On Combustion in the CNG-Diesel Dual Fuel Engine

FREDRIK KÖNIGSSON
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Fredrik Königsson

Doctoral thesis

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“Messen ist Wissen”

- Werner von Siemens, 1816-1892
Abstract
Currently there is a large interest in alternative transport fuels. There are two underlying reasons for this interest: the desire to decrease the environmental impact of transports and the need to compensate for the declining availability of petroleum. In the light of both these factors, the CNG-diesel dual fuel engine is an attractive concept. The primary fuel of the dual fuel engine is methane, which can be derived both from renewables and from fossil sources. Methane from organic waste, commonly referred to as biomethane, can provide a reduction in greenhouse gases unmatched by any other fuel. Furthermore, fossil methane, natural gas, is one of the most abundant fossil fuels. The dual fuel engine is, from a combustion point of view, a hybrid of the diesel and the Otto-engine and it shares characteristics with both.

From a market standpoint, the dual fuel technology is highly desirable; however, from a technical point of view it has proven difficult to realize. The aim of this project was to identify limitations to engine operation, investigate these challenges, and, as much as possible, suggest remedies. Investigations have been made into emissions formation, nozzle-hole coking, impact of varying in-cylinder air motion, behavior and root causes of pre-ignitions, and the potential of advanced injection strategies and unconventional combustion modes. The findings from each of these investigations have been summarized, and recommendations for the development of a Euro 6 compliant dual fuel engine have been formulated. Two key challenges must be researched further for this development to succeed: an aftertreatment system which allows for low exhaust temperatures must be available, and the root cause of pre-ignitions must be found and eliminated.
Sammanfattning


Preface

The material presented in this thesis was generated as part of an AVL-project to develop understanding of CNG-diesel dual fuel combustion. Partner in the project was Scania CV. The work was performed at the group of Internal Combustion Engines at the Royal Institute of Technology, Stockholm, Sweden. Partial funding was provided by the Swedish Energy Agency. The thesis consists of an overview of dual fuel combustion and the challenges and opportunities associated with it. The overview is mainly based on experimental results from the project but also on available literature.

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Fredrik Königsson
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Outline & Readers guide

The thesis summarizes the experiences from the dual fuel project and can be read independently of the appended papers. The papers provide additional information about certain topics which are outlined in the summary of publications. There are also references in the thesis text when additional information is available from the papers. The thesis is organized into the following chapters:

Chapter 1: Introduction
The first section summarizes the history of the internal combustion engine. The section about fuels does the same for automotive fuels and discusses the driving forces behind the diversification of the fuel market that is currently taking place. Finally methane is put into perspective from an environmental and a supply point of view.

Chapter 2: Emissions from combustion engines
Most of the development work performed within combustion engines is driven by emissions legislation. This chapter provides an overview of the legislated emissions and the main mechanisms for their formation in combustion engines. Also included is a brief discussion of the two main global emissions which contribute to the greenhouse effect, CO₂ and methane.

Chapter 3: Overview of methane engines
Chapter 3 presents the main pathways currently available for propulsion when developing a methane road vehicle and introduces the diesel dual fuel engine.

Chapter 4: The dual fuel engine
This chapter provides an overview of the combustion process in the diesel dual fuel engine and outlines the main challenges. Also included are a literature survey and an overview of the effects of the different control parameters available.

Chapter 5: Experimental setup
Details of the equipment used to record the data are found in this chapter.

Chapter 6: Results
The results chapter constitutes the main part of the thesis and is divided into sections based on the main challenges of dual fuel operation.

Chapter 7: Discussion
The discussion attempts to tie the results from the project together into recommendations for the design of a competitive dual fuel engine.

Chapter 8: Summary & Conclusions
A summary of the main findings and suggestions for future work.
List of publications


SAE Technical Paper 2011-01-2223, presented at the 2011 SAE Commercial Vehicle Engineering Congress in Chicago, USA

SAE Technical Paper 2012-01-0826, presented at the 2012 SAE World Congress in Detroit, USA

Published in SAE International Journal of Engines 6(2):2013, SAE Technical Paper 2013-01-0848, presented at the 2013 SAE World Congress in Detroit, USA


1 Introduction

1.1 The internal combustion engine

For the past 300 years, man has wrestled with the task of converting chemical energy stored in various fuels into mechanical work. The first solution to this problem that saw widespread use was the steam engine during the first decades of the 18th century. Originally used to evacuate water from the coal mines of England, the steam engine went on to become the power plant of choice and a driving factor during the industrial revolution. The steam engine is an external combustion engine, where the combustion takes place outside the cylinder or the turbine, and a different medium, water vapor, is used to transfer the work. This stands in contrast to the internal combustion engine, where the fuel-air mixture in its unburned and burned state is also the working fluid which transfers work to the piston. In 1860 the first practical internal combustion engines became available. Burning a mixture of coal-gas and air without compression and attaining an efficiency of 5%, these early engines were manufactured in relatively small numbers. It was not until 1876, when Nicolaus A. Otto ran his first four-stroke engine, that the internal combustion engine received the breakthrough it needed to become the predominant power source that it is today [1].

Recognizing that the greater the expansion of the post combustion gases, the greater the work transferred; attempts were made to increase the expansion ratio. Continuing on the theoretical work of Alphonse Beau de Rochas, James Atkinson constructed an engine where the expansion stroke was longer than the compression stroke. His engine, however, was plagued by mechanical problems and this course of action was more or less abandoned. Since then, for most practical purposes, the expansion and compression ratios have been linked, and the desire to increase the expansion ratio translates into the need to increase the compression ratio.

The fuels available at the time did not permit compression ratios larger than four before the onset of knock. While improvements were made to fuels, carburetors and ignition systems to help address this problem, Rudolf Diesel proposed a different approach [1]. In his patent from 1890 Diesel outlined a new type of combustion engine, where the fuel was not added until after the compression, and the temperature of the hot compressed air was sufficient to ignite the fuel. Diesel improved on the concept by further increasing the compression ratio and, due to his success in patenting his ideas, he is generally considered the inventor of the engine that bears his name. This new concept, the diesel engine, would permit much greater expansion ratios and thus greater efficiency. For the past hundred years, wars, economic interests and lately emissions legislation, have driven the development of both the Otto and the diesel engine into their current respective forms. Advances have been made in many fields: from the materials used, the aftertreatment system and the engine geometry, to complex injection strategies, unconventional combustion modes and the software used for engine control.

1.2 Fuels

In the early days of internal combustion engines, the fuels used were various: coal gas, coal powder, cleaning agents, lamp oil, kerosene and different petroleum distillates. As the demand from manufacturers of Otto- and diesel engines for high octane and high cetane fuels, respectively, coincided with the supply of cheap petroleum from USA and the Middle East, the choice of
automotive fuels for most of the 20th century was in all essence reduced to two: gasoline and diesel fuel from petroleum feedstock.

The year of writing is 2014 and a combination of environmental concerns and high oil price is driving the return to a diversified fuel market. The environmental concerns consist of the fear that burning fossil fuels, and thus reintroducing carbon into the atmosphere which has been stored in our planets crust during millennia, can cause a change in the climate.

The shift to a higher oil price is illustrated in Figure 1, where oil price for the past 30 years is shown along with net oil exports for the same time period.

![Figure 1. World net oil exports and annual average oil price for the time period 1982-2014 [2].](image-url)

Global net exports is the world production minus the consumption of the oil producing countries, the result is the oil available to the market and the oil importing countries. Global net exports reached a peak in 2005 and have not surpassed that level in spite of record prices. This indicates that oil supply is currently inelastic and that the high price is likely here to stay.

In summary: climate change and high oil price are stirring great interest in alternative transport fuels, both fossil and renewable.

### 1.2.1 Methane as fuel

One fuel, or energy carrier, that can address both the challenge of climate change and of supply constraints, is methane. Methane in the form of biogas is a renewable fuel, which compares very favorably in well-to-wheel analyses to other alternatives, renewables as well as fossil. This is shown in Figure 2.
Figure 2. Greenhouse gas emissions of different fuels per km versus energy consumption per km for passenger cars from a well-to-wheel perspective [3].

It is seen that while it is possible to reduce the amount of CO$_2$ emitted, this reduction needs to be paid for by an increase in primary energy consumption compared to gasoline and diesel. This generally translates into higher cost for the alternative fuels. Depending on the source of biomass for the biogas production, different levels of CO$_2$ reduction are achieved. If manure, where the fermentation would occur without human intervention, is used as feedstock, negative emissions of greenhouse gases are achieved. This is possible since methane is a more powerful greenhouse gas than CO$_2$. Therefore, the burning of methane that would otherwise be released into the atmosphere results in ‘negative’ emissions of greenhouse gases. The potential for biogas to displace conventional fuels is a matter of debate. If biogas must compete with fossil fuels from an economic point of view, which is the current situation, the potential for biogas to replace conventional transport fuels in Europe is only a few percent [3]. Different incentives can be used to increase this number and it also increases as oil becomes more expensive.

Natural gas is relatively abundant and will be available at competitive prices for the upcoming decades. Although it is a fossil fuel, it offers a 20% reduction in greenhouse gas emissions compared to gasoline or diesel because of the greater H to C ratio of methane. However, the drivers behind a shift towards natural gas vehicles are mainly price and strategic concerns.

Methane has many properties making it a suitable automotive fuel. Since it is gaseous, exceptional mixing can be achieved and soot emissions are for this reason unlikely. It is a very simple and stable molecule. Because of this stability, it has a high autoignition temperature and good resistance to knock, enabling high compression ratios and resulting benefits in thermal efficiency. However, the stability of the methane molecule also presents a challenge for the aftertreatment system since
methane requires a high light-off temperature. The conversion efficiency of an oxidation catalyst for different hydrocarbons is shown in Figure 3.

![Figure 3. Conversion efficiency of an aged oxidation catalyst versus gas temperature for CO and different hydrocarbons [4].](image)

In addition to the high light-off temperature, the conversion efficiency of methane does not reach 100% for the catalyst shown, even at high temperatures. For this reason, minimizing engine-out emissions of unburned methane is of great importance.

Methane has a stoichiometric air to fuel ratio, AFR, of 17, compared to approximately 14.6 for diesel and gasoline. Since the mass of air inducted into the engine is limited, a higher AFR means less energy into the engine and lower peak power. This effect is to some extent moderated by the higher calorific value of methane but a penalty to power per air mass compared to gasoline remains. Storage on board the vehicle is also a challenge since methane is gaseous at ambient temperatures, and liquid storage in cryogenic tanks is required if energy density comparable to diesel or gasoline should be achieved.
2 Emissions from combustion engines

The emissions from engines, both spark ignited, SI, and compression ignited, CI, can be divided into two subcategories: local emissions, which negatively affect the environment close to the emitter and global emissions, which affect the entire planet equally. Currently legislation and certification of vehicles take only local emissions into account, and therefore these receive the most attention from engine developers. Global emissions, the main one being CO₂, are generally analogous to poor fuel economy and are therefore minimized in order to have a competitive product. The task when developing an engine is therefore to minimize engine fuel consumption, development cost and production costs with the hard constraint that the legislated emission limits need to be fulfilled. An overview of the main local- and global emissions follows.

