Characterization of the Combustion of Light Alcohols in CI Engines

Performance, Combustion Characteristics and Emissions

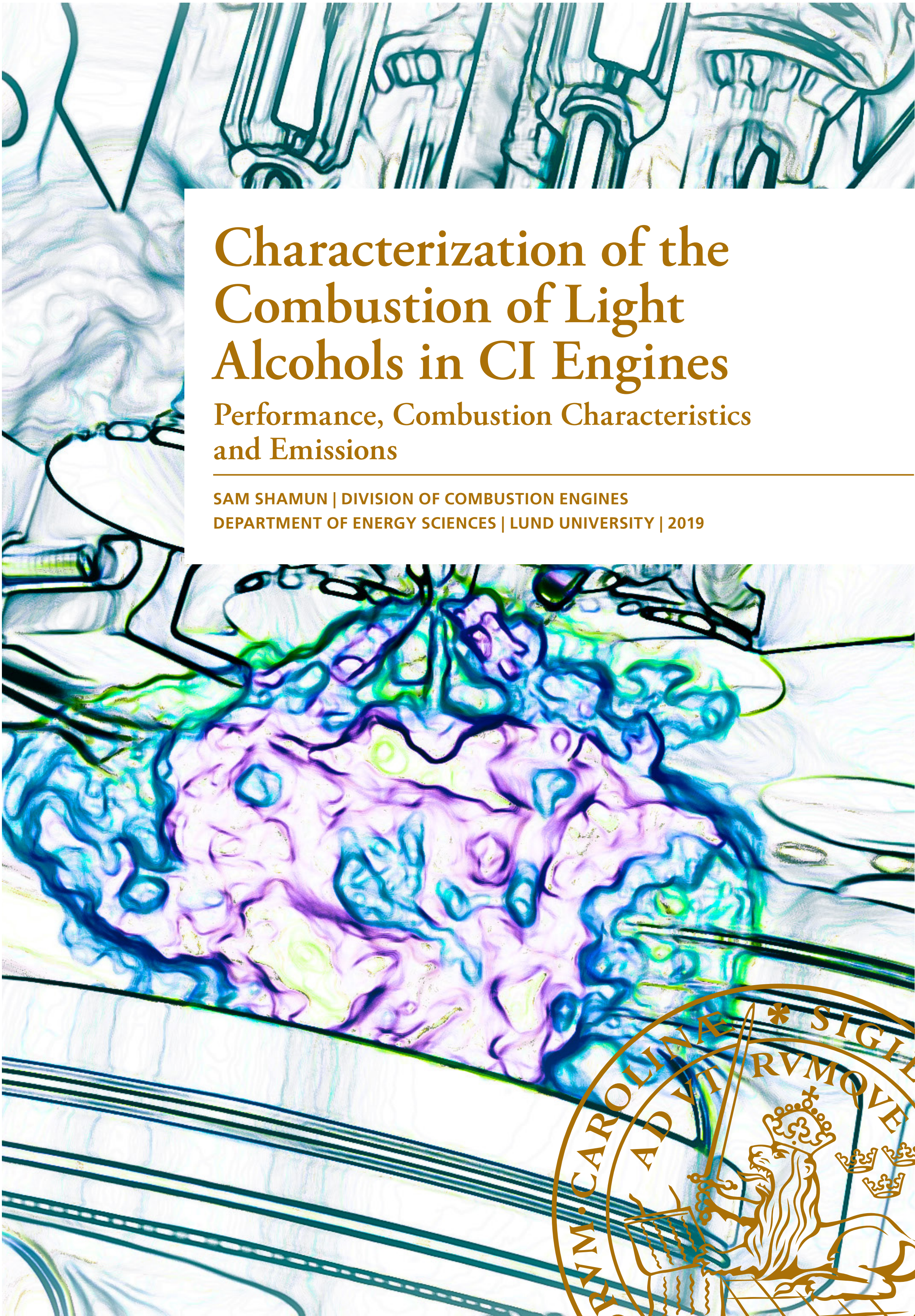
Shamun, Sam

2019

Characterization of the Combustion of Light Alcohols in CI Engines

Performance, Combustion Characteristics and Emissions

SAM SHAMUN | DIVISION OF COMBUSTION ENGINES
DEPARTMENT OF ENERGY SCIENCES | LUND UNIVERSITY | 2019



Characterization of the Combustion of Light Alcohols in CI Engines

Performance, Combustion Characteristics and Emissions

Characterization of the Combustion of Light Alcohols in CI Engines

Performance, Combustion Characteristics and Emissions

by Sam Shamun

LUND
UNIVERSITY

Thesis for the degree of Doctor of Philosophy
Thesis advisors: Prof. Martin Tunér and Assoc. Prof. Sebastian Verhelst
Faculty opponent: Assoc. Prof. Bart Somers

To be presented, with the permission of the Faculty of Engineering of Lund University, for public criticism in the M:B lecture hall at the M-Building, LTH on Friday, the 1st of March 2019 at 10:00.

Organization
LUND UNIVERSITY

Department of Energy Sciences
Box - 211 00 Lund
Sweden

Document name
DOCTORAL DISSERTATION

Date of issue
2019-03-01

Sponsoring organization
MOT-2030 Project 3572-1
KCFP

Author(s)
Sam Shamun

Title and subtitle
Characterization of the Combustion of Light Alcohols in CI Engines: Performance, Combustion Characteristics and Emissions

Abstract

Alternative fuels for combustion engines are becoming increasingly popular as society is pushing to phase out fossil energy to reduce CO_2 emissions. The compression ignition (CI) engine has a high overall efficiency which makes it a valuable option for the transport fleet, despite the well known NO_X and soot pollutants it emits. These two pollutants are emitted due to a combination of high local combustion temperature and low level of premixing prior to the combustion of diesel fuel. Based on previous work, it is well known that high research octane number (RON) fuels, such as gasoline, can be used in an CI engine to increase the premixing thus reducing the engine-out soot emissions, and to a certain extent, also NO_X. Apart from reducing the regulated emissions, the automotive industry is also focusing on developing CI engines that run at a higher efficiency and emit less CO_2, which can be achieved by using biomass based fuel, either neat or in blends. Methanol and ethanol are two good examples of such fuels. The idea of using light alcohols to run a CI engine did not arise recently; in Sweden, ethanol has been used in this engine type to run city buses since the mid 1980's. However, it is worth mentioning that the research of their use in CI engines has not been extensive. This work aims to investigate the performance, combustion characteristics and emissions of CI engines running on light alcohols, either neat or in blends with diesel, to study the advantages and drawbacks. The purpose is to better understand how the potential of these fuels can be further exploited while simultaneously finding ways to minimize the drawbacks of their use. The light alcohols, and in particular methanol, have a high heat of vaporization in combination with a low heating value. This contributes to a cooler combustion which also causes an extensive enleanment of the charge. The cooler combustion increases the efficiency by reducing the heat losses. The excessive enleanment, on the other hand, increases the total hydrocarbon (THC) and CO emissions. Moreover, the combustion instability increases. The findings of this work suggests that it is possible to counter these drawbacks, by increasing the intake temperature, T_{IN}. This could be achieved by using a turbocharger without extensive intercooling. The higher T_{IN} reduces the premixing period and improves the stability, resulting in increased oxidation of THC and CO. The drawback of this strategy is, however, an increased formation of NO$_X$. For similar intake conditions, methanol combustion resulted in a 50 % reduction of NO_X in comparison to iso-octane due to the its charge cooling effect. A double injection strategy can be used to reduce the required T_{IN}, however, this will come at a cost of lower thermal efficiency due to the longer combustion duration. Another viable option to reduce the required T_{IN} is by using a high compression ratio, r_c. The resulting increase in NO_X can be countered with EGR. However, if r_c is too high, operating flexibility is reduced due to restrictions in structural integrity; for example, high lambda alongside high EGR rates will be limited to lower loads. The light alcohols do not produce black carbon soot when combusted, thus significantly lower particulate matter (PM) emissions, which makes them a good alternative to the heavier diesel fuels. On the other hand, the particle number (PN) emission is generally higher than that of conventional gasoline or diesel. It is worth noting that the emitted PN only consist of particles with a diameter 30 nm as measured with a fast particle analyzer. Furthermore, an observation of the emitted PM under a transmission electron microscope, using energy dispersive X-ray, strongly suggested that the origin of the PM was the lubrication oil rather than the combustion products of the light alcohols. The light alcohols have shown some noteworthy results in terms of efficiency and emissions. In this work, a gross indicated efficiency of approx. 53 % was achieved by using a high r_c=27 piston and 50 % EGR at 6 bar $IMEP_G$.

Key words: Methanol, Ethanol, Oxygenated Fuel Blends, Compression Ignition Engine, Partially Premixed Combustion, Combustion Characteristics, Efficiency, Emissions, Particulate Matter

Classification system and/or index termes (if any):

Supplementary bibliographical information:

Language
English

ISSN and key title:
0282-1990

ISBN
978-91-7753-986-5 (print)
978-91-7753-987-2 (pdf)

Recipient's notes

Number of pages

Price

Security classification

Characterization of the Combustion of Light Alcohols in CI Engines

Performance, Combustion Characteristics and Emissions

by Sam Shamun

LUND UNIVERSITY

A doctoral thesis at a university in Sweden takes either the form of a single, cohesive research study (monograph) or a summary of research papers (compilation thesis), which the doctoral student has written alone or together with one or several other author(s).

In the latter case the thesis consists of two parts. An introductory text puts the research work into context and summarizes the main points of the papers. Then, the research publications themselves are reproduced, together with a description of the individual contributions of the authors. The research papers may either have been already published or are manuscripts at various stages (in press, submitted, or in draft).

Funding information: MOT-2030 Project with grant number 35721-1, and the KCFP Engine Research Center

Faculty of Engineering, Department of Energy Sciences

ISBN: 978-91-7753-986-5 (print)
ISBN: 978-91-7753-987-2 (pdf)
ISSN: <0282-1990>
ISRN: LUTMDN/TMHP-19/1142-SE

Printed in Sweden by Media-Tryck, Lund University, Lund 2019

There is no Knowledge that is not Power†

Acknowledgments

The decision making process, of whether becoming a Ph.D. is a good idea or not, might be quite time consuming and difficult. For me, it was easy. This process was very much simplified for me since Bengt Johansson, alongside Martin Tunér, already made that specific decision for me. I would like to show my deep gratitude to Bengt and Martin for giving me an opportunity which has - and will continue - to affect my life in only positive ways. What made Bengt and Martin stand out from the crowd was their ability to give me both good and bad criticism without the fear of being politically incorrect. Per Tunestål and Sebastian Verhelst, my co-supervisors, always had their doors open for me and supported me when I needed it, something which I am very grateful for. Gabriele, my supervisor in CNR-Istituto Motori, is completely unforgettable in every aspect. It was very productive working with you, using the MOMC method! Having these supervisors, possessing the intellect of Einstein and the humbleness of Gandhi, is truly a privilege.

Ashish, Maja and Yann, were the first people that became my friends at the division. I would like to express my gratitude to them for listening to my scientific reasoning, my (bad) jokes, my troubles as well as my anxiety. Changle, Pravesh and Vikram, the three nicest guys on the entire planet; I look up to your ambition, intellect and kindness. You all teach me much more than science.

I can not have an acknowledgement section without mentioning Mateusz, my old friend and colleague. Thanks a lot for the arguments we had regarding the definition of PPC. Although we left the arguments disagreeing with each other, you always managed to convince me after a period of reflection.

Giacomo and Luigi, thanks for giving me a great introduction to one of the best cities on the planet!

My knowledge as a PhD in Combustion Engines would not be complete without the time I spent in the lab and all the help I got from the technicians! For sure, they taught me everything I know about engine hardware. Mats, Thommy, Thomas, Patrik, Anders and Martin; thanks a lot for taking your time to answer all my (silly) questions and fixing test cell 4 after I completely devastated it beyond recognition. What is an engine without a control system? I would like to thank Pablo for every time he healed my LabView related headache!

Moreover, I would like to show gratitude to the rest at the Division of Combustion Engines, Division of Ergonomics and Aerosol Technology, Division of Combustion Physics, Division of Solid State Physics.

I would like to thank my parents and sister, Adnan, Ban and Sayle, for convincing me - and partially forcing me - to study hard and always do my best. I guess you all saw a potential in me which I did not recognize myself.

Abstract

Alternative fuels for combustion engines are becoming increasingly popular as society is pushing to phase out fossil energy to reduce CO_2 emissions. The compression ignition (CI) engine has a high overall efficiency which makes it a valuable option for the transport fleet, despite the well known NO_X and soot pollutants it emits. These two pollutants are emitted due to a combination of high local combustion temperature and low level of premixing prior to the combustion of diesel fuel. Based on previous work, it is well known that high research octane number (RON) fuels, such as gasoline, can be used in an CI engine to increase the premixing thus reducing the engine-out soot emissions, and to a certain extent, also NO_X. Apart from reducing the regulated emissions, the automotive industry is also focusing on developing CI engines that run at a higher efficiency and emit less CO_2, which can be achieved by using biomass based fuel, either neat or in blends. Methanol and ethanol are two good examples of such fuels. The idea of using light alcohols to run a CI engine did not arise recently; in Sweden, ethanol has been used in this engine type to run city buses since the mid 1980's. However, it is worth mentioning that the research of their use in CI engines has not been extensive. This work aims to investigate the performance, combustion characteristics and emissions of CI engines running on light alcohols, either neat or in blends with diesel, to study the advantages and drawbacks. The purpose is to better understand how the potential of these fuels can be further exploited while simultaneously finding ways to minimize the drawbacks of their use. The light alcohols, and in particular methanol, have a high heat of vaporization in combination with a low heating value. This contributes to a cooler combustion which also causes an extensive enleanment of the charge. The cooler combustion increases the efficiency by reducing the heat losses. The excessive enleanment, on the other hand, increases the total hydrocarbon (THC) and CO emissions. Moreover, the combustion instability increases. The findings of this work suggests that it is possible to counter these drawbacks, by increasing the intake temperature, T_{IN}. This could be achieved by using a turbocharger without extensive intercooling. The higher T_{IN} reduces the premixing period and improves the stability, resulting in increased oxidation of THC and CO. The drawback of this strategy is, however, an increased formation of NO_X. For similar intake conditions, methanol combustion resulted in a 50 % reduction of NO_X in comparison to iso-octane due to the its charge cooling effect. A double injection strategy can be used to reduce the required T_{IN}, however, this will come at a cost of lower thermal efficiency due to the longer combustion duration. Another viable option to reduce the required T_{IN} is by using a high compression ratio, r_c. The resulting increase in NO_X can be countered with EGR. However, if r_c is too high, operating flexibility is

reduced due to restrictions in structural integrity; for example, high lambda alongside high EGR rates will be limited to lower loads. The light alcohols do not produce black carbon soot when combusted, thus significantly lower particulate matter (PM) emissions, which makes them a good alternative to the heavier diesel fuels. On the other hand, the particle number (PN) emission is generally higher than that of conventional gasoline or diesel. It is worth noting that the emitted PN only consist of particles with a diameter 30 nm as measured with a fast particle analyzer. Furthermore, an observation of the emitted PM under a transmission electron microscope, using energy dispersive X-ray, strongly suggested that the origin of the PM was the lubrication oil rather than the combustion products of the light alcohols. The light alcohols have shown some noteworthy results in terms of efficiency and emissions. In this work, a gross indicated efficiency of $\sim$53 % was achieved by using a high r_c=27 piston and 50 % EGR at 6 bar $IMEP_G$.

Popular science summary

The low fuel consumption and high torque output of the diesel engine makes it a very attractive power source, both in the private sector as well as in the transport industry. Except for the mentioned advantages, diesel engines emit very small amounts of carbon monoxide and hydrocarbons.

The disadvantage of this engine type is the emissions of high concentrations of oxides of nitrogen (NO_X) and soot, which are both harmful for health and the environment. Because of this, the industry as well as various research institutes have focused on minimizing the emissions of these pollutants while simultaneously reducing the fuel consumption by investigating different advanced combustion concepts for this engine type.

One of these combustion concepts is partially premixed combustion, PPC. This concept uses an injection which is timed slightly earlier in the cycle in comparison to conventional diesel engines, while introducing cooled exhaust gases together with the fresh air. This causes a higher level of premixing which increases the local oxygen availability and maintains a lower combustion temperature. This strategy reduces the concentration of soot and NO_X in the exhaust gases.

To ease the process of premixing of the air-fuel mixture in this combustion concept, the use of a gasoline, with a research octane number rating of 70, is preferred over diesel fuel. Although the soot emissions of gasoline are lower than those of diesel, the disadvantage of gasoline, similarly to diesel, is that there is a trade-off between soot and NO_X. In this work, methanol and ethanol are used as fuels - both neat and in blends with other fuel types - to circumvent this trade-off, due to the soot formation tendency of these alcohols is extremely small. Moreover, these alcohols can be produced from biomass which in turn reduce the carbon dioxide emissions. Another advantage with methanol and ethanol is their cooling effect which further reduces the combustion and thus also reduces the heat losses to the combustion chamber walls as well as the NO_X emissions.

In this work, the research have contributed to achieving an operating condition in which all the regulated emissions have been maintained below current legal restrictions without the use of any exhaust after treatment systems. Meanwhile, the experiments in this work have contributed to an increase in efficiency and reduction in emissions in both light- and heavy-duty engines by using methanol and ethanol

Populärvetenskaplig sammanfattning

Låg bränsleförbrukning och högt vridmoment gör att dieselmotorn är en väldigt attraktiv kraftkälla för transportfordon, både för privatpersoner och transportindustin. Förutom de nämnda fördelarna, släpper dieselmotorer ut väldigt låga halter av både kolmonoxid och kolväten.

Nackdelen för denna motortyp är att den istället släpper ut avgaser med höga halter av kväveoxider och sot, som är skadliga för både hälsa och miljö. På grund av detta har industrin såväl som olika forskningsinstitut, genom att undersöka olika avancerade förbränningskoncept, riktat in sig på att minska utsläppet av dessa föroreningar samtidigt som man försöker minska bränsleförbrukningen för denna motortyp.

Ett av dessa förbränningskoncept är partiellt förblandad förbränning (eng. partially premixed combustion, PPC). Detta koncept innebär att man injicerar bränslet lite tidigare i cykeln än konventionella dieselmotorer samtidigt som man introducerar kylda avgaser tillsammans med friskluften. Detta medför en högre grad av förblandning som ökar den lokala tillgängligheten av syre samt ser till att förbränningstemperaturen bibehålls låg. Denna strategi leder till lägre halter av både sot och kväveoxider i avgaserna.

För att underlätta förblandningen av bränsle och luft i detta förbränningskoncept, använder man hellre bensin med ett oktantal kring 70 istället för diesel. Problemet med bensin, såväl som diesel, är att man måste avväga mellan sot- och kväveoxidemissioner trots att sotemissionerna när man kör bensin är lägre än diesel. För att kringgå denna avvägning används i detta arbete istället metanol eller etanol, både i dess rena form såväl som blandningar med andra bränslen, på grund av att förbränningen av dessa alkoholer inte producerar sot. Dessutom kan dessa alkoholer produceras från biomassa vilket i sin tur minskar utsläppet av koldioxid. En annan fördel med metanol och etanol är deras kylningseffekt som ytterligare kyler ner förbränningen och därmed minskar värmeförlusterna till förbränningsrummets väggar och kväveoxid emissionerna.

Forskningen kring dessa alternativa bränslen har bidragit till att man har, i detta arbete, lyckats köra motorn i ett läge där alla de reglerade emissionerna har bibehållts under rådande begränsningar utan att använda sig av några som helst avgasefterbehandligssystem. Samtidigt har experimenten i detta arbete även bidragit med att öka verkningsgraden och minska avgasutsläppen i både personbils- och lastbilsmotorer genom att använda metanol och etanol.

List of publications

I **Exhaust PM Emissions Analysis of Alcohol Fueled Heavy-Duty Engine Utilizing PPC**
Shamun, S., Shen, M., Johansson, B., Tunér, M., Pagels, J., Gudmundsson, A. and Tunestål, P.
SAE Int. J. Engines 9(4):2142-2152, 2016.

II **Experimental investigation of methanol compression ignition in a high compression ratio HD engine using a Box-Behnken design**
Shamun, S., Hasimoglu, C., Murcak, A., Andersson, Ö., Tunér, M. and Tunestål, P.
Fuel 209:624-633, 2017.

III **Detailed Characterization of Particulate Matter in Alcohol Exhaust Emissions**
Shamun, S., Novakovic, M., Malmborg, V.B., Preger, C., Shen, M., Messing, M.E., Pagels, J., Tunér, M. and Tunestål, P.
The Ninth International Conference on Modeling and Diagnostics for Advanced Engine Systems 2017.9, B304, 2017.

IV **Performance and emissions of diesel-gasoline-ethanol blends in a light duty compression ignition engine**
Belgiorno, G., Di Blasio, G., **Shamun, S.**, Beatrice, C., Tunestål, P. and Tunér, M.
Fuel 217:78-90, 2018.

V **Performance and emissions of diesel-biodiesel-ethanol blends in a light duty compression ignition engine**
Shamun, S., Belgiorno, G., Di Blasio, G., Beatrice, C., Tunér, M. and Tunestål, P.
Applied Thermal Engineering 145:444-452, 2018.

VI **Quantification and Analysis of the Charge Cooling Effect of Methanol in a Compression Ignition Engine utilizing PPC**
Shamun, S., Zincir, B., Shukla, P., Valladolid, P.G., Verhelst, S. and Tunér, M.
The American Society of Mechanical Engineers - Internal Combustion Engine Fall Technical Conference, ICEF2018-9657, 2018.

Other publications

Sensitivity analysis of partially premixed combustion (PPC) for control purposes *SAE Technical Paper 2015-01-0884, 2015.*

Potential Levels of Soot, NOx, HC and CO for Methanol Combustion *SAE Technical Paper 2016-01-0887, 2016*.

Modelling of Methanol Combustion in a Direct Injection Compression Ignition Engine using an Accelerated Stochastic Fields Method Shamun, S. *Energy Procedia 105:1326-1331, 2017*.

The Effect of Injection Pressure on the NOx Emission Rates in a Heavy-Duty DICI Engine Running on Methanol *SAE Technical Paper 2017-01-2194, 2017*.

Effect of Start of Injection on the Combustion Characteristics in a Heavy-Duty DICI Engine Running on Methanol *SAE Technical Paper 2017-01-0560, 2017*.

Investigation of Particle Number Emission Characteristics in a Heavy-Duty Compression Ignition Engine Fueled with Hydrotreated Vegetable Oil (HVO) *SAE Technical Paper 2018-01-0909, 2018*.

Transition from HCCI to PPC: the Sensitivity of Combustion Phasing to the Intake Temperature and the Injection Timing with and without EGR *SAE Technical Paper 2016-01-0767, 2018*.

Heat Loss Analysis for Various Piston Geometries in a Heavy-Duty Methanol PPC Engine Shamun, S. *SAE Technical Paper 2018-01-1726, 2018*.

Alternative Fuels for Particulate Control in CI Engines, Shamun, S. *Engine Exhaust Particulates pp. 181-197, Agarwal, A.K., Dhar, A., Sharma, N. and Shukla, P.C. (Ed.), Springer, 2018*.

Gasoline Exhaust Particulate Matter Emissions Measurement in a Wide Range of EGR in a Heavy-Duty Diesel Engine *SAE Technical Paper 2019-01-0761, 2019*.

