



LUND UNIVERSITY

Investigation of PPC in an Optical Engine

With focus on Fuel Distribution and Combustion Characterization

Lönn, Sara

2019

Document Version:

Publisher's PDF, also known as Version of record

[Link to publication](#)

Citation for published version (APA):

Lönn, S. (2019). *Investigation of PPC in an Optical Engine: With focus on Fuel Distribution and Combustion Characterization*. Department of Energy Sciences, Lund University.

Total number of authors:

1

General rights

Unless other specific re-use rights are stated the following general rights apply:

Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

- Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
- You may not further distribute the material or use it for any profit-making activity or commercial gain
- You may freely distribute the URL identifying the publication in the public portal

Read more about Creative commons licenses: <https://creativecommons.org/licenses/>

Take down policy

If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.

LUND UNIVERSITY

PO Box 117
221 00 Lund
+46 46-222 00 00

Investigation of Partially Premixed Combustion in an Optical Engine with
focus on Fuel Distribution and Combustion Characterization

Investigation of PPC in an Optical Engine

With focus on Fuel Distribution and
Combustion Characterization

Sara Lönn



LUND
UNIVERSITY

DOCTORAL DISSERTATION

by due permission of the Faculty of Engineering, Lund University, Sweden.
To be defended in M:A at M-building, LTH, Ole Römers väg 1. Date 20190605
and time 10:15.

Faculty opponent

Prof. Dr. Sebastian Kaiser, University of Duisburg-Essen, Germany

Organization LUND UNIVERSITY Sara Lönn		Document name DOCTORAL THESIS
Author(s) Sara Lönn		Date of issue 2019-06-05
		Sponsoring organization KCFP Engine Research Center
Title and subtitle Investigation of PPC in an Optical Engine with focus on Fuel Distribution and Combustion Characterization		
<p>The global society is dependent on transportation to function and a big part of these transports are driven by an internal combustion engine. Heavy-duty vehicles utilizes conventional diesel combustion is a combustion concept where the fuel is often ignited during the injection event and while it gives a high combustion efficiency, the combustion mode is also characterized by high emission of NOx and soot. The transports also contributes to greenhouse gas emissions and to emissions of substances harmful to humans. These negative effects have driven research in advanced combustion concepts that combine high efficiency with low emissions. A number of advanced combustion concepts have been developed utilizing low temperature combustion and partially premixed combustion is one of them.</p> <p>Partially premixed combustion utilizes dilution of the charge to lower the combustion temperature, which means that the ignition delay is longer. The ignition delay for PPC can be further delayed by adding EGR. The advantages with keeping a low combustion temperature is reduced thermal NOx-emissions and lower heat transfer. The increased ignition delay will lead to reduced the soot emissions. Another concept that utilizes low temperature combustion is HCCI, where a homogeneous mixture is created before ignition. Since the charge is homogeneous the combustion will progress rapidly through a series of auto-ignition events. The limitations with HCCI is that this rapid heat release limits the possibility of reaching higher loads and at lower loads the emissions of HC and CO increases which limits the efficiency. The PPC concept is a way to combine the benefits, and to limit the disadvantages, of HCCI combustion and conventional diesel combustion.</p> <p>PPC is thus an intermediate concept and the focus of this thesis is to investigate the transition from HCCI-like combustion to more stratified combustion. The studies are mainly carried out in an optical engine with optical diagnostics, including laser-based diagnostics. This includes investigation the effects of the piston geometry and spray angle on the combustion. The thesis links the combustion characteristics with the fuel distribution and the mixing process for a sweep of injection timings. The studies are carried out with fuel having a high octane rating in order to increase the ignition delay. In most of the studies, the PPC transition was carried out with primary reference fuels, but the thesis also includes characterization of methanol combustion in PPC mode for a variety of injection strategies. The methanol study has shown that by injecting a small amount of methanol as a pilot can trigger the combustion of the main injection. The study also shows that by adopting a double injection strategy the inlet temperature required to ignite the fuel can be reduced significantly</p>		
Key words Optical engine, PPC, Fuel distribution, Combustion, Methanol		
Classification system and/or index terms (if any)		
Supplementary bibliographical information		Language English
ISSN and key title 0282-1990		ISBN 978-91-7895-135-2 (print) 978-91-7895-136-9 (pdf)
Recipient's notes	Number of pages 172	Price
	Security classification	

I, the undersigned, being the copyright owner of the abstract of the above-mentioned dissertation, hereby grant to all reference sources permission to publish and disseminate the abstract of the above-mentioned dissertation.

Signature



Date 2019-05-07

Investigation of PPC in an Optical Engine

With focus on Fuel Distribution and
Combustion Characterization

Sara Lönn



LUND
UNIVERSITY

Copyright (Sara Lönn)

Faculty of Engineering
Department of Energy Sciences

ISBN 978-91-7895-135-2 (print)
ISBN 978-91-7895-136-9 (pdf)
ISRN LUTMDN/TMHP-19/1150-SE
ISSN 0282-1990

Printed in Sweden by E-huset, Lund University
Lund 2019

Abstract

The global society is dependent on transportation to function and a big part of these transports are driven by an internal combustion engine. The vehicles on the road today are usually equipped with a compression ignition engine or a spark ignition engine. Heavy-duty vehicles mostly use a compression ignition engine running on diesel since it is more fuel efficient. Conventional diesel combustion is a combustion concept where the fuel is often ignited during the injection event and while it gives a high combustion efficiency, the combustion mode is also characterized by high emission of NO_x and soot. Transports contribute to greenhouse gas emissions and to emissions of substances harmful to humans. The negative effects have driven research in combustion engines to develop advanced combustion concepts that combine high efficiency with low emissions. A number of advanced combustion concepts have been developed utilizing low temperature combustion and partially premixed combustion is one of them.

Partially premixed combustion utilizes dilution of the charge to lower the combustion temperature, which means that the ignition delay is longer than compared to conventional diesel combustion, CDC, where the fuel is ignited during the injection event. The ignition delay for PPC can be further delayed by adding EGR. The advantages with keeping a low combustion temperature is reduced thermal NO_x-emissions and lower heat transfer. The increased ignition delay will also reduce the soot emissions compared to CDC. Another concept that utilizes low temperature combustion is HCCI, that relies on an even longer ignition delay to create a homogeneous mixture before ignition. Since the combustion is homogeneous it will progress rapidly through a series of auto-ignition events. The limitations with HCCI is that this rapid heat release limits the possibility of reaching higher loads and at lower loads the emissions of HC and CO increases which limits the efficiency. The PPC concept is a way to combine the benefits, and to limit the disadvantages, of HCCI combustion and conventional diesel combustion.

PPC is thus an intermediate concept and the focus of this thesis is to investigate the transition from HCCI-like combustion to more stratified combustion. The studies are mainly carried out in an optical engine with optical diagnostics, including laser-based diagnostics. This includes investigation the effects of the piston geometry and spray angle on the combustion. The thesis links the combustion characteristics with the fuel distribution and the mixing process for a sweep of injection timings. The studies are carried out with fuel having a high octane rating in order to increase the ignition delay. In most of the studies, the PPC transition was carried out with primary reference fuels, but the thesis also includes characterization of methanol combustion in PPC mode for a variety of injection strategies. The methanol study has shown that by injecting a small amount of methanol as a pilot can trigger the

combustion of the main injection. The study also shows that by adopting a double injection strategy the inlet temperature required to ignite the fuel can be reduced significantly.

Populärvetenskaplig sammanfattning

Dagens globala samhälle är beroende av transporter för att fungera. Mat, varor, kemikalier och människor transporteras dagligen världen över med bilar, lastbilar, fraktfartyg, tåg och flyg. En hel del av dessa transporter sker med fordon som drivs med en förbränningsmotor. Dessa transporter ger dock upphov till problem. Ett av problemen är att utsläpp av koldioxid från förbränningsmotorn bidrar till växthuseffekten som påverkar klimatet. Andra problem är utsläpp av skadliga ämnen från förbränningsmotorn som påverkar människors hälsa negativt. De skadliga ämnena är till exempel sotpartiklar, kväveoxider och oförbränt bränsle. Världshälsoorganisationen uppskattar att ca 4 miljoner människor dör en för tidig död till följd av luftföroreningar från bland annat transporter. Enligt IPCC, FNs klimatpanel, kommer det att vara en utmaning att minska de globala växthusgaserna från transportsektorn eftersom trenden för antalet transporter är ökande. Det kommer därför att krävas en kombination av flera tekniska lösningar och beteendeförändringar för att ha en möjlighet att uppnå de uppsatta klimatmålen i Parisavtalet. De tekniska lösningarna innebär att fordonsflottan behöver diversifieras så att det blir fler elektriska bilar, hybrider, mer bränsleeffektiva fordon och fordon som drivs på förnybara bränslen.

Forskningen inom förbränningsmotorer har därför fokuserat på att ta fram förbränningskoncept som både är bränslesnåla och som ger låga utsläpp av skadliga ämnen. En metod som kan kombinera dessa egenskaper är lågtemperaturförbränning. Denna avhandling fokuserar på partiellt förblandad förbränning, PPC, som är ett lågtemperaturförbränningskoncept. Partiellt förblandad förbränning innebär att hela bränslemängden sprutas in i en kompressionsantänd motor innan bränslet antänder så att bränslet har tid att blandas med luften, vilket leder till minskade sotutsläpp. Förbränningstemperaturen kan dessutom sänkas genom utspädning med återcirkulerade avgaser och det minskar kväveoxidutsläppen eftersom dessa bildas vid höga temperaturer. Partiellt förblandad förbränning är dessutom energieffektiv eftersom den sker under ett kort tidsförlopp.

Ett speciellt fokus i avhandlingen var att få en bättre förståelse för övergången från partiellt förblandad förbränning till förbränning med mer homogen bränsleblandning. Vid tidig insprutning av bränsle får blandningen längre tid att blandas, vilket leder till en mer homogen bränsleblandning. När bränsleblandningen är mer homogen kommer förbränningen i huvudsak att styras av kemin. När bränslet sprutas in vid en senare tidpunkt, får man istället bränslerika och bränslefattiga zoner i motorn. När bränsle-luftblandningen är mer heterogen blir förbränningen mer diffusionstyrd. Diffusionsförbränning är den förbränningsprocess som dominerar i traditionella dieselmotorer. Denna avhandling använder sig av optiska och laserbaserad mätmetoder för att få en bättre förståelse för förbränningen av både

metanol och bensenliknande bränslen PPC-förhållanden. Genom mätningar i motorn har bränslefördelningen studerats både kvalitativt och kvantitativt för olika insprutningstider. Detta har gett insikt om hur förbränningen påverkas av interaktionen mellan bränsleinsprutningen och kolvens geometri. Slutligen så har det visats att en mindre mängd bränsle som sprutas in tidigare än huvudinsprutningen kan underlätta förbränningen av metanol eftersom metanol kräver mycket värme för att förångas.

List of Publications

- Paper I: Transition from HCCI to PPC by means of Start of Injection
Published**
Shen, M., Lonn, S., and Johansson, B.
SAE Technical Paper 2015-01-1790, 2015
- Paper II: Optical Study of the Fuel Spray Penetration and Initial
Combustion Location under PPC Conditions
Published**
Sara Lonn, Alexios Matamis, Martin Tuner, Mattias Richter, Oivind
Andersson
SAE Technical Paper 2017-01-0752
- Paper III: Transition from HCCI to PPC: Investigation of Fuel
Distribution by Planar Laser Induced Fluorescence
Published**
Zhenkan Wang, Sara Lonn, Alexios Matamis, Öivind Andersson,
Martin Tuner, Marcus Alden, Mattias Richter
SAE International Journal of Engines 2017-01-0748
- Paper IV: Effect of Injection Timing on the ignition and mode of
Combustion in a HD PPC Engine running low load
Published**
Ibron, C., Jangi, M., Lonn, S., Matamis, A., Öivind Andersson,
Martin Tuner, Mattias Richter Xue-Song Bai
SAE Technical Paper 2019-01-0211, 2019
- Paper V: PLIF of fuel distributions at the time of auto-ignition in an
optical PPC engine
Manuscript to be submitted to a journal -**
Sara Lönn, Zhenkan Wang, Christian Ibron, Mehdi Jangi, Xue-Song
Bai, Mattias Richter, Öivind Andersson
- Paper VI: Evaluation of injection strategies with methanol in an optical
engine
Manuscript submitted to Fuel**
Sara Lönn, Mateusz Pucilowski, Alexios Matamis, Ludovica Luise,
Mattias Richter, Xue-Song Bai, Öivind Andersson

Acknowledgements

I would like to thank everyone at combustion engines, both former and present colleagues, for making my PhD studies .so interesting and fun. I am particularly grateful for all the assistance and support given by my supervisor Öivind Andersson during this project. I would also like to extend a thanks to Bengt Johansson for giving me this opportunity to carry out this project. I would also like to thank my co-supervisor Martin Tunér for his help during my time at combustion engines. I would like to extend a thanks to Per Tunestål, Sebastian Verhelst and Marcus Lundgren for always being generous with their knowledge. I am also grateful to the technical team at the division for repairing the optical engine.

Furthermore, a special thanks goes to the current and former colleagues (Ted Lind, Marcus Lundgren, Mike Denny, Xinda Zhu, Miao Zhang, Slavey Tanov) working with optical engines for all of the interesting discussions and help. I would also like to extend my thanks to a former colleague Pablo Garcia for his help with LABVIEW.

I would also like to thank Zhenkan Wang and Alexios Matamis from Combustion Physics for their collaboration in the experiments. It has been fun working with you and learning more about the cameras and lasers from you. In addition, I would like to thank Mateusz Pucilowski and Christian Ibrón for interesting discussions about experiments and simulations.

And a big thanks goes to my family and friends for their support and encouragement during these years.

List of abbreviations

ATDC	After Top Dead Center
BDC	Bottom Dead Center
BEV	Battery Electric Vehicle
CAD	Crank Angle Degrees
CDC	Conventional Diesel Combustion
CFD	Computational Fluid Dynamics
CI	Compression Ignition
DICI	Direct Injected Compression Ignition
EOI	End of Injection
HCCI	Homogeneous Charge Compression Ignition
ICE	Internal Combustion Engine
LTC	Low Temperature Combustion
LTHR	Low Temperature Heat Release
LTR	Low Temperature Reactions
MK	Modulated Kinetics
PPC	Partially Premixed Combustion
PPCI	Partially Premixed Compression Ignition
PPR	Pressure Rise Rate
PRF	Primary Reference Fuel
RPM	Revolutions per minute
RoHR	Rate of Heat Release
SOC	Start of Combustion
SOI	Start of Injection
TDC	Top Dead Center
UNIBUS	Uniform Bulky Combustion System

Table of Contents

Abstract	7
Populärvetenskaplig sammanfattning.....	9
List of Publications.....	11
Acknowledgements	12
List of abbreviations.....	13
1. Introduction	17
1.1 Background.....	17
1.2 Objective.....	19
1.3 Approach	20
2 Combustion concepts and fuels.....	21
2.1 Overview of chapter	21
2.2 Overview of combustion modes	21
2.3 Direct injection compression ignition engine	23
2.4 Conventional diesel combustion.....	25
2.5 Low temperature combustion	26
2.5.1 HCCI.....	27
2.5.2 MK.....	28
2.5.3 UNIBUS	28
2.5.4 Partially premixed combustion	29
2.5.5 RCCI.....	30
2.6 Fuel.....	31
3 Experimental equipment and optical diagnostics.....	33
3.1 Single cylinder metal engine	33
3.2 Heat Release analysis	34
3.3 Optical research engine – Bowditch design.....	36
3.4 Natural luminosity imaging	38
3.5 Mie Scattering.....	39

3.6	PLIF	39
4	Results	43
4.1	Transition from PPC towards HCCI.....	43
4.1.1	Metal engine	43
4.1.2	Optical engine.....	45
4.1.3	Fuel distribution from injection to combustion	49
4.2	Methanol.....	57
4.2.1	Characterization of the methanol combustion process	57
5	Thesis contribution.....	63
5.1	Transition from PPC to HCCI	63
5.2	Methanol combustion	64
6	References	65
	Scientific publications	73

1. Introduction

1.1 Background

Internal combustion engines (ICE) have had and still have a dominating position in the transportation sector. However, the technology development within the sector is becoming more diverse as the technology content in the vehicles keep increasing. This diversity in technology is also driven by the challenges facing the current transport sector and society. For example, the UN [1] lists climate change as one of the global issues in the world and the WHO estimates that 4.2 million deaths are caused by ambient air pollution [2] [3].