2.1 Local emissions

2.1.1 Unburned hydrocarbons

Emissions of unburned hydrocarbons, HC, consist of fuel, partly reacted fuel and lubrication oil which escaped combustion for various reasons. Emitted into the atmosphere, HC can cause photochemical smog and diseases. In the SI engine where a premixed charge is compressed and subsequently ignited, there are three main sources of HC emissions: crevice losses, adsorption and quenching.

The largest contribution can be attributed to crevice losses; parts of the premixed charge are forced into narrow regions, such as the piston ring pack and head gasket crevice. Conditions in these narrow, cold, regions are unfavorable for flame propagation. As the expansion progresses and the pressure in the cylinder drops, the unburned mixture from the crevices return to the combustion chamber and are expelled with the exhaust gases.

Adsorption works in a similar way; at high pressure, fuel hydrocarbons are adsorbed into the oil film on the cylinder walls and into the deposits in the combustion chamber. As the expansion progresses and the pressure drops, the adsorption reverses, and the HC is returned to the combustion chamber.

Finally, there is quenching; close to the cylinder wall, a boundary layer exists, with lower temperature compared to the rest of the charge. The flame therefore quenches before it reaches the wall, and some HC in the boundary layer may escape combustion. The extent to which quenching affects the emissions at stoichiometric operation is considered to be small, since some of the HC will diffuse into the hot mixture and oxidize.

Lean operation will aggravate each of these mechanisms since it leads to lower temperatures. As dilution increases, the quench layer close to the cylinder wall will increase in thickness. Additionally, since the temperature in the bulk gas is lowered, the fuel which diffuses into the bulk gas from the quench layer finds it increasingly difficult to oxidize. If the mixture is sufficiently diluted, the point is reached where partial quenching occurs, meaning that the flame is extinguished before it has traversed the entirety of the combustion chamber. For the SI engine, the HC emissions are handled by an oxidation catalyst and depending on the fuel used, the demands on the catalyst differ. Since CI engines are direct injected, they are therefore not affected by any of the
aforementioned mechanisms. Instead, HC emissions from the CI engine are largely derived from diesel fuel remaining in the injector sac. These contributions are small, and HC emissions are not considered a problem for CI engines. Hydrocarbon emissions from the dual fuel engine are treated further in section 6.1.2.

2.1.2 Carbon monoxide
Carbon monoxide, CO, is formed as an intermediate step in combustion of hydrocarbons. Failure of CO to oxidize into CO$_2$ can be caused by unavailability of oxygen or insufficiently high temperatures. CO is toxic to humans and can cause symptoms ranging from light headaches to death. In a stoichiometric engine, CO emissions will be substantial due to less than perfect mixing and local oxygen deficits. At lean conditions, CO emissions can occur, both in the CI and the SI engine, because of low temperatures. CO emissions are effectively handled by an oxidation catalyst and are therefore not as problematic as emissions of HC or NO$_x$.

2.1.3 Nitrogen oxides
Mono-nitrogen oxides, NO and NO$_2$, commonly referred to as NO$_x$, are formed through various mechanisms, where atmospheric nitrogen, or nitrogen from the fuel, is fused with oxygen with the aid of high temperatures. NO$_x$ emissions can cause respiratory diseases. At ground level NO$_x$ contributes to the formation of ozone while, paradoxically, it destroys the ozone at high altitudes damaging the ozone layer.

The most influential NO$_x$ producing mechanism in combustion engines is believed to be thermal NO$_x$. This route was proposed by Zel’dovich in 1948 and has since been appended to. The extended Zel’dovich mechanism consists of three reversible reactions:

1. $O^* + N_2 \leftrightarrow k_1 \rightarrow NO + N^*$
2. $N^* + O_2 \leftrightarrow k_2 \rightarrow NO + O^*$
3. $N^* + OH \leftrightarrow k_3 \rightarrow NO + H^*$

The name thermal NO$_x$ refers to the high activation energy of the first, rate-limiting reaction which makes this mechanism highly temperature dependent. The rate constants in the mechanism have been revised recently and substantially increased. According to current theory, thermal NO$_x$ is responsible for 90-95% of the NO$_x$ emissions from combustion engines [5].

Prompt NO$_x$, or Fenimore NO$_x$, is a mechanism where atmospheric N$_2$ reacts with CH radicals. Due to the need for these radicals, prompt NO$_x$ is dependent on locally fuel rich areas and decreases quickly at lean mixtures. At lean conditions and high pressures, the first reaction of the Zel’dovich mechanism can be stabilized through collision with a third party so that N$_2$O is formed instead of NO; the N$_2$O is then oxidized to 2 NO molecules. Both Fenimore NO$_x$ and the N$_2$O route are characterized by large uncertainties. Finally, there is the possibility for nitrogen bound in the fuel to form NO, however, engine fuels contain very little N$_2$, and this contribution is therefore considered irrelevant.

Due to the strong temperature dependence of the Zel’dovich mechanism, NO$_x$ formation in engines is dependent on the conditions in the hottest parts of the combustion chamber. In the SI engine, this corresponds to the volume around the spark plug, which burns first and is further compressed by the
burning of the remainder of the charge. In CI engines, most of the NO\textsubscript{x} forms in the edge of the fuel plume where conditions are close to stoichiometry.

2.2 Global emissions

2.2.1 Carbon dioxide
Carbon dioxide, CO\textsubscript{2}, is the inevitable byproduct of hydrocarbon combustion. It is an inert gas, not harmful to plants or animals. It is, however, a greenhouse gas. Burning fossil fuels, stored beneath the ground during past millennia, increases the concentration of CO\textsubscript{2} in the atmosphere. As the concentration of CO\textsubscript{2} increases, more of the sun’s heat is retained, potentially warming the planet and changing the climate.

2.2.2 Methane
Methane, CH\textsubscript{4}, while a hydrocarbon by definition, it is also inert, and does not contribute to smog production or respiratory diseases. However, methane is a very potent greenhouse gas, and if not carefully addressed, methane slip from engines can nullify any environmental benefits of switching from gasoline or diesel.
3 Overview of methane engines

3.1 Methane monofuel engines

Methane is mainly used in SI engines. In light duty applications, these engines are usually bi-fuel engines; they must be able to operate on gasoline fuel as well. This limits the compression ratio, and effectively limits the efficiency of the engine as well. For medium and heavy duty applications, SI gas engines are typically mono-fuel, meaning that they can be optimized for methane operation, providing better efficiency. These engines still suffer from the known Achilles heel of the Otto engines, pumping losses and poor part load efficiency. The pumping losses can be addressed to some extent, by running the engine lean and by adding EGR. Lean operation of an SI engine is the source of many complications since the three way catalyst can no longer be used to reduce NO\textsubscript{x}, and a lot of strain is put on the ignition system.

Examples of mono fuel CI engines operating on methane exist. These are currently not widespread due to the high autoignition temperature and corresponding demand for compression ratio and ignition aids.

3.2 Methane dual fuel engines

One solution to using methane in a CI engine is to introduce a pilot-fuel with higher cetane number, which initiates combustion and ignites the methane, thus giving rise to the methane-diesel dual fuel, DDF, engine. Work is currently carried out, both in the field of direct injected DDF, as well as port injected DDF.

Direct injected DDF involves a special DI injector, which handles both the diesel and the methane fuel. A small pilot amount of diesel fuel is injected ahead of the main injection to raise the temperature and allow for ignition of the methane [6]. The methane is then injected and burns in a diffusion flame, similar to diesel combustion. This concept enables unthrottled operation with little methane slip, but particles and NO\textsubscript{x} present a problem. The system is also complex and expensive, and it is not possible to run the vehicle on diesel only, so fuel flexibility is lost.

In port injected DDF, the methane is injected into the intake manifold and is premixed with the air during induction and compression. A small diesel pilot is used to initiate combustion. The work presented in this thesis focuses solely on port injected DDF, and henceforth this is what is referred to when the acronym DDF is used. Port injected DDF has the potential to maintain diesel capability in case methane is unavailable. This is an important factor when bringing the technology to market.
4 The dual fuel engine

In this chapter, the port injected DDF engine is described in further detail. An introduction to DDF combustion is given, followed by an overview of the main challenges involved in dual fuel operation. These challenges comprise the objectives of the thesis.

4.1 Introduction to DDF combustion

DDF heat release shares characteristics with both CI and SI combustion and can be considered to consist of four parts:

1. Combustion of the diesel pilot
2. Combustion of methane in the premixed pilot-region
3. Flame propagation through the methane-air mixture
4. Possible bulk ignition of the end gas

The four parts are demonstrated in the form of a hypothetical heat release curve in Figure 4.

![Figure 4. Schematic image showing possible contributions to dual fuel heat release.](image)

The traditional description of dual fuel combustion is that it consists of three parts [7]. Based on experience from measurements and work of other authors [8], a fourth possible contribution to the heat release is included, bulk ignition in the end gas region. Depending on load and a number of parameters, each of these parts will contribute to a different degree to the accumulated heat release. The term diesel substitution rate is used to quantify the contribution from each fuel to the total amount of energy supplied. It is further explained in Figure 5.
Figure 5. Diesel substitution rate; percent of the fuel, on an energy basis, that is replaced by methane.

The diesel substitution rate, or $\text{E}\%\ \text{CH}_4$, determines the combustion characteristics and where the DDF engine fits on the scale between the SI- and the CI engine.

Images of dual fuel combustion of syngas, with diesel fuel as ignition source, are shown in Figure 6 [9]. The frames are shot with a high speed camera, through a Bowditch type piston.

Figure 6. Dual fuel combustion of syngas and diesel [9].

The diesel injector used in this study had four holes, and the amount of diesel injected was 2 mg. It is clearly seen how the combustion starts in the regions where the diesel spray is present. The combustion then progresses by means of flame propagation until the entire combustion chamber is engulfed. Glowing soot particles from the diesel pilot are present during the entire combustion event.

4.2 Literature on the dual fuel engine

A lot of literature can be found pertaining to conversion of existing engines to dual fuel operation. A typical investigation includes a simple conversion of a diesel engine to dual fuel operation with
the aid of port injectors or a gas mixer. The engine is operated, and the effect of a few, easily controlled parameters, typically pilot amount, pilot timing, intake temperature, load and speed with respect to emissions and efficiency are investigated [10], [11], [12], [13], [14], [15], [16]. While these publications serve to confirm the general trends for DDF operation, they commonly lack depth and do not advance the understanding of the combustion process. The results in these studies are also quite specific to the engine setup used and may for this reason appear contradictory if this fact is not taken into account. Results of more general investigations are presented in [17], [7], [18]. However, most of the data presented is acquired at λ between 2 and 10, which is hardly relevant.