List of Abbreviations

η_B	Brake efficiency
η_C	Combustion efficiency
η_{GIE}	Gross indicated efficiency
η_{NIE}	Net indicated efficiency
λ	Lambda, $\frac{(A/F)}{(A/F)_S}$
Φ	Lambda, $\frac{(A/F)_S}{(A/F)}$
BSFC	Brake specific fuel consumption
CAD	Crank angle degrees
CDC	Conventional diesel combustion
CI	Compression ignition
CMD	Count mean diameter
CN	Cetane number
EATS	Exhaust after treatment system
FAME	Fatty acid methyl
HD	Heavy duty
HVO	Hydrotreated vegetable oil
ICE	Internal combustion engine
IMEP$_G$	Gross indicated mean effective pressure
IMEP$_N$	Net indicated mean effective pressure
LD	Light duty
MP	Mixing period
NO$_X$	Oxides of nitrogen: NO, NO_2
OEM	Original equipment manufacturer
ON	Octane number
PAH	Polycyclic aromatic hydrocarbon
PM	Particulate matter
PPC	Partially premixed combustion
PPR$_{MAX}$	Maximum pressure rise rate
RON	Research octane number
SI	Spark ignition
SME	Soybean methyl ester
SOC	Start of combustion
SOI	Start of injection
TDC	Top dead center
TEM	Transmission electron microscope
THC	Total hydrocarbon
XEDS	Energy dispersive X-ray spectroscopy

Contents

Chapter 1

Introduction

1.1 Background

The ever increasing global population alongside the living standards puts high demands on the energy production. A majority of the demanded energy is produced by combustion of fossil fuels and a considerable portion of that energy, about 27 %, is consumed by the transportation sector [1, 2]. The transportation sector constitutes about 23 % of the global CO_2 emissions [3].

The vast majority of the transport sector uses reciprocating internal combustion engines (ICE's) as powertrains. The development and utilization of the ICE has a history preceeding Otto's and Diesel's time [4]. Nevertheless, modern day engines are more or less still based on Otto's spark ignition (SI) engine as well as Diesel's compression ignition (CI) engine. Engines of today, however, work at considerably higher efficiencies while simultaneously emitting less exhaust emissions than their predecessors [5]. While the fossil fueled piston engine has the drawback of emitting different greenhouse gases, its simplicity and relatively low price has contributed vastly to the modern day society in several aspects [6].

Since the utilization of fossil fuels increases the concentration of atmospheric CO_2, the use of biomass based fuels has recently become increasingly popular [7]. The production of ethanol, methanol as well as different types of biodiesel is increasing substantially in Brazil, China as well as in the U.S. [8]. In Europe, ethanol is blended in regular pump gasoline while fatty acid methyl esters (FAME) as well as hydrotreated vegetable oil (HVO), are blended into diesel fuel to increase the renewable fuel fraction [9, 10]. Biofuels, which are produced from biomass, do not contribute with a net addition of atmospheric CO_2 during the combustion process itself because the emitted amount of CO_2 is equal to the absorbed amount during the plants lifetime. However, depending on the production and distribution of these fuels, there might be a net CO_2 addition to the atmosphere [11].

Despite the simultaneous increase of popularity of fully electric vehicles (FEV's), the vast majority of the power propulsion systems used in road vehicles are combustion engines, since FEV's are experiencing obstacles in the form of driving range, high cost, high need of rare materials as well as new infrastructure [12, 13]. Moreover, FEV's can still emit a high amount of CO_2, both during the production of the vehicle and its components as well as during its service life [14, 15]. For this reason, the internal combustion engine is still a very viable option, however, there are challenges.

The survival of the combustion engine depends heavily on its development in terms of fuel efficiency, but maybe more importantly, the reduction of harmful exhaust gases as well as particulate emissions (PM). By utilizing methanol and ethanol, which both can be produced sustainably from biomass, a significant reduction in gaseous emissions, and in particular CO_2, can be achieved [16]. Also, methanol can be produced from captured carbon making it a so called e-fuel, since electric energy can be used to produce and store methanol for later use [17, 18]. These fuels, when powering CI engines, reduce the PM emissions to close to zero levels. One of the most abundant regulated emissions from conventional diesel combustion (CDC), NO_X, can be reduced depending on the operating conditions of the engine.

The engines have to be redesigned and recalibrated to be able to run on these oxygenated fuels in a smooth and reliable manner, which puts pressure on the original equipment manufacturers (OEM). The pressure on the OEM's does not only come from the technical challenges, but also from the uncertainty arising from releasing a vehicle running on a fuel not commercially available on an international level. To deal with this uncertainty, the infrastructure must be adapted to handle the new fuels and the availability of different environmentally friendly fuels must increase.

In 2013, an agenda was established in Sweden with the goal of reducing the dependency of fossil fuels in the national transport fleet. The target is to eventually reach zero net CO_2 emissions to the atmosphere [19, 20, 21]. A part of this project, MOT-2030, aimed to investigate the possibility of running CI engines on methanol, neat and/or blended [22]. In Sweden, ethanol have a history of powering heavy duty (HD) CI engines in city buses since the mid 1980's [23]. Moreover, the forest resources in Sweden are more than enough to cover the current and future fueling needs of the transport fleet, while simultaneously being able to expand. The challenges, seen from this aspect, are to be able to exploit the forest wealth as efficiently as possible in a sustainable manner [24].

Most of the studies performed on CI engines using methanol and ethanol, are in blends in low concentration in diesel fuel or other high cetane number

(CN) fuels. In other studies, methanol and ethanol are dual fueled. One of the first significant papers was published in 1990 by Richards [25], where neat methanol was used in two heavy duty (HD) engines. The paper showed that, with the aid of a glow plug, the modified methanol vehicles performed as well or better than diesel fuel in terms of component life. Moreover, laboratory tests showed reduced PM as well as NO_X emissions. Goetz et al. [26] suggested that the NO_X emissions were reduced due to the fuel impingement on the combustion chamber wall, while the same phenomenon also caused unacceptably high CO and THC emissions. It is worth noting that studies conducted utilizing neat methanol in CI engines during the 21stcentury are not common.

Neat ethanol utilization in CI engines, as earlier mentioned, has been more common both in research studies as well as for commercial applications. More recent studies have also been conducted by independent research institutes. The higher efficiency of neat ethanol engine operation, similar to methanol, is due to a decreased heat transfer loss which will be discussed further in Section 4.2.2 [27, 28]. The NO_X emissions do not show the same consistency as the soot emissions and depend more on the operating conditions. The underlying factors, higher intake pressures and temperatures which are required for neat ethanol to combust, could be the causes for higher NO_X emissions [28, 29]. Other factors affecting the NO_X emissions are short combustion duration and high maximum pressure rise rate, PRR_{MAX} [30]. Two independent studies conducted by Li et al. and Rakopoulos et al., showed that a higher concentration of ethanol in diesel fuel reduced the NO_X emissions [31, 32]. This is explained by the later combustion phasing due to the constant start of injection, SOI, for all the tested fuels. Simultaneously, the charge cooling effect of ethanol plays a significant part in lowering the combustion temperature [33]. The charge cooling effect of alcohols has shown to be both advantageous and detrimental. Since the ignition delay is prolonged and combustion temperature is lower in comparison to diesel fuel, the probability of wall impingement and combustion instability is increased. These factors generally increase the CO and THC emissions from alcohol combustion [28].

There are, however, some concerns regarding the utilization of neat alcohols in ICE's. Among the general public, the most repeated arguments against their use are high brake specific fuel consumption (BSFC), incompatibility with the engine due to corrosivity and toxicity, whichof the latter applies for methanol only. While there is a definitive truth behind the higher fuel consumption argument in terms of volume, it has to be mentioned that this is offset by the lower price, per energy unit, as well as the higher efficiency of engine operation [34, 17]. The corrosivity issue has been well studied and tackled with the long experience of running engines on methanol and ethanol. The toxicity of methanol applies only when it is ingested. In California, methanol vehicle tests, exceeding a total of 300 million km, were conducted without a single case of poisoning [35]. Moreover, it has to be highlighted that methanol is easily biodegradable and does not cause severe adverse effects on the environment in case of a major leak or contamination [17].

The aim of this thesis is to examine methanol and ethanol as alternative fuels in a CI engine. Utilizing methanol or ethanol, in CI engines, presents some challenges for the manufacturers since the physico-chemical properties of the light alcohols are very different from those of diesel fuel. The first consideration that has to be taken into account is the high research octane number (RON) for methanol and ethanol, since this property can introduce great difficulties in the initiation of the ignition process in CI ICE's., which will be dealt with in this thesis alongside other combustion related issues.

1.2 Motivation and Scope

The motivation of choosing to investigate the effects of methanol and ethanol in the CI engine combustion process, is the potential of reducing the non-regulated green house gas, CO_2, emission to the atmosphere. To deal with the ignitability and combustion stability (COV_{IMEP_N}) issues, mentioned in Section 1.1, this thesis will provide different possible paths to circumvent them, while also presenting their advantages as well as disadvantages. Moreover, the regulated emissions will be analyzed and presented as a function of operating conditions as well as issues arising from the attempt of reducing these emissions. This thesis will discuss the engine parameters that affect the efficiency, performance, emissions and combustion characteristics through experiments, data post processing and analysis, when writing on methanol and ethanol, to better understand the general trends of using such fuels in a CI combustion process. The tests performed with methanol, used chemical grade 99.85 vol.% methanol, while the tests with ethanol were performed with both neat alcohol 99.7 vol.% as well as in blends with diesel fuel. Since ethanol and diesel fuel separate above an ethanol concentration above 5 vol.% ethanol, gasoline and FAME were used as emulsifiers. These parameters will be compared to those of gasoline and diesel fuel from different perspectives. The experimental work conducted in this dissertation was formulated to answer the following questions:

- How does the engine perform when running on oxygenated fuels in terms of stability and ignitability?
- How does the charge cooling effect influence the emissions and performance?

- How and why do the regulated engine-out emissions differ between the use of neat alcohol or alcohol-diesel blends in a CI engine?
- Methanol and ethanol are known to emit virtually smokeless exhaust, but how about PN?
- How do the light alcohol fuels differ from regular high CN fuels in terms of the different efficiency parameters?

1.3 Thesis Contribution

The main contributions of this thesis work is the complimentary knowledge of methanol and ethanol utilization in both light duty (LD) and heavy duty (HD) CI engines both neat and as in emulsified blends. The thesis discusses the advantages and drawbacks of the use of the above mentioned light alcohols on performance, combustion characteristics as well as the emissions. Also, some brief suggestions are given of how the engine hardware could be altered to ease the path to methanol and ethanol utilization in CI engines. The main findings of this work are, summarized:

- Methanol and ethanol can both be used to power a CI engine utilizing a generally low compression ratio, r_c, given that the intake temperature is high enough to make the fuel auto-ignite at the firing top dead center (TDC). Using a lower r_c enables the engine to be more flexible, in terms of combustion phasing, exhaust gas recirculation (EGR) utilization and intake pressure, at a wider range of loads. A higher r_c will, in contrast to low r_c, ease the ignitability of the fuel. However, this advantage comes with the drawback of reducing the operation flexibility.
- High octane fuels have a rather unstable combustion when utilized in a CI engine and methanol and ethanol are no exceptions to this behavior. This instability has been circumvented in this work by using a high intake temperature. Moreover, it was also shown that a split injection strategy enables the use of a lower intake temperature than that of a single injection. The controllability also increases with the split injection approach since the combustion mode changes from premixed to diffusion. However, this advantage comes with the cost of lower efficiency.
- The charge cooling effect of the light alcohols, and in particular methanol, can be used to a limited extent to reduce the NO_X emissions. The greater advantage of the charge cooling effect of these fuels is evident during the

combustion event. Since the in-cylinder temperature is reduced significantly during the evaporation of the fuel, the combustion itself occurs at a lower temperature as well. While this could, in theory, limit the use of a turbocharger, it also reduces the heat transfer. With the combination of methanol, high r_c and high EGR utilization, about 53 % gross indicated efficiency, η_{GIE}, was achieved.

- The particulate emissions of methanol and ethanol, are significantly different from those of diesel fuel and gasoline. While the soot mass concentration for the light alcohols is negligible, the particle number concentration is significantly higher than that of diesel fuel and gasoline. The particulates count mean diameter of methanol and ethanol on the other hand, never exceeds 30 nm. It is hypothesized that the higher particle number concentration, which originates from the lubrication oil, is due to the lack of accumulation mode particles on which the nucleation mode particles can agglomerate to.

- It is commonly known that CO and THC engine-out emissions from CI engines are generally negligible. However, with heavy use of EGR to suppress NO_X emissions, the CO and THC rise to significant levels. It was shown that with specific operating conditions, utilizing PPC combustion, all three regulated gaseous emissions were kept below the EU VI limitation, without the aid of any emission aftertreatment systems.

Bibliography

[1] International Energy Agency, CO_2 emissions from fuel combustion - Highlights, 2017.

[2] U.S. Energy Information Adminstration, Annual Energy Outlook 2018 - with projection to 2050, 2018.

[3] P. Nejat, F. Jomehzadeh, M. M. Taheri, M. Gohari, and M. Z. A. Majid. A global review of energy consumption, CO_2 emissions and policy in the residential sector (with an overview of the top ten CO_2 emitting countries). *Renewable and Sustainable Energy Reviews*, 43:843–862, 2015, DOI:https://doi.org/10.1016/j.rser.2014.11.066.

[4] L. Cummins. *Internal Fire: The Internal Combustion Engine 1673-1900*. SAE International, 1989, ISBN:978-0898837650.

[5] *National Research Council, Transitions to Alternative Vehicles and Fuels*. The National Academies Press, 2013, ISBN:978-0-309-26852-3.

[6] J. P. Womack, D. T. Jones, and D. Roos. *The Machine That Changed the World*. Free Press, 2007, ISBN:978-0743299794.

[7] K. Araújo, D. Mahajan, R. Kerr, and M. Silva. Global Biofuels at the Crossroads: An Overview of Technical, Policy, and Investment Complexities in the Sustainability of Biofuel Development. *Agriculture*, 7(4):32, 2017.

[8] *Renewable Fuels Association, 2017 Ethanol Industry Outlook, 2017*. URL:http://www.ethanolrfa.org/wp-content/uploads/2017/02/Ethanol-Industry-Outlook-2017.pdf.

[9] *Global Agricultural Information Network, EU Biofuels Annual 2017, 2017*. URL:https://gain.fas.usda.gov/Recent

[10] EU: Fuels: Automotive Diesel Fuel. *Accessed 2018-07-26*, URL:https://www.dieselnet.com/standards/eu/fuel_automotive.php.

[11] R. Edwards, H. Hass, J. F. Larivé, L. Lonza, H. Maas, and D. Rickeard. Well-To-Wheel Analysis Of Future Automotive Fuels And Powertrains In The European Context. *Joint Research Center*, 2014, DOI:10.2790/95533.

[12] F. Un-Noor, S. Padmanaban, L. Mihet-Popa, M. N. Mollah, and E. Hossain. A Comprehensive Study of Key Electric Vehicle (EV) Components, Technologies, Challenges, Impacts, and Future Direction of Development. *Energies*, 10(8):1217, 2017, DOI:10.3390/en10081217.

[13] N. Nitta, F. Wu, J. T. Lee, and G. Yushin. Li-ion battery materials: present and future. *Materials Today*, 18(5):252–264, 2015, DOI:https://doi.org/10.1016/j.mattod.2014.10.040.

[14] Y. Bicer and I. Dincer. Comparative life cycle assessment of hydrogen, methanol and electric vehicles from well to wheel. *International Journal of Hydrogen Energy*, 42(6):3767–3777, 2017, DOI:https://doi.org/10.1016/j.ijhydene.2016.07.252.
[15] T. Peng, X. Ou, and X. Yan. Development and application of an electric vehicles life-cycle energy consumption and greenhouse gas emissions analysis model. *Chemical Engineering Research and Design*, 131:699–708, 2018, DOI:https://doi.org/10.1016/j.cherd.2017.12.018. Energy Systems Engineering.
[16] A. Molino, V. Larocca, S. Chianese, and D. Musmarra. Biofuels Production by Biomass Gasification: A Review. *Energies*, 11(4), 2018, DOI:10.3390/en11040811.
[17] S. Verhelst, J. W. G. Turner, L. Sileghem, and J. Vancoillie. Methanol as a fuel for internal combustion engines. *Progress in Energy and Combustion Science*, 70:43–88, 2019, DOI:https://doi.org/10.1016/j.pecs.2018.10.001.
[18] D.-K. Nguyen, L. Sileghem, and S. Verhelst. Exploring the potential of reformed-exhaust gas recirculation (R-EGR) for increased efficiency of methanol fueled SI engines. *Fuel*, 236:778–791, 2019, DOI:https://doi.org/10.1016/j.fuel.2018.09.073.
[19] *Statens Offentliga Utredningar 2013:84, fossilfrihet på väg, del 1, 2013*. URL:https://www.regeringen.se/49bbab/contentassets/7bb237f0adf546daa36aaf044922f473/fossilfrihet-pa-vag-sou-201384-del-12.
[20] *Statens Offentliga Utredningar 2013:84, fossilfrihet på väg, del 2, 2013*. URL:https://www.regeringen.se/49bbab/contentassets/7bb237f0adf546daa36aaf044922f473/fossilfrihet-pa-vag-sou-201384-del-22.
[21] *Regeringskansliet, Handlingsplan Agenda 2030: 2018-2020, del 2, 2013*. URL:https://www.regeringen.se/49bbab/contentassets/7bb237f0adf546daa36aaf044922f473/fossilfrihet-pa-vag-sou-201384-del-22.
[22] MOT-2030: Highly Efficient Methanol Engine Systems for Fossil Free Transportation 2030. *Accessed 2018-11-28*, URL:http://www.lth.se/mot2030/.
[23] P. Stålhammar. Scania - ED95 Developement, Alcohol CI Combustion, 2015. *Accessed 2018-07-26*, URL:http://www.lth.se/fileadmin/mot2030/filer/11._Stalhammar_-_Scania_ED95_development.pdf.
[24] T. B. Johansson. A fossil free vehicle fleet in Sweden 2030: reflections and priorities from a government investigation, 2015. *Accessed 2018-07-26*, URL:http://www.lth.se/fileadmin/mot2030/filer/2._Johansson_-_Fossil_Free_Fleet_in_Sweden_2030.pdf.

[25] B. G. Richards. Methanol-Fueled Caterpillar 3406 Engine Experience in On-Highway Trucks. *Journal of Fuels and Lubricants*, 99(4):1033–1045, 1990, DOI:https://doi.org/10.4271/902160.
[26] W. A. Goetz, C. G. Barringer, and M. A. Bozzelli. Utilizing Neat Methanol and Glow Plug Ignition in DI Diesels: Laboratory Testing of a Single and Multi-Cylinder Engine. *Journal of Fuels and Lubricants*, 103(4):711–726, 1994, DOI:https://doi.org/10.4271/941044.
[27] G. Nagarajan, A. N. Rao, and S. Renganarayanan. Emission and performance characteristics of neat ethanol fuelled Dl diesel engine. *International Journal of Ambient Energy*, 23(3):149–158, 2002, DOI:10.1080/01430750.2002.9674883.
[28] M. Shen, M. Tunér, B. Johansson, and W. Cannella. Effects of EGR and Intake Pressure on PPC of Conventional Diesel, Gasoline and Ethanol in a Heavy Duty Diesel Engine. *SAE Technical Paper 2013-01-2702*, 2013, DOI:https://doi.org/10.4271/2013-01-2702.
[29] M. Shen, M. Tunér, and B. Johansson. Close to Stoichiometric Partially Premixed Combustion - The Benefit of Ethanol in Comparison to Conventional Fuels. *SAE Technical Paper 2013-01-0277*, 2013, DOI:10.4271/2013-01-0277.
[30] X. Shi, Y. Yu, H. He, S. Shuai, J. Wang, and R. Li. Emission characteristics using methyl soyate-ethanol-diesel fuel blends on a diesel engine. *Fuel*, 84(12):1543–1549, 2005, DOI:https://doi.org/10.1016/j.fuel.2005.03.001.
[31] D. C. Rakopoulos, C. D. Rakopoulos, E. C. Kakaras, and E. G. Giakoumis. Effects of ethanol-diesel fuel blends on the performance and exhaust emissions of heavy duty DI diesel engine. *Energy Conversion and Management*, 49(11):3155–3162, 2008, DOI:https://doi.org/10.1016/j.enconman.2008.05.023.
[32] D. Li, H. Zhen, L. Xingcai, Z. Wu-gao, and Y. Jian-guang. Physico-chemical properties of ethanol-diesel blend fuel and its effect on performance and emissions of diesel engines. *Renewable Energy*, 30(6):967–976, 2005, DOI:https://doi.org/10.1016/j.renene.2004.07.010.
[33] N. Yilmaz. Comparative analysis of biodiesel-ethanol-diesel and biodiesel-methanol-diesel blends in a diesel engine. *Energy*, 40(1):210–213, 2012, DOI:https://doi.org/10.1016/j.energy.2012.01.079.
[34] Methanol Facts: China: The Leader In Methanol Transportation. *Accessed 2018-10-17*, URL:http://www.methanol.org/wp-content/uploads/2016/06/China-Methanol-Fact-Sheet-1.pdf.
[35] L. Bromberg and W. K. Cheng. Methanol as an alternative transportation fuel in the US: Options for sustainable and/or energy-secure transportation. *Sloan Automotive Laboratory, Massachusetts Institute of Technology*, 2010.

Chapter 2

Engines and Fuels

2.1 Combustion Strategies

The basic operation concept of the CI engine boils down to the auto-ignition of directly injected fuel into the combustion chamber. The auto-ignition occurs either due to the high in-cylinder temperature the fuel is injected into or with the assistance of a glow plug, if the in-cylinder temperature is insufficient. There are, however, some parameters which can be adjusted to alter the combustion characteristics and therefore change the combustion concept [1]. The most commonly used commercial strategy, CDC, alongside the more recently studied partially premixed combustion (PPC) will be briefly discussed below.

2.1.1 CDC Operation and Emissions

The common diesel engine utilizes a combustion concept, commonly referred to as CDC. This combustion concept consists of the injection of diesel fuel, which could be fossil diesel fuel or a mix of fossil diesel fuel and FAME or HVO. Since diesel fuel have a high cetane number, CN[1], this fuel is ignited shortly after injection. There are often one or several injections close to the end of the compression stroke in terms of timing, which after a couple of crank angle degrees (CAD) of delay, are ignited and consumed with a very high combustion efficiency, $\eta_C \geq 99.7$ % [3].

One of the advantages of this combustion strategy is that it is easy to control the combustion event which is a crucial part of any ICE. The high level of controllability is due to the combustion phasing, load and the injection event being tightly linked, with the injection continuously which is controlled on a cycle-

[1] CN is a measure of the required r_c for a fuel to autoignite. Also, it is inversly related to RON [2].

to-cycle basis. The high r_c which contributes to a high efficiency, and therefore also torque, output for this engine type, alongside the high reliability, makes it very efficient and in the end cost effective. For these reasons, the CI engine is the best option for heavy duty applications [4].

The drawback of the CDC CI engine, on the other hand, is the exhaust gas; more specifically, the NO_X and soot emissions. The combustion concept of CDC produces significant amounts of NO_X due to the high combustion temperatures at low equivalence ratios, Φ. The high combustion temperature is often associated with the high r_c which CI engines utilizes, the same parameter which generally increases its efficiency. Moreover, since the fuel is injected and combusted with a very short ignition delay, the combustion occurs in a very stratified manner. The short ignition delay, alongside the typical molecular structure of diesel fuel, does not allow a high level of air entrainment into the fuel jet, which then burns in locally rich zones. The locally rich zones are the formation sites of the soot emissions, as illustrated below, in Figure 2.1. Soot oxidation occurs to a certain extent after the combustion event, however, not enough to reduce the soot emissions to a satisfactory level, in terms of the health and environmental aspect.