Global warming has led to higher temperatures in the atmosphere, since the increased levels of greenhouse gases in the atmosphere absorb more heat radiation emitted from the surface of the earth. Furthermore, it is clear that greenhouse gas concentration in the atmosphere has increased based on emission from human activities. One of the more important greenhouse gases is CO₂, and the CO₂-concentration in the atmosphere has increased with 40% since before the industrial era. [4] The main global sources of anthropogenic greenhouse gases are electricity and heat production, agriculture and forestry, industry, and transportation. The transport sector stands for 14% of these emissions, as is seen in Figure 1 [5]. In the US and European Union, the transport sector is responsible for an even greater portion of the greenhouse gas emissions: 28.8% (2017) [6] and 24% (2016) [7], respectively.

The transport sector is also responsible for emitting other pollutants into the atmosphere, *i.e.* NO_x, CO, SO₂ and particulate matter. Air pollution is a great health concern worldwide. In particular, inhaled particulate matter having a diameter of less than 2.5 or 10 microns have been related to cardiovascular and pulmonary diseases, as well as premature deaths. The transport sector is estimated to be responsible for a substantial portion of the particulate emissions. In European cities, 30% of the particulate matter is estimated to originate from road transport. In OECD countries, up to half the particulate matter is estimated to come from road transport. [8]

The goal to reduce global warming and health risks has driven development of new technology for the transport sector. Apart from development of new ICE technology, this includes new solutions for transport system management, autonomous driving systems, electric roads and electric vehicles or hybrid electric vehicles.

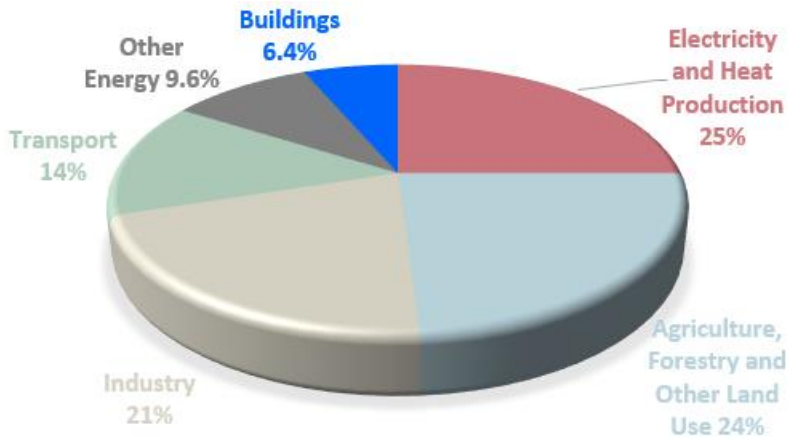


Figure 1 Anthropogenic global greenhouse gas emissions divided in economic sectors. The data is from 2010 and is reproduced from [5]

The goal to reduce global warming and health risks has driven development of new technology for the transport sector. Apart from development of new ICE technology, this includes new solutions for transport system management, autonomous driving systems, electric roads and electric vehicles or hybrid electric vehicles. An example of transport management is to arrange several trucks to drive closely after each other to reduce air resistance and thereby reduce fuel consumption. [9] Electric roads are another technology that has mainly been implemented in pilot projects. Electric roads can charge the electric vehicle while driving by providing electricity through an electrical connection between the road and the vehicle. A more expensive solution is wireless charging where induction is utilized. [10]

Electric propulsion is a technology that has gained momentum and been commercialized. Light-duty vehicles and car manufactures are shifting the production from full ICE-cars towards hybrid vehicles systems and battery electric vehicles (BEV). In 2017, the number of electric cars, including BEV, plugin hybrid-electric vehicles (PHEV) and fuel cell electric vehicles (FCEV), exceeded 3 million. [11] However, in 2015, the worldwide fleet of personal cars in use was estimated to be around 1 billion by the international organization of motor vehicle manufacturers. [12] With these figures in mind, electric vehicles still only hold a small part of the market worldwide. This could change over time, since several car manufacturers have started to transition their car models towards hybrid and electric cars. Even though electric vehicles do not have a tail pipe, it is not the same as being free of emissions. The emissions are moved from the vehicle upstream in the production chain. The battery production and the electricity production for charging are substantial emitters of CO₂ [13]. The electricity mix used to charge the car is

therefore important. This means that it is more advantageous to drive an electric vehicle in Norway compared to China, for example.

The electrification trend is not as clear for the heavy-duty side and electric heavy duty trucks can mainly be found in demonstration and pilot projects [14, 15], since there are remaining limitations such as power density, lifetime, and cost of batteries. However, electric buses are an exception to this and have been realized in several cities around the world. [11] Even as the transportation fleet moves towards a more varied composition of vehicle technology, it is still expected for compression ignition engines to remain the main propulsion system for heavy-duty transportation by road or sea for the foreseeable future.

The heavy-duty trucks on the market today are mainly driven by conventional diesel combustion (CDC) in compression ignition (CI) engines. CI-engines are used because they are more fuel-efficient than spark-ignition engines and will therefore use less fuel and emit less CO₂. [16] Spark ignition engines are more often used in light duty cars since it is possible to use a three-way catalyst to reduce emission such as CO, hydrocarbons (HC), and NO_x. [16] The high fuel efficiency of the conventional diesel engine is, however, weighed down by high levels of NO_x and particulate matter (PM). This has led to an emission legislation that has become stricter in multiple steps during the recent decades and this has also driven the engine research towards reducing emissions.

The development for the internal combustion has focused on improving engine efficiency and reducing tailpipe emissions to both mitigate climate change caused by emission of greenhouse gases and reduce harmful emission to humans. One approach has been to research new advanced combustion concepts, such as low temperature combustion, that simultaneously yield high efficiency and low emissions. There are a number of combustion concepts within low temperature combustion; HCCI (homogeneous charge compression ignition), RCCI (reactivity controlled compression ignition) and PPC (partially premixed combustion). Among these, this thesis focuses on PPC combustion.

1.2 Objective

The objective of this thesis has been to provide better understanding of the partially premixed combustion concept and the mixing process with the use of optical and laser measurements techniques in a compression ignition heavy duty engine. The concept of partially premixed combustion is based on an increased ignition delay to provide better mixing between the fuel and air before the combustion. The focus has therefore been on understanding the fuel mixing and combustion event in the cylinder when transitioning from different combustion modes, in particular the transition from HCCI to PPC. The scope of the thesis includes the use of primary reference fuel mixtures that display two-stage ignition behavior and methanol with a single stage ignition behavior. An additional aim of this study is to provide engine,

spray, and combustion data to improve CFD modeling of partially premixed combustion. An additional objective has been to characterize methanol combustion in an optical compression ignition engine.

1.3 Approach

The thesis mainly comprises of experimental measurement in a single cylinder optical engine in PPC mode. Optical diagnostic measurement techniques have been used during the experiments. The combustion has been visualized with high speed video while the liquid spray has been detected with Mie Scattering and the fuel distribution has been visualized with laser induced fluorescence. Furthermore, a single cylinder metal engine and CFD simulations have been used to provide additional knowledge. The work has involved the use of PRF fuels and methanol.

2 Combustion concepts and fuels

2.1 Overview of chapter

As an introduction, this chapter will present a brief overview of the different combustion modes in an internal combustion engine and a short description of the compression ignition engine. The chapter will then describe different combustion concepts used in a compression ignition engine with a particular focus on the partially premixed combustion (PPC) concept. The chapter is concluded with a brief discussion of how certain fuel properties affect the combustion with a focus on diesel, gasoline and alcohols fuels in PPC.

2.2 Overview of combustion modes

The combustion process is not the same in all engine types. It can be divided into three main combustion modes: flame propagation, diffusion combustion and kinetic combustion. [17]

Flame propagation, shown in the upper left corner of Figure 2.1, is a combustion mode that requires an external ignition source to create a flame kernel that will grow and propagate as a flame through a homogeneous mix of fuel and oxidizer. The flame will propagate by preheating the cold premixed fuel and oxidizer ahead of hot products in the reaction zone.

In diffusion combustion, the fuel and oxidizer are initially separated but mix during combustion. The combustion is limited by the mixing rate of fuel and oxidizer and, in engines, the mixing rate is driven by the turbulence. [18]

The final combustion mode is kinetic combustion, which leads to a rapid combustion process compared to the other combustion modes. The combustion occurs in homogeneous mixtures of fuel and oxidizer through spontaneous auto-ignition. As the mixture is homogeneous that will lead to multiple auto-ignition events for conditions with the same pressure and temperature. This auto-ignition events will lead to a rapid combustion event, characterized by sequential autoignition rather than a flame front.

Vehicles on the road today are mainly equipped with either compression ignition or spark ignition engines and these engine types operate using separate combustion

modes. The combustion concepts are distributed along the lines of the triangle in Figure 2.1. The position of the combustion concepts is determined by which combustion mode/s are included in the concept. The combustion mode in spark-ignition engines is flame propagation, where the combustion is initiated by a spark plug. The fuel used in spark ignition engines are resistant to auto-ignition, but in some instances spark-ignition engines can experience knock, which is a form of undesired kinetic combustion that can be harmful to the engine. Compression ignition engines are found in regular diesel cars, which utilizes conventional diesel combustion (CDC). The CDC concept is a mix of more than one combustion mode. Even though diffusion combustion is the dominant mode, the combustion will start through a kinetic combustion event where a portion of the fuel and air will auto-ignite before shifting to spray-driven and mixing controlled combustion. HCCI is a combustion concept that in principle only utilizes kinetic combustion. But, several combustion concepts utilize a mix of at least two combustion modes, for example PPC, RCCI [19, 20, 21], SACI, and DISI. CDC, PPC, HCCI will be discussed further in section 2.4, 2.5.4 and 2.5.1. RCCI will also be briefly mentioned in section 2.5.5.

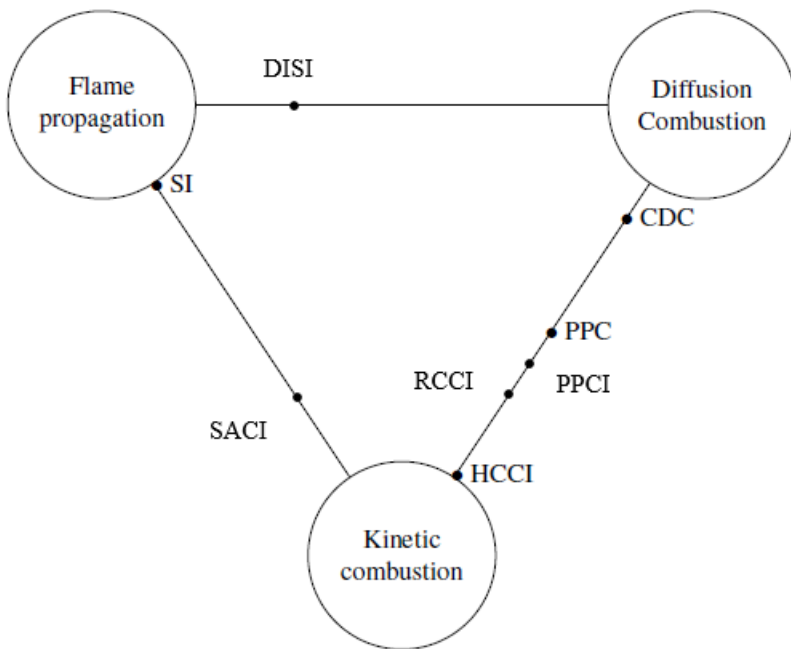


Figure 2.1 A schematic illustration of the different combustion modes and how the combustion concepts fit to the respective combustion modes. Figure reproduced and adapted from [22]

2.3 Direct injection compression ignition engine

Compression ignition engines can be divided into 2-stroke or 4-stroke engines and in this thesis the focus is on 4-stroke engines. The different strokes relate to the valve positions and the piston movement from top dead center (TDC) to the bottom dead center (BDC) and vice versa. [16]

The first stroke is the intake stroke where the inlet valves are open, allowing gas to enter the cylinder as the piston moves from TDC to BDC, as seen in figure 2.2. During the compression stroke, the inlet valves will close and the gas will be compressed as the piston moves from BDC to TDC. As the volume decreases, the pressure and the temperature in the cylinder will increase. In the direct injected compression ignition engine, the fuel is most often injected during the compression stroke and air will be entrained in the fuel spray. The spray will atomize and break up into smaller fuel droplets that will vaporize due to the increasing temperature. As the piston moves up towards TDC, an auto-ignition event will be initiated at certain in-cylinder conditions depending on the temperature, pressure, and fuel/air composition. When the piston moves past TDC, the compression stroke will shift to the expansion stroke where the piston moves from TDC to BDC with closed inlet and exhaust valves. After the auto-ignition event the combustion will spread through one or more of the combustion modes releasing heat through exothermic reactions. The heat released from the combustion will increase the pressure in the cylinder, which is why it is preferred that the auto-ignition event takes place close to TDC so that the increased gas force from the combustion can be applied to the piston during as large a portion of the expansion stroke as possible. This displacement of the piston will produce mechanical work. This is also the reason why the expansion stroke sometimes is referred to as the power stroke. In the final stroke the piston moves from BDC to TDC while the exhaust valves are open, to allow the combustion products to leave the cylinder.

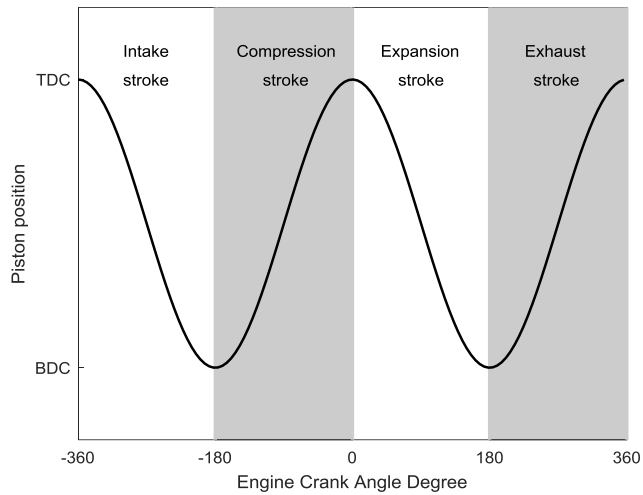


Figure 2.2 Schematic illustration of the different strokes in a 4-stroke compression ignition engine.

An advantage with compression ignition engines is that the efficiency is higher compared to SI-engines. The higher efficiency of the compression ignition engine can be explained by the relationship for the theoretical efficiency for an ideal Otto cycle [16].

$$(2.1) \quad \eta_t = 1 - \frac{1}{r^{\gamma-1}}$$

Compression ignition engines are operated with an excess of air, while spark ignition engines throttle the inlet to limit the air supply to operate under stoichiometric conditions. The excess of air in compression ignition engines lead to higher gamma values, which also gives a higher efficiency as seen from equation 2.1. In addition, higher compression ratios are used in compression ignition engines compared to SI-engines. This is because the combustion relies on auto-ignition for compression ignition engines, while SI-engine uses an external ignition source. Spark ignition engines are also limited from using too high compression ratios due to knock. According to equation (2.1), higher compression ratios also improves the efficiency. Therefore, compression ignition engines are often found in heavy duty applications such as trucks, construction equipment, and ships, where fuel economy is of central importance for the total cost of ownership.

2.4 Conventional diesel combustion

Currently, the vehicles on the market with a compression ignition engine are operated with conventional diesel combustion. Even though the name implies that diesel is used as fuel, the CDC concept is not limited to run on diesel fuel. But as the combustion occurs through auto-ignition it will be easier to start the combustion for a fuel with a high cetane number such as diesel fuel, since high cetane ratings reduces the ignition delay.