More in-depth investigations into the effect of air motion and gas supply method, by means of optical observation through an endoscope, are performed in [19], [20]. A high speed camera was used, each frame was converted to gray scale and the average luminosity across the entire frame was computed, resulting in a curve of luminosity versus crank angle. Considering the very strong luminosity of soot particles, compared to that of a propagating flame, and the limited field of view provided by an endoscope, the results could be seen as an indication of how much glowing soot transported in front of the window by the air motion, rather than an indication of combustion quality. The authors also fail to provide information about λ during the tests, making the results impossible to put into context.

Several attempts to model dual fuel combustion have been made [21], [22], [23], [24], [25], [26], [27], [28], [29], [30], [31]. These publications all have in common that the models are validated against small datasets and no conclusions can be made regarding the validity over a wider range of conditions. Some of the models are validated for irrelevant operating conditions, [21], [23] while others show very poor congruence with measurements, [26], [22], [31]. Based on the available literature, it appears that dual fuel combustion is not sufficiently understood to enable predictive modeling.

The effects of the different control parameters available in the DDF engine, as reported in the literature, are summarized in the following section. Data from the current project is also presented to support the conclusions when necessary. The focus of the discussion will be on HC and NOx emissions.

4.2.1 Injection timing

Most of the literature on DDF combustion pertains to diesel engines which have been converted to DDF operation by relatively simple means. These engines are unthrottled and operate at very lean conditions, especially at light load. For this reason, several authors report a reduction in unburned hydrocarbons when advancing the injection timing; combustion closer to TDC leads to higher temperatures and higher combustion efficiency at lean mixtures [32], [33], [34], [35], [36]. This is supported by the results at λ = 1.8 in Figure 7, where HC emissions are shown as a function of combustion phasing for different λ. CA50 refers to the crank angle where 50% of the charge has burned and is based on heat release calculations.
Figure 7. HC emissions versus combustion phasing for different $\lambda$.

However, at $\lambda=1.6$ this is no longer true; the HC emissions appear largely unaffected by combustion phasing. At even richer mixtures, the relationship is in fact the opposite; the emissions of unburned hydrocarbons decrease with retarded combustion phasing. The explanation for this is that late combustion phasing results in higher temperature during the expansion, which in turn promotes oxidation of HC returning from crevices. At richer mixtures, this mechanism is of greater importance than at leaner conditions, since combustion efficiency in the bulk gas is good, and a larger fraction of the HC is derived from the crevices. This is shown in Paper IV. NO$_x$ emissions initially increase as injection timing is advances [33], [34], [32], [36], [35]. This is readily explained by the increased temperatures. Additional discussion regarding advanced combustion phasing is found in Paper II. If the injection timing is advanced to the point that the combustion mode changes, the emissions trends will behave differently. This is discussed further in section 6.2.1 regarding unconventional combustion modes.

4.2.2 Diesel substitution rate, E% CH$_4$

The emissions of HC increase monotonously with increased diesel substitution rate and decreased amount of pilot injection [7], [35], [37]. At low $\lambda$, in spite of efficient flame propagation, the effect of crevices will lead to increased HC emissions when the amount of fuel supplied as a premixed charge increases. The emissions of NO$_x$ show a more complex behavior, they decrease for lean mixtures when the substitution rate is increased [7], [37], [35]. The pilot zone is where the highest temperatures exist and where most of the NO$_x$ is formed. At lower $\lambda$, however, the behavior is entirely the opposite, increased diesel substitution rate instead leads to more NO$_x$. This is further discussed in Paper II. At high substitution rates, soot formation is not considered a problem, but it remains to be demonstrated whether future emissions levels can be reached without a DPF. If fuel flexibility is desired, a DPF will be needed anyway to manage the soot from diesel operation.
4.2.3 Inlet temperature
Increased inlet temperature enhances the flame propagation and reduces the emissions of HC and CO [7]. Predictably, it also leads to higher emissions of NO\textsubscript{x}. High inlet temperature increases the likelihood of knock and pre-ignition at high load, and it is therefore something which can be utilized primarily at light load to reduce the amount of throttling needed. Results from test runs with raised intake temperature and further discussion is located in Paper II.

4.2.4 EGR
EGR dilutes the mixture and lowers the compression- and combustion temperature. From the dilution, the O\textsubscript{2} concentration is reduced. The reduction in temperature and O\textsubscript{2} concentration affects combustion efficiency negatively. For this reason, CO and HC emissions increase. It also lowers the combustion temperature, which drastically reduces the formation of NO\textsubscript{x} and reduces the risk of pre-ignitions [32], [38], [39]. The strong suppressing effect of EGR on pre-ignitions has been shown in this project, and cooled EGR may be a tool needed to reach desired power density. Hot EGR combines the NO\textsubscript{x}-reducing effect with the improved flame propagation from high inlet temperature, allowing a simultaneous reduction of HC, CO and NO\textsubscript{x} [33]. The use of EGR to reach stoichiometric conditions, while minimizing throttling loss, is also a strong possibility in the DDF engine, thus enabling a very cost efficient aftertreatment system through the use of a three-way-catalyst [38]. The EGR tolerance is very large compared to an SI engine due to the powerful ignition source. EGR as a tool to reach stoichiometry is discussed further in Paper II.

4.2.5 Diesel common rail pressure
The diesel common rail pressure has a small effect on combustion, with the notable exception that reduced common rail pressure helps facilitate reliable pilot injection at operating points with substantial throttling. Results from this project show that, reducing the injection pressure decreases the dispersion of the diesel pilot and enables further throttling, with maintained diesel ignition. This is illustrated in Figure 8.
Figure 8. Lowest possible loads attainable in DDF operation versus common rail pressure for three different diesel substitution rates.

The data shown in Figure 8 is generated by setting a limit for HC emissions. Since HC emissions increase with $\lambda$, this effectively imposes a limit on intake manifold pressure. A larger diesel pilot extends the lean limit, which means less throttling is needed for a given load. At loads below the ones shown, the remaining option is to resort to diesel operation.

4.2.6 Engine speed

No significant effect of engine speed on emissions has been found in literature. Experiences from this project show that the combustion duration, in crank angle degrees, increases only slightly with engine speed; this is shown in the left graph in Figure 9. For this to occur, the combustion rate versus time must increase greatly; this is shown in the right graph in Figure 9.
As the engine speed increases, so does in cylinder turbulence. Higher rates of small scale turbulence increase the flame propagation speed, an effect which is well known from SI engines. This strong effect of in cylinder air motion on dual fuel combustion rate indicates that a broader operating band might be possible compared to diesel operation.

Increased engine speed does mean that the heat flux power increases, which will aggravate thermal issues such as high nozzle tip temperatures, something which is treated further in Paper III.

4.2.7 Air excess ratio, $\lambda$.

The emissions from DDF combustion as a function of $\lambda$ are discussed in Papers II, IV and V. CO is high at $\lambda=1$, and has a minimum around $\lambda=1.1$. Both CO and HC increase as $\lambda$ increases and combustion becomes colder. NOx has a maximum around $\lambda = 1.2$ for high substitution rates and decrease with higher $\lambda$. The same trends are confirmed in literature [7], [40], [34], [17].

4.2.8 Coolant temperature

A cold engine causes more heat to be transferred from the gas to the combustion chamber walls; this reduces combustion temperatures and increases emissions of CO and HC, while emissions of NOx decrease. As the engine heats up, the influence of post oxidation of HC and CO becomes greater. Of minor effect is also the fact that the piston expands and crevice volumes decrease [41].

4.2.9 Valve timing

Since the DDF engine is premixed, variable valve timing can be used to reduce the pumping losses from throttling at light load. However, variable valve timing has not yet made its way into production diesel engines and is therefore unlikely to be available to the DDF engine. This is reflected in the literature; very little work is presented pertaining to this subject.

Negative exhaust valve overlap has been used successfully to reduce engine-out HC emissions [42]. An exhaust valve timing advance of 19° CA resulted in a reduction of HC emissions by approximately 25% at high substitution rates and $\lambda=2$. An advance of 38° CA resulted in a 50% reduction of HC. For the 38° CA advance, the in-cylinder temperature increased by 20° C due to
increased amount of residual gases. The increase in temperature can be expected to improve combustion efficiency and probably accounts for some of the 50% reduction. However, since the 19° CA advance had the same in-cylinder temperature at IVC, the reduction in HC could be from trapping HC emerging from crevices late during the expansion and during the blowdown. Negative valve overlap with fixed cams would likely severely limit maximum load due to knock and is for this reason unlikely to be utilized in production.

The possibility to reduce the effective compression ratio at high load through Miller timing, and thus mitigate knock and pre-ignition, is a promising application for a variable valve train in the DDF engine.

4.2.10 Summary of influence of control parameters
A rule of thumb is that parameter changes which increase combustion temperature decrease HC and CO emissions but increase the emissions of NOx. Hot EGR is the exception to the rule since it has the potential to simultaneously reduce HC, CO and NOx.

4.3 Objectives and challenges
Dual fuel combustion is not a mature technology when compared to CI and SI combustion. Because of this, it is expected that not all challenges and limitations have been encountered and documented, much less fully understood. This project is preceded by student projects. Based on information from those projects, the principal challenges which hinder the introduction of this engine type are hydrocarbon emissions and knock. The main focus of the project is therefore to investigate these two challenges and propose solutions, but also to identify, investigate and attempt to overcome limits to dual fuel operation which are not yet known. For this reason, a screening of the engine operating range, where the most influential control parameters are varied, was carried out during the initial phase of the project. The results from this screening are presented in Paper I and Paper II.

In Paper I, operating conditions were carried over from a production diesel engine, and different injection strategies for the diesel injection during DDF operation were evaluated. It was found that injection strategies were not sufficient to enable the conversion to dual fuel operation with acceptable emissions. Further investigations were carried out under the assumption that a throttle was available and that λ-control, similar to SI engines, would be required. Several control parameters were evaluated, mainly with regards to their influence on the lean limit for operation. The findings from this investigation were published in Paper II. Based on these screenings and on available literature, the main challenges of CNG-diesel dual fuel operation are:

- HC emissions:
  - At light load, unthrottled operation at high λ will result in poor flame propagation and incomplete combustion. Different injection strategies for the diesel fuel are known to affect lean performance. However, how these compare to baseline diesel and standard DDF operation with regards to emissions, efficiency and stability is not fully investigated. Additionally, the potential of in-cylinder air motion to improve the tolerance for lean operation is largely unknown. These topics were investigated, and the results are presented in Papers I and V.
Methane slip from crevices needs to be managed regardless of load. The contribution to total HC emissions from crevices, during lean operation and using methane as fuel, is not known. Furthermore, the effect, on the relative importance of different HC-sources, from varying engine operating conditions, is also unknown. A thorough investigation of methane slip from crevices is presented in Paper IV.

