of the operating points compensate the efficiency loss of the unburned fuel.

7.2.5 Soot

In this thesis soot production has not been measured, even though it is of major importance for engine research. A major point of attention for future research is not only the amount of soot produced in weight, but also in numbers and particle size. Recent developments in time-resolved measurement devices for soot particle size and distribution (e.g., Cambustion [101]) makes it possible to combine optical diagnostics closely with soot measurements. Quantification of soot formation by using optical diagnostics is an interesting field of research. However, the relation of soot formation with engine out soot is also influenced by the soot oxidation late in the combustion cycle [26, 102].
7.3 Conclusion

Developments in the last decade have proven that optical diagnostics is an important field in engine research. As shown in this thesis, flow fields, temperatures, chemiluminescence and OH formation can be determined in a running engine. This will undoubtedly encourage engine researchers to apply these diagnostic techniques for an increased range of fuels, engine geometries and operating ranges to improve future fuels and engines. However, there is still along way to go to measure temperature and fuel concentration in real-time routinely.
References


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References


References


References


References


Dankwoord

Het schrijven van dit hoofdstuk is waarschijnlijk het lastigst, mede omdat het Limburgs en Nederlands de laatste jaren is omgezet in Engels.

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Dankwoord

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Eindhoven, juli 2012.
Ron
Curriculum Vitae

Ron Zegers was born on June 5th, 1982 in Heerlen, the Netherlands. He grew up in Eys and finished his secondary education (atheneum) in 2000 at the S.G. Sophianum in Gulpen, the Netherlands. After obtaining his propaedeutic in Automotive Engineering at HAN University of Applied Sciences in Arnhem in 2001, he started studying Mechanical Engineering at Eindhoven University of Technology in the same year. During this study, he performed an internship at Corus, Ijmuiden. His MSc project was performed at the Combustion Technology group of prof. Philip de Goey at the department of Mechanical Engineering, Eindhoven University of Technology in cooperation with Philips Lighting and focused on oxy-fuel burners for glass production. In this group, he started his PhD project in December 2007, which focused on optical diagnostic techniques for advanced compression ignition combustion, of which the results are presented in this thesis. During his PhD research, he also participated in the Dutch Academic Year Prize, with the clean combustion project called "Brandschone Verbranding". Since April 2012, he was appointed as researcher in the same group in cooperation with Progression-Industry, focusing on fuel flexibility of cyclic oxygenates.

Linkedin: http://nl.linkedin.com/in/ronzegers
Facebook: http://facebook.com/ronzegers
Twitter: https://twitter.com/ronzegers