The conventional diesel combustion concept generally involves a fuel injection close to TDC, and the combustion will be initiated before the end of injection (EOI), as seen in figure 2.3. As previously mentioned, the auto-ignition event for CDC occurs in premixed conditions but as more fuel is injected the combustion will shift to spray driven combustion, which relies on mixing. The final part of the combustion is the late mixing controlled combustion that takes place after the end of injection. This is characterized by slower combustion than during the spray-driven part through, as the turbulence generated by the spray quickly dissipates after the end of injection. This means that the mixing and hence the combustion rate then slows down. [16] [23]

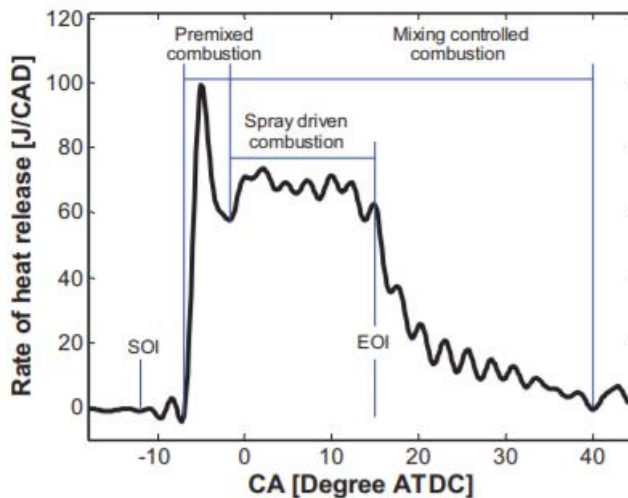


Figure 2.3 Rate of heat release of typical conventional diesel combustion. Figure reproduced from [24]

One of the problems with CDC is the high NO_x and soot emissions. The ϕ - T diagram in figure 2.4 describes the pathway for CDC and illustrates at what conditions the

NO_x and soot are created. When the fuel is injected into the cylinder air is entrained into the spray which creates a mixture of fuel and oxidizer that is initially rich. The rich fuel mixture will ignite and soot will begin to form from the products of the combustion of the rich mixture. [25] As seen in figure 2.4, soot is created at equivalence ratios greater than two and temperatures are greater than roughly 1600 K. Furthermore, a diffusion flame can be found at the periphery of the spray, where the product gas and the air are mixed. The diffusion flame will raise the temperature into the region of thermal NO_x production [26], as seen in figure 2.4.

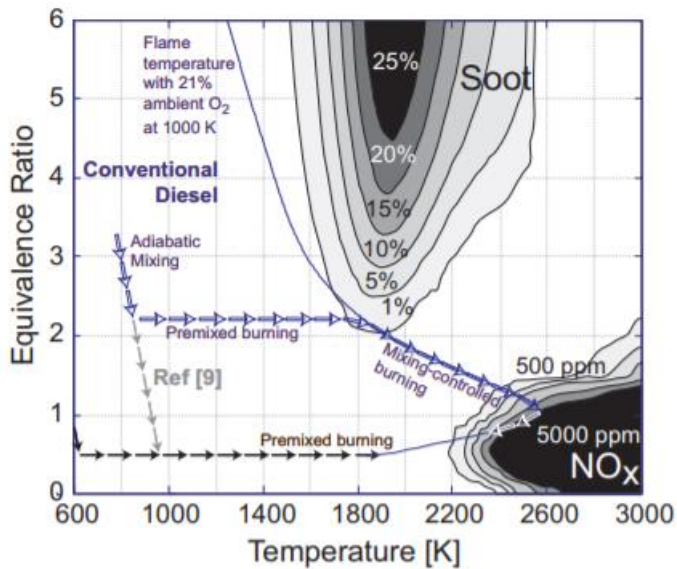


Figure 2.4 shows a schematic of Φ and T during the CDC process. Figure reproduced from [27].

2.5 Low temperature combustion

The emission legislation has become stricter in several steps in the last few decades and there is an ongoing discussion among legislators of how to limit the emissions from the transport sector even further. The automotive industry is also interested in finding a concept that would reduce the need for after-treatment systems and therefore reduce the production cost. Research has therefore gone into finding a concept that combines high efficiency and low emissions. Several concepts have been developed that utilizes low combustion temperature. As seen in Figure 2.4, by lowering the combustion temperature, it is possible to reduce the production of thermal NO_x. The combustion temperature can be lowered by diluting the charge

with for example EGR and this also reduces the heat losses in the engine. Furthermore, the soot production can be avoided by a combination of decreased combustion temperature and increased premixing. In figure 2.5, the rate of heat release for a typical low temperature combustion event can be seen. The heat is released quickly in a Gaussian shape. The fast combustion will provide a high effective expansion ratio that will increase the efficiency for the combustion concept. Combustion rates are limited by the engine durability, but dilution can be used to moderate the combustion rate. [28]

In figure 2.5, low temperature reactions give a contribution to the heat release before the main heat release event. This contribution is not seen for all fuels and is limited to the fuels with a two-stage ignition process. The low temperature reactions will cease when the temperature increases and the reactions enter the negative temperature coefficient regime. However, the main combustion reactions will be initiated when the temperature reaches a critical level. The position of the low temperature heat release will therefore be affected by the degree of premixing and temperature of the charge.

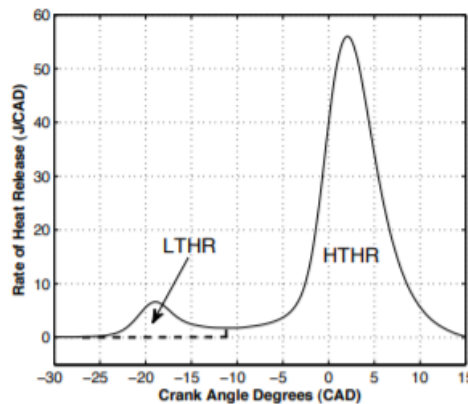


Figure 2.5 Rate of heat release of typical low temperature heat release. Figure reproduced from [29]

2.5.1 HCCI

One of the first low temperature combustion concepts was HCCI, which was developed in the 1970's. The research continued into the early 2000s. The HCCI concept is based on kinetic combustion. The fuel and air is mixed to homogeneous conditions which enables for a fast combustion event, where the combustion is spread through a series of auto-ignitions events. The combustion temperature is kept low due to the amount of excess air which reduces the formation of thermal NO_x

and soot is avoided by mixing the fuel air mixture to fuel lean conditions. [30] The low emissions of NO_x and soot are the main advantages with HCCI. One of the main disadvantages with HCCI is the problem with controlling the combustion timing. Problems with controlling the combustion event can damage the cylinder and reduce the lifespan of the engine due to the fast release of heat from the combustion.

Another disadvantage with HCCI is higher emission of hydrocarbons and CO due to that some of the fuel mixture is trapped in the crevice volume. In the expansion phase, when the piston is moving down the fuel mixture in the crevice goes back into the cylinder, but by then the temperature is too low to oxidize the fuel or CO to CO₂. The required temperature to oxidize fuel to CO₂ is 1500K. [31]

The load range is also limited for HCCI, both for low and high loads. The low load limit gives a low combustion efficiency because the combustion temperature is so low that some combustion reactions are quenched, which gives an unstable and incomplete combustion. The high load limit is caused by limitations in the total in-cylinder peak pressure and the maximum pressure rise rate for the engine. Higher loads can be reached from using boosted inlet pressure. EGR can also be used to control the temperature in the cylinder and combustion characteristics. The limitations of the HCCI concept have led to the development of other low temperature combustion concepts. Some of these concepts have combined HCCI with SI, such as the SACI concept [32], while other low temperature combustion concepts are combinations of HCCI with diffusive combustion. In the following sections some of the LTC concepts combining HCCI and CI will be discussed. The first two LTC concepts, MK and UNIBUS are concepts that were developed early. RCCI and PPC are advanced concepts that are of current research topics within the engine research community.

2.5.2 MK

MK is a concept that was developed by researchers at Nissan Motor Company where high levels of EGR was used to dilute the charge and fuel was injected after TDC to keep the combustion during the expansion phase and to provide time for the fuel and air to mix. [33] [34] However, this causes the combustion to occur late in the expansion phase which will give low combustion efficiency. As the fuel is injected so late it will be difficult to reach higher loads and the diesel fuel hindered more advanced injection timings as diesel is ignited too easily.

2.5.3 UNIBUS

UNIBUS is a concept developed by Toyota Motor Corporation. [35] The principle with UNIBUS is to divide the diesel fuel into at least two injections. The first injection is to instigate low temperature reactions to prepare the in-cylinder charge

for a second the fuel injection closer to TDC that will trigger the main combustion event. [36] [37]

2.5.4 Partially premixed combustion

Partially premixed combustion (PPC) also goes under the acronyms PCCI or PPCI. One of the reasons for the many names for a similar concept is that the boundaries are fluid. PPC is a concept that utilizes fuel stratification to achieve the desired ignition delay and combustion phasing. However, there have been attempts to divide and categorize partially premixed combustion in terms of fuel stratification level. One way to define the PPC concept is to keep a positive separation between the end of injection and the start of injection.

In the broad sense PPC is the transition between HCCI and CDC and the aim with the concept is to combine the advantages while avoiding the problems with HCCI and CDC. For PPC, the mix of the cylinder charge is of great importance since too rich pockets of fuel leads to soot formation and too dilute and homogeneous mixtures can lead to high pressure rise rates and rapid combustion or a reduction of combustion efficiency. The aim of PPC is to balance between these extremes to create a charge that is premixed enough. Injection strategies in combination with different injection timings can create a desired level of “premixedness”.

EGR is used to keep temperature low, but too high levels of EGR will eventually lead to higher soot emissions. The PPC process has been studied in optical engines with optical measurements techniques in order to understand and characterize the combustion concept. Musculus *et al.* created a conceptual model for diesel PPC with EGR-dilution in a low load case. [38] In the conceptual model, the injected fuel will first vaporize and mix. After this, the fuel will start to react in the first-stage ignition producing formaldehyde and similar combustion intermediates. In figure 2.6 it can be seen that a second stage ignition is initiated for both intermediate fuel mixtures producing OH and rich fuel mixtures that will produce soot. In addition, intermediate ignition regions can also be found that produces CO and UHC due to low temperature regions.

Partially premixed combustion was first operated with diesel fuel, but the ignition delay for diesel is short, which requires excessive EGR amounts to dilute the mixture in order to increase the ignition delay to provide a partially premixed charge. The high EGR levels will also contribute to increasing soot levels. These problems led Kalghatgi *et al.* to investigate the possibility of operating PPC with gasoline. By utilizing a fuel with a higher resistance to auto-ignition, it is easier to reach a positive delay between the end of injection and the start of combustion. This positive separation enables the charge to mix to a “premixed enough” mixture before combustion which lowers the phi-levels and thereby leads to reduced soot levels. [39] [40] The use of gasoline in PPC also enables standard compression ratios to be used for compression ignition engines.

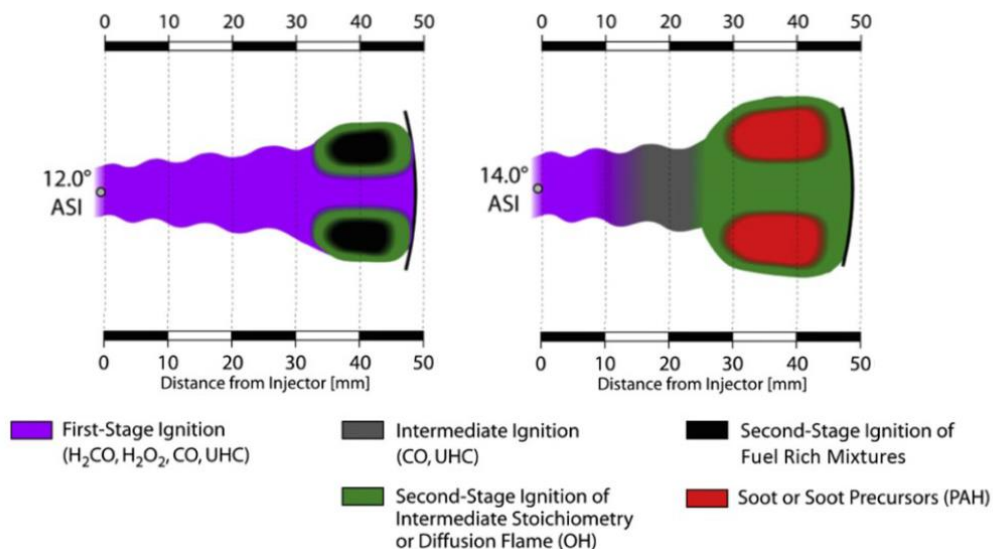


Figure 2.6 Conceptual model by Musculus et. al. Reproduced from [38].

The advantages made fuels resistant to auto-ignition a focus for the PPC research field. Manente *et al.* showed in a single cylinder engine that PPC could be operated in the entire load range with gasoline. [41] The research in PPC has been carried out in compression ignition engines that have not been designed for LTC, but have instead been designed for CDC. However, in 2010 the first single cylinder engine for gasoline direct compression ignition, GDCI, was designed and built by Delphi. [42] The engine development has continued to create a complete engine concept for combustion of gasoline in compression ignition engines. [43]

The PPC research has also involved studies on fuel effects, efficiency, soot emission, how to control the PPC combustion, simulation of PPC, optical engine research of PPC and using alternative fuels in PPC [44].

2.5.5 RCCI

Reactivity controlled compression ignition is a competing concept to PPC and it is a concept that uses fuels with different ignition characteristics to control the ignition and combustion. [45] [46] An ignition resistant fuel is introduced early to mix with the ambient gas in the cylinder and to create a stratified charge and, to instigate the combustion, a less ignition resistant fuel is injected late into the cylinder. The concept has several controlling parameters. Some of those are when each fuel should be injected and how much of each. This opens up possibilities to control both the

equivalence ratio stratification and the ignitability of the fuel mix. However, it also adds complexity to the system that will increase production cost. [46]

2.6 Fuel

In the beginning, diesel fuel was used in LTC concepts, but it is not an appropriate fuel for PPC since it is developed for CDC which aims for a fuels that ignites easily. Then there was a transition from high cetane fuels towards fuels with a higher resistance to auto-ignition in PPC. However, it has also been found that by using higher octane rated fuels it becomes more difficult to run low load PPC, due to the resistance to auto-ignition that can lead to misfire or unstable combustion. Studies using alternative fuels in combination with PPC has also appeared in recent years. In particular light alcohol fuels have been of special interest [47] [44] [48]. It is important to investigate the possibilities for PPC with renewable fuels as there are discussion among legislators of banning the sales of new cars running on fossil fuels in certain countries. [49] In table 1, the differences between methanol, gasoline and diesel can be seen for some fuel properties.

Table 1. Fuel properties for diesel [50] [51] [52], gasoline, and methanol [53]

Fuel	Methanol	Gasoline	Diesel
Density (STP) [kg/m ³]	790	740	833-837
Heat of vaporization [kJ/kg]	1100	180-350	250
Oxygen content by mass [%]	49.93	0.00	0.00
RON	108.7	91-93	-
MON	88.6	81-84	-
Vol. Energy Content [MJ/m ³]	15871	31746	35812
Stoichiometric AFR [kg/kg]	6.5	14.7	14.8

Autoignition temperature [K]	738	465-743	~588.7
Adiabatic flame temperature [K]	2143	~2275	

Methanol is a simple molecule and is a common chemical in the industry. Methanol can be produced through several pathways including renewable production processes. [53] One of the advantages with light alcohol fuels is that the oxygen content in the fuel is high, which reduces the risk of combustion of rich mixtures where soot is produced. However, alcohol fuels are difficult to ignite due to a high octane number and a high heat of vaporization. This will lead to difficulties with cold starts, but the high heat of vaporization will also reduce the in-cylinder temperature which can increase the efficiency, due to reduced heat losses. [44]

Both diesel and gasoline has a two-stage ignition process. However, alcohol fuels display a single stage ignition process. The two-stage ignition process will in its first stage form formaldehyde. In the second ignition stage the formaldehyde is consumed and OH radicals are formed indicating the main combustion event. [54] Single stage ignition fuels do not display the low temperature heat release reactions as seen in the two-stage ignition fuels. For the single stage ignition fuels the oxidation begins around 950 and 1050 K. [55]

3 Experimental equipment and optical diagnostics

3.1 Single cylinder metal engine

The studies for the thesis have been carried out in a Scania D13 single cylinder engine. Most of the studies have been carried out in an optical version of the single cylinder engine, but the initial study was carried out in a conventional “metal” single cylinder engine. The fuel injection system was an XPI injection system. An external compressor was used to supply boost pressure. The inlet pressure was measured with pressure sensors and the inlet pressure was regulated by controlling the compressor speed. In order to adjust the ignition delay, the inlet temperature could be adjusted with an inlet heater. The temperature was measured with thermocouples in different positions in the engine. The in-cylinder piezo-electric pressure sensor and charge amplifier was used to measure the in-cylinder pressure. In the table 3.1 the properties of the single cylinder engine and the injector properties are listed that have been used in most of the studies. When the engine operated with methanol, the compression ratio was increased and other injectors were used.