- **Thermal issues**
  - At high load, overheating of the diesel injector tip is an issue which may lead to nozzle hole coking and problems with durability. Currently no work is fund which addresses these issues. The temperatures which occur are unknown, as is the effect that these temperatures have on rate of nozzle hole coking. These related issues were investigated in Papers III and VI.

- **Pre-ignitions & knock**
  - From the first screenings it was found that pre-ignitions impose stricter limits to engine operation than knock. The root cause for pre-ignitions at high compression ratios using methane fuel is unknown. Investigations focused on pre-ignition and knock have been performed since and are presented in this thesis but have not yet been published in other forums.

In addition to the Papers; the current state of each of these phenomena and the contribution from this project to their understanding will be explored in further detail in chapter 6, Results.
5  Experimental setup
This section details the experimental setup used. If no other information is given, all data shown in this publication is recorded using this equipment.

5.1  Engine
The measurements have been carried out on a single cylinder Scania lab engine, available at the Royal Institute of Technology in Stockholm, Sweden. The engine has a displacement of 2 l and is equipped with a high pressure common rail system, Scania XPI, as well as gas injectors from Keihin, placed in the intake runners. A photograph of the engine, showing the installation of the gas injectors, is located in Figure 10.

![Figure 10. Single cylinder engine used in the experiments, the installation of the gas injectors is visible on the left.](image)

The rail-pressure for the methane was maintained at 2.8 bars above intake pressure by an automatic pressure regulator. The piston and piston crown are from Scania’s current line of Euro 5 engines.

Auxiliary systems are in place for heating or cooling of intake air, oil, fuel and water. A supercharged SI-engine runs in an adjacent room, supplying the EGR when desired. When evaluating the data, the stoichiometric EGR from the SI-engine is translated into the equivalent EGR ratio, had the EGR been generated at the current operating conditions.
Table 1. Engine specifications.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement [cm³]</td>
<td>1950</td>
</tr>
<tr>
<td>Bore [mm]</td>
<td>127</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>154</td>
</tr>
<tr>
<td>Connecting rod length [mm]</td>
<td>255</td>
</tr>
<tr>
<td>Geometric compression ratio</td>
<td>17.3</td>
</tr>
<tr>
<td>IVO[°ATDC]</td>
<td>346</td>
</tr>
<tr>
<td>IVC[°ATDC]</td>
<td>-154</td>
</tr>
<tr>
<td>EVO[°ATDC]</td>
<td>145</td>
</tr>
<tr>
<td>EVC[°ATDC]</td>
<td>355</td>
</tr>
</tbody>
</table>

5.2 Control system

Both data acquisition and communication with the test bed were performed, using an in-house software, which runs on a PC and is supported by several PIC-processors. For direct control of the injection system, and indirect control of the test bed through the cell control software, the AVL Rapid Prototyping Controller was used. The Rapid Prototyping Controller is an open source rapid prototyping system for engine control, which is based on Simulink models and executed on dSPACE hardware [43]. The control system can also be simulated offline together with an engine/vehicle model. This enables the developer to easily test, calibrate and verify new concepts and ideas before they are tried on an actual engine. The complete system consists of:

- ECM controller
- Engine/Vehicle model
- dSPACE hardware
- Auxiliary tools to help analyze data and calibrate the engine

Figure 11 shows a screenshot of the Simulink model, when used for offline simulation. The same model is used both for simulation and for online control. The transition and variant management are handled by GUI and scripts, which provide ease of use. For this project, the drive stages built into the Rapid Prototyping Controller handled the Keihin gas injectors, as well as the Scania XPI diesel injector. A rapid self-tuning heat release script was implemented into the controller, which calculates the heat release in real-time and enables automatic cycle-to-cycle control of the combustion phasing [44].
5.3 Fuel
The gaseous fuel used throughout most of the project was chemically pure methane. During the pre-ignition tests biogas according to Swedish standards was used, it consists of 97% +/- 2% methane, the remainder being CO₂. The properties of the biogas are very similar to the chemically pure methane. In addition to the biogas, two tailored gas blends with methane number, MN, 70 and 80 were used. These are detailed in Table 2.

<table>
<thead>
<tr>
<th>Methane number</th>
<th>CH₄ [%]</th>
<th>C₃H₈ [%]</th>
<th>Inert(CO₂, N₂, O₂) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>92.5</td>
<td>7.5</td>
<td>~0</td>
</tr>
<tr>
<td>80</td>
<td>96.4</td>
<td>3.4</td>
<td>~0</td>
</tr>
<tr>
<td>~100</td>
<td>&gt;96</td>
<td>0</td>
<td>&lt;4</td>
</tr>
</tbody>
</table>

Petroleum diesel mixed with 10% RME was used as ignition fuel. During the nozzle coking tests, the ignition fuel was contaminated using zinc neodecanoate, resulting in 1 ppm zinc in the fuel. This is the same contaminant used in the DW10 test [3].

5.4 Lubrication oil
The standard lubrication oil, used for the majority of the testing, was a 10W40 HD long life, based on ACEA E7. During the pre-ignition studies, oil samples of different viscosity and different calcium content were also used, in order to evaluate the effect of these parameters on the frequency of pre-ignition events. The properties of these oils are detailed in Table 3.
Table 3. Oil formulations.

<table>
<thead>
<tr>
<th>Oil no.</th>
<th>Oil</th>
<th>Ca</th>
<th>Zn</th>
<th>Ash [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Scania 10W40</td>
<td>4300</td>
<td>1300</td>
<td>1.9</td>
</tr>
<tr>
<td>2</td>
<td>E4 10W40</td>
<td>4870</td>
<td>1360</td>
<td>1.9</td>
</tr>
<tr>
<td>3</td>
<td>E6 5W30</td>
<td>2330</td>
<td>790</td>
<td>0.99</td>
</tr>
<tr>
<td>4</td>
<td>NA 15W40</td>
<td>0</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>E4 5W30</td>
<td>4920</td>
<td>1340</td>
<td>1.9</td>
</tr>
<tr>
<td>6</td>
<td>E6 10W40</td>
<td>2350</td>
<td>820</td>
<td>0.98</td>
</tr>
<tr>
<td>7</td>
<td>E4 10W40 Repeat</td>
<td>4870</td>
<td>1360</td>
<td>1.9</td>
</tr>
</tbody>
</table>

5.5 Instrumented diesel injector

To measure the temperature in the injector nozzle tip, a set of special DI injectors were manufactured. The injectors were machined to allow for a K-type thermocouple to be inserted close to the tip. The thermocouple placement is shown in Figure 12.

![Figure 12. DI injector tip showing the thermocouple placement.](image)

Efforts were made to place the thermocouple as close to the injector tip as possible. Naturally, the temperature presented is not the actual nozzle tip temperature but the temperature of the tip of the thermocouple. For convenience, the measured temperature will be referred to as the nozzle tip temperature.

5.6 Instrumented glow plug

To investigate the root cause of pre-ignitions, it was desirable to introduce a known hotspot, the temperature of which could be controlled independent of engine operating conditions. For this purpose, two instrumented glow plugs were purchased. These were fitted with a K-type thermocouple, mounted in the first windings of the spiral in the tip of the heater rod. An adapter was made so the glow plugs could be fitted in the location of the flush mounted pressure transducer, and the cylinder head was machined to incorporate a channel mounted pressure transducer instead. Each glow plug was delivered with a calibration sheet, showing the correlation between thermocouple reading and actual surface temperature. Using closed loop feedback and an in-house designed
driver, the surface temperature of the glow plug could then be controlled in the interval 500°C-1060°C.

5.7 Variable valve train
For the tests investigating the effects of in-cylinder air motion, the engine was fitted with cylinder head incorporating a fully variable valve train from Lotus Engineering [10]. The actuators, placed on top of the cylinder head, are shown in Figure 13.

![Figure 13. The four hydraulic actuators of the Lotus AVT system, one actuator for each valve.](image)

The Lotus system allows for different valve lift profiles and hence enables control of the air motion in the cylinder. The cylinder head had special maskings installed in the intake valve seats to increase the maximum swirl number, SN. The SN could then be controlled between 0.4 and the maximum value of approximately 7. This was achieved by varying the lift height or profile of the valves, or by deactivating one valve entirely. The deactivation of one valve typically results in an increase of the SN by a factor 2.

The airflow parameters, swirl and tumble, were evaluated for each valve lift height using measurements performed in a steady-state flow rig. A 1D engine simulation program, GT-POWER v7.1, was then used to integrate the flow over a full intake event and a full valve lift curve. This way, the resulting flow parameters could be calculated for each different valve profile tested in the single cylinder engine. Six sample valve lift curves are shown in Figure 14.
STD denotes standard valve profile and TPZ trapezoid valve profile. The method, used by the 1D simulation tool for calculating swirl, differs slightly from the commonly used Thien-method [45]. See [46], [47] and [48] for further details. The predictions of tumble are more approximative, and the values for tumble number, TN, presented should therefore be interpreted with care. For more detailed information regarding the simulations, the experimental setup and the methodology, for the test using different air motion, see Paper V and [49].

5.8 Heat transfer instrumentation

For the tests when heat transfer was investigated, four thermocouples were installed in the cylinder head. Three of the thermocouples were of a conventional type, installed at approximately 2 mm depth. These were used to record a time averaged steady-state temperature. Two of these were installed in the bridge between the exhaust valves, which is a critical location with regards to thermal loading. A special, rapid response, type-J surface-thermocouple was also used. The surface-thermocouple was flush mounted, allowing direct contact between the hot gases and the surface of the thermocouple. This setup enabled a crank angle resolved measurement of the temperature. The thermocouple had similar properties to the cylinder head, so the measured temperature should be representative of the cylinder head surface temperature.

The engine was also fitted with heat exchangers on the oil and coolant circuit. By controlling the coolant flow through the heat exchangers, a large temperature difference between the coolant entry and exit point could be achieved. The entering coolant was at room temperature. The flow of coolant was measured using a Sartorius balance. The measured flow, multiplied by the temperature difference across the heat exchangers, enables accurate calculation of the steady state heat transfer power.
5.9 Piston modifications

In order to provoke a response and investigate the sensitivity of HC emissions to the volume of the topland crevice, a set of pistons with known increases in topland volume were manufactured. A circumferential cut was made in the topland, as illustrated in Figure 15.

![Figure 15. Top of pistons A-D, illustrating the modifications to the topland.](image)

The cut was made in three depths, on three separate pistons, resulting in the topland crevice volumes listed in Table 4.