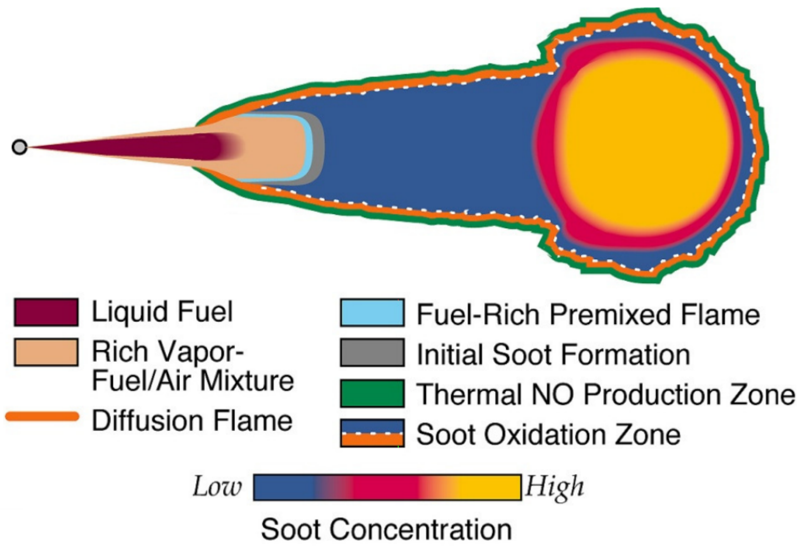


Figure 2.1: John Dec's phenomenological description of the combustion of a diesel fuel jet [5].

There are ways to reduce the emissions from the CI engine, however, they often come as trade-offs. For example, EGR could be used to suppress NO_X, however, exceeding a certain level of EGR will result in a penalty in soot, CO and THC emissions [6, 7]. The SOI timing can be set earlier to achieve a faster combustion with a higher level of premixing and lower soot formation as well as higher brake efficiency, η_B, or set later to compromise the η_B to obtain lower NO_X emissions. As a result of these kinds of trade-offs, the CI engine running on diesel fuel requires the utilization of advanced emissions aftertreatment systems (EATS) to be able to meet the EU VI and EU 6 standards [8, 9, 10].

2.1.2 PPC Operation and Emissions

The increased emissions stringency has led to the development of several new, cleaner and more efficient, combustion concepts for the CI engine. One of the combustion concepts is referred to as PPC. This concept, not being clearly defined, generally consists of the use of a low research octane number (RON) gasoline instead of diesel fuel, or low CN fuels in combination with high levels of EGR use in a CI engine. A common "recipe" for PPC generally consists in keeping λ=1.5 and EGR=50 % [11, 12]. The optimal fuel RON for a given engine depends to a large extent on the r_c, however, other parameters also play a role. For example, given a r_c of 17:1, the gasoline used to achieve PPC should preferably have a RON of approximately 70 to be able to run the engine over a wider range of loads [13]. The reasoning behind this is that the engine is suppose to have a hardware[2] which allows the fuel to ignite, without the need of extreme parameter settings, at the lower loads while keeping a certain premixing when going to the higher loads.

The use of a fuel with a significantly lower CN than that of diesel fuel, for example 70 RON gasoline, often results in a significant increase of the mixing period (MP), which in this work is defined as the difference between the SOI and start of combustion (SOC) according to Equation 2.1, seen below.

$$MP = EOI - SOC \tag{2.1}$$

This prolonged MP allows more air to be entrained into the fuel jet prior to the auto-ignition event, which in turn results in a more premixed combustion, suppressing the local rich areas and reducing the soot formation. Often, the use of high levels of EGR helps to increase the ignition delay and thus the premixing. Since exhaust gases are reintroduced into the combustion chamber, the overall specific heat ratio, γ, of the charge increases, reducing the combustion temperature and therefore also suppressing the NO_X formation.

It should be mentioned that in contrast to CDC, PPC generally runs with a longer positive MP. This results in a combustion phasing that is hard to control accurately, due to the fact that MP is a function of many variables such as the

[2] Mainly by using a piston with an appropriately chosen r_c.

intake temperature, T_{IN}, internal EGR, intake pressure, P_{IN}, SOI etc. In Figure 2.2, the NO_X and THC emissions can be seen as a function of SOI.

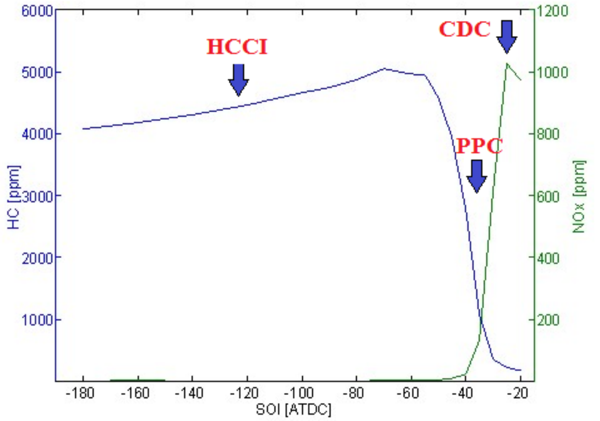


Figure 2.2: PPC injection strategy is an SOI located between truly homogeneous combustion, HCCI, and diffusion controller combustion, CDC [14].

As a result of using a fuel with a similar RON of gasoline or higher, the injection is set earlier to achieve a combustion phased around the same timing as CDC. The earlier injection will give the fuel slightly more time to impinge on the combustion chamber walls, which generally increases the THC and CO emissions. However, in previous works conducted at Lund University, this was circumvented by the use of injectors with narrower umbrella angles to avoid fuel impingement on the piston crown, the top land and cylinder liner when advancing the SOI [15, 8, 13].

As mentioned earlier, a good fuel for PPC operation is gasoline. Gasoline, having an average molecule length of 7-11 carbon atoms, will also form soot, however, at a lower level than that of diesel fuel, which consists of heavier compounds. Although soot emissions from gasoline combustion are significantly lower than those of diesel fuel, the NO_X-soot trade-off is still a reoccurring issue when running gasoline PPC.

For this reason, in combination with the lower CO_2 footprint, the use of light alcohols, such as methanol or ethanol has been proposed and examined in several studies to eliminate the trade-off, since these fuels do not produce soot in the same manner as diesel fuel and gasoline.

2.2 Influence of Fuels on Combustion

As mentioned in Section 2.1, the utilized fuel type plays an extremely important role and can alter the combustion strategy completely. In the following Sections, three kinds of fuels will be discussed along with their effect on the combustion process.

2.2.1 Diesel Fuel and Biodiesel

Diesel fuel is a high CN fuel, containing hydrocarbons with chain lengths in the range of 8-21 carbon atoms [16]. The chemical composition as well as the physico-chemical properties of this fuel can vary from one country to another. However, there are standards which regulates such fuel properties and in Europe it is the EN 590:2004-2009 Specifications (Euro 4/5). Properties such as minimum CN, density, maximum polycyclic aromatic hydrocarbon (PAH) content, sulfur and FAME content, etc. [17].

Since diesel fuel has a high CN and a certain concentration of PAH's, the fuel tends to produce a significant amount of soot during the combustion process. As mentioned in Section 2.1.1, this is an effect mainly attributed to the long carbon chain causing a very short ignition delay and a stratified two stage combustion, consisting of low temperature reactions and the main combustion. There are combustion strategies in place to reduce the emission of particulate matter, however, the common drawback is higher NO_X emissions instead.

Biodiesel can be any kind of methyl ester derived from vegetable oil. Biodiesel in general has longer average carbon chains than that of diesel fuel, and consequently also has a higher CN. Intuitively, this would reduce the ignition delay significantly and increase the engine out soot emissions. The main difference between diesel fuel and biodiesel, in terms of chemical composition, is the oxygen content. Diesel fuel consists mainly of saturated paraffins and naphthenes while having a significant aromatic content, about 25 vol.%, and no oxygenates [18]. Biodiesel, on the other hand consists of longer carbon chains which may be branched and unsaturated with a carboxyl group, giving biodiesels about 10-12 mass.% of oxygen [19, 20]. The oxygen content of biodiesels give the fuels an advantage during the combustion process; a faster and more complete oxidation. It is not only CO and THC that are better oxidized during the combustion process, but also soot [21].

The higher oxidiation level, which help to oxidize a higher amount of CO and THC, has the drawback of increasing the in-cylinder temperature, which in its turn increases the engine out NO_x emissions [22]. There are also other theories to why NO_X is increased when using biodiesel as a fuel. One of them is the radiative heat losses, which for biodiesel are smaller due to the lower amount of produced soot. The reduced radiation increases the gas temperature and thus also the NO_X formation. Also, the shorter ignition delay advances the combustion to a phasing which is located closer to the TDC, which in turn increases pressure and temperature and, in turn, the NO_X emissions [23]. The effect of diesel-biodiesel blends on engine out soot and NO_X emissions can be observed in Figure 2.3, below.

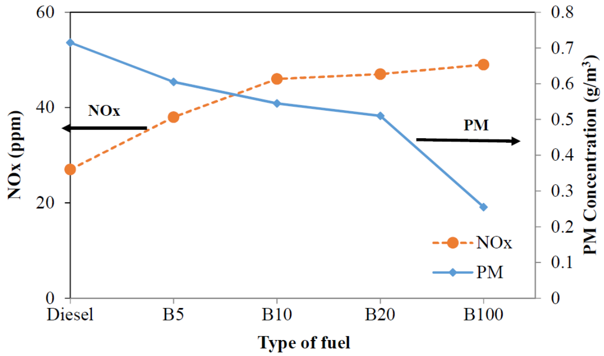


Figure 2.3: PM-NO_X under 20 % engine load for different diesel-biodiesel blends at 2500 rpm [24].

Since the ignition delay is very short when utilizing fuels like diesel and biodiesel, the combustion phasing can be controlled easily using the SOI, which results in a longer combustion duration. The disadvantage of the mixing controlled combustion, however, is that the combustion duration is increased, which to a certain extent reduces the gross indicated mean effective pressure, $IMEP_G$, and in turn the net indicated efficiency, η_{NIE}.

2.2.2 Gasoline and Naphtha

Gasoline and naphtha are petroleum based fuels consisting of shorter carbon chain than diesel fuel (4-12 carbon atoms), giving them a lower CN rating, or a higher octane number, ON [25]. Regarding the emissions, gasoline and naphtha would produce more HC and CO at lower loads due to longer MP at these conditions, contributing to a higher level of fuel impingement on the piston crown and combustion chamber walls. Moreover, the longer ignition delay causes overmixing in some regions of the combustion chamber, which become leaner than the flammability limits and therefore increases the CO and THC emissions [26]. At higher loads, however, at which the in-cylinder temperatures are higher, the MP is reduced while the oxidation process is improved. This tends to reduce the CO and THC emissions significantly [27].

The NO_X emissions are generally lower in comparison to the NO_X emissions of high CN fuels, given that the load is constant and EGR is not used. The higher level of premixing reduces the local Φ, which in turn reduces the local temperature and thus the NO_X [28]. EGR utilization when running an CI engine on fuels with higher ON can be used to both further reduce the NO_X emissions as well as cool down the combustion and reduce the maximum pressure rise rate, PRR_{MAX}, making the engine less noisy. As a result of a cooler combustion, the heat transfer losses can be somewhat decreased and a higher overall engine efficiency can be achieved.

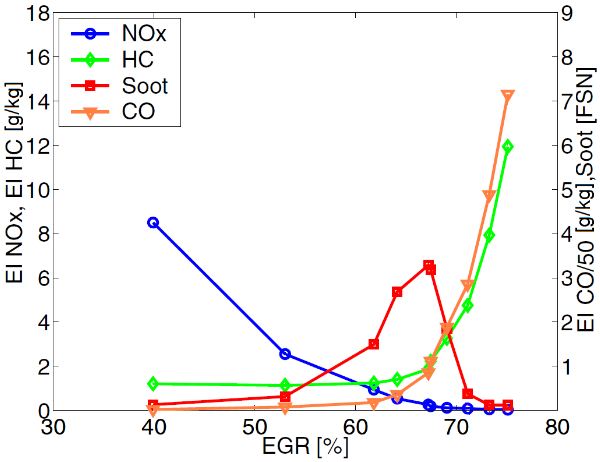


Figure 2.4: Emission index for THC, CO, NO_X and soot emissions as a function of EGR at 8 bar $IMEP_G$ [7].

The use of EGR can, however, in some cases be detrimental to the efficiency as well as emissions and must therefore be regulated with caution. As mentioned earlier, increasing the EGR decreases the heat transfer losses, however, the oxidation process deteriorates rapidly after exceeding a certain level. This causes the lower level of energy extraction from the fuel. Simultaneously, the oxidation level of the soot is also reduced and higher engine out soot emissions will be measured. Moreover, given similar operating conditions for gasoline and diesel fuel, gasoline will always produce less soot due to the shorter carbon chain.

The combustion of gasoline-like fuels is significantly more premixed than that of diesel fuel and biodiesel. This leaves a positive MP between the SOI and SOC, which makes combustion phasing slightly more difficult to keep under control. Since an engine in real life is also subjected to transients, this becomes an issue to take into serious consideration.

2.2.3 Methanol and Ethanol

Methanol and ethanol are the two simplest alcohols with one and two carbon atoms, respectively, attached to a OH^--group. The RON for these fuels is high, $>$106, due to the simple chemical structure and single stage ignition. The bonds and short length of methanol and ethanol molecules makes them quite hard to break. The short carbon chains of these fuels has the advantage of producing, more or less, a negligible amount of particulate matter and soot emission. Moreover, the CO_2 emissions, well-to-wheel, can be reduced dramatically if these fuels originate from feed-stock and produced using an efficient process [29, 30].

The CO and THC emissions, when running a CI engine utilizing ethanol or methanol, are higher than those of low CN fuels. The THC emissions are partially caused by the higher level of fuel impingement on the piston crown and combustion chamber walls, resulting in more unburned fuel leaving through the exhaust valves. The charge cooling effect of these fuels also causes a more unstable combustion, which increases the CO and THC emissions. Intuitively, the NO_X emissions should be reduced due to the charge cooling effect, however, this is not necessarily the case. Depending on the fraction of fuel burning in premixed mode and diffusion mode, which in turn depends on the ignition delay of the operating point, a violent premixed combustion can setup a high temperature for the diffusion mode, which increases can increase the NO_X formation.

The above mentioned charge cooling effect of alcohols can be detrimental to the ignition process itself, however, good in terms of reducing heat transfer losses from the charge to the surrounding combustion chamber walls. Moreover, a fast burning partially premixed charge has the potential to reduce the heat transfer further, in the sense that the time frame for heat transfer to occur is less than that of a fuel burning at a slower rate. This also has the potential to reduce the NO_X emissions since less time is spend at higher temperatures, and a larger quantity of the fuel charge burns in leaner, and cooler, regions [31]. The downside, similar to gasoline and naphtha, is the controllability. However, for alcohols the inability to control the combustion at normal intake temperatures is more severe than that of gasoline-like fuels. In Table 2.1, below, some properties of the light alcohols, diesel fuel, naphtha gasoline as well as soybean methyl ester (SME) can be observed.

Table 2.1: Fuel specifications [4, 32, 33, 34, 35].

	Diesel fuel	Naphtha Gasoline	SME	MeOH	EtOH
RON [-]	-	69	-	107-109	108-109
MON [-]	-	66	-	92	89
CN [-]	53	-	51	-	-
H/C [-]	1.8	1.98	1.79	4	3
O/C [-]	0	0	0.11	1	0.5
Q_{LHV} [MJ/kg]	43.2	43.68	37.0	19.9	26.9
$(A/F)_S$ [-]	14.5	14.68	12.4	6.5	9.0

Bibliography

[1] B. Challen and R. Baranescu. *Diesel Engines Reference Book*. Society of Automotive Engineers, 1999, ISBN:9780768004038.

[2] W. Dabelstein, A. Reglitzky, A. Schütze, K. Reders, and A. Brunner. *Automotive Fuels*, pages 1–41. American Cancer Society, 2016.

[3] M. Tunér. Review and Benchmarking of Alternative Fuels in Conventional and Advanced Engine Concepts with Emphasis on Efficiency, CO_2, and Regulated Emissions. *SAE Technical Paper 2016-01-0882*, 2016, DOI:https://doi.org/10.4271/2016-01-0882.

[4] J. B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988, ISBN:0-07-028637-X.

[5] J. E. Dec. A conceptual model of di diesel combustion based on laser-sheet imaging*. In *SAE Technical Paper*, 1997.

[6] M. Yao, Q. Zhang, H. Liu, Z.-q. Zheng, P. Zhang, Z. Lin, T. Lin, and J. Shen. Diesel Engine Combustion Control: Medium or Heavy EGR? *SAE Technical Paper 2010-01-1125*, 2010, DOI:https://doi.org/10.4271/2010-01-1125.

[7] C. Noehre, M. Andersson, B. Johansson, and A. Hultqvist. Characterization of Partially Premixed Combustion. *SAE Technical Paper 2006-01-3412*, 2006, DOI:https://doi.org/10.4271/2006-01-3412.

[8] C. Li, L. Yin, S. Shamun, M. Tunér, B. Johansson, R. Solsjo, and X.-S. Bai. Transition from HCCI to PPC: the Sensitivity of Combustion Phasing to the Intake Temperature and the Injection Timing with and without EGR. *SAE Technical Paper 2016-01-0767*, 2016, DOI:https://doi.org/10.4271/2016-01-0767.

[9] R. Vallinayagam, S. Vedharaj, Y. An, A. Dawood, M. Izadi Najafabadi, B. Somers, and B. Johansson. Combustion Stratification for Naphtha from CI Combustion to PPC. *SAE Technical Paper 2017-01-0745*, 2017, DOI:https://doi.org/10.4271/2017-01-0745.

[10] A. K. Agarwal, D. K. Srivastava, A. Dhar, R. K. Maurya, P. C. Shukla, and A. P. Singh. Effect of fuel injection timing and pressure on combustion, emissions and performance characteristics of a single cylinder diesel engine. *Fuel*, 111:374–383, 2013, DOI:https://doi.org/10.1016/j.fuel.2013.03.016.

[11] V. Manente, C.-G. Zander, B. Johansson, P. Tunestål, and W. Cannella. An Advanced Internal Combustion Engine Concept for Low Emissions and High Efficiency from Idle to Max Load Using Gasoline Partially Premixed Combustion. 2010.

[12] V. Manente, B. Johansson, P. Tunestål, and W. J. Cannella. Influence of Inlet Pressure, EGR, Combustion Phasing, Speed and Pilot Ratio on High Load Gasoline Partially Premixed Combustion. 2010.

[13] V. Manente. *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*. PhD thesis, Lund University, 2010.

[14] H. Nordgren, A. Hultqvist, and B. Johansson. Start of injection strategies for hcci-combustion. SAE International, 2004.

[15] M. Shen, M. Tunér, B. Johansson, P. Tunestål, and J. Pagels. Influence of injection timing on exhaust particulate matter emissions of gasoline in hcci and ppc. *SAE Technical Paper 2016-01-2300*, 2016, DOI:https://doi.org/10.4271/2016-01-2300.

[16] C. D. Collins. *Implementing Phytoremediation of Petroleum Hydrocarbons*, pages 99–108. Humana Press, 2007.

[17] EU: Fuels: Automotive Diesel Fuel. *Accessed 2018-09-12*, URL:https://www.dieselnet.com/standards/eu/fuel_automotive.php.

[18] C. Song. *Chemistry of Diesel Fuels*. CRC Press, 2000, ISBN:9781560328452.

[19] K. Kohse-Höinghaus, P. Oßwald, T. Cool, T. Kasper, N. Hansen, F. Qi, C. K. Westbrook, and P. R. Westmoreland. Biofuel Combustion Chemistry: From Ethanol to Biodiesel. *Angewandte Chemie International Edition*, 49(21):3572–3597, 2010, DOI:10.1002/anie.200905335.

[20] G. Belgiorno, G. D. Blasio, S. Shamun, C. Beatrice, P. Tunestål, and M. Tunér. Performance and emissions of diesel-gasoline-ethanol blends in a light duty compression ignition engine. *Fuel*, 217:78–90, 2018, DOI:https://doi.org/10.1016/j.fuel.2017.12.090.

[21] F. Soto, M. Alves, J. C. Valdés, O. Armas, P. Crnkovic, G. Rodrigues, A. Lacerda, and L. Melo. The determination of the activation energy of diesel and biodiesel fuels and the analysis of engine performance and soot emissions. *Fuel Processing Technology*, 174:69–77, 2018, DOI:https://doi.org/10.1016/j.fuproc.2018.02.008.

[22] M. Gumus, C. Sayin, and M. Canakci. The impact of fuel injection pressure on the exhaust emissions of a direct injection diesel engine fueled with biodiesel-diesel fuel blends. *Fuel*, 95:486–494, 2012, DOI:https://doi.org/10.1016/j.fuel.2011.11.020.

[23] S. K. Hoekman and C. Robbins. Review of the effects of biodiesel on NOx emissions. *Fuel Processing Technology*, 96:237–249, 2012, DOI:https://doi.org/10.1016/j.fuproc.2011.12.036.

[24] A. F. Yusop, R. Mamat, T. Yusaf, G. Najafi, M. H. M. Yasin, and A. M. Khathri. Analysis of particulate matter (pm) emissions in diesel engines using palm oil biodiesel blended with diesel fuel. *Energies*, 11(5), 2018.

[25] W. Dabelstein, A. Reglitzky, A. Schütze, K. Reders, and A. Brunner. *Automotive Fuels*, pages 1–41. American Cancer Society, 2016.

[26] S. Imtenan, H. Masjuki, M. Varman, M. Arbab, H. Sajjad, I. R. Fattah, M. Abedin, and A. S. M. Hasib. Emission and Performance Improvement Analysis of Biodiesel-diesel Blends with Additives. *Procedia Engineering*, 90:472–477, 2014, DOI:https://doi.org/10.1016/j.proeng.2014.11.759.

[27] C. Sayin. Engine performance and exhaust gas emissions of methanol and ethanol-diesel blends. *Fuel*, 89(11):3410–3415, 2010, DOI:https://doi.org/10.1016/j.fuel.2010.02.017.

[28] M. Pucilowski, M. Jangi, S. Shamun, M. Tuner, and X.-S. Bai. The Effect of Injection Pressure on the NOx Emission Rates in a Heavy-Duty DICI Engine Running on Methanol. *SAE Technical Paper 2017-01-2194*, 2017, DOI:https://doi.org/10.4271/2017-01-2194.

[29] S. Verhelst, J. W. Turner, L. Sileghem, and J. Vancoillie. Methanol as a fuel for internal combustion engines. *Progress in Energy and Combustion Science*, 70:43–88, 2019, DOI:https://doi.org/10.1016/j.pecs.2018.10.001.

[30] C. Wyman. *Handbook on Bioethanol: Production and Utilization*. Routledge, 2018, ISBN:9781351441773.

[31] G. Valentino, F. E. Corcione, S. E. Iannuzzi, and S. Serra. Experimental study on performance and emissions of a high speed diesel engine fuelled with n-butanol diesel blends under premixed low temperature combustion. *Fuel*, 92(1):295–307, 2012, DOI:https://doi.org/10.1016/j.fuel.2011.07.035.