Table 3.1 Single cylinder metal engine Scania D13 and injector properties

Displacement Volume [cm ³]	2124
Stroke [mm]	160
Bore [mm]	130
Connecting Rod [mm]	255
Compression Ratio [-]	15:1

Swirl Ratio [-]	2.1
IVC [CAD ATDC]	-141
EVO [CAD ATDC]	137
Umbrella Angle [deg]	120
Orifices [-]	8

3.2 Heat Release analysis

Heat release analysis of the combustion is a useful tool that can provide interesting information of the combustion process. The heat release is calculated from the pressure data and the results of the calculations can be visualized as the rate of heat released per CAD or as the accumulated heat release over a CAD range. Certain parameters, such as start of combustion (SOC) and combustion phasing (CA50) that can be extracted from the heat release, can be used to during analysis of the combustion characteristics and as control parameters in combustion control. The start of combustion is defined as the crank angle position where enough heat is released to give a positive contribution. CA50 is the crank angle position where 50 % of the heat from the combustion has been released.

The heat release calculations are based on first law of thermodynamics that states that energy can only shift from one form to another. To start the heat release calculations, the combustion chamber is considered a closed system.

$$(3.1) \quad \frac{dQ}{dt} = \frac{dU}{dt} + \frac{dW}{dt}$$

The term Q describes the added heat to the combustion chamber, while W is the work carried out in the system and U is the internal energy. The internal energy can be expressed as

$$(3.2) \quad U = mC_v T,$$

where m is the mass of the system, C_v is the specific heat of a constant volume and T is the temperature to the assumptions of a closed system there is no change in mass. Assuming no change in the total mass due to the closed system, the change of the internal energy can be expressed as

$$(3.3) \quad \frac{dU}{dt} = mC_v \frac{dT}{dt}.$$

It is assumed that the gas in the combustion chamber can be treated according to the ideal gas law. This means that the temperature is assumed to be the same for the entire combustion chamber. The temperature can be described by:

$$(3.4) \quad pV = mRT$$

Where p is the in-cylinder pressure, V is the volume of the combustion chamber, m is the number of moles in the gas and R is the gas constant. The ideal gas law can be differentiated before inserting the expression into equation (3.3),

$$(3.5) \quad \frac{dU}{dt} = \frac{C_v}{R} \left(p \frac{dV}{dt} + V \frac{dp}{dt} \right),$$

The mechanical work can be formulated as

$$(3.6) \quad \frac{dW}{dt} = p \frac{dV}{dt}.$$

The specific gas constant can be expressed with the specific heat constants according to:

$$(3.7) \quad R = C_p - C_v$$

and the specific heat ratio is given by:

$$(3.8) \quad \gamma = \frac{C_p}{C_v}.$$

This gives the following relation

$$(3.9) \quad \frac{C_v}{C_p - C_v} = \frac{1}{\gamma - 1}. \quad (3.9)$$

By inserting expressions (3.5) to (3.8) into equation (3.1), the following equation can be found

$$\frac{dQ}{dt} = \frac{1}{\gamma-1} \left(p \frac{dV}{dt} + V \frac{dp}{dt} \right) + p \frac{dV}{dt}, \quad (3.9)$$

which is the heat release equation. Note that the derivatives are usually taken with respect to crank angles rather than to time. Also note that this equation does not take heat losses into account. This would require additional terms based on a heat loss model.

3.3 Optical research engine – Bowditch design

The optical experiments have been carried out in an optical engine with a Bowditch design. In the optical engine, shown in Figure 3.1, the combustion chamber is separated from the engine block through a piston extension. The cylinder head is supported by four pillars and the pillars are in turn attached to the block. A window liner together with a metal plate are mounted between the cylinder head and the pillars. The engine provides optical access through the quartz piston, which is attached to the piston extension, and three side windows in quartz that are incorporated in the window liner. The piston extension is a hollow cylinder with an opening in the cylinder wall to be able to mount a mirror in the centre of the piston extension. The 45-degree mirror will enable optical access from beneath the piston. A cylinder liner with integrated cooling is connected to the metal plate and liner is easily disconnected to allow cleaning of the optical parts within the combustion chamber. The cylinder liner is used to guide the piston extension and to seal the combustion chamber together with the piston rings surrounding the base of the piston. In order to balance the engine, the non-used pistons were equipped with tungsten weights.

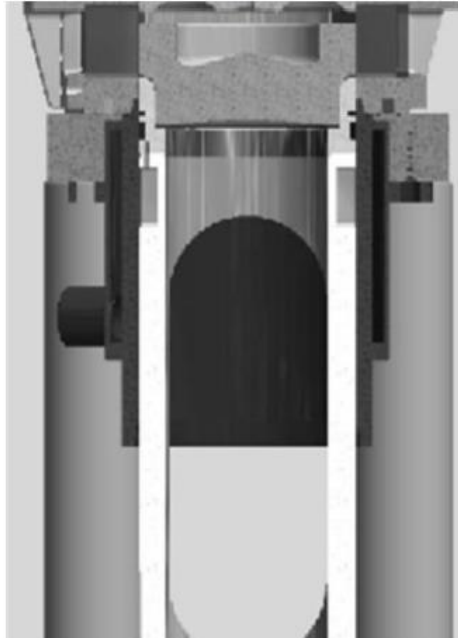


Figure 3.1 Optical engine of Bowditch design.

In this thesis, the two piston designs shown in Figure 3.2 were used. The studies regarding the transition from HCCI to PPC were conducted with the bowl-in-piston design shown on the right. The piston walls are vertical to reduce unwanted refraction when sending in a laser beam. In the methanol study, a flat “pancake” piston was used. The side-view windows provide 35 mm optical access in the vertical direction and the horizontal dimension of the window is 60 mm. However, in the horizontal direction, the field of view is broader due to the curvature of the inner surface of the window. In the vertical direction the optical access starts from the cylinder head.

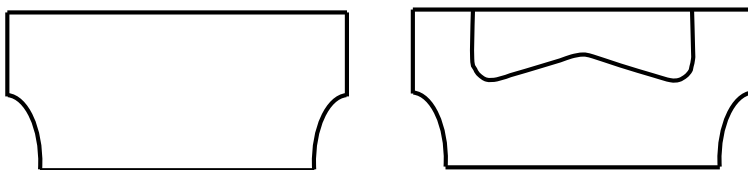


Figure 3.2 Schematics of the geometry of the flat piston (left) and the piston bowl (right)

There are some differences between metal and optical engines that needs to be taken into consideration when making comparisons between data from an optical engine and a metal engine. One of the more important aspects is the difference in heat transfer between quartz and steel (or aluminum). The quartz in the optical engine transmits heat poorly and, to maintain surface temperatures at levels typical of a metal engine, it will therefore be operated by skipping fuel injection in a sequence of cycles between each cycle with fuel injection. Quartz is used because it transmits a broad range of wavelength down in UV-range.

In the metal engine, not the all of the exhaust will be scavenged from the cylinder during the gas exchange. That is not the case in an optical engine, due to the multiple cycles without fuel injection in the skip-fire mode. In the optical engine an external burner is used to provide "EGR". The burner operates with a stoichiometric mixture of diesel and air and thus produces no particles.

Furthermore, the piston extension will introduce an increased spring motion due to the extra added mass. The elongation of the piston extension was evaluated during the gas exchange phase when the pressure is at its lowest to ensure that the piston does not hit the valves or cylinder head.

The optical engine does not use piston rings in metal because the piston extension is not lubricated with oil. The piston ring material are made out of a low friction material. Two types of material have been used in this work; either a rulon material of PTFE, or a mixture of PTFE, 55 % bronze, and 5 % molybdenum disulphide.

The optical engine will give a lower compression ratio compared to a metal engine with the same squish height, due to increased crevice volumes.

3.4 Natural luminosity imaging

The combustion in the optical engine is captured by high speed camera. The light emitted from the combustion (the natural luminosity) consists of chemiluminescence and black body radiation.

Chemiluminescence is the light emission excited molecules originating from fuel molecules that have started to react. Examples of these molecules that emit chemiluminescence are CH^* , OH^* and C_2^* . [56] Blackbody radiation is a broadband radiation distribution that varies depending on the temperature of the black bodies. In cases with a lot of soot content in the combustion, black body radiation will be dominant. It should be noted that soot is not a perfect black body. [57]

3.5 Mie Scattering

Both Mie scattering and Rayleigh scattering are elastic scattering events, and the elastic scattering depends on the size of the scattering particle. Mie scattering is dominant when the particle is approximately of the same size as the wavelength of the incident light or larger, while Rayleigh scattering is dominant when the scattering particle is smaller than the wavelength. [58] [59] In this thesis, Mie scattering was used to image the liquid portion of the fuel spray. The combustion chamber was illuminated with a 3 W continuous laser diode with a wavelength of 452 nm. As the liquid in the fuel spray consists of small droplets, these will scatter the light in the combustion chamber. Some of the scattered light is detected by a high-speed camera to investigate the development of the liquid portion of the fuel spray. The distribution of Mie scattering is angularly dependent, but the intensity of the scattering is not as dependent on the wavelength as for Rayleigh scattering, which is commonly used to study the gas phase.

3.6 PLIF

Laser induced fluorescence (LIF) can be used to measure different species, species concentration, and temperatures. In this thesis, PLIF has been used to detect the fuel distribution.

The laser pulses are used to excite the atom or molecule of interest through absorption of light. The wavelength of the laser is selected based on the atom or molecule of interest. The energy of the light will correspond to the distance between two energy levels of the atom or molecule. As the fuels used in this thesis are non-fluorescent, acetone has been selected as a fuel tracer in order to identify where the fuel is located. When the molecule absorbs the light from the laser, the molecule will increase its energy and be transferred to an excited state. However, the molecule will not remain stable in the excited state and will transfer back to a lower energy state, for example by spontaneous emission of light. (The molecule can also deexcite through a non-radiative process, often called collisional quenching.) The spontaneous light emission has a longer wavelength because of shifts in rotational and vibrational states of the molecule and due to allowed transitions in energy level. It should be noted that there are some exceptions to this, for example when two-photon LIF is used or for anti-stokes shift.

Acetone was selected as a tracer for the fuel distribution since it is advantageous if the fuel properties and the tracer properties are similar so that the tracer can mimic the behavior of the fuel. [60] According to Yip *et al.* acetone have physical and chemical properties comparable to those of hydrocarbon fuels, at least if very light fuels such as methane are avoided. [61, 62] The pressure dependence of the LIF signal has also been shown to be weak at pressures comparable to those found in

an engine. [63] In addition to those, acetone is non-toxic and it has a relatively low cost.

In the experiments, 10% (vol.) of acetone was added to the fuel mixture as a fuel tracer. A pulsed 10 Hz Nd-YAG laser with a wavelength of 266 nm was used to excite the acetone molecule. The Nd-YAG laser originally emits a wavelength of 1064 which was sent through two separate non-linear crystals that doubles the frequency of the light. The remaining longer wavelengths were diffracted with the help of a Pellin-Broca prism and blocked by a beam dump, only to let the 266 nm through. The laser is sent through a set of lenses to create a laser sheet that is directed through a window in the optical engine. In front of the camera, 2 liquid filters were used to block the camera from capturing scattered light from the laser wavelength.

It is possible to determine the equivalence ratio based on the LIF-signal. The intensity of the LIF-signal can be expressed in the following terms [64] [65]:

$$(3.10) \quad I_{LIF} = \frac{E}{hv} \frac{\chi_{tracer} P dV}{kT} \sigma(T, P, \chi, \nu) \eta(T, P, \chi, \nu) \varepsilon,$$

where E is the laser fluency [J/m^2], hv is the energy of the photon, χ_{tracer} is the mole fraction of the tracer, P is the pressure [Pa], dV is the volume depicted in one pixel [m^3], and T is the local temperature [K]. The remaining factors are σ , the cross section for absorption [m^2], η , the fluorescence quantum yield [-], and ε , the efficiency of the optics and detector. The absorption cross section and the fluorescence quantum yield are dependent on the local temperature (T), pressure (P), local molecular composition (χ), and the frequency of the light (ν).

By assuming a stable setup and laser, some simplifications can be made. Under non-reacting conditions, changes in local molecular composition can be neglected. Furthermore, acetone is relatively pressure independent at pressures above 5 bar. [66] This allows us to write equation (3.10) in the form:

$$(3.11) \quad I_{LIF} = CE \frac{\chi_{tracer} P}{T} \sigma(T) \eta(T) = CE \chi_{tracer} P F(T).$$

The constant C and the temperature dependence can be determined through calibration measurements with varied fuel amounts and temperatures, enabling the equivalence ratio to be determined.

The expression for the equivalence ratio is the following:

$$(3.12) \quad \phi = \frac{\chi_{tracer} + \chi_{fuel}}{\chi_{air}} / \left(\frac{\chi_{fuel} + \chi_{tracer}}{\chi_{air}} \right)_{stoichiometric}$$

For a given temperature, the absorption cross-section and fluorescence quantum yield is constant, but to evaluate the equivalence ratio more accurately, the variations in temperature from the fuel concentration needs to be taken into account. Adiabatic mixing of the gas and fuel is assumed, meaning that the mixture temperature is only affected by the initial temperatures of the two components, and not by heat exchange with the walls. The temperature of the liquid fuel during the injection is much lower compared to the gas temperature, but will increase during the mixing with the surrounding gas. The temperature of various mixtures of fuel and air for different temperatures can be calculated and tabulated. Determining the temperature for the mixture will improve the estimation of the ϕ -values. The temperature of the mixture can be calculated according to [67]:

$$(3.17) \quad \int_{T_{mix}}^{T_{a,ini}} c_p dT = \frac{F}{A} \left[\int_{T_{liquid\ fuel}}^{T_{int}} c_p dT + h_{g,T_{int}} + \int_{T_{ini}}^{T_{mix}} c_{p,g} dT \right]$$

In this thesis, fully quantitative evaluation of the equivalence ratio is not used, since neither the temperature dependence of the absorption cross-section and the quantum yield, nor the adiabatic mixture temperature are considered. The signal strength is evaluated to make relative comparisons of fuel concentrations between squish plane and the piston bowl plane. The LIF-images from the measurements are divided by a flat field image in order to remove effects of variations of laser power across the sheet, the efficiency of the detector and filters, and so forth, as seen in eq (3.18-19). With this method it is not possible to compare the signal strength between different cases, but is sufficient for the experiment to compare the signal strength between the squish plane and the piston bowl plane. The background is subtracted from both the LIF-image and the flat field image before dividing the images with each other, see eq. (3.18-19).

$$(3.18) \quad S_{squish_SOI_case} = \frac{I_{SOI_case_squish} - I_{Background_SOI_case_squish}}{I_{Flatfield_squish} - I_{Background_squish}}$$

$$(3.19) \quad S_{bowl_SOI_case} = \frac{I_{SOI_case_bowl} - I_{Background_SOI_case_bowl}}{I_{Flatfield_bowl} - I_{Background_bowl}}$$

The resulting signal S only depends on the fuel concentration and the local temperature. The local temperature is uniquely coupled to the fuel concentration if adiabatic mixing is assumed, making relative comparisons of the fuel concentrations between the two laser planes possible.

4 Results

The result from the experiments will be presented and discussed in this chapter. The results are divided into two main parts. The first part includes the investigation of the transition from PPC to HCCI. This is further divided into sections based on the experimental campaigns and simulation work. The results will begin from the first investigation of this transition in a metal engine to continue with the experimental campaigns in the optical engine. In the second part, the characterization of methanol combustion will be presented together with how pilot injections can facilitate the combustion of methanol.