<table>
<thead>
<tr>
<th>Piston version</th>
<th>Hot topland volume[cm3]</th>
<th>Topland volume relative to standard[%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A(Standard)</td>
<td>4.1</td>
<td>100</td>
</tr>
<tr>
<td>B</td>
<td>5.75</td>
<td>140</td>
</tr>
<tr>
<td>C</td>
<td>7.65</td>
<td>186</td>
</tr>
<tr>
<td>D</td>
<td>10.5</td>
<td>256</td>
</tr>
<tr>
<td>E</td>
<td>1.83</td>
<td>44</td>
</tr>
</tbody>
</table>

Table 4. Piston modifications.

In addition to the modifications made to the piston topland volume, a fourth piston, piston E, was machined with a 45 degree chamfer around the top edge, as seen in Figure 16.

![Figure 16. Piston E with a 45 degree chamfer.](image)
The amount of material removed from piston E was sufficient to lower the compression ratio of the engine from 17.3 to approximately 16. None of the other modifications were large enough to significantly affect the compression ratio. This is shown in Figure 17.

![Figure 16. Top of piston E, illustrating the modifications to the topland.](image)

**Figure 16. Top of piston E, illustrating the modifications to the topland.**

**Figure 17. Motored cylinder pressure versus crank angle for different topland crevice volumes.**

### 5.10 Data acquisition

The EGR level and the emissions of NOₓ, HC and CO were measured using a Horiba EXSA-1500 exhaust analyzer. Smoke was measured with an AVL Micro Soot Sensor and an AVL 415S smoke meter. For fuel metering, an AVL 733 fuel balance was used for the liquid fuel and an Alicat Scientific 500 SLPM-type gas flow meter for the methane. The pressure history was recorded at 0.1
CA increments, using AVL QC32D and AVL GH12D pressure transducers and a Kistler 5011 charge amplifier.

5.10.1 Crank angle resolved measurements of hydrocarbon emissions
Crank angle resolved hydrocarbon emissions were acquired using a Cambustion HFR400 Fast FID. The sample point was located in the exhaust runner, 15 cm downstream of the exhaust valves. A typical output from the FastFID, including error bars corresponding to one standard deviation, is shown in Figure 18.

![Figure 18. HC emissions versus crank angle, averaged over 100 cycles, sampled in the exhaust runner 15 cm downstream of valves.](image)

The appearance of the curve differs depending on the sample point. When sampling close to the exhaust valve, the measurements are sensitive to the radial placement of the probe [50], [51], but at distances in excess of approximately 5.5 cm no radial gradients remain, [52]. The output from the FastFID also depends on the axial placement of the probe. The measurements presented in this publication have a similar appearance to measurements acquired, at similar distance from the exhaust valves, by other authors [53], [52], [54].

The FastFID is equipped with a constant pressure chamber, which ideally removes the influence of changes in exhaust pressure. However, rapid pressure fluctuations can still affect the measurements. This is seen between 450 and 700°CA in Figure 18; there are standing pressure waves in the exhaust pipe, which cause matching oscillations in the measured concentration of HC.

While the utilization of fast hydrocarbon analyzers has been described by several authors, there is not complete consensus with regards to how the results should be interpreted. There are three distinct peaks seen on the curve. Suggestions for the interpretation of peak 1:

- HC from end of previous cycle residing in the port, [55], [54], [56], [57].
- HC from quench layers and crevices close to exhaust valve, [55], [53], [52], [58], [54], [59], [60], [61].

Considering the pressure sensitivity of the FastFID, a plausible third explanation for the first peak can be postulated: a combination of HC from the previous cycle, intensified by the pressure increase from the blowdown pulse.

The second peak is more controversial; some authors claim that it is influenced by changes to the ring pack [55]. According to this reasoning, the second peak would be a result of turbulent release of HC trapped in second land. However, measurements carried out during the work presented in Paper IV, where the ring gap of the second piston ring was increased, show no influence on the second peak, making this theory unlikely.

Others authors postulate that the reverse flow, which can occur immediately following the blowdown pulse, causes some of the HC from peak 1 to be measured twice [53], [52], [61]. This theory has further support as [56] show a correlation between the height of the second peak and the level of HC present in the port before EVO.

Regarding the third peak, there seems to be a general agreement that it is the result of HC from piston crevices. These hydrocarbons have been deposited along the liner, by the downward motion of the piston during the expansion stroke, and are subsequently collected again in a vortex at the edge of the piston during the exhaust stroke [41] [55]. This is also supported by the tests with increased second ring ring-gap, which have been previously mentioned. Increasing the ring-gap, causing HC in second land to flow into the crank case to a greater extent, instead of back into the combustion chamber, caused the third peak to decrease in magnitude.

Based on the above reasoning, the general conclusion regarding crank angle resolved HC measurements is that HC expelled late during the exhaust stroke can be attributed to piston crevices, and HC expelled during the early part of the exhaust stroke to other sources.
6 Results

The results chapter is organized according to Figure 19, where the challenges of dual fuel operation, based on the screenings presented in Papers I and II, are shown. Each challenge is addressed in a separate section.

![Figure 19. Challenges of dual fuel operation.](image)

6.1 NO\textsubscript{x} & HC emissions

This section concerns NO\textsubscript{x} and HC emissions, which constitute an issue regardless of engine load. The challenge is greatest at light load, where temperatures in the combustion chamber and of the exhaust gases are lower. This increases the levels of HC, and also makes aftertreatment of both NO\textsubscript{x} and HC more difficult. At high load, even though the levels of HC are lower, they are still problematic, considering the challenge of oxidizing methane in a catalyst. In recent years, with the arrival of EPA10 and Euro 6 emissions standards, lean NO\textsubscript{x} aftertreatment has received a lot of attention, and great advances have been made. Considering the extensive experience available for aftertreatment, NO\textsubscript{x} emissions are regarded to constitute less of an issue than HC and will be treated in much less depth.

6.1.1 NO\textsubscript{x} emissions

In Paper II, it is shown that the way NO\textsubscript{x} is formed during DDF combustion varies with the E\% CH\textsubscript{4}. At high rates of methane utilization, the mechanism is similar to that which occurs in SI engines. However, at low rates of methane utilization and large diesel injections, NO\textsubscript{x} is formed the
same way as in diesel engines. NO\textsubscript{x} formation is dependent on high temperature and availability of oxygen. For this reason, in SI engines, a maximum occurs slightly lean of stoichiometry, around $\lambda \approx 1.1$ [1]. NO\textsubscript{x} is mainly formed in regions of the combustion chamber where the highest temperatures occur for the longest time. In the SI engine, this corresponds to the volume close to the spark plug, which burns first and is heated further as it is compressed by combustion of the remainder of the charge [1]. In the DDF engine, this would correspond to the pilot region in the piston bowl. As the pilot amount is increased, several mechanisms can be expected to affect the NO\textsubscript{x} formation. The larger injection will cause increased mixing between the pilot zone and the bulk gas; this may decrease local temperatures. Also, as the diesel amount is increased, the local availability of oxygen in the pilot zone can be expected to decrease, thus reducing the amount of NO\textsubscript{x} formed. A large reduction in NO\textsubscript{x}, as the pilot amount increases, is visible in Figure 20.

![Figure 20. NO\textsubscript{x} and HC emissions versus $\lambda$ for three different diesel substitution rates.](image)

At high E% CH\textsubscript{4}, as $\lambda$ increases, the temperature drops and NO\textsubscript{x} emissions decrease quickly. NO\textsubscript{x} emissions for the 70% CH\textsubscript{4} case remain relatively flat as $\lambda$ changes. The reason for this is that NO\textsubscript{x} formation during diesel combustion occurs in regions close to stoichiometry, in the edge of the flame plume, this mechanism is therefore less sensitive to variations in global $\lambda$. It is seen that the crossover occurs at approximately $\lambda = 1.6$. Below this $\lambda$, the SI mechanism dominates the NO\textsubscript{x} production, while above this $\lambda$ the diesel mechanism is most influential. If low $\lambda$ operation is necessary, for instance during transients, then the diesel amount should be increased slightly to avoid the peak in NO\textsubscript{x} production at $\lambda = 1.2$. For an extended discussion on NO\textsubscript{x} formation during dual fuel combustion, refer to Paper II.

### 6.1.2 HC emissions

HC emissions are problematic throughout the engine operating range. This is also seen in Figure 20, where the HC emissions are shown as a function of $\lambda$ and E% CH\textsubscript{4}. Also included in the figure, is a hypothetical limit of 4 g/kWh for engine-out HC emissions. This limit is based on a legislated limit.
of 0.5 g/kWh for tailpipe emissions, and a conversion efficiency of 90% for a commercial oxidation catalyst [62]. In order to stay below the limit of 4 g/kWh, the option is to either run at relatively low $\lambda$, or to reduce the diesel substitution rate; neither of which is desirable from an economic point of view. For this reason, an investigation into the root causes of these emissions was carried out. The objective was to determine the main sources, their relative importance and how said importance changes with operating conditions. The full investigation is found in Paper IV. From literature it is known that combustion chamber crevices, mainly the topland volume, contribute significantly to the HC emissions. How large this contribution is, has been investigated for stoichiometric SI engines [41], [63], [64]. However, for lean operation with methane fuel, no studies were found. The effect of parameter variations has also received insufficient attention.

In order to provoke a response and investigate the sensitivity of HC emissions to the volume of the topland crevice, a set of pistons with known increases in topland volume were manufactured. The piston modifications are detailed in chapter 5, Experimental setup.

The test plan consists of sweeps in $\lambda$, combustion phasing, diesel substitution rate, coolant temperature and load, for each of the pistons. The baseline settings are outlined in Table 5, and unless any other information is given, those are the conditions that apply.

![Table 5. Baseline settings.](image)

### 6.1.2.1 Influence of $\lambda$ on sources of HC emissions

The HC emissions, as a function of $\lambda$ and topland crevice volume, are shown in Figure 21.
For $\lambda$ between 1 and 1.2, only very small differences in HC emissions can be discerned between the different pistons A-D. However, as the mixture becomes more dilute, differences arise. At $\lambda$ between 1.4 and 2.1, a trend is seen that a larger crevice volume results in higher HC emissions. One exception to this trend is displayed by piston D, the largest crevice volume, at $\lambda = 1.4$, which has similar HC emissions to piston C. This same behavior is also present for $\lambda$-sweeps at 7.5 bar IMEP.