[32] Implementing agreements for advanced motor fuels - ethanol properties. *Accessed 2018-10*, URL:http://www.iea-amf.org/content/fuel_information/fuel_info_home/ethanol/e10-/ethanol_properties.

[33] Implementing agreements for advanced motor fuels - methanol properties. *Accessed 2018-10*, URL:http://www.iea-amf.org/content/fuel_information/methanol.

[34] A. I. Bamgboye and A. C. Hansen. Prediction of cetane number of biodiesel fuel from the fatty acid methyl ester (FAME) composition. *Int. Agrophysics*, 2(1):21–29, 2008.

[35] Z. Habib, R. Parthasarathy, and S. Gollahallo. Performance and emission characteristics of biofuel in a small-scale gas turbine engine. *Applied Energy*, 87(5):1701–1709, 2010.

Chapter 3

Methodology

3.1 Experimental Setups

3.1.1 Scania D13

The Scania D13 engine started out as a straight six cylinder heavy duty (HD), however, it was modified to run on only one cylinder in the test cell. This modification included a heavier flywheel due to the loading from combustion in only one cylinder. To maintain a balanced crankshaft, the pistons were removed and replaced with hollow weights to balance the crank shaft assembly as well as removing the compression work. The single cylinder in operation has a displacement of $\sim$2124 cm^3 and is supplied air from an in-house compressor which has the capability to achieve pressures well over 8 bar. To simulate a real turbocharger, when needed, an exhaust back pressure butterfly valve was located downstream of the exhaust manifold. An EGR valve can be adjusted to recirculate exhaust to the intake, given that the back pressure is higher than the intake pressure. This enabled the use of cooled EGR in different ratios, since the EGR passed a water cooled heat exchanger before being mixed with fresh air just outside the intake manifold.

Pressure sensors were placed in the intake, exhaust as well as inside the cylinder head to measure the crank angle resolved pressure. To measure the temperature, thermocouples were located at the intake, exhaust as well as before and after the EGR cooler. Moreover, several other pressure sensors and thermocouples were located in places either for feedback control purposes or to maintain a safe operation while conducting experiments. In Figure 3.1, the engine setup schematic can observed.

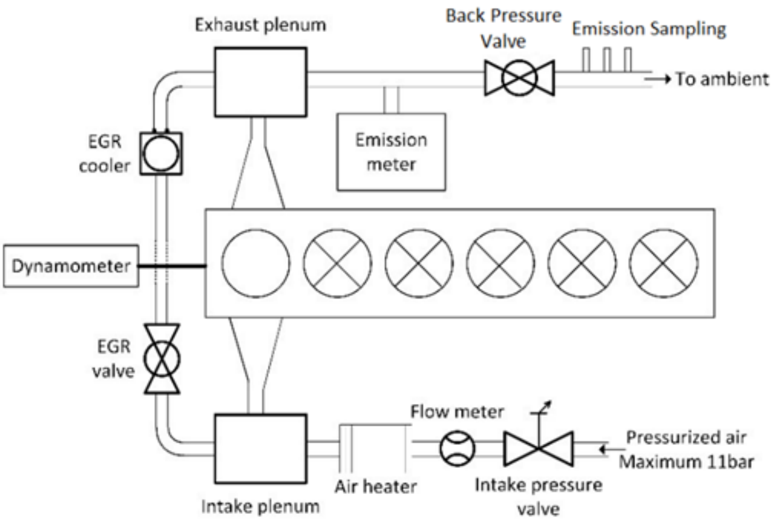


Figure 3.1: Schematic of the Scania D13 single cylinder engine.

The emission sampling probes, seen in Figure 3.1, are not located as close to each other as the Figure in question suggests. Great care was taken when determining the probe locations, since a minimum distance between each probe is required as per instruction from the manufacturers.

The utilized fueling system on the engine is a common rail type, controlled by a solenoid valve and a pulse width modulation signal. Since this engine was to be run only on one cylinder, the common rail system was adjusted to accommodate this kind of operation. The high pressure pump was factory modified both in terms of fuel flow rate as well as changing different gaskets and materials which are in contact with the fuel. This is necessary to keep the pump operation reliable when utilizing low lubricity, low viscosity, corrosive fuels such as gasoline, methanol or ethanol. The injectors used in this work were also modified to be able to withstand methanol corrosivity as well as supplying fuel at a higher flow rate, which accommodates the lower A/F-ratio and heating value of the light alcohols. Since the light alcohols, and gasoline to a certain extent, is harder to ignite than regular diesel fuel, an intake heater had to be utilized in order to achieve a stable combustion.

National Instruments LabView software was utilized to obtain measurement signals and control everything related to the engine itself; injection timing, injection duration, intake and exhaust pressures, EGR level as well as the intake temperature. However, the engine speed was controlled with its separate control system by means of a dynamo to either motor the engine at a set speed when no load is applied, or braking the engine at the set speed if load is applied. Below in Table 3.1, the engine specification can be observed.

Table 3.1: Engine specifications

Displaced volume [cm^3]	2124
Stroke [mm]	160
Bore [mm]	130
Connecting rod length [mm]	255
Number of valves [-]	4
Swirl ratio [-]	2.1
Exhaust valve open [-]	137^o ATDC
Inlet valve open [-]	-141^o ATDC

This engine was used in the works mentioned in Sections 6.1,6.2,6.3 and 6.6. Every study was unique in terms of operating conditions and measurements, thus requiring different hardware. The tested fuels in this setup were methanol, ethanol, naphtha gasoline with RON =69 and Swedish commercial diesel fuel, termed MiljöKlass-1, MK-1. The r_c as well as the piston crown geometry were changed from one study to another depending on the performed experiments and their goals.

3.1.2 Fiat/GM JTE 1.9

Similarly to the Scania D13 engine presented in Section 3.1.1, this engine also started out as a multi-cylinder engine, a four cylinder light duty (LD) engine. However, this engine still utilized the factory piston, with a geometrical r_c of 16.5:1, when converted into a single cylinder engine. The crank shaft as well as the engine block were custom manufactured while the cylinder head, originating from a Fiat/GM JTE 1.9 diesel engine, was modified to fit the single cylinder block. The valvetrain from the Fiat/GM JTE 1.9 engine was used in this engine setup. Pressurized air was supplied to this engine, as for the Scania D13, by a in-house compressor. Back pressure and EGR was regulated in the same manner as for the engine setup described in Section 3.1.1, to be able to simulate the back pressure of a turbocharger.

The fuel delivery system in this engine setup was also a common rail type, similar to the Scania D13. The high pressure pump, supplying fuel to the common rail, was driven by an external electric motor and hence it was completely decoupled from the crankshaft to obtain a higher level of flexibility.

The injectors and high pressure fuel pump were in this setup not were modified for alcohol utilization. The reason for this was that the fuel blends, used in the experiments, consisted of at least 56 vol.% Italian commercial diesel fuel.

With such high concentration of diesel in the fuel blend, it was deemed safe to operate the engine for a certain period of time without increasing the risk of severely damaging the engine itself or the fueling system. In Figure 3.2, below, the engine schematic of the Fiat/GM JTE 1.9 can be observed.

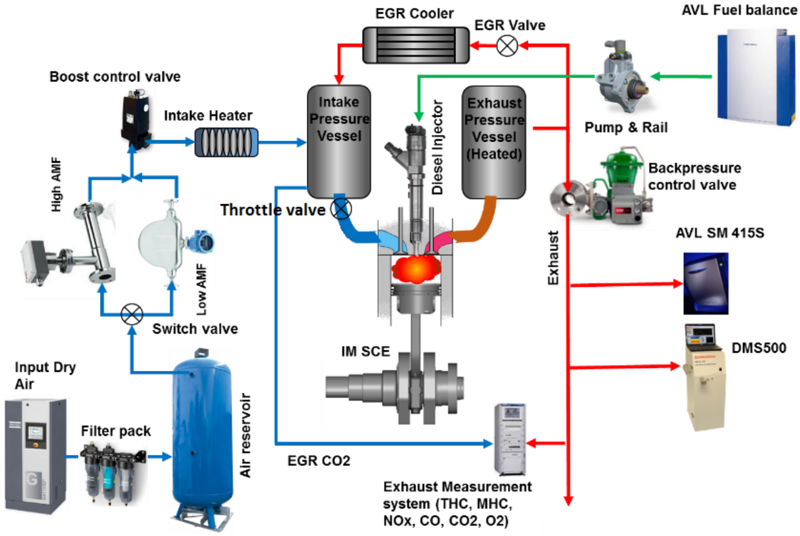


Figure 3.2: Schematic of the Fiat/GM JTE 1.9 single cylinder engine.

The engine control system used the partially modified stock engine control unit (ECU) to control the different engine parameters related to the injection. The injection pressure, engine speed, EGR, intake heating, among other parameters, were controlled individually and were outside the control of the ECU. As for the engine setup in Section 3.1.1, the engine crankshaft was connected to an electrical AVL high speed dynamo. The data acquisition was obtained and post processed in real time using Indicom, which is software provided by AVL. Below in Table 3.2, some of the important engine specifications can be observed. This engine was used in the works mentioned in Sections 6.4 and 6.5.

Table 3.2: Engine specifications

Displaced volume [cm^3]	478
Stroke [mm]	90.4
Bore [mm]	82
Connecting rod length [mm]	145
Number of valves [-]	4
Swirl ratio [-]	2.1
Exhaust valve close [-]	152° BTDC
Diesel Injection System [-]	Common rail, 1800 bar
Diesel Injector [-]	Solenoid, 7 hole microsac

In the Fiat/GM JTE 1.9 engine experiments, Italian commercial diesel fuel was tested as a baseline. The results were then compared with four diesel-ethanol blends. Since diesel fuel does not blend with ethanol, an emulsifier was required. Two emulsifiers were used; biodiesel and commercial gasoline. The first diesel-biodiesel-ethanol emulsion consisted of 68 vol.% diesel, 17 vol.% biodiesel and 15 vol.% ethanol, while the second consisted of 56 vol.% diesel, 14 vol.% biodiesel and 30 vol.% ethanol. They are termed DBE15 and DBE30, after the ethanol content. The second emulsion blend type consisted of the same ratios of the different fuels, except that the biodiesel was exchanged with gasoline.

3.1.3 Measurement systems

In this work, various measurement systems were used to obtain and analyze data. Some of the more important measurement systems are presented below.

3.1.3.1 Gaseous Emissions and In-Cylinder Pressure

The gaseous emission measurements were conducted by three different emission measurement systems, however, they used the same measurement principles. The CO and CO_2 were measured dry by a non-dispersive infrared detector while the NO - and NO_2 when measurement was available - where quantified using a chemiluminescence detector. A flame ionization detector was used to measure the wet concentration of the THC emissions, while a paramagnetic detector was used to measure the oxygen concentration.

The pressure sensor inside the cylinder is a piezoelectric transducer which outputs an electrical charge. The electrical charge must be converted to a voltage and amplified. After the amplification, the signal is sent to the data acquisition system which converts the voltage to a pressure value - every 0.2 CAD -

depending on the magnitude of the voltage.

The crank position measurement is conducted with a photoelectric scanning technique. This measurement technique consists of a disk which rotates at the same speed as the crank shaft of the engine. An output signal is generated every time two slits move relative to scanning the reticle, which in the case of this work, is every 0.2 CAD.

3.1.3.2 Particulate Emissions

The particulate matter were measured using mainly the following three systems: AVL Micro Soot Sensor, Cambustion DMS500 M177 and the JEOL 3000F Transmission Electron Microscope.

AVL Micro Soot Sensor
The AVL Micro Soot Sensor (MSS) utilizes a photoacoustic technique to measure the soot mass concentration in the exhaust gases. The measurement technique assumes that the soot particles are completely black and are able to strongly absorb modulated light with a wavelength of λ=808 nm. This modulated light is directed towards the flow path of the diluted exhaust gas, and the containing particles will start to fluctuate in size due to the heating and cooling. The size fluctuation causes a sound wave, which is detected by sensitive microphones and then translated into a concentration of soot. The MMS has a measurement range from 0.001-1000 mg/m^3 with a sensitivity in $\mu m/m^3$ [1, 2, 3].

Cambustion DMS500 M177
Using a corona charger, the DMS500 gives particles a positive charge. The particle's obtained charge is approximately proportional to its surface area. The charged particles are passed through a strong radial electrical field and depending on their electrical mobility, they are deposited on different electrometer detectors. The currents generated from every particle deposition, on one of the 22 electrometers, is detected. By using an inversion matrix, the signals are transformed into a number particle size distribution. With a maximum sampling rate of 10 Hz, the DMS500 can measure PN in the size range of $\sim$4.6-1 μm [3, 4].

JEOL 3000F Transmission Electron Microscope
Exploiting the wave-particle duality of electrons, a transmission electron microscope (TEM) utilizes a beam of electrons instead of light to visualize objects. A TEM generally consists of an electron emission source instead of a

light source, electromagnetic lenses instead of optical lenses and an electron detector replaces the observer's eye. An electron beam is produced, accelerated and then focused on the sample by the electromagnetic lenses. The beam is modified when passed through the sample and then imprinted on the electron detector which produces an image of the sample [5].

To determine the chemical composition of the specimen, a X-ray energy dispersive spectroscopy (XEDS) is generally utilized in parallel with the TEM. When electrons pass the specimen, electrons from the specimen are excited creating an electron hole. When electrons from the outer shell, with higher energy, fills the electron hole, X-rays are emitted. Since the energy of the emitted and then measured X-rays are characteristic to the energy difference between the two shells and the atomic structure of the emitting element, the XEDS allows the determination of the elemental composition [5].

The JEOL is a 300 kV analytical high resolution TEM utilizing a field emission electron source. It has a video camera as well as a 2x2 k charge coupled device camera for high resolution imaging. The resolutions for the microscope is 0.17 nm when utilizing conventional TEM mode. The microscope utilizes an Oxford XEDS and a Gatan imaging filter for analysis of chemical composition with a spatial precision of $>$1 nm [6].

3.2 Post Processing Procedure

The following Subsection will present the main procedure of the calculation process. The calculation of the most common/important engine output variables will be presented. The main references of this Section are [7, 8].

3.2.1 Mean Effective Pressures and Efficiencies

In Figure 3.3 below, the flowchart of the energy flow can be observed. The energy flow, however, is not expressed in energy, but is normalized with the engine displacement volume. This is conducted to enable the comparison between different engines with varying sizes.

The calculation starts by calculating the fuel indicated mean effective pressure (FuelMEP) with Equation 3.1, observed below.

$$FuelMEP = \frac{m_f \times Q_{LHV}}{V_D} = \frac{\dot{m}_f \times n_T \times Q_{LHV}}{N \times V_D} \tag{3.1}$$

where $\dot{m}_f$ is the fuel flow, n_T is the stroke factor, Q_{LHV} is the energy density of the fuel, N is the engine speed and V_D is the engine displacement. Since the combustion is not complete, the energy extracted from the combustion process is expressed as $QMEP$, as follows.

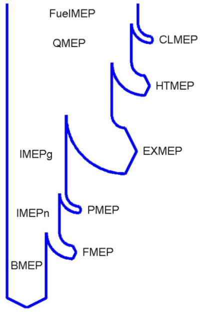


Figure 3.3: Energy input from fuel on top to the break energy output at the bottom.

$$QMEP = FuelMEP \times \eta_C \tag{3.2}$$

where η_C is the combustion efficiency as calculated with Equation 3.10. The combustion loss, $CLMEP$, then is obtained with the following formula:

$$CLMEP = FuelMEP - QMEP \tag{3.3}$$

The $IMEP_G$ calculation is conducted only during the compression and expansion process in the region between $-180 \leq CAD$
≤ 180.

$$IMEP_G = \frac{1}{V_D} \int_{-180}^{180} \mathbf{P} d\mathbf{V} \tag{3.4}$$

where $\mathbf{P}$ is the pressure vector and $\mathbf{V}$ is the volume vector calculated according to Equation 3.32, in Subsection 3.2.3. The $IMEP_N$ is the mean effective pressure for the whole cycle and is calculated in the same manner as $IMEP_G$, however, for the whole cycle: -360≤CAD≤360.

$$IMEP_N = \frac{1}{V_D} \int_{-360}^{360} \mathbf{P} d\mathbf{V} \tag{3.5}$$

The difference between $IMEP_G$ and $IMEP_N$ is the gas pumping loss, $PMEP$.

$$PMEP = IMEP_G - IMEP_N \tag{3.6}$$

By subtracting $IMEP_G$ from $QMEP$, the total heat loss is obtained. The total heat loss consists of the heat transfer loss, $HTMEP$, as well as the exhaust loss, $ExMEP$, according to the expression seen in Equation 3.7.

$$HTMEP + ExMEP = QMEP - IMEP_G \tag{3.7}$$

The $HTMEP$ can then be separated when calculating the exhaust energy per cycle and normalizing it with V_D using Equations 3.8 and 3.9 seen below.

$$W_{EX} = \dot{m}_f \frac{c_{p,IN} + c_{p,EX}}{2}(T_{IN} - T_{EX}) \tag{3.8}$$

$$ExMEP = \frac{n_T \times W_{EX}}{N \times V_D} \tag{3.9}$$

The W_{EX} is the exhaust energy flow, $c_{p,IN}$ and $c_{p,EX}$ are the specific heats for the gas in the intake and exhaust respectively. When calculating these values, consideration is taken of the composition of the gas: depending on the composition and temperature of the intake and exhaust gas, the c_p for each is estimated using the exhaust gas emission measurements. The brake mean effective pressure, $BMEP$, as well as the brake efficiency are not calculated throughout this thesis since only single cylinder engines were used in the experiments.

In this work, mainly four part efficiencies will be discussed: combustion efficiency, η_C, thermal efficiency, η_T, gross indicated efficiency, η_{GIE}, and net indicated efficiency, η_{NIE}. The thermal efficiency can be divided into two parts. The first part, η_{HT}, represents the energy which has not dissipated through heat transfer to the combustion chamber and exhaust ports. The second part of the η_T represents the loss in form of hot exhaust gases.

The η_C is calculated from the exhaust gas emission and its calculation procedure follows the Equation 3.10, below.

$$\eta_C = \frac{\sum \frac{M_i}{M_p} x_i^* (1 - x_{H_2O}) Q_{LHV,i}}{\frac{Q_{LHV,f}}{1+A/F}} \tag{3.10}$$

where M_i is the molar mass for the exhaust gas denoted i, M_p is the molar mass for all gas emissions, x_i^* is the dry exhaust gas fraction, x_{H2O} is the water fraction, $Q_{LHV,i}$ and $Q_{LHV,f}$ are the lower heating values for each exhaust gas component and the fuel respectively. Finally, the A/F is the air to fuel ratio. Equation 3.10 generally applies for $i = H_2$, THC, CO and occasionally PM. For further calculations regarding the exhaust gas emissions, the reader is referred to Subsection 3.2.2.

3.2.2 Exhaust Gas Analysis

The calculation below assumes the measurement of the following exhaust gases: THC, CO, CO_2, $O_{2,p}$ and NO_X. NO_X is assumed to be $NO + NO_2$ while the THC is summed to have the chemical formula $C_aH_bO_c$ or $C_{\frac{a}{a}}H_{\frac{a}{b}}O_{\frac{a}{c}}$. The combustion equilibrium is simplified to

$$\begin{aligned} C_aH_bO_c + \lambda \times n_{O_2,r}(O_2 + 3.773N_2) = n_{C_aH_bO_c} \times C_aH_bO_c + n_{CO_2} \times CO_2 \\ + n_{H_2O} \times H_2O + n_{CO} \times CO + n_{H_2} \times H_2 + n_{O_2,p} \times O_2 + n_{N_2} \times N_2 \\ + n_{NO} \times NO + n_{NO_2} \times NO_2 \end{aligned} \tag{3.11}$$

where n is the number of moles for component i. With the total mole number for all the products, n_p, the gas concentration is expressed as x, which follows the expression in Equation 3.12, below.

$$x_i = \frac{n_i}{n_p} \tag{3.12}$$

Equation 3.11 can then be rewritten to the following expression

$$\begin{aligned} C_aH_bO_c + \lambda \times n_{O_2,r}(O_2 + 3.773N_2) = n_p(x_{C_aH_bO_c} \times C_aH_bO_c \\ + x_{CO_2} \times CO_2 + x_{H_2O} \times H_2O + x_{CO} \times CO + x_{H_2} \times H_2 + x_{O_2} \times O_2 \\ + x_{N_2} \times N_2 + x_{NO} \times NO + x_{NO_2} \times NO_2) \end{aligned} \tag{3.13}$$

In Equation 3.13, a, b, c, $n_{O_2,r}$, n_p, x_{H_2O}, x_{N_2} and x_{H_2} are all unknown and must be obtained to conduct the analysis. This is done by doing an equilibrium calculation for each element, C, H, O and N.

$$a = n_p(a \times x_{C_aH_bO_c} + x_{CO} + x_{CO_2}) \tag{3.14}$$

$$b = n_p(b \times x_{C_aH_bO_c} + 2x_{H_2O} + 2x_{H_2}) \tag{3.15}$$

$$c + 2\lambda n_{O_2} = n_p(c \times x_{C_aH_bO_c} + 2x_{CO_2} + x_{H_2O} + x_{CO} + 2x_{O_2} + x_{NO} + 2x_{NO_2}) \tag{3.16}$$

$$2 \times 3.773\lambda \times n_{O_2} = n_p(2x_{N_2} + x_{NO} + x_{NO_2}) \tag{3.17}$$

Moreover, the total concentration of all components at the exhaust side should always be equal to unity. From this we obtain Equation 3.18.

$$x_{C_aH_bO_c} + x_{CO_2} + x_{H_2O} + x_{CO} + x_{H_2} + x_{O_2} + x_{N_2} + x_{NO} + x_{NO_2} = 1 \tag{3.18}$$

The CO and CO_2 concentration in the exhaust gas can be related to the H_2O as well as the H_2 with the equilibrium Equation 3.19, seen below.

$$K(T) = \frac{x_{CO} \times x_{H_2O}}{x_{CO_2} \times x_{H_2}} \tag{3.19}$$

where $K(T)$ is generally assumed to be 3.5, which corresponds to evaluating the combustion equilibrium at ~1740 K. From Equations 3.11 to 3.19, it is possible to obtain the values of all the unknown variables mentioned above. However, it is necessary to take the measurement method of each exhaust component into consideration since some species are measured together with the water content, wet, and some without, dry. The relation between wet and dry exhaust gas concentration can be seen in Equation 3.20, below.

$$x_i = x_i^*(1 - x_{H_2O}) \tag{3.20}$$

where x_i and x_i^* are the wet and dry concentration of exhaust gas i.