4.1 Transition from PPC towards HCCI

4.1.1 Metal engine

In order to investigate the transition from PPC to HCCI, a SOI-sweep was carried out in a single cylinder metal engine. The combustion timing was kept constant for the SOI-sweep by adjusting the inlet temperature.

Single injection and double injection

As seen in Figure 4.1, the inlet temperature required to keep the combustion phasing constant shows a complex dependency on the injection timing for both single and double injection cases. The single injection in figure 4.1 corresponds to Case 1. For injection timings earlier than -80 CAD ATDC, the required inlet temperature is rather constant, which indicates that the conditions are homogeneous. When moving to later injection timings between -80 and -50 CAD ATDC, a dip in the required inlet temperature can be noticed. The required inlet temperature decreases steeply when approaching later injection timings between roughly -50 and -25 CAD ATDC, then increases again for later injection timings than -25 CAD. The gasoline fuel needs time to vaporize and mix before auto-ignition. When the mixing time is

reduced too much due to retarded injection timing, a raised inlet temperature may be needed to speed up the vaporization and reduce the ignition delay.

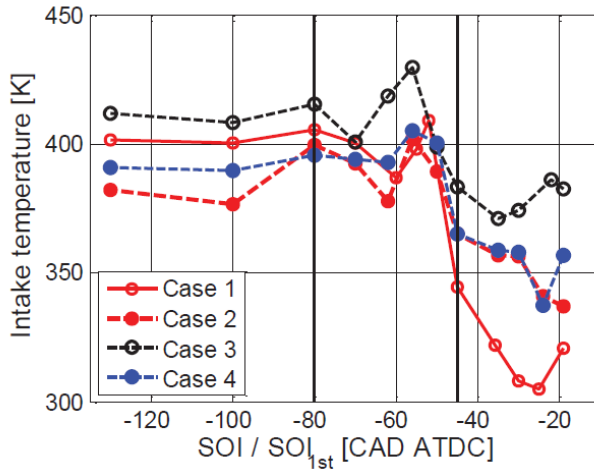


Figure 4.1 Required inlet temperature to keep the combustion phasing constant for an SOI-sweep.

The emissions of CO and UHC shows the same type of complex behaviour as the inlet temperature, as shown in Figure 4.2. The study of the transition from PPC to HCCI shows that an SOI-sweep can be utilized to shift the combustion towards either more PPC-like combustion or more homogeneous conditions. Three regions were identified. HCCI-like combustion was found for injection timings between -100 and -80 CAD ATDC. In the injection timing region between -80 and -45 CAD ATDC, a more complex behaviour was found for the required inlet temperature as well as for the HC and CO emissions. The third and final region occurred for injections starting later than -45 CAD ATDC, where most of the fuel is expected to go into the piston bowl.

The findings in this study initiated several of the following studies in order to find a better understanding of the complex behaviour of the inlet temperature and emission such as CO and HC.

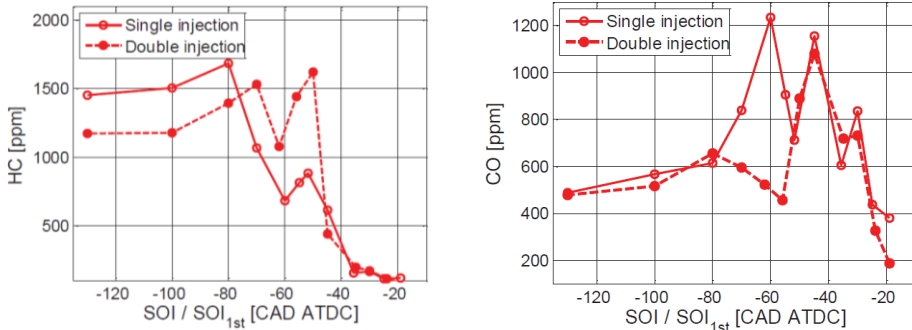


Figure 4.2 Emissions of HC, CO, NOx and soot emissions for the SOI-sweep.

4.1.2 Optical engine

Spray Characterization

The previous study in the metal engine showed interesting behaviours between the PPC combustion and more HCCI-like combustion. This initiated the interest for optical studies of the spray and combustion to gather more information of what happens in the cylinder during the SOI-sweep. The liquid fuel in the sprays was visualized in order to find whether the sprays interact with the piston and, if so, in what way and for which injection timings. It was also of interest to visualize the emission of light from the combustion, to evaluate how the combustion changes during the SOI-sweep.

The spray interaction with the piston can be evaluated based on analysis of the spray images, see Figure 4.3. The SOI-sweep is divided into 3 different regimes, based on where the spray is directed in relation to the piston position. In the first regime, here called the piston bowl regime, the fuel is injected into the piston bowl. In the transition regime the spray is divided between the piston bowl and the squish region. The third regime is the squish regime where all of the fuel is injected into the squish region.

Required inlet temperature

Figure 4.4 shows the inlet temperature required to maintain constant combustion phasing in the optical engine. It displays the same trend as seen in figure 4.3 for the SOI-sweep in the metal engine

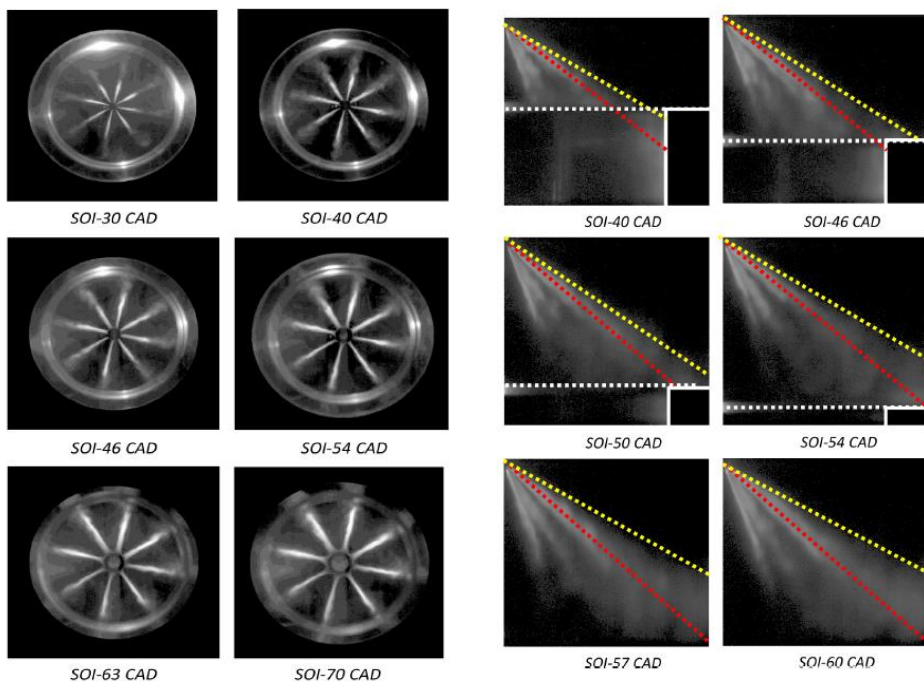


Figure 4.3 Spray images for different SOI-timings taken from beneath the piston (left) and from the side (right).

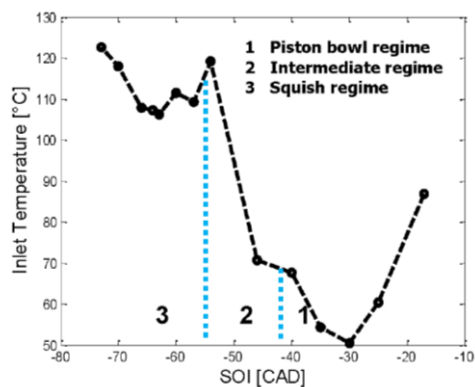


Figure 4.4 The required inlet temperature for the SOI-sweep.

Combustion characteristics

The single shot images in figure 4.5 shows that the combustion is affected by where the fuel is injected. For the late injection timings, -17 and -30 CAD ATDC, the combustion is concentrated to the piston bowl. This can be understood from the spray images as all of the fuel enters the piston bowl during the injection. For the latest injection case, there is an indication that the combustion area is divided into 8 semi-separated zones making the combustion more stratified. With injection timing -30 CAD, the fuel and air has had more time to mix which also shows in that the combustion is distributed over the entire combustion chamber. In the transition regime the combustion is still concentrated to the piston bowl. However, in the squish regime the combustion is focused on the periphery of the piston. The results also indicate that more of the fuel has gone into the piston bowl for the -70 CAD injection timing compared to the rest of the timings in the squish regime.

Combustion location

During the sweep from early to late injection timings, the radial combustion location shifts from the piston bowl to the squish region and then back to the piston bowl again. The radial location of the combustion was determined from 50 cycles for each injection timing. The combustion location can be seen in figure 4.6. Combustion starts in the squish region for injection timings around SOI-63 CAD ATDC. It can be seen that there is a spread in where the combustion starts, but even though the combustion starts in the piston bowl for some of the cycles, the majority of the cases start in the squish region. For timings more advanced than -70 CAD and more retarded than -54 CAD ATDC, the initial combustion locations are back in the piston bowl. It could also be noted that for the -54 CAD timing, the distribution in the combustion location is wide, which is a further indication of the transition regime of the spray.

A hypothesis aiming to explain the initial combustion location was formulated: Considering where most of the fuel seems to be located at the time of auto-ignition, the combustion seems most likely to start in fuel-rich regions.

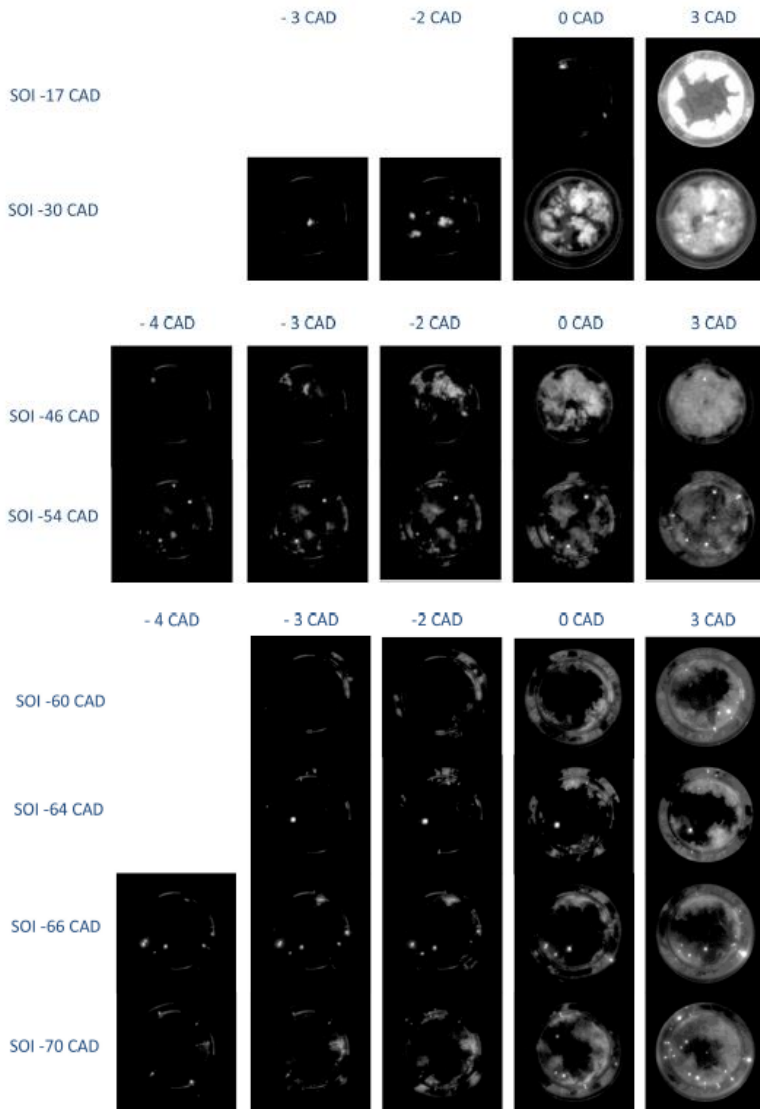


Figure 4.5 Single shot images of the natural luminosity for selected SOI-timings separated in the piston bowl regime, transition regime and the squish regime.

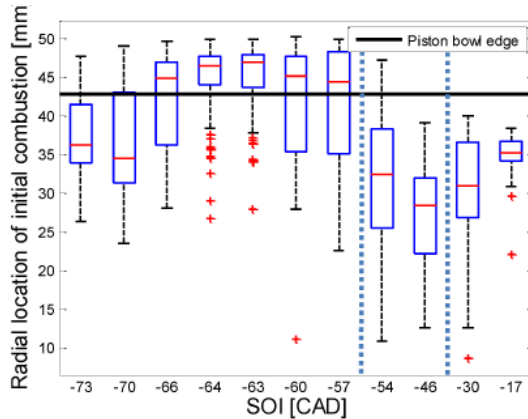


Figure 4.6 Radial combustion location for the SOI-sweep.

4.1.3 Fuel distribution from injection to combustion

The fuel distribution in the engine was visualized by detecting laser-induced fluorescence of the fuel tracer (acetone), as seen in Figures 4.7-4.8. The laser was shaped into a vertical laser sheet using a combination of lenses and sent into the engine through a side window. Qualitative LIF-measurements were captured every 5 crank angle degree. The LIF measurements were carried out in order to gain more information of the mixing process of the fuel.

In Figure 4.7, the LIF signal of the fuel is displayed 10 CAD after the injection timing. It can be seen even more clearly for the PLIF measurements that the fuel is broken up by the corner of the piston bowl wall for the -46 and -54 CAD timings. While a part of the fuel is pushed into the squish region, the rest enters the piston bowl. When the fuel spray is divided by the piston and is injected in the squish volume, it can be seen in the emission measurements in the metal study that the CO and HC increases. This causes the efficiency of the combustion to decrease and it is therefore desired to avoid this region. This shows the effect that the piston geometry and the umbrella angle of the injector can have on the combustion efficiency.

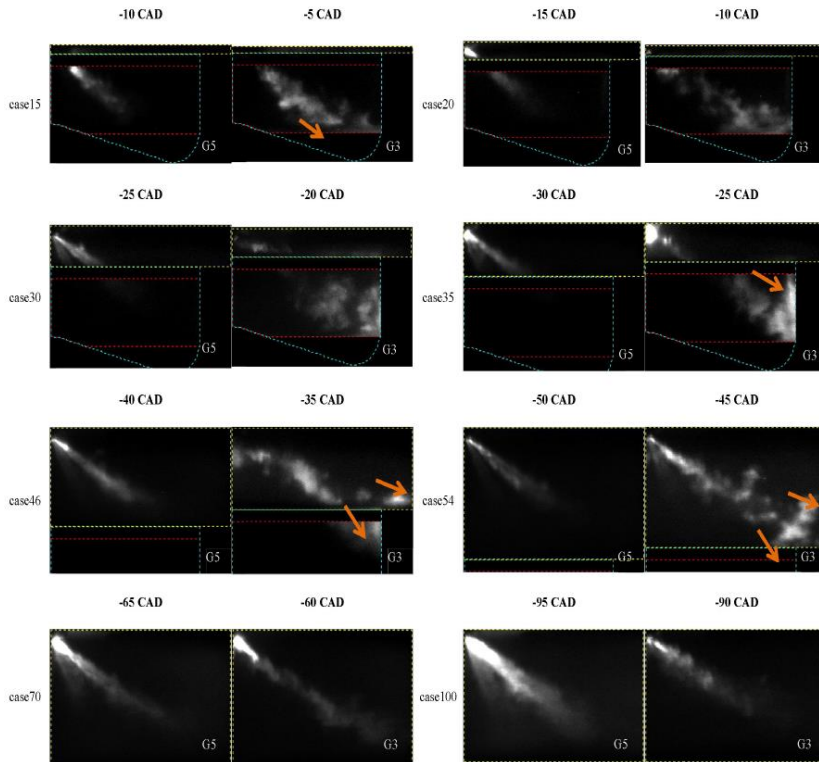


Figure 4.7. LIF-images of spray 5 and 10 CAD after injection timing.

The single shot images of the fuel distribution at various crank angle degrees are shown in figure 4.8. It can be seen that for injection timings between -54 and -100 CAD the signal is weaker and more evenly distributed which indicates that the fuel is more well mixed.