At low $\lambda$ the combustion is very rapid, causing a sharp gradient in pressure and temperature in the combustion chamber. From the ideal gas law, the mass flow into the crevice volume can be calculated. Additionally, the maximum, choked, mass flow through the narrow slit between piston and cylinder liner can also be calculated. By comparing these two, it is seen that the mass flow into the crevice becomes choked for certain operating points. At $\lambda = 1$ and $\lambda = 1.2$ this occurs for pistons B, C and D, and for $\lambda = 1.4$ this occurs for piston D. For this reason, the filling of the crevice is not completed when the flame arrives, and burned products are forced into the crevice along with the fresh charge. Since the flame arrives at the same time for all pistons, and the remainder of the crevice is filled with burned products, the mass of unburned charge in the crevice is independent of crevice volume. Since the method of increasing the crevice volume, to investigate the contribution from crevices to engine-out HC, relies on the assumption that the crevices are in pressure equilibrium with the combustion chamber, the method is not valid for the operating points where choking occurs. To avoid the problem of restricted flow into the crevice, and to enable investigation of the sensitivity of HC emissions to crevices at low $\lambda$; a piston with 45° chamfer of the edge was machined. This way, the crevice volume was decreased instead. The chamfer removed approximately 56% of the topland crevice.

If the data from Figure 21 is normalized to the standard piston and straight lines are fitted to the data from each piston and extrapolated to a theoretical crevice volume of 0, the results can be seen in Figure 22. The data where choked flow occurred is not included.
According to this data, the topland crevice accounts for approximately 70% of total HC emissions for $\lambda = 1.4$ and $\lambda = 1.6$. For $\lambda = 1.9$ and 2.1, the contribution from the topland crevice is 40% and 20% respectively. If the contributions to HC emissions from the topland crevice, acquired from the linear fit in Figure 22, are plotted as a function of $\lambda$, the result can be seen in Figure 23 along with CoV for the different cases. The data points at $\lambda = 1$ and $\lambda = 1.2$ are generated in a similar way, by fitting a line to the data from piston E and the standard piston, piston A.

![Figure 22](image_url)

**Figure 22.** HC emissions normalized to standard piston versus topland crevice volume at four different $\lambda$ and extrapolated to a theoretical topland volume of zero.

![Figure 23](image_url)

**Figure 23.** Fraction of HC emissions from topland crevice and CoV versus $\lambda$. 

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At $\lambda < 1.2$, crevices contribute approximately 50% of HC emissions at 10.5 bar IMEP and 35% at 7.5 bar IMEP. The results are in agreement with what is presented in [63]. As conditions become leaner, the contribution from crevices increases to a maximum of 70% of engine-out HC emissions, at $\lambda = 1.65$ for 10.5 bar IMEP. As the amount of excess air increases, combustion quality deteriorates, as illustrated by CoV of IMEP in Figure 23, and crevices are no longer the dominating source of HC emissions. CoV of IMEP is an average between pistons A-D for each $\lambda$. The difference between 7.5 and 10.5 bar IMEP is that combustion quality improves slightly at the higher load; flame extinction in the bulk gas is reduced, and the fraction HC emissions from the topland crevice consequently increases.

6.1.2.2 Influence of combustion phasing on sources of HC emissions
The influence of combustion phasing on the origins of HC emissions is illustrated in Figure 24.

As combustion phasing is retarded, two mechanisms can be expected to influence the HC emissions:

- Peak combustion temperatures are decreased and combustion quality deteriorates. The contribution to HC emissions from wall quenching and incomplete combustion can be expected to increase.
- Temperatures during the later part of the expansion increases, and hydrocarbons returning from crevices are expected to oxidize to a greater extent.

These factors combined, give the result shown in Figure 24. It appears that, while total HC emissions remain relatively constant, the origin of HC emissions has shifted from crevices to quenching. In order to confirm this reasoning, and to separate HC from the two main sources,
crevices and quenching, a fast FID was used, which provides crank angle resolved measurements of HC emissions. Measurements from the fast FID are shown in Figure 25 for early and late combustion.

![Figure 25. HC emissions versus crank angle for early and late combustion using Piston C. The blue line corresponds to early combustion and the grey line to late combustion.](image)

As discussed in the section regarding FastFID measurements in chapter 5, emissions during the late part of the exhaust stroke correspond to crevice hydrocarbons, deposited along the liner during expansion and then collected by the piston in a ‘scrape-up-vortex’. In this case, early combustion yields significantly higher emissions during the late part of the exhaust stroke, compared to late combustion, which indicates more HC from crevices. The same results are also seen for pistons A and B.

The conclusion with regards to combustion phasing is that at lean conditions, where combustion is sensitive to changes in temperature, a large shift in the origin of the emissions, from crevices to quenching, occurs as the combustion phasing is retarded.

### 6.1.2.3 Influence of E% CH\textsubscript{4} on sources of HC emissions

The influence of diesel substitution rate, on HC emissions is illustrated in Figure 26.
As shown, the HC emissions increase with E% CH₄. However, the relative importance of the topland crevice is reduced. It contributes almost 70% of HC emissions at 50% CH₄, but just above 50% at 95% CH₄. This trend can be explained by considering the improved combustion in the bulk gas due to the large diesel injection. A larger diesel pilot results in less HC from flame extinction, and hence a larger fraction from crevices.

6.1.2.4 Influence of swirl on sources of HC emissions

Calculation of the amount of HC in the topland crevice during the cycle, show that methane returning from the crevice during the expansion escapes oxidation. Closed cycle simulations, based on the ideal gas law, and measured HC concentrations from the exhaust, indicate that the average temperature in the cylinder when oxidation of crevice-gas stops is approximately 1600 K. However, this temperature should be sufficient to rapidly oxidize methane. This contradiction can be explained if the methane returning from crevices is deposited in the boundary layer, along the cylinder liner, by the downward motion of the piston. This mechanism is possible since the speed of the outflow of gas from the crevice is lower than the piston speed, as shown in [41]. Based on this reasoning, it should be possible to increase oxidation of unburned methane returning from the crevices, if mixing is improved. To investigate this hypothesis, pistons A and C were used, together with a high swirl cylinder head and a fully variable valve train. This setup made it possible to vary the swirl and tumble over a wide range. The equipment is described further in chapter 5, Experimental setup, in section 6.2.2 regarding in-cylinder air motion and also in Paper V. The results of these tests with regards to HC emissions, relative to the standard piston, are seen in Figure 27.
Figure 27. HC emissions for piston C relative to HC emissions from the standard piston versus swirl and tumble at $\lambda = 1.7$. The topland volume ratio of piston C relative to the standard piston was 186%. The numbers on the surface show the actual value for each case.

Figure 27 shows the HC emissions for piston C as a percentage of the emissions of piston A. The relative increase in emissions is greatest at low SN. This is also where the highest absolute levels of HC emissions are found. It appears that swirl is helpful in stripping this thin layer of returning hydrocarbons from the liner and mixing them with the bulk gas, thus improving oxidation. Based on these results, a SN of approximately 3 is sufficient, and no added benefit with regards to crevice HC is gained by increasing SN above this level. Additional information regarding the effect of swirl and tumble is found in Paper V.

6.1.3 Conclusions regarding NO$_x$ & HC emissions

An experimental investigation into the sources of HC emissions during DDF operation has been carried out. NO$_x$ formation is also discussed based on experimental data. The main conclusions from this section are:

NO$_x$ formation is mostly dependent on the combustion of the diesel pilot. With the exception of low $\lambda$ operation, it is beneficial to employ smallest possible pilot injection to keep NO$_x$ levels to a minimum. Since efficient NO$_x$ aftertreatment has become readily available, NO$_x$ is considered less of a problem than HC.

Figure 28 shows the HC emissions for the $\lambda$-sweep at 10.5 bar IMEP, using the standard piston.
Figure 28. HC emissions versus λ, illustration shows sources of increase in emissions.

The total HC emissions are broken down to show the contribution from the topland crevice and from other sources, mainly quenching. It is seen that in order to meet a limit of 4 g/kWh, in the λ-range between 1.4 and approximately 1.8, the topland crevice is the first priority, and efforts must be made to reduce it in volume and to improve oxidation of returning hydrocarbons.

At λ > 1.8, quenching becomes the main source of HC, and combustion efficiency must be improved, either by increasing the amount of diesel, or by promoting flame propagation in other ways. Methods for improving the combustion efficiency in the bulk gas at high λ are discussed further in section 6.2. For additional information on NOx formation, refer to Paper II. For full details on the investigations of crevice hydrocarbons and in-cylinder air motion, refer to Papers IV and V respectively.

6.2 Throttling and incomplete combustion

At light load, the challenge of dual fuel operation consists largely of managing the conflicting needs to keep HC emissions and exhaust temperature within the limits of the aftertreatment system, with the desire to operate at high diesel substitution rates and to keep throttling at a minimum to improve brake efficiency. To minimize throttling it is desired to operate the engine as lean as possible. As the mixture becomes more dilute, poor combustion efficiency and flame quenching lead to high emissions of HC.

Two main pathways to increase performance during lean operation have been investigated: using advanced diesel injection strategies to achieve unconventional combustion modes and using in-cylinder air motion, swirl and tumble, to promote efficient combustion. The information regarding unconventional injection strategies is mostly a summary of the results in Paper I, while the section on in-cylinder air motion is based on Paper V.
6.2.1 The effect of advanced injection strategies

Unconventional combustion modes such as RCCI, HCCI and PPCI have received a lot of attention during the past decade. They are commonly referred to as low temperature combustion, LTC, concepts. From a combustion point of view, the results are encouraging, but the enthusiasm has, with a few notable exceptions, not resulted in any production engines. The main obstacles to realize these combustion modes can be summarized into two categories: the difficulty to obtain information about combustion phasing and the difficulty to control it, once information is made available.

Information about combustion phasing can be gained by several methods: ion current measurements, fast torque measurements, engine speed variations and cylinder pressure transducers to name a few. Currently, only cylinder pressure measurements are demonstrated to give information of sufficient quality for combustion phasing close to TDC. However, cylinder pressure transducers have not yet made their way into production heavy duty engines. At the time of writing, this problem remains to be solved.

Since combustion phasing and combustion speed during LTC combustion are governed to a large extent by chemical kinetics, the methods available for control differ from the SI and the CI engine. Several control methods have been proposed: fast control of inlet temperature, fast compression ratio changes, fast EGR adjustments, to name a few. In addition to these methods, it is also possible to mix two fuels, and in this way create a compound fuel with a cetane number tailored for the current operating conditions. This method has recently been referred to as fuel reactivity controlled compression ignition, RCCI. This was suggested for DME and methane in [65]. The main difference, between RCCI operation and HCCI with two fuels, is that one fuel is direct injected during RCCI operation while both fuels are port injected during HCCI operation to achieve homogenous mixing. This means that, for RCCI, the reactivity can be influenced by in-cylinder stratification, in addition to the combined cetane number. In the dual fuel engine, a direct injected high cetane fuel and a port injected low cetane fuel is readily available on board the vehicle, making this approach very suitable.

To approach the subject of unconventional combustion modes, a known phenomenon for DDF operation is shown in Figure 29. The graph shows rate of heat release versus crank angle for different injections of the diesel pilot.
Figure 29. Rate of heat release versus crank angle for different start of energizing.