3.2.3 Heat Release Analysis

Application of the first law of thermodynamics yields the following expression:

$$\partial Q = \partial U + \partial W + \partial Q_{HT} + \partial Q_{Crevice} + \partial Q_{Blowby} \tag{3.21}$$

where ∂Q is the heat released from combustion, ∂Q is the change in the internal energy, ∂W is work conducted by the system, ∂Q_{HT} is the heat transferred to the combustion chamber walls, $\partial Q_{Crevice}$ is the heat loss due to flow in and out from small crevice volumes and ∂Q_{Blowby} is the heat loss due to blowby. Throughout this work, it is assumed that $\partial Q_{Crevice} = \partial Q_{Blowby} = 0$. The internal energy, U, can be expressed as

$$U = m \times u = m \times c_v \times T \tag{3.22}$$

where m is the mass inside the cylinder, c_v is the specific heat at a constant volume and T is the temperature. From this we obtain that

$$\partial U = m \times \partial u + u \times \partial m = m \times c_v \times \partial T + u \times \partial m \tag{3.23}$$

With the assumption that the system is completely insulated and there is no flow of gas into or out from the cylinder, the following expression is obtained for ∂W

$$\partial W = P \times \partial V \tag{3.24}$$

where P is the cylinder pressure and V is the cylinder volume at a given CAD. The ideal gas law, with constant m and R can be expressed as

$$\frac{\partial P}{P} + \frac{\partial V}{V} = \frac{\partial T}{T} \tag{3.25}$$

which can be rewritten to

$$\partial T = T\left(\frac{\partial P}{P} + \frac{\partial V}{V}\right) \tag{3.26}$$

With Equations 3.23, 3.24 as well as 3.26 inserted into 3.21, the following expression is obtained:

$$\partial Q = m \times c_v \times T\left(\frac{\partial P}{P} + \frac{\partial V}{V}\right) + P \times \partial V + \partial Q_{HT} \tag{3.27}$$

By rewriting the ideal gas law as $mT = \frac{PV}{R}$, Equation 3.27 can be expressed as observed below.

$$\partial Q = \left(1 + \frac{c_v}{R}\right) P \times \partial V + \frac{c_v}{R} V \times \partial P + \partial Q_{HT} \tag{3.28}$$

In Equation 3.28, the expressions $1 + \frac{c_v}{R}$ and $\frac{c_v}{R}$ can be rewritten to $\frac{\gamma}{\gamma-1}$ and $\frac{1}{\gamma-1}$ respectively, due to the fact that for ideal gases, the two following relations apply:

$$R = c_p - c_v \tag{3.29}$$

$$\gamma = \frac{c_p}{c_v} \tag{3.30}$$

If the change in energy occurs for every CAD increment, $\partial\theta$, the final expression becomes:

$$\frac{\partial Q}{\partial \theta} = \frac{\gamma}{\gamma - 1} P \frac{\partial V}{\partial \theta} + \frac{1}{\gamma - 1} V \frac{\partial P}{\partial \theta} + \frac{\partial Q_{HT}}{\partial \theta} \tag{3.31}$$

For this calculation to be possible, the functions $V(\theta)$ and $V'(\theta)$ must be obtained. These are presented as Equations 3.32 and 3.33, below.

$$V(\theta) = V_c + \frac{\pi \times B^2 \times t}{4}\left[l/t + 1 - \cos(\theta) - \sqrt{(l/t)^2 - \sin^2(\theta)}\right] \tag{3.32}$$

$$V'(\theta) = \frac{V_d \times \sin(\theta)}{2}\left[1 + \frac{\cos(\theta)}{\sqrt{(l/t)^2 - \sin^2(\theta)}}\right] \tag{3.33}$$

where V_c and V_d are the volume at TDC and the displaced volume, t is the crank radius, l is the connecting rod length, B is the bore. Woschni's heat transfer model was used to estimate $\frac{\partial Q_{HT}}{\partial \theta}$. The model includes a heat transfer coefficient, h, which has been empirically determined. The heat transfer rate is presented as Equation 3.34.

$$\frac{\partial Q_{HT}}{\partial \theta} = A_w \times h \times (T_g - T_w) \tag{3.34}$$

where A is the combustion chamber wall area, T_g and T_w is the gas and wall temperature respectively. h follows the relation seen below.

$$h = C \times B^{-0.2} \times P^{0.8} \times T^{-0.55} \times w^{0.8} \tag{3.35}$$

where C is some tunable constant and w is given by Equation 3.36, below.

$$w = C_1 \times \overline{S} + C_2 \frac{V_d \times T_r}{P_r \times V_r}(P - P_m) \tag{3.36}$$

where $\overline{S}$ is the average piston velocity. T_r, P_r and V_r is the temperature, pressure and volume at a reference point while P_m is the motored pressure.

Bibliography

[1] W. Schindler, M. Arndt, and A. P. B. Giechalskiel. AVL Micro Soot Sensor Applications. *AVL Tech Days 2012*, 2012.

[2] MSS$^{\text{plus}}$ AVL Micro Soot Sensor. *Accessed 2018-09-06*, URL:https://www.avl.com/-/mssplus-avl-micro-soot-sensor.

[3] S. Shamun, M. Shen, B. Johansson, M. Tuner, J. Pagels, A. Gudmundsson, and P. Tunestal. Exhaust PM Emissions Analysis of Alcohol Fueled Heavy-Duty Engine Utilizing PPC. *SAE Int. J. Engines*, 9(4):2142–2152, 2016, DOI:10.4271/2016-01-2288.

[4] DMS500 MkII Fast Engine Particulate Analyzer. *Accessed 2018-09-06*, URL:https://www.cambustion.com/products/dms500/engine.

[5] J. Goldstein, D. Newbury, D. Joy, C. Lyman, P. Echlin, E. Lifshin, L. Sawyer, and J. Michael. *Scanning Electron Microscopy and X-Ray Microanalysis*. Springer, 2003, ISBN:978-1-4615-0215-9.

[6] Description of Our Instruments - JEOL 3000F. *Accessed 2018-09-06*, URL:http://www.polymat.lth.se/nchrem/microscopes/.

[7] J. B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw Hill, 2013, ISBN:0-07-028637-X.

[8] B. Johansson. *Förbränningsmotorer*. Division of Combustion Engines, Lund University, 2006.

Chapter 4

Results and Discussion

In this Section, the results in terms of combustion characteristics, efficiencies and emissions - both gaseous and particulate matter - will be presented and discussed.

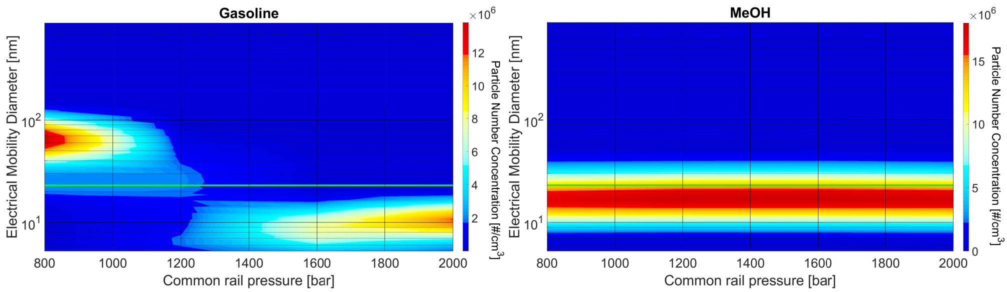
4.1 Combustion Characteristics

The combustion characteristics of methanol and ethanol in a CI engine are quite different from those exhibited by commercial diesel fuel. As earlier mentioned, the first and foremost difference is the ignition resistance between alcohols and diesel fuel. For the reader to comprehend the effects of fuel RON on ignitability, the required intake temperature to achieve combustion in the Scania D13 engine, is ~105 °C for ethanol and ~155 °C for methanol. To put these numbers in perspective, diesel fuel would ignite with a very short ID with an intake temperature equal to - or lower than - room temperature. The intake temperatures mentioned for the alcohols applies when utilizing a piston with r_c=15 and the engine coolant already heated to ~65 °C. The requirement of high intake temperature is a result of the high heat of vaporization in combination with a low stoichiometric $(A/F)_S$ for the light alcohols in comparison to other fuels. Since more fuel has to be injected to achieve a combustible AF-mixture, more energy is drawn from the air inside the combustion chamber and thus the cooling effect becomes more significant. Depending on the injection strategy used, the cooling effect, in its turn, affects many other combustion parameters.

4.1.1 Injection Pressure

For a CI engine running on diesel fuel , the injection pressure can be used to regulate the injected fuel mass for any given constant injection duration. Generally, at higher loads, the injection pressure is increased to increase the fuel flow through the injector and at lower loads, the injection pressure is decreased. There is a possibility, at higher loads, to increase the injection duration instead of the injection pressure, however, the drawback of doing so is high PM emissions. When increasing the injection pressure, the amount of air entrained into the fuel jet becomes higher. This leads to better premixing and a faster combustion at a higher temperature which in turn gives more time for higher level of soot oxidation to occur, thus reducing the soot formation. It should be noted that the energy required to compress the fuel is taken directly from the crank shaft, meaning that higher injection pressures introduce higher losses.

In terms of injection pressure, the advantage of methanol and ethanol is that the PM emissions do not have to be taken into consideration when setting the injection pressure as Figure 4.1, below, suggests. Instead, given a single injection strategy is utilized, this engine parameter can be set to maintain a reasonable PRR_{MAX} while also controlling the NO_X emissions. Since a larger amount of fuel is needed to obtain similar load for the light alcohols in comparison to diesel fuel, the injector needs larger holes to maintain a higher fuel flow, which commonly causes PM emissions to increase if commercial diesel fuel would be used.



(a) Particle size distribution vs. injection pressure for naphphta gasoline.

(b) Particle size distribution vs. injection pressure for methanol.

Figure 4.1: Particle size distribution vs. common rail pressure for 4.1a naphphta gasoline and 4.1b methanol at 6 bar $IMEP_N$.

An issue which could arise when running a CI engine on light alcohols at low injection pressures is insufficient fuel evaporation. If the fuel is not broken up properly into small droplets, the mixture will be too rich, at least locally, to ignite. A numerical study, conducted by Svensson et al. [1] suggests that the ignition of methanol in CI engines occurs at the leaner locations of the charge. The issue at lower injection pressures is generally not an issue at higher loads due to higher in-cylinder temperatures which counteract the deteriorated fuel evaporation.

It is evident that methanol utilization requires a higher r_c than a conventional CI engine running on commercial diesel fuel, if preheating the air is excluded. High injection pressures can potentially give rise to high THC emissions and thus reduced η_C. When the fuel is injected at high pressures, wall wetting becomes an issue due to the piston crown being rather close to the injector nozzle. Wall wetting in this work refers to fuel impinging on any surface of the combustion chamber. Apart from wall wetting, a part of the fuel will end up in the volume between the piston crown and the cylinder liner as well as in the squish volume, in which fuel cannot be oxidized properly.

4.1.2 Ignition Delay

Ignition delay is defined as the difference in time, often expressed in CAD, between the start of first injection and start of combustion and can be observed as Equation 4.1, below. In this section, however, the ID is discussed considering only the main injection while disregarding the pilot injection. The ID depends on several factors, such as fuel properties, intake temperature, number of injections, injection timing, combustion phasing, intake oxygen concentration etc.

$$ID = SOI - SOC \tag{4.1}$$

Single Injection The first noticeable effect after the injection event is the ID. If all engine parameters are constant, the ID is prolonged significantly due to the cooler in-cylinder conditions when utilizing methanol or ethanol in comparison to, for example, the majority of the consistuents in gasoline. Since methanol and ethanol have lower boiling points in combination with a constant distillation curve, these fuels evaporate at a lower temperature than the conventional petroleum based fuels, making the cooling effect a factor to be considered seriously [2]. A high intake temperature is required to even out the cooling effect and eventually keeping the in-cylinder temperature equal to, or above, the auto ignition temperature of the fuel [3]. In Figure 4.2 (b), below, the ID can be seen for three different fuels: diesel fuel, DBE15 and DBE30, consisting of diesel fuel, biodiesel and ethanol in the ratios 68:17:15 and 56:14:30, respectively [4].

Split Injection A pilot injection will absorb less energy from the compressed gas during evaporation and will therefore combust with a shorter ignition delay. Since the pilot increases the in-cylinder temperature post injection, the ignition delay for the main injection will be reduced significantly. With this injection strategy, it is possible to have a lower intake temperature in comparison to the single injection strategy, given that load, speed and all other control parameters are set equally. The ID is significantly reduced with a double injection strategy, a trend of the MP can still be seen between different fuels. Below in Figure 4.2 (a), the ID can be seen for diesel fuel, DBE15 and DBE30, when utilizing a double injection strategy.

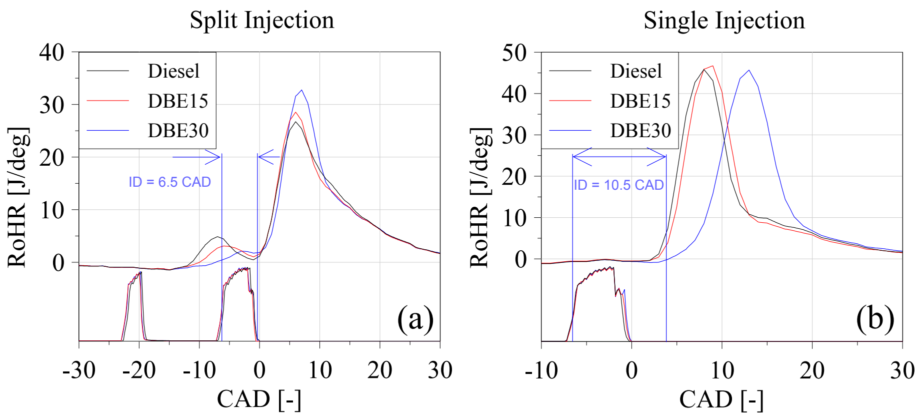


Figure 4.2: Heat release rate with ID - with the main injection as a reference - for DBE30 when utilizing a **(a)** double injection and **(b)** single injection [4].

The engine conditions from Figure 4.2 (a) and (b) are identical in all aspects except for the number of injection events. The transition between single and double injections is noticed mainly on the heat release rates premixed flame, which is much smaller for the double injection than for the single injection. This kind of change is observed due to the flame becoming more mixing controlled in the double injection strategy, with a larger diffusion flame tail. Note also, that the combustion from the pilot for the blends with higher alcohol content are also more delayed and have a lower absolute value.

4.1.3 Combustion Duration

The combustion event, CA_{10-90}, in this work defined according to Equation 4.2,

$$CA_{10-90} = CA90 - CA10 \tag{4.2}$$

will occur fast for methanol and ethanol. This is due to the higher flame speed of the light alcohols in comparison to the petroleum based fuels, given that the operating conditions are similar [5]. Moreover, when running the engine in the

PPC strategy, the fuel has more time for premixing, and therefore the charge will be, locally, more homogeneous. This also helps the flame to propagate faster through the combustion chamber and decreases the combustion duration [6, 7].

The general idea of the combustion duration is that it should be kept moderate, considering the operation condition. If the combustion duration is too long, there will be losses in form of heat transfer, due to the longer time frame in which the heat transfer can occur. Simultaneously, having a too long combustion duration increases the exhaust losses due to the reduced expansion possibility, limiting the potential efficiency. On the other hand, if the combustion duration is too short, heat transfer to the combustion chamber walls will not occur to the same extent, however, the PRR_{MAX} will increase significantly. A too high PRR_{MAX} can cause standing waves inside the cylinder, thus transferring heat more efficiently from the charge to the combustion chamber walls, which is eventually dissipated as a heat transfer loss to the engine coolant. Apart from the standing waves and high ringing intensity, a high PRR_{MAX} can be detrimental to the structural integrity of the internal components of the engine as well as causing the engine to become noisy. The combustion duration cannot be altered by one single variable, but is a product of many variables combined, making it a parameter hard to control. Like any other fuel, it is possible to control the combustion duration with the utilization of EGR. However, the light alcohols, and in particular methanol, are very sensitive to higher levels of EGR, as observed in Figure 4.3, below. This sensitivity to EGR has the drawback of reducing the in-cylinder temperature during combustion thus also impacting the COV_{IMEP_N} in a negative manner.

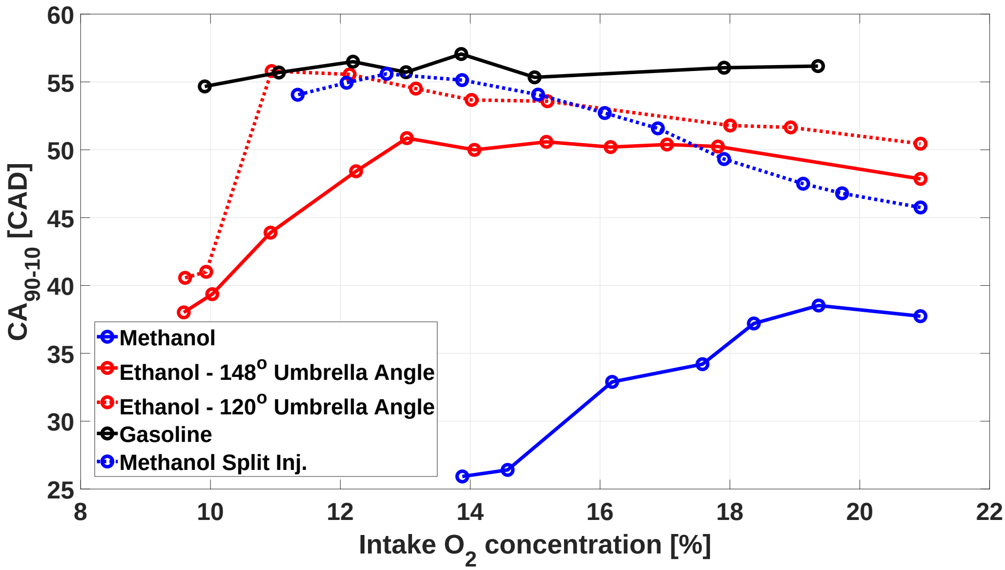


Figure 4.3: CA_{10-90} vs. intake oxygen concentration for methanol single injection, methanol split injection, ethanol single injection and 69 RON gasoline single injection. The data represents a constant FuelMEP as well as CA50 and is acquired from [8].

As can be seen from Figure 4.3, the combustion duration for regular gasoline PPC does not respond significantly to a increased EGR, or reduced intake oxygen concentration. The combustion duration fluctuates slightly around 56 CAD, which is probably due to the lower RON of this naphtha gasoline in contrast to the oxygenates. Methanol and ethanol single injection are, however, highly sensitive to EGR. To keep the CA50 constant, the SOI was advanced, which increases the premixing and therefore also reducing the combustion duration. The advancing of the injection timing is performed to avoid a retarded combustion phasing, which increases the COV_{IMEP_N} as will be discussed further in Subsection 4.1.4.

The combustion will occur faster in the case of an earlier double injection strategy as well. The effects of SOI, when utilizing this injection strategy, are not as significant as for the single injection strategy, due the combustion becoming more mixing controlled, as mentioned in Subsection 4.1.2. The ability to control the combustion phasing, and the combustion in general, is simpler when utilizing this strategy due to the shortened MP. The drawback, however, is the higher heat transfer losses to the combustion chamber walls, due to the prolonged combustion duration.

Ethanol was tested at the same operation conditions with two different injectors, which apart from flow, also differed in umbrella angle, where the injector with the higher umbrella angle had a higher flow due to two more holes as well as larger hole diameters in the nozzle. During the intake O_2 concentration sweep, observed in Figure 4.3, the common rail pressure was held constant at 1200 bar. The 120^o umbrella angle injector, having fewer and smaller hole diameters, injects the fuel at a slower rate which slows down the evaporation and increases the combustion duration [9].

4.1.4 Coefficient of Variation - $\mathbf{COV_{IMEP_N}}$

Since there is a turbulent flow occurring into and out of the cylinder, the combustion event is not identical every cycle, and there is cycle to cycle variation. The variation is calculated using Equation 4.3,

$$COV_{IMEP_N} = \frac{\sigma_{IMEP_N}}{\overline{IMEP_N}} \times 100 \tag{4.3}$$

where σ_{IMEP_N} is the standard deviation of $IMEP_N$, generally for 300 cycles, and $\overline{IMEP_N}$ is the mean of $IMEP_N$. The maximum limit of COV_{IMEP_N} is generally set to 5 %. For methanol and ethanol in CI engines this can be an issue depending on the operating conditions, and in particular when utilizing a single injection strategy. The parameters affecting the COV_{IMEP_N} when running gasoline PPC, will also affect the COV_{IMEP_N} of methanol and ethanol PPC operation. The difference is that methanol and ethanol combustion stability is much more sensitive than other fuels, due to the RON number, and in particular the charge cooling effect of these fuels. The charge cooling effect reduces the temperature and therefore increases the ID which results in a excessive enleanment and reduced level of fuel oxidation. These factors affects the COV_{IMEP_N} in a negative manner. Looking at Figure 4.4, below, it is possible to observe the COV_{IMEP_N} as a function of intake oxygen concentration, which is regulated by the EGR. Although gasoline is running with a lower intake temperature, the COV_{IMEP_N} is lower than that of methanol- and ethanol- split injection at high intake oxygen concentrations. In fact, with the methanol single injection, it was not possible to reach intake oxygen concentrations below $\sim$14 % due to the cooling effect on the combustion which comes with further addition of EGR. If a higher intake temperature would be utilized, the combustion would become stable enough to allow an increase in EGR. The double injection strategy, as expected, showed a lower COV_{IMEP_N} than the rest, since this mode of operation is generally stable.

Another factor also influencing the COV_{IMEP_N} is the combustion phasing. If the CA50 is set earlier, the COV_{IMEP_N} is lower. An early combustion phasing, however, tends to increase the PRR_{MAX} and might not be optimal in terms of

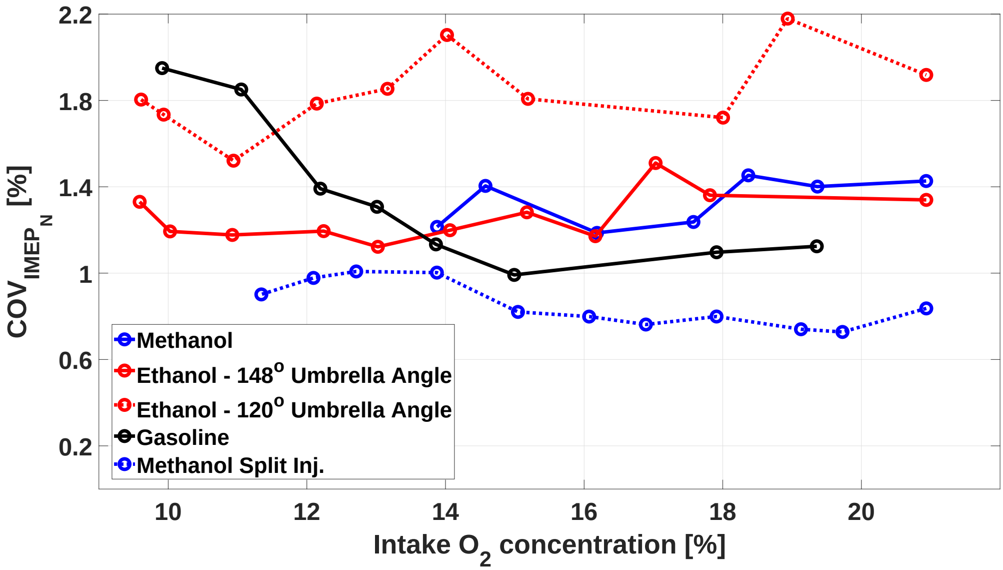


Figure 4.4: COV_{IMEP_N} as a function of intake O_2 concentration at 6 bar $IMEP_N$. The load for methanol split injection was set to 10 bar $IMEP_N$. Note that the intake temperature for gasoline PPC was set to 35 ^{o}C lower than the rest of the tested fuels and injection strategies.

η_B. A too retarded CA50, on the other hand, might also reduce the efficiency while increasing the COV_{IMEP_N}.

In the case where operation on methanol or ethanol results in a lower combustion duration, the COV_{IMEP_N} tends to be lower for these fuels. The combustion event becomes more predictable, and therefore also more stable, when the combustion duration is decreased.

4.1.5 EGR

The utilization of EGR, as earlier mentioned, has a significant importance in terms of NO_X control. When running a CI engine on diesel fuel or a low RON gasoline, the combustion event becomes less violent which results in a lower PRR_{MAX} while NO_X emissions are reduced. The effects of EGR on the combustion acts in mainly two ways by:

- maintaining a cooler combustion temperature.
- reducing the reaction speed due to an oxygen concentration decrease.