Figure 4.9 shows the radial distribution of the LIF signal at -5 CAD ATDC for all of the injection cases. It seems that for the late injection cases (-20 to -35 CAD ATDC), most of the fuel will be located close to the piston bowl wall. When the injections timings become more advanced, the fuel becomes more evenly distributed in the piston bowl, as seen in the -46 CAD case. But in the even earlier injection cases, it can be seen that the amount of fuel is slightly increasing when moving outwards in the piston bowl.

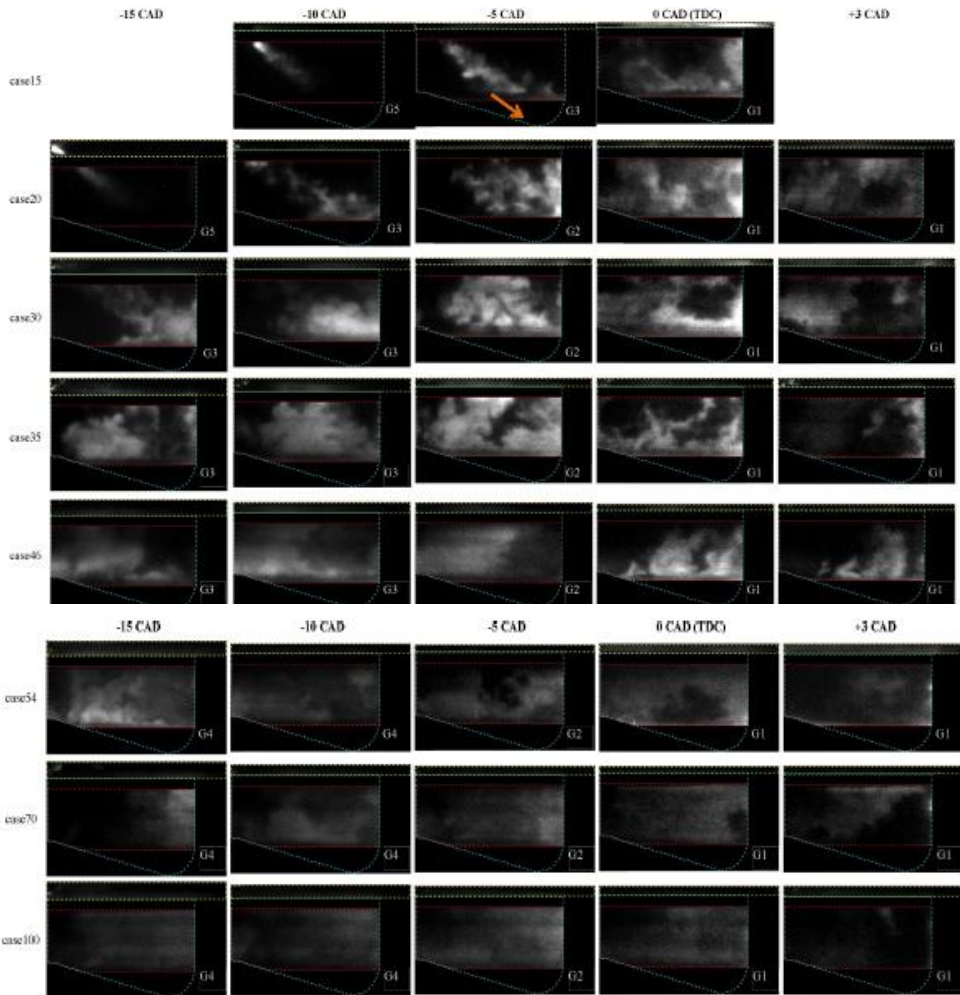


Figure 4.8 LIF-signal of the fuel tracer

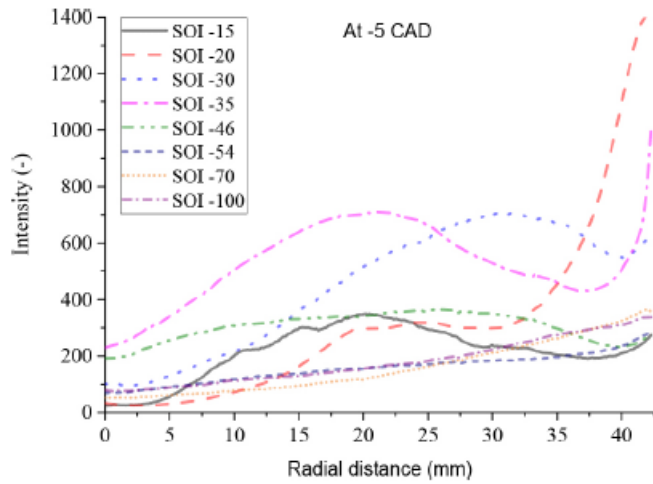


Figure 4.9 Averaged LIF-signal at -5 CAD ATDC for each injection timing.

PLIF and CFD simulations

The fuel distribution was measured at two different heights in the engine, see figure 4.10. The purpose with this study is to evaluate the fuel distribution when the combustion starts for a number of selected injection timings.

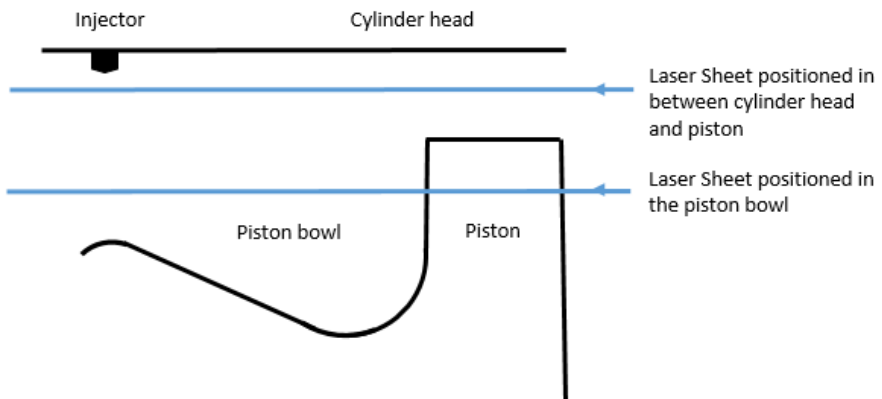


Figure 4.10 Schematic illustration of the positions of the laser sheet.

The LIF-data is then compared with CFD simulations (not performed by the author) of the same injection timings, shown in Figures 4.11 and 4.12. The first shows the fuel distribution in the piston bowl, 14 mm from the cylinder head, whereas the other shows the fuel distribution in the squish area, 1.3 mm from the cylinder head. Similar fuel distributions are seen in the PLIF measurements and the CFD simulations. The hypothesis that the combustion starts in the richer fuel mixtures seem to correlate well with measurements for the late injection cases. Furthermore, SOI-63 CAD ATDC appears to have the more fuel in the squish region which supports the hypothesis that richer fuel mixtures are easier to combust. However, when the fuel is distributed more evenly as in SOI-70 CAD ATDC for the squish plane, it seems that the local temperature determines the ignition delay since it is expected that the squish region has a higher heat transfer. In the earlier injection cases it shows that the combustion is not only sensitive to higher equivalence ratios but is also affected by the heat transfer.

Piston bowl – 14 mm from cylinder head

soi

LIF-measurements CFD

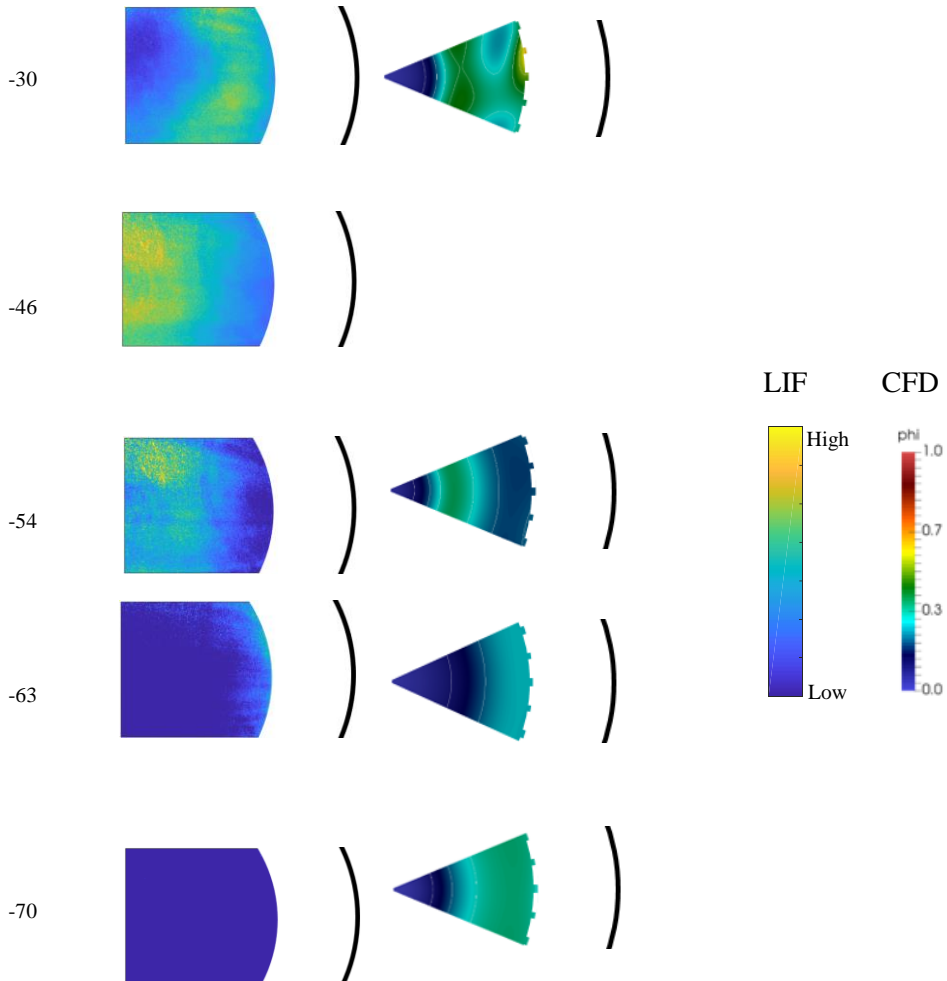


Figure 4.11 Fuel-distribution evaluated from LIF measurements and from CFD in the piston bowl.

Squish – 1.3 mm from cylinder head

soi LIF-measurements

CFD

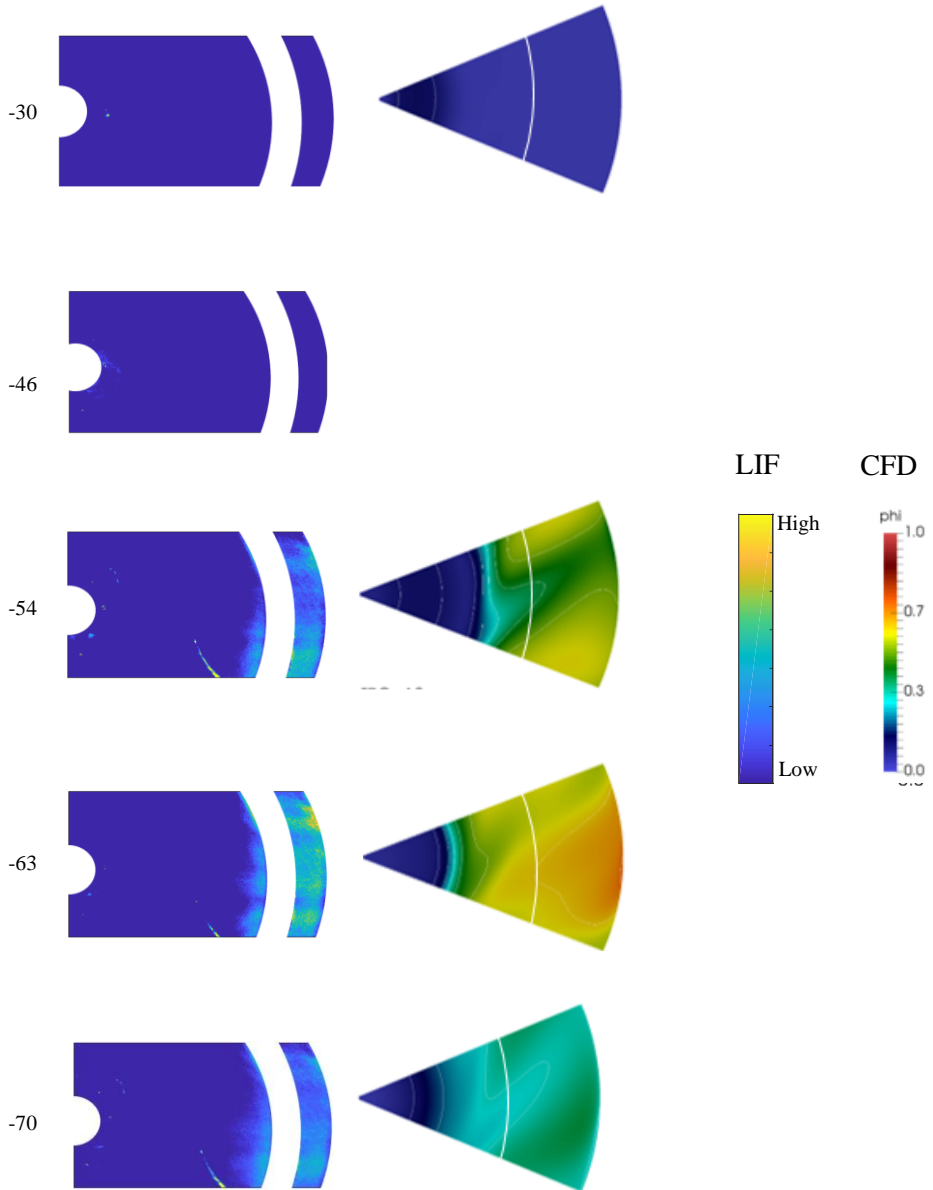


Figure 4.12 Fuel-distribution evaluated from LIF measurements and from CFD in the piston bowl.

By combining the information from Figures 4.11-4.13, it can be seen that the combustion does not only start in the regions with the highest fuel equivalence ratio. It can be seen that the temperature also provides an important role from the CFD simulations and the results of the LIF measurements. The results indicate that the heat transfer is more pronounced in the squish region. The CFD simulations (not performed by the author) did not display that the combustion started in the squish region for any of the injection timings. It should be noted that the CFD has another wall temperature compared to the experiments. The difference in wall temperature can explain some of the differences in combustion location for CFD and LIF-measurements. However, for the SOI -63 CAD ATDC, the combustion started close to the squish region. This is a difference compared to what was seen in the optical engine, where the combustion location for SOI-63 CAD ATDC started in the squish region. It seems that in the RANS simulations the heat losses in the squish region becomes more important compared to the experimental data.

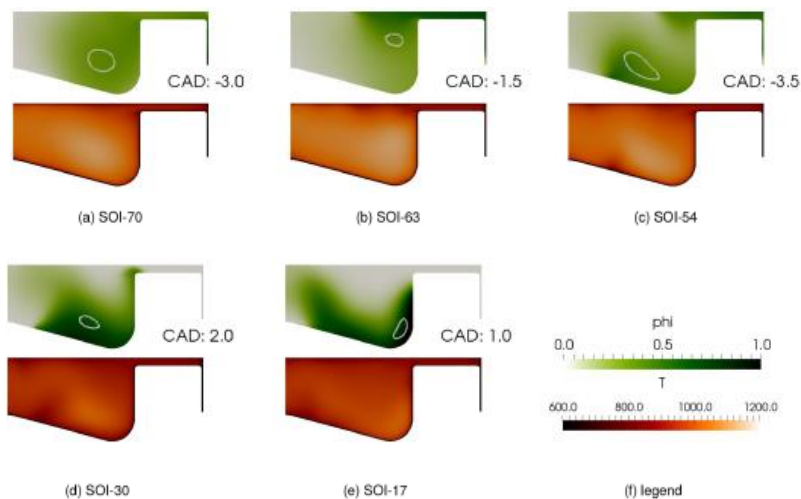


Figure 4.13 Vertical ϕ -distribution based on CFD simulations for selected injection timings.