As the injection timing for the pilot injection is advanced, the combustion advances. However, once a certain limiting value is reached, further advancement of the injection timing will instead cause the combustion to retard. The DI injector does not allow for a needle lift sensor, for this reason, start of injection is unknown, and start of injector coil energizing, SOE, is used instead. Actual, hydraulic, SOI occurs approximately 4° CA after SOE at this engine speed. Detailed settings for the operating point are located in Table 6.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Setting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>1400 RPM</td>
</tr>
<tr>
<td>Common Rail Pressure</td>
<td>1200 bar</td>
</tr>
<tr>
<td>Diesel amount</td>
<td>8 mg/comb</td>
</tr>
<tr>
<td>Methane amount</td>
<td>59 mg/comb</td>
</tr>
<tr>
<td>Diesel substitution rate</td>
<td>~90 %</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>~1.65</td>
</tr>
</tbody>
</table>

In this case, the limiting value for SOE is between -25° and -32.5° ATDC. The likely reason for this behavior is that, when the injection timing is sufficiently advanced, the spray begins to miss the
piston bowl and becomes increasingly dispersed in the combustion chamber. For the current geometry of the piston bowl, and an injector umbrella angle of 146°, this is predicted to happen for injection angles of approximately -30° ATDC, which corresponds well with the observed behavior. As the pilot spray is increasingly dispersed in the combustion chamber, the ignition source weakens, and the combustion mode changes. A smoother more homogenous heat release is achieved. The corresponding emissions data and CoV of IMEP are located in Figure 30. SOE of -11.5° corresponds roughly to MBT combustion phasing and CA50 at 8° ATDC, and it is therefore considered the reference case.

Advancing SOE initially leads to advanced combustion phasing and higher peak temperatures. This in turn explains the reduction in HC and CO emissions and the increase in NOx. As SOE is advanced beyond -25° ATDC, and the diesel spray becomes more and more dispersed, smoke and NOx are reduced simultaneously without any penalty to HC or CO. However, at SOE of -40° ATDC, combustion stability deteriorates and emissions of HC and CO increases due to complete or partial misfire. From this introduction, it is seen that allowing better mixing between the diesel pilot and the methane has the potential to reduce NOx, CO, HC and smoke simultaneously, if combustion phasing and combustion stability can be controlled. Two methods for achieving this are proposed: using a single injection, while varying the diesel substitution rate to tailor the cetane number, or adding a second injection which controls the combustion phasing. The first method will be referred to as Reactivity Controlled Compression Ignition, RCCI, the second one as Partially Premixed Compression Ignition, PPCI.

Figure 30. Emissions of HC, CO, NOx, Smoke and CoV of IMEP versus start of energizing.
6.2.1.1 RCCI/HCCI

As the injection timing in Figure 30 is advanced to -40° ATDC, near zero emissions of NOx and smoke are achieved, but HC and CO emissions deteriorate along with combustion stability. This can be addressed by increasing the size of the diesel injection and advancing it even further, to allow longer time for mixing. This results in RCCI combustion with low HC and CO emissions, satisfactory combustion stability and near zero emissions of NOx and smoke.

In RCCI combustion, the combustion timing and speed are governed mainly by chemical kinetics and to some extent by in-cylinder stratification of the direct injected fuel. Combustion is therefore very sensitive to the small changes in local temperature, which occur stochastically in combustion engines. There is also a strong possibility of positive feedback; if combustion occurs early one cycle, more heat is transferred to the combustion chamber walls, thus heating valves and piston crown. Because of this, the gases during the next cycle are exposed to higher temperatures, and the probability increases that combustion will advance even further. This situation can quickly escalate and, within fractions of a second, lead to intolerable peak firing pressures and rates of pressure rise. This behavior is aggravated as the load increases, and it is highly unlikely that RCCI combustion can be controlled without closed loop cylinder pressure feedback.

In the DDF engine, RCCI combustion phasing is controlled by varying the ratio between the high cetane fuel, diesel, and the low cetane fuel, methane. More diesel for a given load advances the combustion phasing and more methane delays combustion. Rate of heat release for 4 cycles of RCCI combustion using methane-diesel dual fuelling is shown in Figure 31.

![Figure 31. Rate of heat release versus crank angle for RCCI combustion, 6.5 bar BMEP, 1400 RPM and 70% methane.](image)

At around -20° ATDC some cool flame activity can be observed. Depending on pressure and temperature of the operating point, the main heat release may be preceded by cool flame oxidation of parts of the diesel fuel. During this phase, radicals and CO are formed which accelerate the main
heat release. For this load point the combustion is relatively stable though no closed loop feedback was used. Maximum rate of heat release varies by approximately 10%. More on the performance of RCCI combustion, in relation to diesel, DDF and PPCI, is found in section 6.2.1.3.

6.2.1.2 PPCI
To accomplish PPCI, the settings used to generate Figure 30 are maintained, but the pilot injection is split into two. An early injection of 6 mg with SOE at -40° ATDC and a late injection of 2 mg with varied SOE are used instead of a single, 8 mg injection. The results are shown in Figure 32.

![Figure 32. Rate of heat release versus crank angle for different start of energizing for the second injection.](image)

It is clear that control over combustion phasing is now exercised by the timing of the second injection, and that the heat release is much more rapid than what is shown in Figure 29 for single injection and SOE at -40° ATDC. Advancing the combustion phasing by 8°CA increases the maximum rate of heat release, from approximately 150 J°CA to 250 J°CA. This large increase indicates that the combustion is governed by chemical kinetics to a larger extent than is typical for flame propagation and that a shift in combustion mode has occurred. The emissions results from the sweep in combustion phasing are illustrated in Figure 33.
Figure 33. Emissions of HC, CO, NO\textsubscript{x}, Smoke and CoV of IMEP versus start of energizing for the second injection.

No significant trend can be seen with regards to CO, Smoke and CoV. This indicates that the combustion is relatively robust. HC emissions increase with retarded combustion phasing and NO\textsubscript{x} emissions decrease. For comparison, rate of heat release for two cases with similar combustion phasing, close to MBT timing, where one case uses single injection and one case uses split injection, is shown in Figure 34. For the single injection SOE is at -11.5° ATDC, and for the double injection SOE is at -12.5° ATDC.
The initial, premixed part of the heat release corresponds to the late diesel pilot injection. This part of the heat release is reduced when the injection is split, since 75 percent of the diesel is removed and instead injected in the first part of the injection. The early injection is given much more time for premixing, becomes much more diluted and is therefore combusted along with the methane in the smooth main part of the heat release, which has increased in magnitude compared to single injection. The reduction of the premixed peak at -1° ATDC removes the ringing visible in the late part of the heat release. The emissions from these operating points are shown in Figure 35.
Allocating most of the diesel to the early injection, and allowing it to better mix with the methane, helps the combustion of the lean mixture. This is the reason for the decrease in HC, CO and CoV. Since the entire injection takes place during the ignition delay, both for single and split injection, the combustion is premixed. Hence, reducing the size of the late part of the pilot injection means that the premixed combustion of the mixture, in the pilot region in the piston bowl, will occur at leaner conditions. This reduces the amount of NO\textsubscript{x} and smoke that is formed in this region of the combustion chamber. For these reasons, it can be said that splitting the pilot injection in two decreases $\lambda$ in the bulk where problems with lean mixture exists, and increases $\lambda$ in the pilot region where problems occur because of overly rich mixture. The data presented in Figure 32 implies that the early diesel injection causes the main combustion to change from flame propagation into a premixed combustion controlled by chemical kinetics. The combustion is relatively robust and can be controlled by the timing of the second part of the pilot. If the diesel injection system provides enough freedom to split the pilot injection, there appears to be no downside to using it. The benefits of unconventional combustion modes are even greater at leaner conditions; this is demonstrated in the following section.

### 6.2.1.3 RCCI/HCCI and PPCI at highly diluted conditions

In this section, RCCI and PPCI combustion are compared to diesel and DDF combustion for the operating point defined in Table 7. The conditions are derived from a production Scania diesel engine, operating at 25% load.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed [RPM]</td>
<td>1400</td>
</tr>
<tr>
<td>BMEP [bar]</td>
<td>6.5 bar</td>
</tr>
<tr>
<td>Common Rail Pressure [bar]</td>
<td>2500</td>
</tr>
<tr>
<td>Inlet pressure, absolute [bar]</td>
<td>1.4</td>
</tr>
<tr>
<td>Exhaust pressure, absolute [bar]</td>
<td>1.6</td>
</tr>
<tr>
<td>Inlet temperature [°C]</td>
<td>30</td>
</tr>
<tr>
<td>Diesel substitution rate [E%]</td>
<td>0-90 %</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>~2.7</td>
</tr>
</tbody>
</table>

In steady state at light load, the diesel engine operates at very lean conditions. When attempting to replace diesel with methane under these conditions, the result is poor to nonexistent flame propagation and high emissions of HC from the regions of the combustion chamber not in contact with the diesel spray. This is shown in Figure 36.
Figure 36. Emissions of HC and CO versus diesel substitution rate for different combustion modes at $\lambda = 2.7$. Blue color on lines and markers correspond to HC emissions while grey correspond to CO.

As the diesel substitution rate is increased above 60% and the diesel pilot is reduced in size, methane which was previously partly oxidized into CO now escapes combustion entirely and exits the engine as HC. RCCI and PPCI combustion reduces both HC and CO emissions by ~50% compared to DDF operation at the corresponding diesel substitution rate. It is likely that the bulk of the remaining emissions can be attributed to crevice losses. They are therefore difficult to address without changing the geometry of the piston and the piston-ring assembly. The NO$_x$ emissions are located in Figure 37.

Figure 37. Emissions of NO$_x$ versus diesel substitution rate for different combustion modes at $\lambda = 2.7$. 
For DDF combustion, the NO\textsubscript{x} emissions decrease steadily as the diesel pilot becomes smaller. This is discussed previously in section 6.1.1. For \(\lambda\) larger than 1.6, the diesel pilot zone is the dominating source of NO\textsubscript{x} emissions. This is especially true for these highly diluted conditions. This case is further strengthened since the PPCI combustion, where a late diesel pilot is utilized, shows similar NO\textsubscript{x} emissions as DDF combustion while RCCI combustion, where the diesel is dispersed and highly mixed with the air and methane, results in near zero NO\textsubscript{x}. The results from these tests, with regards to efficiency, are summarized in Figure 38.

![Figure 38](image)

**Figure 38.** Combustion efficiency and brake efficiency versus diesel substitution rate for different combustion modes at \(\lambda = 2.7\). Blue color on lines and markers corresponds to combustion efficiency while grey corresponds to brake efficiency.