The cooler combustion temperature is maintained due to the higher c_p of exhaust gas, which can absorb more of the released heat than fresh air. Also, prior to the introduction into the intake, the exhaust gases are generally passed through a heat exchanger for cooling, thus, increasing the cooling effect of the combustion. The oxygen concentration is reduced to levels significantly below ambient when EGR is utilized thus decreasing the overall reactivity of the charge [10, 11, 12].

In the case of the light alcohols, and in particular methanol, the cooling effect of EGR utilization can introduce some difficulties. For any given load, methanol and ethanol cools down the charge about 8.9 and 2.4 times more than diesel fuel, respectively, for a given amount of chemical energy unit. Intuitively, this is good in terms of NO_X reduction, however, maintaining a consistently low COV_{IMEP_N} has proven to be difficult when such excessive charge cooling occurs. The charge cooling effect of the light alcohols does not reduce the NO_X formation by itself, since the combustion speed of these fuels is high, resulting in high PRR_{MAX} while a high intake temperature is required to keep the COV_{IMEP_N} low. A high intake temperature in its turn has a negative impact on the NO_X emissions. The negative impact of EGR on the combustion stability is reduced when a double injection strategy is utilized due to its lower temperature sensitivity, as can be observed from Figure 4.3 and 4.4.

4.2 Efficiency

4.2.1 Combustion Efficiency, η_C

The combustion efficiency, η_C, is a parameter that is closely related to the emissions; more specifically, the emissions that can oxidize. The ideal case would be if all the fuel reacts with the available O_2 and result in exhaust products consisting only of CO_2, H_2O and N_2. In reality the variety of species that exists in the resulting exhaust gases are vast, however, in the calculation of η_C, the chemical equilibrium of the combustion is simplified to Equation 3.11. Of these products, there are three emissions, listed in Table 4.1 below, that are taken into consideration when determining the η_C.

Table 4.1: Engine specifications

Species	Energy density [MJ/kg]
THC	Q_{LHV}
CO	10.1
H_2	120
Soot	32.8

From the species listed in Table 4.1, the η_C is mostly sensitive to THC and

CO, in that particular order. H_2 emissions are calculated from Equation 3.19 and contribute only to a minor part of the η_C, while the soot emissions are rarely taken into account [13]. An example of this can be observed in Figure 4.5, below.

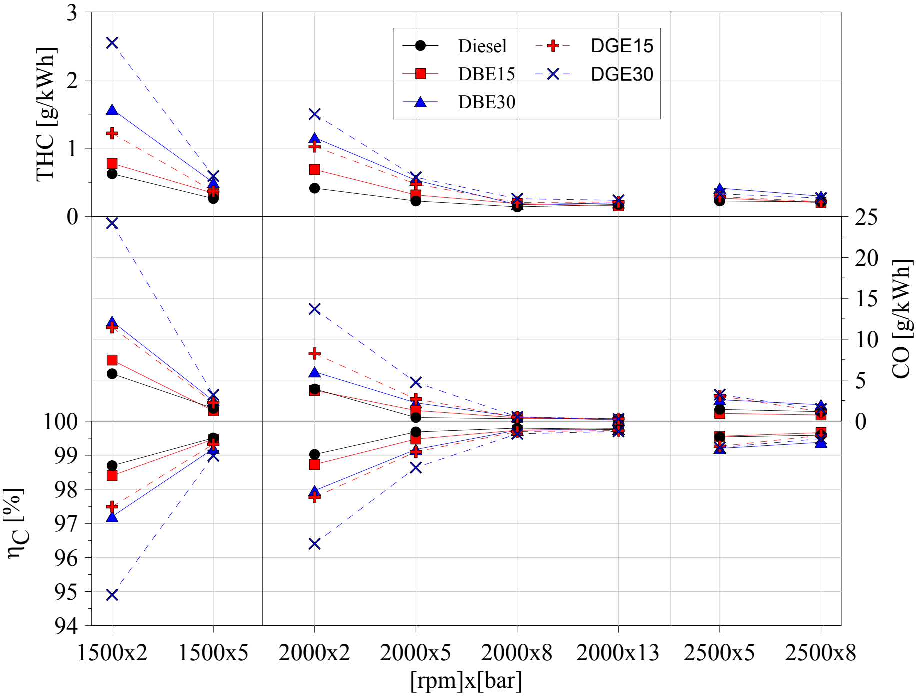


Figure 4.5: THC and CO emissions and η_C plotted as a function of Speed-Load in a 1.9 liter LD diesel engine [4, 14].

When calculating the η_C, all the gases are considered to be ideal. This makes it difficult to include the soot emissions in a accurate way, since soot is not even a gas. Since the η_C only depends on the emissions, the trends of this efficiency will be inversely proportional to the CO, and in particular THC emissions.

4.2.2 Thermal Efficiency, η_T

The thermal efficiency, η_T, consists of two part effieciencies: the heat transfer to the combustion chamber walls and the losses in form of hot exhaust gases. When the charge burns, a temperature gradient arises between the hot gases and the combustion chamber walls, which transfers the heat to the surrounding combustion chamber walls. This energy cannot be used in the expansion process and therefore inevitably contributes to a loss in η_T. The difficulty of igniting ethanol, and particularly methanol, due to their high charge cooling effect is definitely an advantage in terms of η_T. Prior to the autoignition, the fuel is vaporized and causes a reduction of the charge temperature which in turn reduces the energy required to compress the air-fuel mixture. The saving in compression work is, however, negligible since it only corresponds to a couple of J [15]. The reader should note that only PPC SOI were used in [15], and conclusions regarding charge cooling effects of HCCI SOI can therefore not be ruled out. Later injection timings, such as that of PPC, will have considerably less time to cool down the charge prior to the SOC, while HCCI injection timings - in theory at least - will result in a more readily evaporated fuel which has a significantly higher charge cooling effect.

The advantage of the cooling effect of methanol injection only presents itself during the combustion process, which occurs mainly after firing TDC. PPC, which consists of several instantaneous combustion modes, allows some of the fuel to evaporate and burn simultaneously. The evaporation of the fuel reduces the combustion temperature significantly, which in turn reduces the heat transfer to the combustion chamber walls, resulting in a higher η_T. In Figure 4.6, the reduced heat transfer for methanol can be observed in relation to iso-octane at the lower loads.

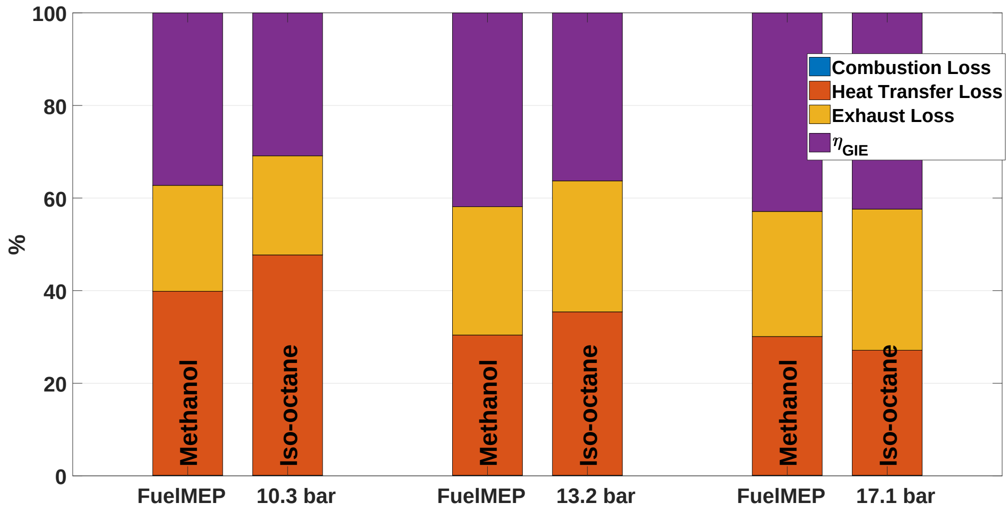


Figure 4.6: Energy balance for methanol in comparison to iso-octane [15].

As can be observed in Figure 4.6, experiments were conducted for methanol and iso-octane. Great care was taken when setting the operating conditions so that the intake conditions and injected fuel amount in terms of energy, FuelMEP, were set as similar as possible. With the combustion loss being negligibly low, η_C >99.7 %, it could be assumed that the energy released from the combustion of both fuels were very similar. It was noted that the firing pressure was higher, while the combustion duration was longer, for methanol than that of iso-octane when running the engine at similar FuelMEP. This suggests that a higher proportion of the energy released was utilized to exert pressure on the piston instead of being dissipated to the combustion chamber walls as heat transfer losses. This behavior was observed despite the fact that the methanol charge ignited at lower temperatures than that of iso-octane. This occurrence is in line with the result observed in 4.6. In Figure 4.7, the η_T can be seen plotted against various speed-load points for diesel fuel and other fuel blends containing ethanol. Two fuel blends were used in these experiments: diesel-biodiesel-ethanol, DBE, and diesel-gasoline-ethanol, DGE, where the biodiesel was SME. The volumetric ratio of diesel fuel to emulsifier, i.e. biodiesel and gasoline, was kept constant at 4:1 in all the blends, while the ethanol content was increased from 15 to 30 vol.%. The numbers after the fuel blends' abbreviations represent the ethanol content in vol.%.

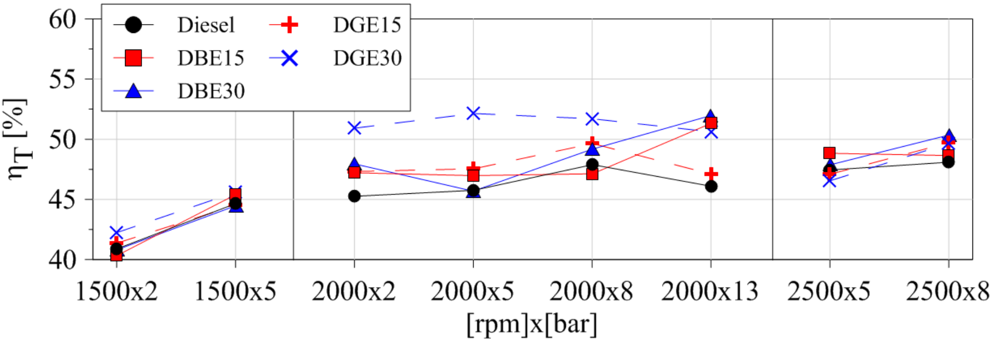


Figure 4.7: η_T as a function of Speed-Load in a 1.9 liter LD diesel engine.

As can be observed in Figure 4.7, the η_T for the oxygenated fuels is constantly higher than that of diesel fuel, except at two operation points. This suggests that the charge cooling potential that ethanol has is also significant. Given that all the operating parameters are kept constant, as far as possible, it is intuitive that the exhaust gas temperature will be reduced when a fuel has a higher heat of vaporization.

4.2.3 Indicated Efficiencies, η_{GIE} and η_{NIE}

In this work, the efficiency is represented mainly as η_{GIE}. Since the experimental setups only consisted of single cylinder engines, η_B is not an appropriate measure of efficiency due to the unknown losses in friction and other auxiliary components. η_{NIE}, which includes the pumping losses, was presented as a measure of efficiency in the LD engine, Publication IV and V observed in Section 6.4 and 6.5 respectively. The η_{NIE} was used in the LD engine since the pumping loops for the original engine were supplied by the OEM, and therefore it could be repeated in the experiments to obtain realistic pumping losses. In the HD engine, the pumping loops for the engine were not given, thus the efficiency only up to η_{GIE} is presented for this engine. In Figure 4.6, Section 4.2.2, a comparison between iso-octane and methanol in terms of $_GIE$ can be observed while Figure 4.8 below shows the η_{NIE} for diesel as well as the tested oxygenated blends DBE and DGE.

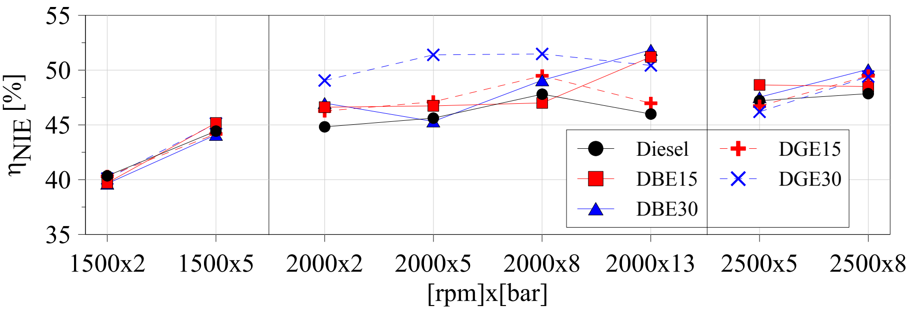


Figure 4.8: η_{NIE} as a function of Speed-Load in a 1.9 liter LD diesel engine.

Ultimately, the η_{GIE} represent the energy that is used to exert pressure on the piston before the η_C, η_T and pumping losses are counted in. This measure therefore depends heavily on the η_T, due to the η_C generally being slightly less than 1. Therefore, the trends of η_{GIE} as well as η_{NIE} will follow that of η_T.

4.3 Regulated Gaseous Emissions

4.3.1 CO

The emissions of CO are generally affected by two factors: Φ and combustion temperature [16, 17]. When the fuel is oxidized, the carbon reacts with the oxygen to form CO, which eventually reacts further and finally forms CO_2, given that the combustion temperature is high enough and oxygen supply is adequate. In CDC, the high temperature and abundance of oxygen results in a very low concentration of CO in the exhaust gas and, therefore, this regulated emission is considered to be unimportant in CI engines.

The oxygen atom in methanol and ethanol fuel decreases the overall Φ of the charge which results in more complete combustion. Nevertheless, it should be noted that a too lean charge can freeze the reactions at CO, due to combustion instability and fuel not being oxidized properly due to fuel impingement on the combustion chamber walls, which can be the case when running a CI engine on light alcohols [18]. As expected, this issue originates in the high RON number of these fuels in combination with the high heat of vaporization. In Figures 4.9 and 4.10, the CO emissions for two engine configurations and different fuels and fuel blends can be observed.

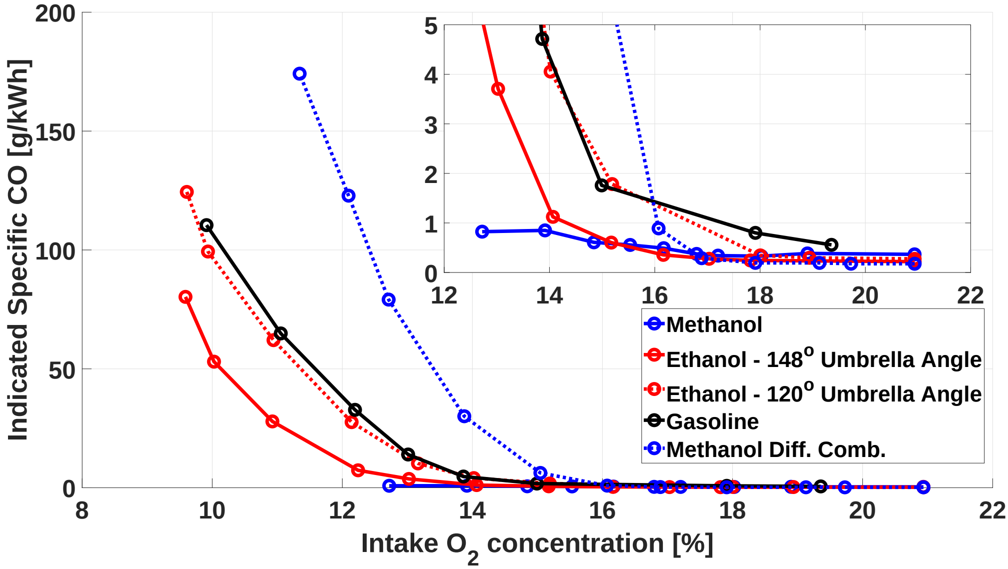


Figure 4.9: Indicated Specific CO emissions as a function of intake O_2 concentration. Methanol diffusion combustion was running at 10 bar $IMEP_G$, while the rest of the fuels and configurations were running at 6 bar $IMEP_G$.

From Figure 4.9 it can be seen that the CO emission from methanol and ethanol are lower than that of gasoline, given that the operating conditions are constant. However, it has to be noted that gasoline was running with a lower intake temperature than the light alcohols. The higher level of oxidation of CO to CO_2 seem to be highly dependent on the oxygen content of the fuel: higher fuel oxygen content results in lower CO emission.

There are, however, cases, in which this does not apply. Observing Figure 4.10, it can be seen that at lower loads, the CO emissions from the oxygenated fuels are higher than those of diesel fuel. In this case, the longer ID occurring at lower loads as well as the EGR utilization at all loads below 13 bar BMEP, result in a higher fraction of the fuel hitting the combustion chamber walls prior to combustion. Yu et al. [19] performed a numerical investigation, showing that the majority of the emitted CO is formed in the region in which the fuel impinges on the combustion chamber walls, which is more evident for the DGE blends as they have a lower CN rating. The study also showed that altering the umbrella angle of the injector could help reduce the CO emissions, as confirmed by Figure 4.9.

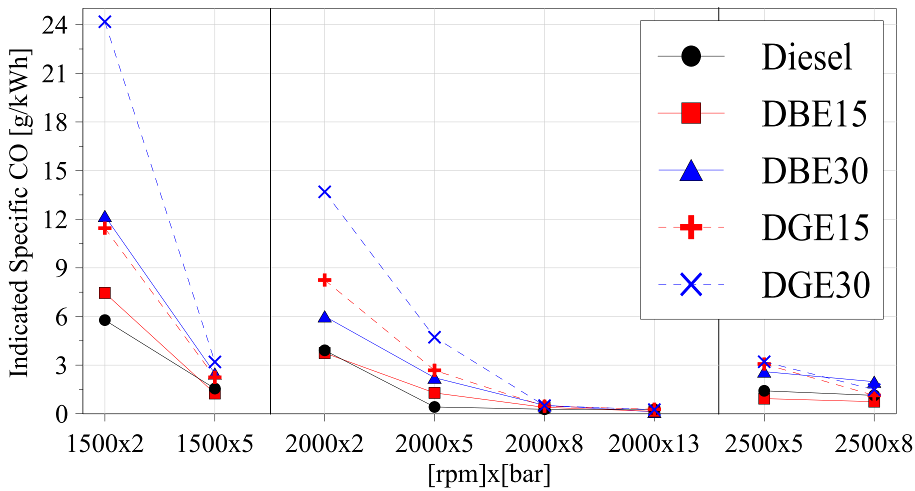


Figure 4.10: Indicated Specific CO emissions as a function of Speed-Load in a 1.9 liter light duty diesel engine.

When the load and speed are increased, the CO emission of the DBE and DGE fuel blends merges with the diesel fuel CO emission. This is because, the higher load and speed increases the in-cylinder temperature, which not only decreases the fuel impingement due to a shorter ID, but also makes sure to oxidize all the formed CO into CO_2. Moreover, Sjöberg et al. [20] suggested that the low engine speed can be an obstacle for the conversion of CO into CO_2, due to the expansion stroke cooling down the combustion temperature - below the required 1500 K - and to a certain extent, halts the conversion process.

4.3.2 THC

The emissions of THC's generally follow a trend similar to those of CO emissions, however with significantly lower amounts emitted. As for the case of the CI engine CO emission, THC are also considered to be less important than

NO_X and soot. Nevertheless, when the combustion temperature drops, fuel is not readily oxidized and THC emissions increase. A similar effect can also be noticed when combustion instability is increased [4, 14, 6]. This can be seen in Figures 4.11 and 4.12, below.

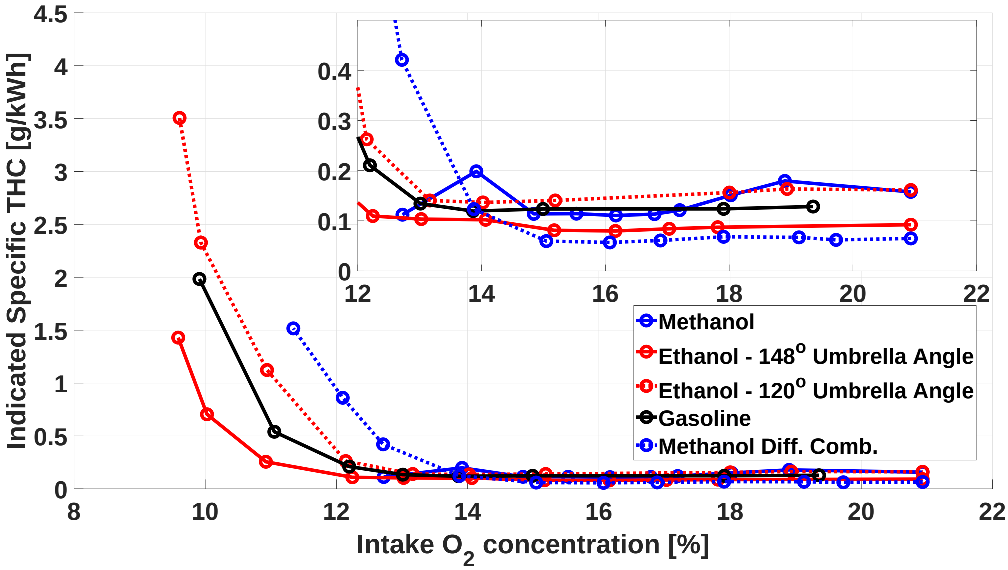


Figure 4.11: Indicated Specific THC emissions as a function of intake O_2 concentration. Methanol diffusion combustion was running at 10 bar $IMEP_G$, while the rest of the fuels and configurations were running at 6 bar $IMEP_G$.

The in-cylinder temperature and O_2 availability are important factors to control the THC emissions, however, they are not the only factors playing a role. For CI engines, there are two major causes of THC emissions:

- overleaning.
- undermixing.

Overleaning occurs when the local Φ decreases below the lean combustion limit at $\Phi_L \sim$0.3. This happens at the site where the fuel spray boundary is located. The mixture in these regions can only oxidize slowly and will therefore be incomplete. Meanwhile, undermixing normally occurs when the fuel is mixed slowly with the air. The sac volume of the injector is an important source of this mechanism [17].

For methanol, and most probably also for ethanol, overleaning can become an important source of THC emissions due to the in-cylinder Φ distribution. Pucilowski et al. [21] and Svensson et al. [1] both showed that, in contrast

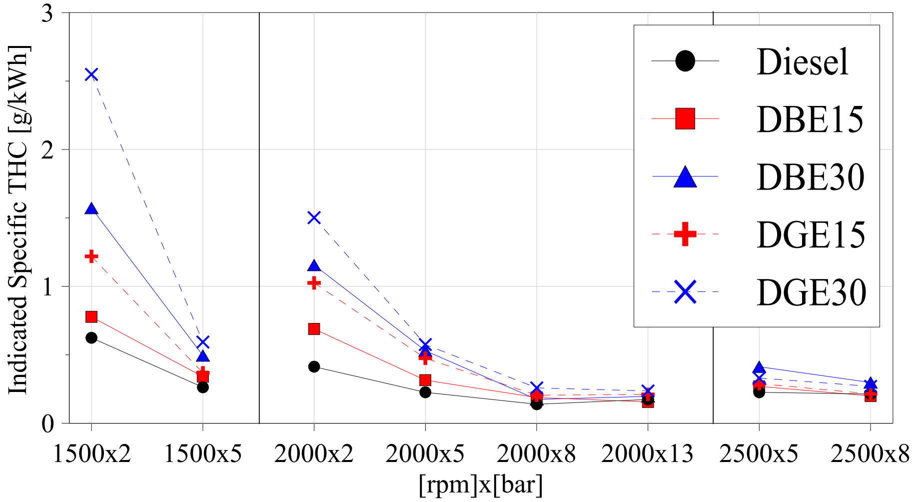


Figure 4.12: Indicated Specific THC emissions as a function of Speed-Load in a 1.9 liter light duty diesel engine.

to diesel fuel combustion, the ignition of methanol is initiated in the fuel lean regions of the combustion chamber. The evaporation of methanol and ethanol fuel requires a considerable amount of energy that is drawn from the surrounding gas. Therefore these fuels will mix with air for a longer time period - in which they evaporate - and simultaneously lean out as the temperature decreases. This could be a potential source of the THC emissions from methanol, and most probably ethanol, combustion in a CI engine [22].