4.2 Methanol

4.2.1 Characterization of the methanol combustion process

Single Injection Strategy with fixed CA50

In the first part, a single injection strategy was used to explore methanol's reluctance to autoignition by determining the inlet temperature required to keep CA50 fixed. Two different injection timings were also used to capture differences in combustion modes between injections around TDC to slightly earlier.

Figure 4.14 shows the natural luminosity of the methanol combustion for injection at -15 CAD ATDC. In this case the injection of methanol lasted until roughly TDC and the combustion started close to the end of injection. However, the fuel seems to have had time to mix, as indicated by the absence of clearly defined combustion "clouds" at the end of each fuel spray. An example of how the more defined combustion clouds may look can be seen for the -17 CAD case in Figure 4.5, where the fuel has not had sufficient time to mix before combustion.

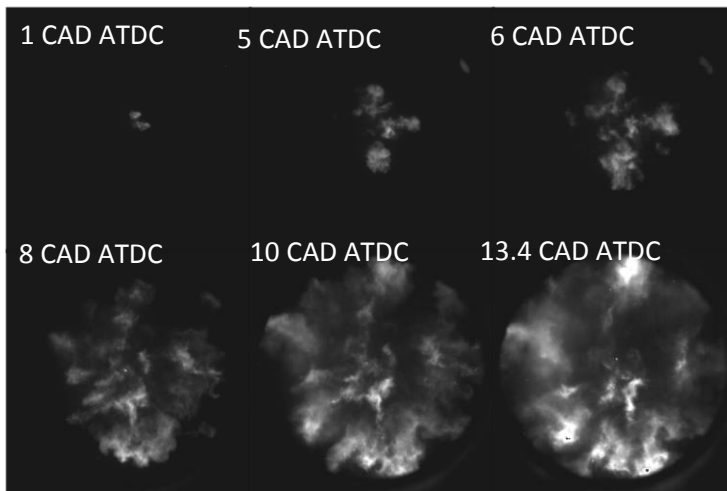


Figure 4.14 Combustion images for SOI -15 CAD ATDC single injection case for methanol.

Figure 4.15 shows the natural luminosity of methanol injected at -6 CAD ATDC. It can be seen that the fuel is injected during the combustion. The combustion starts close to the fuel spray and spreads outward and to the adjacent sprays. It is possible

to see that the combustion starts close to the nozzle for both of the single injection cases.

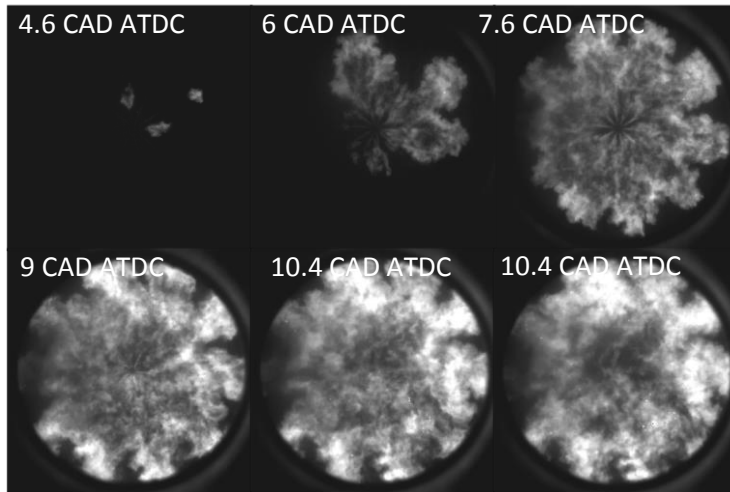


Figure 4.15. Combustion images of methanol single injection case. SOI: -6 CAD ATDC.

The CFD simulation of these injection timings (not performed by the author) show that for both of the cases the combustion starts lean below $\phi=0.5$. The reason for methanol to start so lean could be due to the high heat of vaporization, lowering the temperature in the areas with more fuel and making them more difficult to ignite. The difference in the injection timing between the cases is 9 CADs. The CFD simulations shows that there is a difference in the ϕ -distribution for the single injection cases. This is an indication that lower equivalence ratios can be reached quicker due to the oxygen content in the fuel.

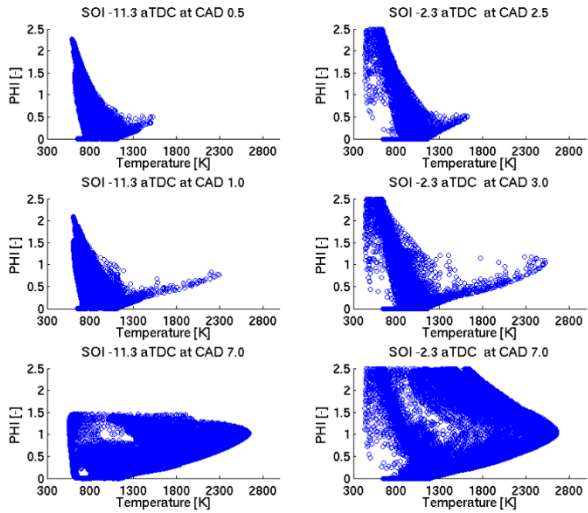


Figure 4.16. Φ -T diagrams at three different crank angle degrees for the single injection cases.

Double Injection Strategy with fixed CA50

In the second part, double injection strategies were adopted to investigate how much the inlet temperature required to keep CA50 constant could be decreased. Figure 4.17 shows the injection timings for the double injection strategies. The best strategy of the four double injection strategies is an advanced first injection while keeping a large separation between the injections. By injecting the first injection at -20 and the second at -6 CAD ATDC, the inlet temperature could be lowered 58°C compared to the worst case for the single injection.

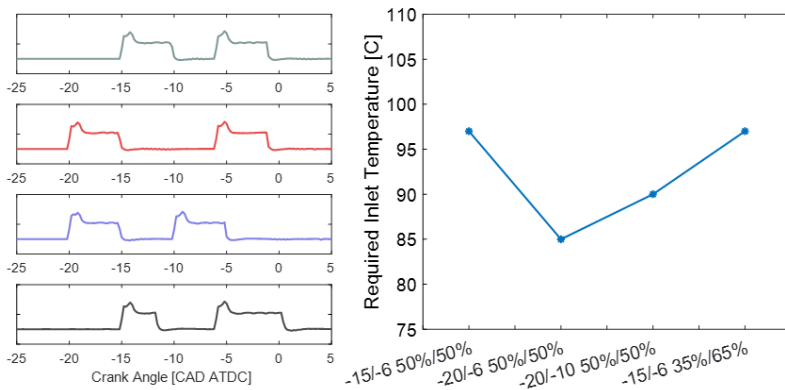


Figure 4.17. Injection timings for double injection strategies.

Pilot/Main Injection Strategy with fixed temperature

In the final part of the methanol study, a pilot/main injection strategy was used. The purpose of the study was to investigate if a small portion of fuel injected such that it gives a small positive contribution to the heat release around TDC could trigger the injection of a non-combusting main injection.

The inlet temperature for the pilot/main injection strategy was fixed at 100 °C since the main injection event did not combust at that inlet temperature by itself. The main injection timings were varied from -10 to +5 CAD ATDC in steps of 5 CAD. The pilot injection event was set to -25 CAD ATDC.

Figure 4.18 shows that the pilot injection enables the combustion of the main injection. An attempt to provide an explanation as to why the combustion is enabled by the pilot injection is shown in Figure 4.19. The in-cylinder temperature is shown for the main injection (only), pilot injection (only), a motored cycle, and the combination of the pilot and main injection. The figure shows that the pilot injection will raise the temperature in the cylinder before the temperature drops due to vaporization of the main injection. This increase in temperature is sufficient to trigger the combustion of the main injection. It could also be noted that the auto-ignition occurs around 1000 K. By dividing the fuel into smaller portions, the temperature will not decrease as much and will therefore enable the combustion.

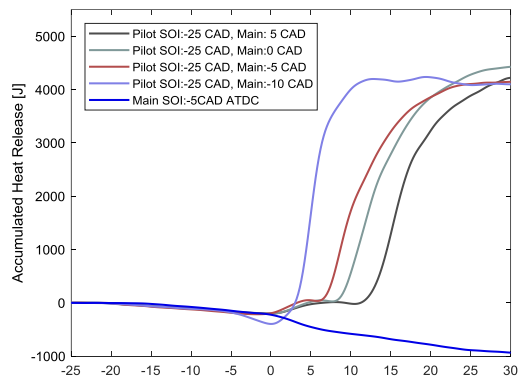


Figure 4.18. Accumulated heat release for four pilot/main injection cases and a single injection case.

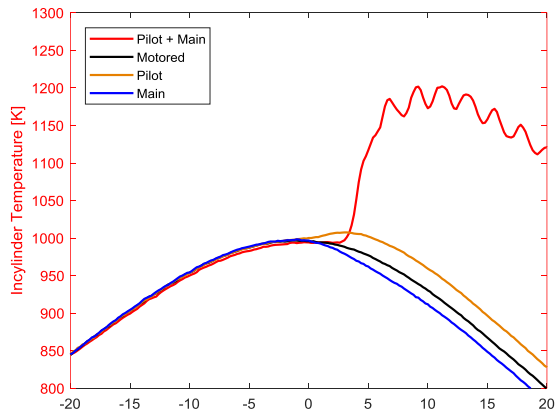


Figure 4.19. In-cylinder temperature for pilot, main and combined pilot/main-injection case and the motored case.

5 Thesis contribution

The thesis has contributed to gaining more knowledge in what happens in the transition between PPC mode and HCCI mode, when more and more fuel is injected into the squish volume. It also provides knowledge of how methanol combustion behaves in a direct injection compression ignition engine running in PPC mode or spray driven combustion.

5.1 Transition from PPC to HCCI

This thesis has identified the complex behaviour of the charge reactivity as the injection timing is advanced to enable the transition between PPC mode to more HCCI-like combustion. The variation in charge reactivity is indicated by the inlet temperature required to maintain a fixed combustion phasing, which varies in a highly non-linear manner during the transition. This behaviour is also seen for CO and HC emissions. The most complex behaviour is observed for injection timings between -70 and -54 CAD ADTC.

Studies in an optical engine have also shown that the combustion location is affected in this period of injection timings. The combustion location shifts towards the squish region from the piston bowl wall when the required inlet temperature is reduced.

The thesis has contributed in gaining more knowledge of the how the fuel is distributed and mixed in the combustion chamber from injection towards TDC by utilizing fuel tracer PLIF. Finally, horizontal fuel PLIF in combination with CFD has been utilized to evaluate if the combustion location is initiated at the richer equivalence ratios.

In summary, the results indicate that the non-linear behaviour in charge reactivity results from the interaction between the spray and the piston bowl edge. As the injection timing is advanced, increasing shares of the fuel will eventually be injected into the squish volume. As the fuel is divided between the bowl and the squish volume, the local equivalence ratios in the two zones will vary accordingly. Broadly speaking, richer mixtures seem more prone to auto-ignite, but there are instances that suggest an influence of the local temperature as well. This is the case when the fuel distribution is relatively even between the squish volume and the adjacent

region at the top of the bowl. In those cases, the fuel in the squish volume tends to auto-ignite in the bowl, probably due to increased heat losses.

5.2 Methanol combustion

This thesis has contributed with high-speed visualization of methanol combustion in an optical compression ignition engine. The combustion mode varied from PPC to more mixing-controlled combustion. In the latter case, the fuel was still injected into the engine when the combustion started. It has been shown that methanol is ignited close to the nozzle. The combustion is fast and that the boundaries between the sprays during combustion are not clearly defined.

Different injection strategies have been applied in order to investigate if the effects of the high heat of vaporization can be mitigated, which increases the inlet temperature required to achieve auto-ignition. It was found that a big separation between the injection timings was advantageous in lowering the required inlet temperature.

Furthermore, it has been found that a pilot injection can facilitate the combustion for the main injection. This was tested by adding a pilot injection to a main injection that by itself would not combust at a constant temperature.

6 References

- [1] UN, “United Nations - Global Issues - Climate Change,” [Online]. Available: <http://www.un.org/en/sections/issues-depth/climate-change/index.html>. [Accessed 08 March 2019].
- [2] World Health Organisation (WHO), “World Health Organisation - How air pollution is destroying our health,” [Online]. Available: <https://www.who.int/air-pollution/news-and-events/how-air-pollution-is-destroying-our-health>. [Accessed 08 March 2019].
- [3] World Health Organisation (WHO), “World Health Organisation - 9 out of 10 people worldwide breath polluted air but more countries are taking action,” 02 05 2018. [Online]. Available: <https://www.who.int/news-room/detail/02-05-2018-9-out-of-10-people-worldwide-breathe-polluted-air-but-more-countries-are-taking-action>. [Accessed 8 March 2019].
- [4] T. Stocker, D. Qin, G.-K. Plattner, M. Tignor, S. Allen, J. Boschung, A. Nauel, Y. Xia and V. B. a. P. M. (eds.), “IPCC, 2013: Climate Change 2013: The physical science Basis. Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change,” Cambridge University Press, 2013.
- [5] Edenhofer, Pichs-Madruga, Sokona, Farahani, Kadner, Seyboth, Adler, Baum, Brunner, Eickemeier, Kriemann, Savolainen, Schlömer, v. Stechow and Zwickel, “IPCC, 2014: Climate Change 2014: Mitigation of Climate Change. Contribution of Working Group III to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change,” Cambridge University Press, 2014.
- [6] “Inventory of U.S. Greenhouse Gas Emissions and Sinks,” U.S. Environmental Protection Agency, 2019.
- [7] Eurostat, “Greenhouse gas emission statistics - emission inventories,” June 2018. [Online]. Available: <http://ec.europa.eu/eurostat/statistics-explained/>

- index.php/Greenhouse_gas_emission_statistics_-_emission_inventories. [Accessed 07 March 2019].
- [8] World Health Organisation (WHO), “WHO - Sustainable development - Transport - Health Risks - Air pollution,” [Online]. Available: <https://www.who.int/sustainable-development/transport/health-risks/air-pollution/en/>. [Accessed 8 March 2019].
- [9] S. van de Hoef, “Coordination of Heavy-Duty Vehicle Platooning”.
- [10] VTI, “Electric roads: a solution for the future,” [Online]. Available: <https://www.vti.se/en/news/electric-roads-a-solution-for-the-future/>. [Accessed 14 March 2019].
- [11] IEA, “Global EV outlook 2018, towards cross-modal electrification,” IEA, 2018.
- [12] OICA, “PC World Vehicles in use,” http://www.oica.net/wp-content/uploads/PC_Vehicles-in-use.pdf, Retrieved 20190129.
- [13] EEA, European Environment Agency, “Electric Vehicles in Europe,” Publications Office of the European Union, 2016, Luxembourg, 2016.
- [14] Ny Teknik, John Edgren, “Tesla Semi har klarat av sin första långkörning med last,” 28 08 2018. [Online]. Available: <https://www.nyteknik.se/fordon/tesla-semi-har-klarat-av-sin-forsta-langkorning-med-last-6927560>. [Accessed 01 05 2019].
- [15] Electrek, Fred Lambert, “Electrek: Mercedes-Benz delivers first 10eActras all-electric heavy duty trucks,” 17 09 2018. [Online]. Available: <https://insideevs.com/news/340380/mercedes-benz-delivers-first-10-eactros-electric-trucks/>. [Accessed 01 05 2019].
- [16] J. Heywood, Internal Combustion Engine Fundamentals, McGraw Hill, 1988.
- [17] I. Glassman, R. Yetter and N. Glumac, Combustion, Academic press, 2014.
- [18] G. Stiech, Modeling Engine Spray and Combustion Processes, Springer, 2013.
- [19] S. L. Kokjohn, M. P. Musculus and R. D. Reitz, “Evaluating temperature and fuel stratification for heat-release rate control in a reactivity-controlled compression ignition engine using optical diagnostics and chemical kinetics modeling,” *Combustion and Flame*, vol. 162, no. 6, pp. 2729-2742, 2015.
- [20] D. T. Klos and S. L. Kokjohn, “Investigation of the Effect of Injection and Control Strategies on Combustion Instability in Reactivity-Controlled

Compression Ignition Engines,” *Journal of Engineering for Gas Turbines and Power*, vol. 12, no. 3, pp. 209-226, 2011.