When running a CI engine on a high RON fuel, or an oxygenated fuel in particular, there are other mechanisms in action. For example, the important THC source of SI engines, wall quenching, becomes a significant source. The cause is the long ID which gives the fuel enough time to hit the combustion chamber walls and stops it from ever reaching the oxidation process. Instead, the unburned fuel vaporizes and leaves the cylinder as THC during the exhaust stroke. This is clearly noticed in Publication II, Section 6.2, when utilizing a high r_c [23]. The fuel spray impinged on the shallow piston and caused higher THC emissions, which increased with injection pressure. This is illustrated by Figure 4.13, below. As the reader might note, the THC emissions are lower for the operating condition utilizing a higher amount of EGR. This initial reduction of THC with a higher level of EGR was noted already in Publication I, Subsection 6.1 [8]. Although this occurrence has not passed the observation stage, it can be explained by the fact that the exhaust gases are included in the combustion to be oxidized again.

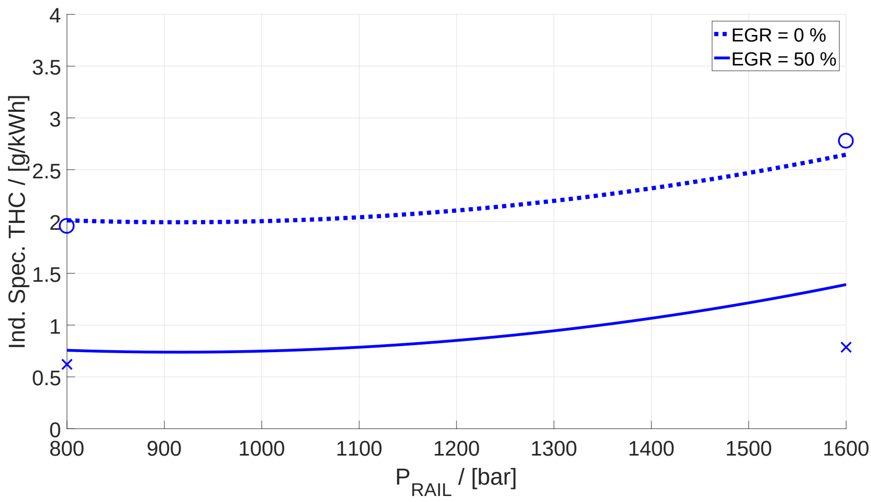


Figure 4.13: Indicated Specific THC emissions as a function of injection pressure in the Scania D13 HD engine at 6 bar $IMEP_G$.

Wall-quenching is also related by the fact that a combustible air-fuel mixture will not burn close to surfaces due to the relation between heat release within a flame and heat loss to the wall. A higher combustion temperature will reduce the wall-quenching distance between the fuel and wall, which in turn will result in reduced THC emissions [24, 25]. For regular diesel fuel, this is not an issue, since diesel fuel has an overall short ID, even when intake temperature is low or when a high level of EGR is utilized. In contrast to diesel fuel, for methanol and ethanol the intake temperature, and eventually, the combustion temperature is important. It needs to be adequately high to get the combustion started and also high enough to keep an ID which does not cause excessive fuel impingement as well as maintaining a small wall-quenching distance to reduce the THC emissions effectively.

4.3.3 NO_X

The NO_X emissions from the CI engine is one of the major drawbacks of this engine type. NO_X emissions are formed according to the extended Zeldovich mechanism, which is a function of temperature, oxygen availability and time. When there is an abundance of available air and temperature is high, NO_X formation will occur and concentration will increase exponentially with time

[17].

As earlier mentioned, the parameters affecting the NO_X emissions from the CDC strategy are quite well studied. Studies conducted with methanol and ethanol in CI engines have shown some varied results. There are mainly two effects that counter each other in this case. When running the engine on alcohols, or fuel blends containing alcohols, the premixing increases alongside the charge cooling. When premixing increases, the charge undergoes a faster combustion which results in higher peak temperatures, given a similar combustion phasing. This by itself causes the NO_X emissions to increase. On the other hand, it is counteracted by the charge cooling effect. This can be seen clearly in Figure 4.14.

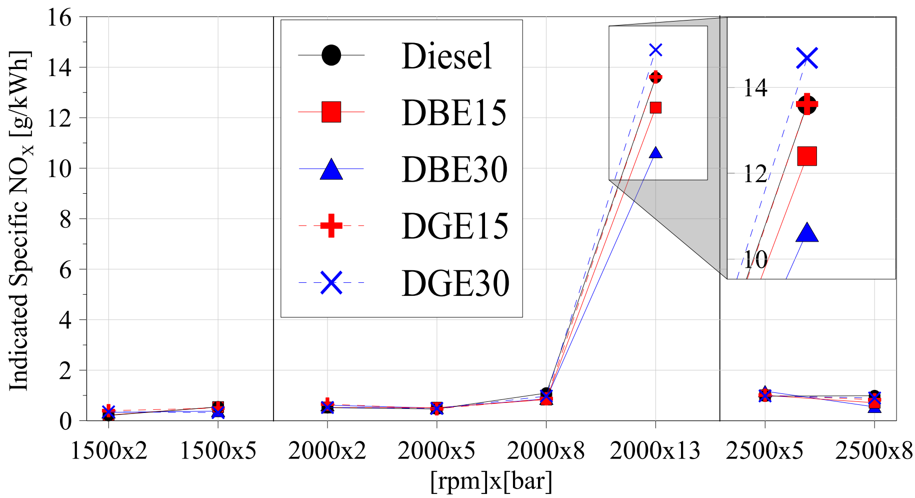


Figure 4.14: Indicated Specific NO_X emissions as a function of Speed-Load in a 1.9 liter light duty diesel engine.

At 13 bar BMEP, where the EGR utilization ceases, a significant rise in NO_X emissions can be observed. The fuels using gasoline as an emulsifier, DGE, form similar or higher levels of NO_X than those of diesel fuel. There is a cooling effect reducing the NO_X, however, the premixing increases the combustion speed enough to increase the NO_X to level higher than those of diesel fuel. In the case of the DBE, where biodiesel acts as an emulsifier, the NO_X emissions are below those of diesel fuel due to less premixing.

The NO_X emissions were also measured in Publication VI, Section 6.6, when using similar FuelMEP, intake temperatures, pressures and combustion phasings [15]. The results can be observed below, in Figure 4.15.

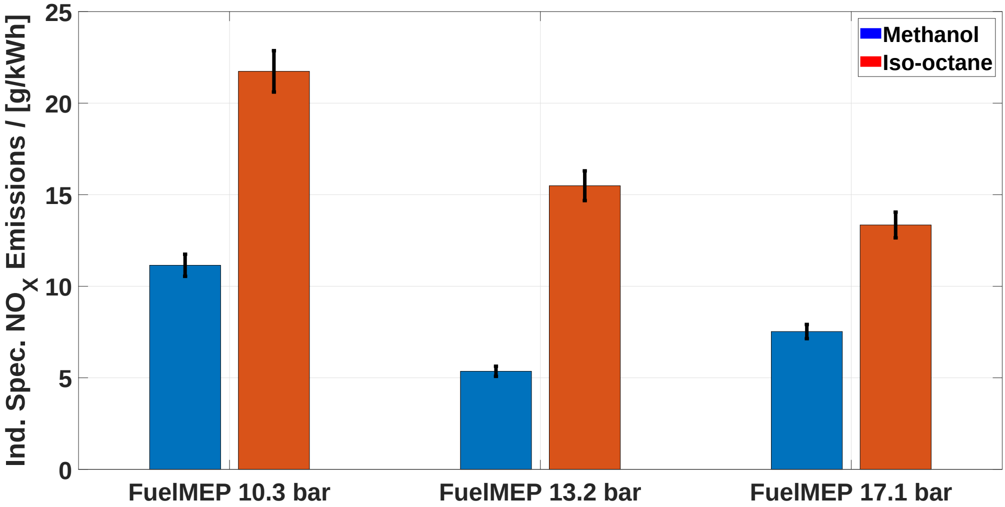


Figure 4.15: NO_X emissions at similar intake conditions comparing methanol and iso-octane at 10.3 bar FuelMEP, 13.2 bar FuelMEP and 17.1 bar FuelMEP.

From Figure 4.15, it can be seen that the same amount of heat release inside the combustion chamber at identical conditions generates more NO_X when iso-octane is combusted instead of methanol. In these operating conditions the charge cooling temperature was isolated where possible, and the results take form of significantly reduced NO_X emissions for methanol. At 10.3 bar FuelMEP, the combustion consisted only of a premixed mode heat release, while at 13.2 and 17.1 bar FuelMEP, there were an apparent diffusion tail for both iso-octane and methanol. This suggests that the cooling effect does not only occur during the fuel evaporation process, but also during the diffusion combustion. Matter of fact, the results obtained in Publication VI suggest that the majority of the cooling effect actually occurs during the combustion process.

4.4 Particulate Matter Emissions

In the PPC strategy, despite running with gasoline, the PM emissions could be significant at the operation point which utilizes a significant amount of EGR. The classical PPC "recipe" consists of keeping λ=1.5 and EGR=50 %. This reduces the NO_X while keeping the heat transfer losses relatively low. However, the production of soot emissions will not be negligible and could at high loads reach up to FSN=5 [26]. This is because gasoline and diesel fuel contain aromatics, which have a high propensity of taking the chemical reaction path of

PAH's, which eventually form soot particles [27].

It is well known that the PM emissions from engines running on methanol and ethanol do not emit PM emissions in the same way as regular gasoline or diesel fuel. This is because the chemical structure of these fuels, that makes it very hard for them to form the PAH precursors [28]. In Publication I, Section 6.1, the particulate matter emissions for methanol and ethanol were investigated, using a electrical mobility particle size distribution measurement system mentioned in more detail in Section 3.1.3.2.

Intake temperature, injection pressure and most importantly, the intake oxygen concentration was swept from ambient down to as low as possible for gasoline, ethanol and methanol. As expected, gasoline formed significant amounts of soot at high intake temperatures, low injection pressures and high EGR levels. Soot, in this case, is referring to black carbon soot, which was simultaneously quantified by an AVL MSS, also explained in more detail in Section 3.1.3.2. In Figures 4.16 and 4.17, the PM emissions for gasoline can be observed while sweeping the intake O_2 concentration.

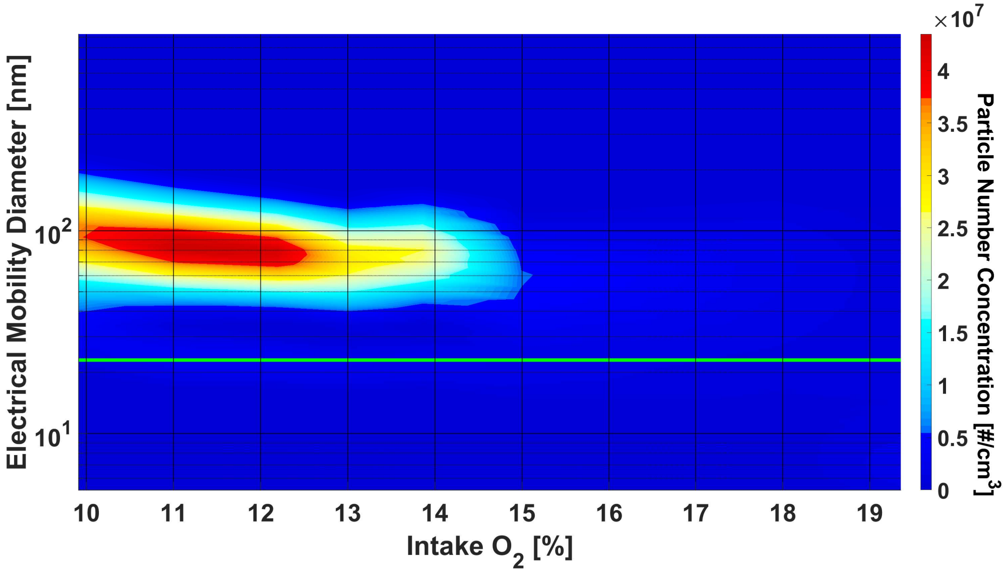


Figure 4.16: Particle number and size distribution vs. intake O_2 concentration for gasoline single injection at 6 bar $IMEP_G$.

When the soot emissions increased for gasoline, as seen in Figure 4.17, a evident response could be observed from the measurement of the DMS500, Figure 4.16, in the form of higher PN as well as larger diameter sizes. This trend was observed for the above mentioned injection pressure and intake temperature sweeps.

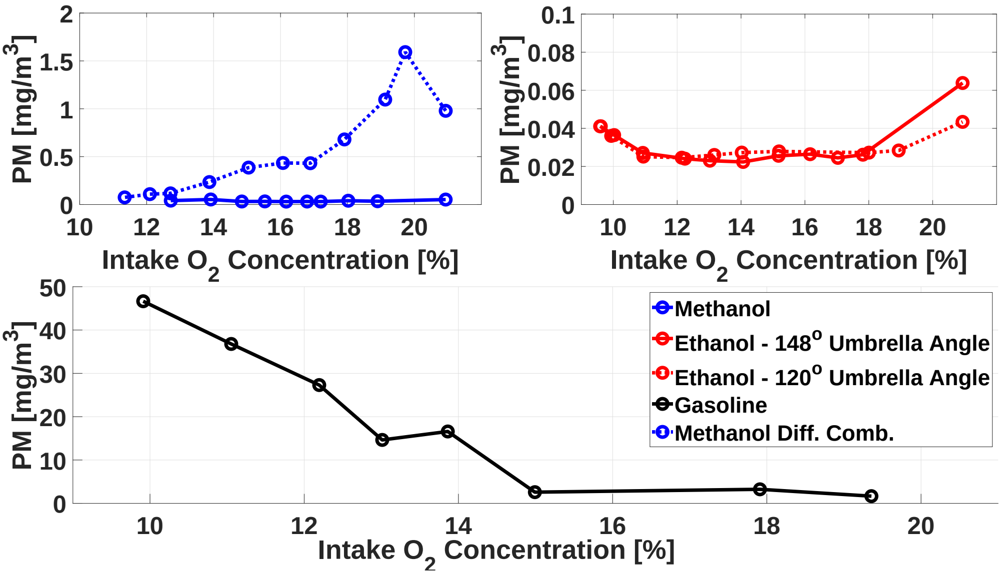


Figure 4.17: Soot mass concentration for the various fuels vs. intake O_2 concentration measured using the AVL MSS.

In the case of the oxygenated fuels, the soot emissions did not exceed 1.6 mg/m^3 for all the tested operating conditions. In the vast majority of the operating conditions, the soot emissions did not exceed 0.05 mg/m^3. The interesting result of this study was the particle size distribution curves for the oxygenated alcohols. Figure 4.18, below, illustrates the particle size distribution curves for different intake O_2 concentrations when running the engine on methanol diffusion combustion.

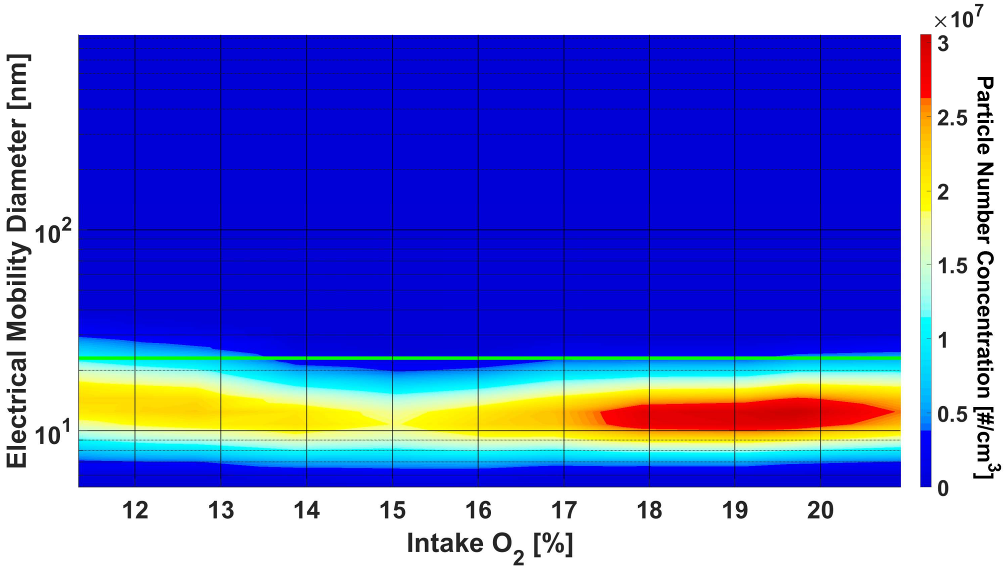


Figure 4.18: Particle number and size distribution vs. intake O_2 concentration for methanol diffusion combustion at 10 bar $IMEP_G$.

Initially, the authors hypothesized that the large particle number emitted from ethanol combustion, and in particular methanol combustion, was an effect of the corrosivity of these fuels. The hypothesis was made on the basis of a study conducted by Mayer et al. [29].

However, further studies found that the corrosivity of the oxygenated fuels was not necessarily the factor in effect in this case. In Publication III, Section 6.3, the PM emissions were measured again for three fuels at two loads: methanol, ethanol and diesel fuel at 6 bar and 10 bar $IMEP_G$. The operating conditions did not utilize EGR, but were running with ambient intake O_2 concentration. Measurements were conducted again with the AVL MSS and the Cambustion DMS500 M177, however, this time exhaust gas was sampled on TEM grids. The results of the TEM analysis are shown in Figure 4.19, below.

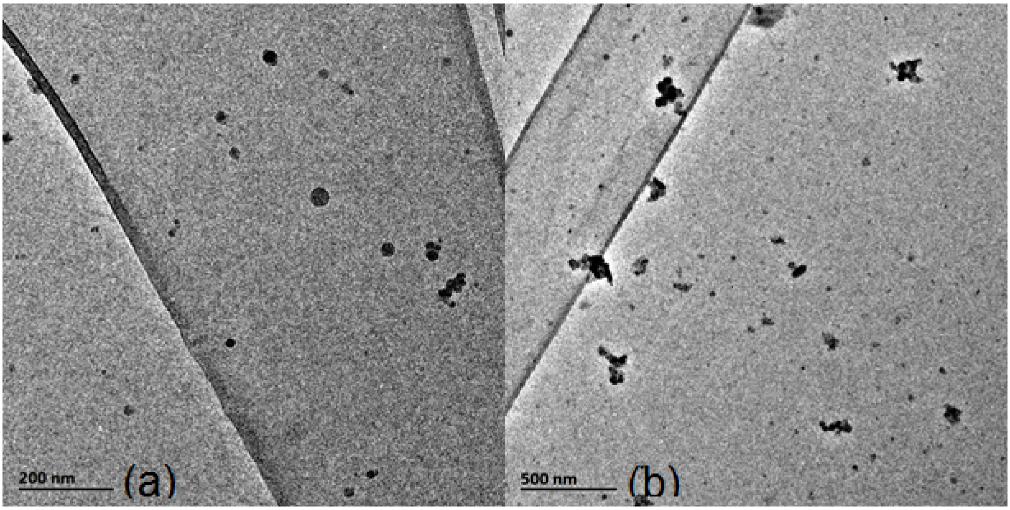


Figure 4.19: TEM results for **(a)** ethanol and **(b)** diesel [30].

The difference between the light alcohols and diesel fuel particulates is significant. The particles emitted from diesel fuel combustion are significantly larger than those of methanol and ethanol (note the scale). As earlier mentioned, three fuels were tested in Publication III. However, no trends were revealed between methanol and ethanol TEM grids during the analysis. They were similar to the extent that it was not possible to distinguish between them. An energy dispersive X-ray analysis was conducted on the TEM grids to determine the elemental constituents and the abundance of the particles. Four elements were found in all the particles smaller than 50 nm; Zinc, Calcium, Sulfur and Phosphor. The abundance varied significantly from one particle to another, however, never descended below 5 mass% in any of them. Chemical analyses of the lubrication oil and fuels showed no content at all of the four above mentioned elements in either methanol or ethanol. The lubrication oil, on the other hand, contained a considerable amount of Zn, Ca, S and P, which suggests that the origin of the nanoparticles and the majority of the nucleation mode particles originates from the lubrication oil rather than from the combustion of the light alcohols [30].

A way to reduce the combustion products of lubrication oil is to reduce the the thickness of the lubricating layer on the cylinder walls. Although difficult, this can be achieved by using scraper rings which control the lubrication oil flow to the top land, thereby limiting the amount of lubrication oil involved in the combustion process [31]. Moreover, a lubrication oil with a lower content of sulfur and ash could be used to reduce the engine out PN [32, 33].

Bibliography

[1] E. Svensson, C. Li, S. Shamun, B. Johansson, M. Tunér, C. Perlman, H. Lehtiniemi, and F. Mauss. Potential Levels of Soot, NO_x, HC and CO for Methanol Combustion. 2016, DOI:10.4271/2016-01-0887.

[2] L. M. Rodríguez-Antón, F. Gutíerrez-Martín, and Y. Doce. Physical properties of gasoline, isobutanol and ETBE binary blends in comparison with gasoline ethanol blends. *Fuel*, 166:73–78, 2016, DOI:https://doi.org/10.1016/j.fuel.2015.10.106.

[3] N. Yilmaz and T. M. Sanchez. Analysis of operating a diesel engine on biodiesel-ethanol and biodiesel-methanol blends. *Energy*, 46(1):126–129, 2012, DOI:https://doi.org/10.1016/j.energy.2011.11.062.

[4] S. Shamun, G. Belgiorno, G. D. Blasio, C. Beatrice, M. Tunér, and P. Tunestål. Performance and emissions of diesel-biodiesel-ethanol blends in a light duty compression ignition engine. *Applied Thermal Engineering*, 145:444 – 452, 2018, DOI:https://doi.org/10.1016/j.applthermaleng.2018.09.067.

[5] Q. Li, W. Jin, and Z. Huang. Laminar Flame Characteristics of C1-C5 Primary Alcohol-Isooctane Blends at Elevated Temperature. *Energies*, 9(7), 2016, DOI:10.3390/en9070511.

[6] C. Li, L. Yin, S. Shamun, M. Tunér, B. Johansson, R. Solsjö, and X. S. Bai. Transition from HCCI to PPC: the Sensitivity of Combustion Phasing to the Intake Temperature and the Injection Timing with and without EGR. *SAE Technical Paper 2016-01-0767*, 2016, DOI:https://doi.org/10.4271/2016-01-0767.