- [21] D. E. Niemann, A. B. Dempsey and R. D. Reitz, “Heavy-duty RCCI operation using natural gas and diesel,” *SAE International Journal of Engines*, pp. 270-285, 2012.
- [22] M. Lundgren, Avhandling, Lund University, 2016?.
- [23] I. M. and K. T., *Conventional diesel combustion*, Springer, 2008.
- [24] U. Aronsson, *Processes in Optical Diesel Engines - Emission Formation and Heat Release*, Lund University, 2011.
- [25] J. Dec, “A conceptual model of DI diesel combustion based on laser sheet imaging,” in *SAE970873*, 1997.
- [26] J. Dec and Canaan, in *SAE 980147*, 1998.
- [27] S. Kook, C. Bae, P. C. Miles, D. Choi and L. M. Pickett, “The influence of charge dilution and injection timing on low-temperature diesel combustion and emissions,” *SAE Technical Paper*, no. 2005-01-3738, 2005.
- [28] S. Dev, H. B Chaudhari, S. Gothekar, S. Juttu and e. al., “Review on Advanced Low Temperature Combustion Approach for BS VI,” in *SAE Technical Paper 2017-26-0042*, 2017.
- [29] I. Truedsson, *The HCCI Fuel Number - Measuring and Describing Auto-Ignition for HCCI Combustion Engines*, Lund University, 2014.
- [30] Zhao, Asmus, Assanis, Dec, Eng and Najt, “Homogeneous Charge Compression Ignition: Key research and development issues,” *Society of Automotive Engineers*, 2003.
- [31] M. Sjoberg and J. Dec, “An Investigation into Lowest Acceptable Combustion Temperatures for Hydrocarbon Fuels in HCCI Engines,” in *Proceeding of the Combustion Insitute*, 2005.
- [32] W. Lin, J. Sterniak and S. Bohac, “NOx Emissions Characterization During Transient Spark Assisted Compression Ignition (SACI) Engine Operation,” in *ASME. Internal Combustion Engine Division Fall Technical Conference*, 2013.

- [33] S. Kimura, O. Aoki, H. Ogawa, S. Muranaka and Y. Enomoto, "New Combustion concept for ultra-clean and high-efficiency small DI diesel engines," in *SAE Technical Paper 1999-01-3681*, 1999.
- [34] S. Kimura, O. Aoki, Y. Kitahara and E. Aiyoshizawa, "Ultra-clean combustion technology combining low-temperature and premixed combustion concept for meeting future emission standards," in *SAE Technical Paper 2001-01-0200*, 2001.
- [35] R. Hasegawa and H. Yanagihara, "HCCI combustion in DI diesel engine," in *SAE Technical Paper 2003*, 2003.
- [36] R. Hasegawa and H. Yanagihara, "HCCI Combustion in DI Diesel Engine," *SAE Technical Paper*, pp. 2003-01-0745, 2003.
- [37] L. Hildingsson, H. Persson, B. Johansson and R. e. a. Collin, "Optical Diagnostics of HCCI and UNIBUS Using 2-D PLIF of OH and Formaldehyde," *SAE Technical Paper*, pp. 2005-01-0175, 2005.
- [38] M. P. Musculus, P. C. Miles and L. M. Pickett, "Conceptual models for partially premixed low-temperature diesel combustion," *Progress in Energy and Science*, vol. 39, pp. 246-283, 2013.
- [39] G. Kalghatgi, P. Risberg and H. and Ångström, "Partially Pre-Mixed Auto-Ignition of Gasoline to Attain Low Smoke and Low NO_x at High Load in a Compression Ignition Engine and Comparison with a Diesel Fuel".
- [40] G. T. Kalghatgi, P. Risberg and H.-E. Ångström, "Advantages of Fuels with High Resistance to Auto-ignition in Late-injection, Low-temperature, Compression Ignition Combustion," *SAE Technical Paper 2006-01-3385*, no. <https://doi.org/10.4271/2006-01-3385>, 2006.
- [41] V. Manente and ..., "Gasoline partially premixed combustion - an advanced internal combustion engine concept aimed to high efficiency, low emissions and low acoustic noise in the whole load range," 2010.
- [42] M. A. Ghadikolaei, "History of Gasoline Direct Compression Ignition (GD CI) Engine - A Review," *International Journal of Research in Engineering and Technology*, vol. 03, no. 01, 2014.
- [43] M. Sellnau, M. Foster, W. Moore, J. Sinnamon and e. al., "Pathway to 50% Brake Thermal Efficiency Using Gasoline Direct Injection Compression Ignition," in *SAE World Congress*, 2019.

- [44] S. Shamun, B. Zincir, P. Shukla, P. G. Valladoid, S. Verhelst and M. Tunér, “Quantification and Analysis of the Charge Cooling Effect of Methanol in a Compression Ignition Engine Utilizing PPC Strategy,” in *ASME International Combustion Engine Division Fall Technical Conference*, San Diego, 2018.
- [45] K. Inagaki, T. Fuyuto, K. Nishikawa, K. Nakakita and I. Sakata, “Dual-Fuel PCI Combustion Controlled by In-Cylinder Stratification of Ignitability,” *SAE Technical Paper*, pp. 2006-01-0028, 2006.
- [46] R. D. Reitz and G. Duraisamy, “Review of high efficiency and clean reactivity controlled compression ignition (RCCI) combustion in internal combustion engines,” *Progress in Energy and Combustion Science*, vol. 46, pp. 12-71, 2015.
- [47] S. Shamun, M. Novakovic, V. Malmberg, C. Preger, M. Shen, M. Messing, J. Pagels, M. Tunér and T. P., “Detailed characterization of particulate matter in alcohol exhaust emissions,” in *COMODIA - 9th International Conference on Modeling and Diagnostics for Advanced Engine Systems*, 2017.
- [48] M. Shen, M. Tuner and B. Johansson, “Close to Stoichiometric Partially Premixed Combustion - The benefits of ethanol is comparison to conventional fuels,” *SAE 2013-01-0277*, 2013.
- [49] M. Dugdale, “European Countries banning fossil fuel cars and switching to electric,” *Road traffic technology*, 01 08 2018.
- [50] Dieselnets, “EU: Fuels: Reference Diesel Fuel,” [Online]. Available: https://www.dieselnets.com/standards/eu/fuel_reference.php. [Accessed 09 04 2019].
- [51] U.S. Department of Energy, “Alternative Fuels Data Center,” [Online]. Available: https://afdc.energy.gov/fuels/fuel_comparison_chart.pdf. [Accessed 09 04 2019].
- [52] K. Rambabu, M. S. R. Murthy, M. Ramji and B. A. Rao, “Study of Combustion and vibration on DI-Diesel Engine Fueled with Rice Bran Methyl Ester Injection and Ethanol Carburetion,” *International Journal of Research in mechanical engineering and Technology*, vol. 4, no. 1, 2014.
- [53] S. Verhelst, J. W. Turner, L. Sileghem and J. Vancoillie, “Methanol as a fuel for internal combustion engines,” *Progress in Energy and Combustion Science*, vol. 70, pp. 43-88, 2019.

- [54] M. Musculus and T. Lachaux, “Two-Stage Ignition and Unburned Fuel Emission for Heavy-Duty Diesel Low-Temperature Combustion of Neat n-Heptane,” in *5th US Combustion Meeting*, San Diego, 2007.
- [55] R. K. Maurya, *Characteristics and Control of Low Temperature Combustion Engines*, Springer, 2017.
- [56] V. Nori and J. Seitzmann, “Evaluation of Chemiluminescence as a Combustion Diagnostic under Varying Operating Conditions,” in *Aerospace Sciences Meeting and Exhibit*, Reno, Nevada, 2008.
- [57] Institution of Mechanical Engineers, IMechE, *Internal Combustion Engines: Improving Performance, Fuel Economy and Emissions*, London: Woodhead Publishing Limited, 2011.
- [58] D. Crolla, *Encyclopedia of Automotive Engineering*, John Wiley & Sons, 2015.
- [59] L. D.J., Rayleigh and Mie Scattering, *Encyclopedia of Color Science and Technology*, Editor Ming Ronnier Luo: Springer, 2016.
- [60] H. Neij, A. Saitzkoff, R. Reinmann, A. Franke and M. Aldén, “Application of Two-Dimensional Laser-Induced Fuel Tracer Fluorescence for Ion Current Evaluation,” *Combustion Science and Technology*, vol. 140, no. 1-6, pp. 295-314, 1998.
- [61] A. Lozano, B. Yip and R. K. Hanson, “Acetone: a tracer for concentration measurements in gaseous flows by planar laser-induced fluorescence,” *Experiments in Fluids*, vol. 13, no. 6, p. 369–376, 1992.
- [62] B. Yip, M. F. Miller, A. Lozano and R. K. Hanson, “A combined OH/acetone planar laser-induced fluorescence imaging technique for visualizing combusting flows,” *Experiments in Fluids*, vol. 17, no. 5, p. 330–336, 1994.
- [63] M. C. Thurber, F. Grisch, B. J. Kirby, M. Votsmeier and R. K. Hanson, “Measurements and modeling of acetone laser-induced fluorescence with implications for temperature-imaging diagnostics,” *Applied Optics*, vol. 37, no. 21, pp. 4963-4978, 1998.
- [64] D. Sahoo, P. C. Miles, J. Trost and A. Leipertz, “The Impact of Fuel Mass, Injection Pressure, Ambient Temperature, and Swirl Ratio on the Mixture Preparation of a Pilot Injection,” *SAE Int. J. Engines 2013-24-0061*, vol. 6, no. 3, pp. 1716-1730, 2013.

- [65] J. Sjöholm, *High Repetition Rate Laser Diagnostics for Combustion Applications*, 2012.
- [66] C. Schulz and V. Sick, "Tracer-LIF diagnostics: Quantitative measurement of fuel concentration, temperature and fuel/air ratio in practical combustion systems," *Progress in Energy and Combustion Science*, vol. 31, no. 1, pp. 75-121, 2005.
- [67] C. Espey, J. E. Dec, T. A. Litzinger and D. A. Santavicca, "Planar Laser Rayleigh Scattering for Quantitative Vapor-Fuel Imaging in a Diesel Jet," *Combustion and Flame*, vol. 109, pp. 65-86, 1997.
- [68] S. Shamun, B. Zincir, P. Shukla, P. Valladolid, S. Verhelst and M. Tunér, "Quantification and analysis of the charge cooling effect of methanol in a compression ignition engine utilizing PPC strategy," in *ASME 2018 Internal Combustion Engine Division Fall Technical Conference*, 2018.

Scientific publications

Author contribution to papers

Paper I: Transition from HCCI to PPC by means of Start of Injection

Published

Shen, M., Lonn, S., and Johansson, B.

SAE Technical Paper 2015-01-1790, 2015

This paper provides an investigation of the transition from HCCI to PPC exploring the combustion and emission characteristics in a heavy duty engine. A single and double injection strategy was employed, respectively. Furthermore, EGR and boost pressure were other parameters used to explore the effect of the transition from PPC to HCCI mode.

I carried out the measurements with Mengqin Shen. Mengqin Shen was the main responsible for writing the paper. I participated in the writing and in the discussion of the paper.

Paper II: Optical Study of the Fuel Spray Penetration and Initial Combustion Location under PPC Conditions

Published

Sara Lonn, Alexios Matamis, Martin Tuner, Mattias Richter, Oivind Andersson

SAE Technical Paper 2017-01-0752

This paper investigates the transition from HCCI to PPC with a single injection SOI-sweep in an optical engine. The paper includes investigation of the spray and combustion in an SOI-sweep. The injection of the liquid fuel spray was investigated with Mie Scattering and a high speed video. Furthermore, the combustion was captured using a high speed video. The spray and combustion images were analyzed

and based on the behavior of the spray and the combustion location the SOI-sweep could be divided into 3 different regimes. It was also found for the SOI-range were most of the fuel was injected into the squish region, the start of the combustion location shifted between the squish and the piston bowl. For the findings in this paper a hypothesis is formed that the combustion location is found in the richer equivalence ratios.

I planned the experiments with Öivind Andersson. I carried out the experiments together with Alexios and I postprocessed most of the data. Alexios contributed with writing the optical setup in the paper and in some of the postprocessing. I wrote the paper together with Öivind Andersson.

Paper III: Transition from HCCI to PPC: Investigation of Fuel Distribution by Planar Laser Induced Fluorescence

Published

Zhenkan Wang, Sara Lonn, Alexios Matamis, Öivind Andersson, Martin Tuner, Marcus Alden, Mattias Richter

SAE International Journal of Engines 2017-01-0748

This study investigates the charge stratification for a SOI-sweep in a heavy duty engine. The fuel stratification was studied by using fuel-tracer planar induced fluorescence (PLIF) imaging. The laser was formed to a sheet through a combination of lenses and a vertical laser sheet was sent through the engine. The PLIF-images were analyzed at different crank angle degrees and the results show how the fuel was distributed in the piston bowl and in the squish region close to the cylinder head. The results in the piston bowl showed that for the 2 early injection cases, SOI-100 and SOI-70 the fuel distribution was even, but with slightly more fuel in the periphery of the piston bowl. For SOI-54 and SOI-46 more fuel could be found in the center of the piston bowl. In the later SOI-cases more fuel was again found closer to the piston bowl.

I planned the experiments together with Zhenkan Wang, Mattias Richter and Öivind Andersson. The experiments were carried out together with Zhenkan Wang. Zhenkan Wang was responsible for writing the paper.

Paper IV: Effect of Injection Timing on the ignition and mode of Combustion in a HD PPC Engine running low load

Published

Ibron, C., Jangi, M., Lonn, S., Matamis, A., Öivind Andersson, Martin Tuner, Mattias Richter Xue-Song Bai

SAE Technical Paper 2019-01-0211, 2019

The study provides an investigation of the fuel stratification effect on ignition and combustion for 5 single injection cases with different injection timing using RANS simulation. The intake temperature has been adjusted to keep CA50 stable for the different cases. The simulation showed that the auto-ignition was sensitive to both equivalence ratio and temperature.

I contributed with optical imaging of the spray and combustion as well as operating conditions for the engine. Christian Ibron carried out the simulations and was the main responsible for writing the paper.

Paper V: PLIF of fuel distributions at the time of auto-ignition in an optical PPC engine

Manuscript to be submitted to a journal -

Sara Lönn, Zhenkan Wang, Christian Ibron, M. Jangi, Xue-Song Bai, Mattias Richter, Öivind Andersson

The study investigated the fuel distribution in the engine with a horizontal laser sheet at two different positions. Planar laser induced fluorescence was used to visualize acetone that was added as a tracer to the fuel. The fuel distribution data is further compared to CFD data of the fuel distribution at the corresponding planes. The aim of the study is to compare the fuel distribution in the squish and the piston bowl in order to find whether the hypothesis, from a previous study, that the combustion starts in the regions of higher equivalence ratios can be disregard.

I planed the experiments with Öivind Andersson. The experiments were carried out together with Zhenkan Wang. I post-processed the data. I have taken the main

responsibility for writing the manuscript. Öivind Andersson have assisted me in the writing process.

Paper V: Evaluation of injection strategies with methanol in an optical engine

Submitted to Fuel

Sara Lönn, Mateusz Pucilowski, Alexios Matamis, Ludovica Luise, Mattias Richter, Xue-Song Bai, Öivind Andersson

This study will focus on methanol combustion with single and multiple injections and how the combustion is affected by different injections strategies. Another objective with the study is to investigate if a pilot injection strategy can facilitate the combustion and reduce the effect of methanol's high heat of vaporization. RANS simulations have also been carried out for the single injection cases. The simulations have shown that methanol starts to react in lean conditions, around $\Phi=0.5$ or leaner. It was also found that it was possible to reduce the required inlet temperatures for single injection by using double injection strategies. Furthermore, in this study it was shown that a pilot injection can enable the combustion of a main injection, that would not combustion without a pilot.

I planned the experiments with Öivind Andersson. I carried out the experiments with Alexios Matamis and Ludovica Luise. Mateusz Pucilowski has carried out the CFD simulations. Mateusz has also contributed in the discussions of the paper and written the experiment methodology for the CFD simulations. I have post-processed the data. I have written the manuscript in cooperation with Öivind Andersson.