[7] T. Ganapathy, R. P. Gakkhar, and K. Murugesan. Influence of injection timing on performance, combustion and emission characteristics of Jatropha biodiesel engine. *Applied Energy*, 88(12):4376–4386, 2011, DOI:https://doi.org/10.1016/j.apenergy.2011.05.016.

[8] S. Shamun, M. Shen, B. Johansson, M. Tunér, J. Pagels, A. Gudmundsson, and P. Tunestål. Exhaust PM Emissions Analysis of Alcohol Fueled Heavy-Duty Engine Utilizing PPC. *SAE International Journal of Engines*, 9(4):2142–2152, 2016, DOI:https://doi.org/10.4271/2016-01-2288.

[9] P. Bergstrand and I. Denbratt. Diesel Combustion with Reduced Nozzle Orifice Diameter. *SAE Technical Paper 2001-01-2010*, 2001, DOI:https://doi.org/10.4271/2001-01-2010.

[10] M. Christensen and B. Johansson. Influence of Mixture Quality on Homogeneous Charge Compression Ignition. *SAE Technical Paper 982454*, 1998, DOI:https://doi.org/10.4271/982454.

[11] M. Christensen and B. Johansson. Supercharged Homogeneous Charge Compression Ignition (HCCI) with Exhaust Gas Recirculation and Pilot Fuel. *SAE Technical Paper 2000-01-1835*, 2000, DOI:https://doi.org/10.4271/2000-01-1835.

[12] M. Yao, B. Zhang, Z. Zheng, Z. Cheng, and Y. Xing. Experimental Study on the Effects of EGR and Octane Number of PRF Fuel on Combustion and Emission Characteristics of HCCI Engines. *SAE Technical Paper 2005-01-0174*, 2005, DOI:https://doi.org/10.4271/2005-01-0174.

[13] S. Shamun. *Evaluation of Ethanol-Gasoline Blends in SI Engine*. Division of Combustion Engines, Lund University, 2014.

[14] G. Belgiorno, G. D. Blasio, S. Shamun, C. Beatrice, P. Tunestål, and M. Tunér. Performance and emissions of diesel-gasoline-ethanol blends in a light duty compression ignition engine. *Fuel*, 217:78–90, 2018, DOI:https://doi.org/10.1016/j.fuel.2017.12.090.

[15] S. Shamun, B. Zincir, P. Shukla, P. G. Valladolid, S. Verhelst, and M. Tunér. Quantification and analysis of the charge cooling effect of methanol in a compression ignition engine utilizing ppc strategy. *ASME, Internal Combustion Engine Division Fall Technical Conference, ICEF2018-9657*, 2018, DOI:https://doi.org/10.1115/ICEF2018-9657.

[16] R. Opat, Y. Ra, M. A. Gonzalez D., R. Krieger, R. D. Reitz, D. E. Foster, R. P. Durrett, and R. M. Siewert. Investigation of Mixing and Temperature Effects on HC/CO Emissions for Highly Dilute Low Temperature Combustion in a Light Duty Diesel Engine. *SAE Technical Paper 2007-01-0193*, 2007, DOI:https://doi.org/10.4271/2007-01-0193.

[17] J. B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988, ISBN:0-07-028637-X.

[18] M. Abu-Qudais, O. Haddad, and M. Qudaisat. The effect of alcohol fumigation on diesel engine performance and emissions. *Energy Conversion and Management*, 41(4):389 – 399, 2000, DOI:https://doi.org/10.1016/S0196-8904(99)00099-0.

[19] H. Yu, Y. Guo, D. Li, X. Liang, G.-Q. Shu, Y. Wang, X. Wang, and L. Dong. Numerical Investigation of the Effect of Spray Cone Angle on Mixture Formation and CO/Soot Emissions in an Early Injection HCCI Diesel Engine. *SAE Technical Paper 2015-01-1070*, 2015, DOI:https://doi.org/10.4271/2015-01-1070.

[20] M. Sjöberg and J. E. Dec. Combined Effects of Fuel-Type and Engine Speed on Intake Temperature Requirements and Completeness of Bulk-Gas Reactions for HCCI Combustion. *SAE Technical Paper 2003-01-3173*, 2003, DOI:https://doi.org/10.4271/2003-01-3173.

[21] M. Pucilowski, M. Jangi, S. Shamun, C. Li, M. Tunér, and X.-S. Bai. Effect of Start of Injection on the Combustion Characteristics in a Heavy-Duty DICI Engine Running on Methanol. *SAE Technical Paper 2017-01-0560*, 2017, DOI:https://doi.org/10.4271/2017-01-0560.

[22] D. C. Rakopoulos, C. D. Rakopoulos, E. G. Giakoumis, R. G. Papagiannakis, and D. C. Kyritsis. Influence of properties of various common bio-fuels on the combustion and emission characteristics of high-speed DI (direct injection) diesel engine: Vegetable oil, bio-diesel, ethanol, n-butanol, diethyl ether. *Energy*, 73:354–366, 2014, DOI:https://doi.org/10.1016/j.energy.2014.06.032.

[23] S. Shamun, C. Haşimoğlu, A. Murcak, and ˙ Experimental investigation of methanol compression ignition in a high compression ratio HD engine using a Box-Behnken design. *Fuel*, 209:624–633, 2017, DOI:https://doi.org/10.1016/j.fuel.2017.08.039.

[24] H. Kosaka, F. Zentgraf, A. Scholtissek, L. Bischoff, and T. H˙ Wall heat fluxes and CO formation/oxidation during laminar and turbulent side-wall quenching of methane and DME flames. *International Journal of Heat and Fluid Flow*, 70:181–192, 2018, DOI:https://doi.org/10.1016/j.ijheatfluidflow.2018.01.009.

[25] O. Ezekoye, R. Greif, and R. Sawyer. Increased surface temperature effects on wall heat transfer during unsteady flame quenching. *Symposium (International) on Combustion*, 24(1):1465–1472, 1992, DOI:https://doi.org/10.1016/S0082-0784(06)80171-2.

[26] V. Manente. *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*. PhD thesis, Lund University, 2010.

[27] M. M. Maricq. Chemical characterization of particulate emissions from diesel engines: A review. *Journal of Aerosol Science*, 38(11):1079–1118, 2007, DOI:https://doi.org/10.1016/j.jaerosci.2007.08.001.

[28] J. Wu, K. H. Song, T. Litzinger, S.-Y. Lee, R. Santoro, M. Linevsky, M. Colket, and D. Liscinsky. Reduction of PAH and soot in premixed ethylene-air flames by addition of ethanol. *Combustion and Flame*, 144(4):675–687, 2006, DOI:https://doi.org/10.1016/j.combustflame.2005.08.036.

[29] A. C. Mayer, A. Ulrich, J. Czerwinski, and J. J. Mooney. Metal-Oxide Particles in Combustion Engine Exhaust. *SAE Technical Paper 2010-01-0792*, 2010, DOI:https://doi.org/10.4271/2010-01-0792.

[30] S. Shamun, M. Novakovic, V. B. Malmborg, C. Preger, M. Shen, M. E. Messing, J. Pagels, M. Tunér, and P. Tunestål. Detailed Characterization of Particulate Matter in Alcohol Exhaust Emissions. *The international*

symposium on diagnostics and modeling of combustion in internal combustion engines, 2017.9:B304, 2017, DOI:10.1299/jmsesdm.2017.9.B304.

[31] B. Thirouard and T. Tian. Oil Transport in the Piston Ring Pack (Part II): Zone Analysis and Macro Oil Transport Model. 2003, DOI:https://doi.org/10.4271/2003-01-1953.

[32] P.-q. Tan and D.-y. Wang. Effects of Sulfur Content and Ash Content in Lubricating Oil on the Aggregate Morphology and Nanostructure of Diesel Particulate Matter. *Energy & Fuels*, 32(1):713–724, 2018, DOI:10.1021/acs.energyfuels.7b03017.

[33] S. Whitacre and W. Van Dam. Enhanced Fuel Economy Retention from an Ultra-Low Ash Heavy Duty Engine Oil. 2019.

Chapter 5

Conclusions

Methanol and ethanol have a couple of interesting properties which contribute to their advantages as well as disadvantages. However, according to the author's obtained knowledge through research experience, the main attribute which distinguishes the light alcohols from diesel fuel, or gasoline in the case of PPC, is the charge cooling effect. The high heat of vaporization and the low $(A/F)_S$ of ethanol, and in particular methanol, affect every aspect of the combustion process; starting from the injection event, ignition, combustion and finally the emission formation process.

From this work it can be concluded that methanol and ethanol use, both neat and in blends, in compression ignition engines is a fully viable option to reduce the net well-to-wheel CO_2 emissions while increasing the efficiency. There are, however, some obstacles.

The use of methanol and ethanol have shown a couple of advantages over diesel fuel. The first and foremost, the close to nonexistent particulate matter emissions from these fuels in comparison to diesel fuel. The results in this work strongly suggests that the particle number emitted from the combustion of the light alcohols originates from the nano particles contained in the lubrication oil rather than the combustion of the fuels themselves.

Due to the higher heat of vaporization of methanol and ethanol and their laminar flame speeds, the combustion occurs at a lower temperature and at a higher rate, which reduces the heat transfer losses significantly. These are the most significant factors behind the increased thermal- and gross indicated efficiencies.

Another effect of the cooler combustion - in comparison with diesel fuel or other high research octane number fuel, such as gasoline or even iso-octane - is the reduced NO_X formation. From publication VI, Section 6.6, the NO_X emissions of methanol were drastically reduced in comparison to iso-octane given similar intake conditions. The increased intake temperature, however, offsets this reduction to a certain extent since a higher intake temperature increases the initial temperature of the charge.

CO and THC emissions are higher when running a CI engine on methanol and ethanol at low load, due to a combination of longer MP - causing a higher degree of bowl wetting and crevice losses - and cooler combustion. These emissions can be reduced to the level of regular diesel fuel combustion by increasing the intake temperature, which leads to a higher level of oxidation and a more complete combustion.

Diesel-Ethanol Blends

Oxygenated blends can be used in diesel engines, with a significant amount - 30 vol.% - of ethanol. If the necessary emulsifier consists of high octane fuel, such as gasoline, the CO, THC and NO_X emissions will be higher than if a high cetane number fuel would be used as an emulsifier. The ignition delay is significantly increased when the overall cetane number of the fuel is decreased, which results in a higher level of bowl wetting and crevice losses. In light duty engines, this issue is compounded due to the small combustion chamber size.

The higher proportion of premixing can also lead to a higher combustion temperature, which is maintained high throughout the mixing controlled combustion, thus resulting in higher NO_X. When using biodiesel as an emulsifier, the too long ignition delay is somewhat countered. NO_X emissions are reduced, in comparison to diesel fuel, while CO and THC are only higher at the lower loads. The CO and THC emissions can be reduced with the use of an intake air heater. On the other hand, the intake air heater will increase the NO_X formation. The use of diesel-ethanol blends is a relatively easy way of reducing overall CO_2 and PM, however, the reduction is not as significant as if neat alcohols would be used.

Hardware

The r_c can be increased to run a compression ignition engine on neat methanol and ethanol. Since the research octane number for these fuels is rather high, the r_c has to be increased significantly. The choice of r_c is a trade-off. If the r_c is high, the ignitability will be easier at the whole load range. On the other hand, the engine operation flexibility will decrease. For example, higher λ values in combination with exhaust gas recirculation is difficult to run efficiently at high load, since this increases the maximum firing pressure. The maximum in-cylinder pressure is limited from an engine structural point of view. This enforces a retarded combustion phasing to the extent that thermal efficiency is reduced due to the increased exhaust losses. Higher r_c is generally also correlated with increased NO_X emissions. The advantage of lower r_c, higher freedom of operating flexibility, comes alongside the cost of poor low load operation. If a lower r_c is used, the fuel will not ignite with ease at the lower load range. To achieve a stable combustion, or any combustion at all, the intake temperature needs to be increased significantly. A possible source of hot intake air is the use of a turbocharger, or compressor, without an intercooler. If EGR is used, it could be possible to cool down the exhaust gas to a lesser extent - or not at all if necessary - before reintroducing it to the intake.

Corrosivity
Corrosivity is a noticeable issue when using ethanol, and methanol in particular. This issue requires carefully chosen materials for the fueling system, and everything else that comes in contact with the fuel. As an anecdote, during the author's Ph.D. period, non of the stock diesel injectors failed due to ethanol utilization. The use of methanol, on the other hand, repeatedly destroyed the stock injector gaskets within a couple of hours of experimentation. It should be mentioned that the ethanol common rail system, called XPI, used by the Scania D13 engine operated as expected when fueling it with methanol. In either case, care should be taken when designing components for the fueling system for both methanol and ethanol.

Recommended Future Work
Since the use of methanol and ethanol fuel in the compression ignition engine have previously been - and still is - very limited, it is worth considering further analysis. Some areas that might be of interest are listed below.

- A consequence of high intake temperatures is increased NO_X emissions. To remedy this issue, it is highly recommended to further investigate the combustion of methanol and ethanol while using a glow plug. The glow plug would be a heat source to initiate the combustion process of the light alcohols without increasing the NO_X emissions.
- The effects of double and triple injection strategies on combustion characteristics, efficiency and emissions should be further investigated, varying injection timings as well as injection mass ratio. It was concluded from this work, among other things, that several injection events ease the ignition process and enable a more predictable, stable as well as controllable combustion, however no detailed scientific publications regarding this matter were published.
- The author recommends that the work of Pucilowski et al. [1] regarding different piston shape geometries should be experimentally investigated. The piston geometry is not only an important factor in terms of

heat transfer, but also in terms of THC and CO emissions, since emissions become more significant for the light alcohols at relatively low cylinder temperatures.

Bibliography

[1] M. Pucilowski, M. Jangi, S. Shamun, M. Tunér, and X.-S. Bai. Heat Loss Analysis for Various Piston Geometries in a Heavy-Duty Methanol PPC Engine. 2018, DOI:10.4271/2018-01-1726.

Chapter 6

Summary of Publications

6.1 Publication I

Exhaust PM Emissions Analysis of Alcohol Fueled Heavy-Duty Engine Utilizing PPC

Sam Shamun[1], Mengqin Shen[1], Bengt Johansson[2], Martin Tunér[1], Joakim Pagels[3], Anders Gudmundsson[3], Per Tunestål[1].
[1]*Division of Combustion Engines, Lund University, Sweden*
[2]*Physical Science and Engineering Division, King Abdullah University of Science & Technology, Saudi Arabia*
[3]*Division of Ergonomics and Aerosol Technology, Lund University, Sweden*

SAE International Journal of Engines 9(4):2142-2152, 2016
DOI: *https://doi.org/10.4271/2016-01-2288*

This publication examines the engine-out emissions, and in particular the particle size distribution and soot emissions, from a heavy duty compression ignition engine utilizing methanol and ethanol and naphtha gasoline. Single injection was used for all the fuels, except for methanol which was also tested with a double injection strategy to achieve a diffusion controlled combustion. The 70 RON gasoline used in this work, having a moderate sooting tendency, showed that a larger count mean diameter, CMD, was the main cause of higher soot emissions when running in conditions promoting soot formation. For the case of methanol and ethanol, the CMD did not exceed 30 nm in contrast to the naphtha gasoline which emitted particles with a CMD=142 nm. This work shows that when running alcohols in a CI engine, high levels of EGR can be utilized while having the ability to completely disregard the NO_X-soot trade-off.

This work was conducted in collaboration with the Division of Ergonomics and Aerosol Technology. The main author designed the experiments and performed them together with Mengqin Shen while Bengt Johansson, Martin Tunér and Per Tunestål supervised the engine experiments, both in form of giving advice as well as discussions regarding the experimental setup. Joakim Pagels and Anders Gudmundsson also helped writing the paper while giving valuable information and feedback in the field of aerosol science.

6.2 Publication II

Experimental investigation of methanol compression ignition in a high compression ratio HD engine using a Box-Behnken design

Sam Shamun[1], Can Haşimoglŭ[2], Ahmet Murcak[2], Öivind Andersson[1], Martin Tunér[1], Per Tunestål[1].
[1]*Division of Combustion Engines, Lund University, Sweden*
[2]*Department of Mechanical Engineering, Sakarya University, Turkey*

Fuel 209:624-633, 2017
DOI: *https://doi.org/10.1016/j.fuel.2017.08.039*

The objective of this publication was to investigate the effects of intake pressure, EGR, CA50 as well as injection pressure on the gross indicated efficiency and regulated emissions in a heavy duty compression ignition engine utilizing methanol. The investigation was approached using a Box-Behnken type design of experiment due to its cautious nature of not going to extreme operations, since the utilized r_c was unconventionally high, r_c=27. It was shown that the efficiency of the engine reached its peak, η_{GIE}=52.8 %, at the highest tested EGR level of 50 %, due to reduction in heat transfer losses. Meanwhile, at EGR=50 %, NO_X emissions are reduced to less that 10 ppm. THC emissions however never descended below ∼150 ppm due to the crevice to volume ratio as well as fuel impingement on the piston crown. Running the engine on methanol with a high r_c enables combustion stability in the low load range, however, decreases the operation flexibility at high loads.

This publication was written by the main author, who was supervised by Martin Tunér and Per Tunestål. The experimental design itself was inspired and supervised by Öivind Andersson. The experiments were conducted by the main author, Can Haşimoglŭ as well as Ahmet Murcak.

6.3 Publication III

Detailed Characterization of Particulate Matter in Alcohol Exhaust Emissions

Sam Shamun[1], Maja Novakovic[1], Vilhelm B. Malmborg[2], Calle Preger[3], Mengqin Shen[1], Maria E. Messing[3], Joakim Pagels[2], Martin Tunér[1], Per Tunestål[1].
[1]*Division of Combustion Engines, Lund University, Sweden*
[2]*Division of Ergonomics and Aerosol Technology, Lund University, Sweden*
[3]*Division of Solid State Physics, Lund University, Sweden*

The Ninth International Conference on Modeling and Diagnostics for Advanced Engine Systems 2017.9 B304, 2017
DOI: *https://doi.org/10.1299/jmsesdm.2017.9.B304*

The motivation behind the work conducted in this publication was to investigate the nucleation mode particles observed in Publication I. Three fuels were tested in the experiments: methanol, ethanol and diesel. In order to analyze the particles, measurements with the AVL MSS and the Cambustion DMS500 M177 were conducted. Moreover, exhaust particles were sampled on carbon coated copper grids in order to observe the particles under a TEM. The result indicated that the particles emitted from combustion of the two alcohols where indistinguishable. Also, the results from the EDX suggested that the particles did not originate from the combustion itself or the combustion products, but from the lubricating oil, since similar elements were found when analyzing the lubricating oil.

The experiments were conducted in collaboration with the Division of Ergonomics and Aerosol Science and the Division of Solid State Physics by the main author, Maja Novakovic, Vilhelm B. Malmborg and Mengqin Shen. The engine experiments and procedure were supervised by Martin Tunér, Per Tunestål and Joakim Pagels. The TEM and EDX analysis was performed by Calle Preger and supervised by Maria E. Messing.

6.4 Publication IV

Performance and emissions of diesel-gasoline-ethanol blends in a light duty compression ignition engine

Giacomo Belgiorno[1], Gabriele Di Blasio[1], Sam Shamun[2], Carlo Beatrice[1], Per Tunestål[2], Martin Tunér[2].

[1]*Consiglio Nazionale delle Ricerche - Istituto Motori, Italy*
[2]*Division of Combustion Engines, Lund University, Sweden*

Fuel 217:78-90, 2018
DOI: *https://doi.org/10.1016/j.fuel.2017.12.090*

In this publication, the impact of two fuel blends - consisting of diesel, biodiesel and ethanol - on performance, characteristics and emissions were investigated in a light duty single cylinder engine. Operating conditions were taken from the original engine ECU and replicated using diesel and the two fuel blends. The result showed that both NO_X and soot emissions were reduced when using the oxygenated fuels, and in particular with the fuel containing the higher concentration of ethanol. There CO and THC emissions increased at the lower loads. The indicated efficiency showed a significant increase over diesel fuel when using the oxygenated fuels.

The experiments were performed by Giacomo Belgiorno, Gabriele Di Blasio and partially by Sam Shamun at CNR-Istituto Motori in collaboration with the Division of Combustion Engines at Lund University. The main parts were written by Sam Shamun and partially written by Giacomo Belgiorno and Gabriele Di Blasio. The experiments and publication were supervised by Carlo Beatrice, Per Tunestål and Martin Tunér.

6.5 Publication V

Performance and emissions of diesel-biodiesel-ethanol blends in a light duty compression ignition engine

Sam Shamun[1], Giacomo Belgiorno[2], Gabriele Di Blasio[2], Carlo Beatrice[2], Martin Tunér[1], Per Tunestål[1].
[1]*Division of Combustion Engines, Lund University, Sweden*
[2]*Consiglio Nazionale delle Ricerche - Istituto Motori, Italy*

Applied Thermal Engineering 145:444-452, 2018
DOI: *https://doi.org/10.1016/j.applthermaleng.2018.09.067*

In this publication, the impact of two fuel blends - consisting of diesel, gasoline and ethanol - on performance, characteristics and emissions were investigated in a light duty single cylinder engine. Operating conditions were taken from the original engine ECU and replicated using diesel and the two fuel blends.

The result showed that particulate matter emissions were reduced, while NO_X increased, when using the oxygenated fuels, and in particular with the fuel containing the higher concentration of ethanol. There was, however, an increase in CO and THC emissions at the lower loads, while at the higher load point, these emissions for the oxygenates evened out with diesel fuel. The indicated efficiency showed a significant increase over diesel fuel when using the oxygenated fuels.

The experiments were performed by Giacomo Belgiorno, Gabriele Di Blasio and partially by Sam Shamun at CNR-Istituto Motori in collaboration with the Division of Combustion Engines at Lund University. The main parts were written by Sam Shamun and partially written by Giacomo Belgiorno and Gabriele Di Blasio. The experiments and publication were supervised by Carlo Beatrice, Per Tunestål and Martin Tunér.

6.6 Publication VI

Quantification and Analysis of the Charge Cooling Effect of Methanol in a Compression Ignition Engine utilizing PPC strategy

Sam Shamun[1], Burak Zincir[2], Pravesh Shukla[1], Pablo Garcia Valladolid[1], Sebastian Verhelst[1], Martin Tunér[1].
[1]*Division of Combustion Engines, Lund University, Sweden*
[2]*Maritime Engineering Department, Istanbul Technical University, Turkey*

ASME Internal Combustion Engine Division Fall Technical Conference, ICEF2018-9657, 2018
DOI: *https://doi.org/10.1115/ICEF2018-9657*

The objective of this work was to investigate the cooling effect of neat methanol in comparison with iso-octane while using the start of injection timings of partially premixed combustion. Each fuel was tested at three different FuelMEP, while keeping the intake pressure and temperature at each operating condition, in order to decrease the compression work. This was done to be able to compare the cooling in relation to the energy density of the two fuels. The result showed that the pressure reduction in the compression stroke was higher for methanol than for iso-octane. However, the pressure reduction did not decrease the compression work to a significant degree. The cooling effect of methanol, instead, was particularly efficient at reducing the heat transfer during the expansion stroke. This was also noticed by significantly reduced NO_X

emissions.

The main author conducted the experiments together with Burak Zincir and Pravesh Shukla. Pablo Garcia Valladolid was responsible for maintaining the experimental set-up while Sebastian Verhelst and Martin Tunér supervised the experiments as well as the authoring of the publication.

Printed by Media-Tryck, Lund 2019 NORDIC SWAN ECOLABEL 3041 0903

LUND UNIVERSITY

ISBN 978-91-7753-986-5
ISRN LUTMDN/TMHP-19/1142-SE
ISSN 0282-1990