From these data, it is shown that PPCI and RCCI combustion allows the DDF engine to operate unthrottled, at diesel-like conditions, without any penalty to combustion efficiency, except for what is mandated by crevice losses and wall quenching. The rapid combustion compensates for this, and brake efficiency for RCCI and PPCI combustion is similar to base diesel operation. The issue of control of combustion remains, but it is reasonable that at least PPCI, if not RCCI, can be mapped and utilized at light load, without closed loop feedback. For additional information on unconventional combustion modes, refer to Paper I.

### 6.2.2 The effect of in-cylinder air motion

In-cylinder air motion is currently used, in both SI and CI engines, to promote efficient combustion and reduce emissions formation. Complex in-cylinder flows are commonly reduced to two circular bulk motions, swirl and tumble.

Swirl can be measured in swirl number, SN, and tumble in tumble number, TN [49]. As the compression stroke progresses, the geometry of the in-cylinder volume flattens close to TDC, causing the tumble to break down into small scale turbulence, which is known to improve combustion speed in SI engines [66] [67].
The swirl is forced into the smaller radius of the piston bowl and conservation of momentum causes the speed to increase close to TDC. The swirl survives both compression and combustion and is used in diesel engines, mainly to promote after-oxidation of soot [68] [69]. For this reason, port designs which enable swirl on diesel engines have historically been of great importance [70].

At high E% CH₄ in a dual fuel engine, combustion consists mainly of flame propagation and can therefore be expected to respond to changes in tumble motion. As with all premixed engines, air motion will affect the mixing between methane and air. In addition, both the combustion speed and the emissions formation are greatly affected by the distribution of the diesel pilot in the combustion chamber. This in turn, is affected by the swirl number. Hence both swirl and tumble can be expected to influence dual fuel combustion, in ways that are difficult to anticipate.

For an investigation into the effects on swirl and tumble on dual fuel combustion, a variable valve train from Lotus Engineering, Lotus AVT, an additional piston with enlarged topland volume and a cylinder head instrumented with thermocouples were used. The equipment and methodology are detailed in chapter 5, Experimental setup. For a full description of the investigation, refer to Paper V.

6.2.2.1 Test matrix
The aforementioned equipment was used to create 32 cases of varying swirl and tumble; these are shown in Figure 39.

![Figure 39. 32 cases with varying swirl and tumble. The two-valve cases are shown in blue and the cases utilizing only one valve in grey.](image)

Three different valve strategies were used: standard profile (STD), trapezoid profile (TPZ) and deactivation of one valve. Deactivation was used in combination with both STD and TPZ valve profiles. Six cases which will be examined in further detail are marked in Figure 39.
These 32 cases were run for three different operating conditions, shown in Table 8.

Table 8. Details of operating points 1, 2 and 3.

<table>
<thead>
<tr>
<th>Operating point</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed [RPM]</td>
<td>1225</td>
<td>1225</td>
<td>1225</td>
</tr>
<tr>
<td>IMEP [bar]</td>
<td>7</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>Global λ</td>
<td>1.7</td>
<td>1.7</td>
<td>1.9</td>
</tr>
<tr>
<td>E% CH4 [%]</td>
<td>95</td>
<td>95</td>
<td>95</td>
</tr>
<tr>
<td>Piston</td>
<td>A</td>
<td>C (186%)</td>
<td>A</td>
</tr>
<tr>
<td>Common rail pressure [bar]</td>
<td>1200</td>
<td>1200</td>
<td>1200</td>
</tr>
<tr>
<td>SOE [°BTDC]</td>
<td>10</td>
<td>10</td>
<td>12</td>
</tr>
<tr>
<td>CH₄ SOI [°BTDC]</td>
<td>350</td>
<td>350</td>
<td>350</td>
</tr>
<tr>
<td>Intake pressure, absolute [bar]</td>
<td>1-1.4</td>
<td>1-1.4</td>
<td>1.2-1.7</td>
</tr>
<tr>
<td>Intake temperature [°C]</td>
<td>30</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Exhaust pressure, absolute [bar]</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

The operating point, OP, at λ = 1.7 is chosen since it is close to this λ that HC emissions start to increase sharply, and it is a realistic goal to bring the emissions down below the 4 g/kWh limit, shown in Figure 20. Since the injector lacks needle lift sensor, start of injection is unknown and start of injector coil energizing, SOE, is used instead. Actual, hydraulic, SOI is delayed by approximately 3.5°CA at this engine speed. The SOE is fixed for each operating point and is chosen so that it results in MBT combustion phasing for the standard valve profiles and 15 mm lift, which corresponds to using the standard fixed cam. The intake pressure was varied to maintain constant air mass, regardless of valve lift profile.

6.2.2.2 Heat transfer as a function of swirl and tumble
The heat transfer to the combustion chamber walls was measured in several ways, as described in chapter 5, Experimental setup. The results from two of these methods, for a subset of the 32 test point matrix, are shown in Figure 40 for operating point 1.
The left graph of Figure 40 shows the power of the heat transfer to the coolant, as a function of swirl and tumble, and the right graph shows the temperature in the exhaust valve bridge. The trends match regardless of method used. It is also seen that heat transfer scales with swirl but is less sensitive to tumble. This is to be expected, since most of the heat transfer occurs during and after combustion. At this point, the tumble motion has been broken down into small scale turbulence and dissipated, while the swirl remains. Figure 41 shows rate of heat release, ROHR, and surface temperature as a function of crank angle for two cases. Both cases are marked in Figure 39. Case 1 is at medium SN of ~2, while case 2 is at maximum SN of ~7.
The first part of the heat release corresponds to the rapid combustion of the pilot fuel and some CH₄, which has been entrained in the pilot zone. The second part of the heat release corresponds to the flame propagation through the CH₄ and air mixture.

The heat release is more intense for Case 1. However, despite this fact, the heat swing, the surface temperature increase during combustion, is greater for Case 2. The temperature increase between -40° and 20° ATDC is 15°C, for case 2. This is a 30%, larger temperature increase compared to case 1. From this it is shown that it is the gas motion in itself, and not changes in the combustion rate, that drives the increase in heat transfer at high SN.

6.2.2.3 Emissions and combustion quality as a function of swirl and tumble at λ = 1.7
In Figure 42, soot emissions at λ = 1.7 are shown as a function of swirl and tumble.
Figure 42. Soot emissions versus swirl and tumble at $\lambda = 1.7$. The numbers show measured value for each case.

To better see the macro trends, and to reduce the influence of measurement noise, a smoothing spline function was applied when fitting the surface to the measurements. This was done for all the surface plots shown, and it means that the surface does not necessarily intersect the actual measurements in each point. The numbers in the figure correspond to the actual measurements. Swirl has a very strong effect on soot. As the SN was increased from 0.5 to 7, soot emissions decreased from 10 to 1.3 mg/kWh; a reduction of approximately 85%. All operating points, except for the one in the bottom left corner, are well below the Euro 6 particle mass limit of 10 mg/kWh. With the possible exception of lubrication oil, the contribution of which is assumed to be small, the diesel pilot is the dominant source of soot emissions. This indicates that the mixing, between the diesel pilot and the $\text{CH}_4$-air mixture, depends almost exclusively on swirl. It even appears that tumble seem to have a negative effect on soot emissions; higher TN generates more soot. This has been previously shown in [49] for diesel combustion.

Air motion can be expected to affect the gas combustion directly. High tumble generates high levels of small scale turbulence, which benefits flame propagation. However, the air motion can also be expected to greatly influence the distribution of pilot fuel in the cylinder, which would in turn affect the flame propagation. Breaking down the results into these two categories, direct and indirect effect on combustion, will likely require detailed CFD analysis but some conclusions and speculations can be made by investigating the heat release.

It is assumed that the intensity of the premixed pilot heat release is depending on the diesel distribution in the cylinder. However, it is not trivial to determine in which way this relationship works. A more distributed pilot injection will yield a higher local $\lambda$ and can for this reason be expected to result in a lower intensity of the premixed heat release. At the same time, a more distributed pilot injection will entrain more $\text{CH}_4$ in the pilot zone, causing more fuel to burn in the initial premixed phase. This can be expected to increase the intensity of the pilot heat release.
The parts of the heat release corresponding to the pilot injection, and the heat release corresponding to the flame propagation, are identified in Figure 41. In Figure 43, the correlation between the intensity of the pilot heat release and the intensity of the flame propagation heat release, is shown for the cases run at $\lambda = 1.7$.

![Figure 43](image.png)

Figure 43. Maximum heat release intensity for the flame propagation versus maximum heat release intensity for the pilot at $\lambda = 1.7$. All cycles are shown in grey and average for each case in blue.

There is a clear correlation between the intensity of the pilot heat release and the intensity of the heat release for the subsequent flame propagation. However, there are also strong variations that are not captured by this trend. The conclusion is therefore that, the distribution of the diesel pilot alone, cannot explain the difference in combustion speed and emissions. Hence, air motion affects the flame propagation, both directly and indirectly.

In Figure 44 the HC emissions for $\lambda = 1.7$, operating point 1, are shown as a function of swirl and tumble.
Figure 44. HC emissions versus swirl and tumble at $\lambda = 1.7$, operating point 1. The numbers on the surface show the actual value for each case.

It is seen that the maximum HC emissions are occurring at low swirl, regardless of tumble. The reason for this is shown to be related to crevice losses, this is further described in section 6.1.2.4. The lowest levels of HC emissions occur at high swirl and also at swirl between 3 and 4 and low tumble.

It is known that high tumble is beneficial since it is broken down into small scale turbulence, which improves flame propagation. This factor, combined with the results in Figure 44, should mean that the best results, with regards to combustion efficiency, should be found at high SN and high TN. This is, however, not so pronounced, see Figure 45.
Figure 45. Combustion efficiency, based on emissions measurements, versus swirl and tumble at $\lambda = 1.7$. The numbers on the surface show the actual value for each case.

It is possible that the pilot injection provides the necessary turbulence close to TDC, and that tumble is, for this reason, redundant. This might also mean that the shape of the combustion chamber is less important for dual fuel, compared to SI operation.

Maximum combustion efficiency, based on emission measurements, occurs at higher swirl levels, but as the heat transfer also is increased at high swirl, as shown in Figure 40, the highest brake efficiency occurs in a region around SN 2. This is shown in Figure 46.
Figure 46. Brake efficiency versus swirl and tumble at $\lambda = 1.7$. The numbers on the surface show the actual value for each case.

6.2.2.4 Emissions and combustion quality as a function of swirl and tumble at $\lambda = 1.9$

When evaluating the results for operating point 3, it became evident that it was not possible to explain all the trends on the basis of large scale air motion. A more detailed investigation of cases 4 and 6 is presented. These cases are marked in Figure 39, and it can be seen that they are quite close to each other with regards to swirl and tumble. Case 4 uses the standard valve profile and 15 mm valve lift. Case 6 is also using 15 mm valve lift but trapezoid valve profile.

Simulations, using 1-d gas dynamics software, show that the high lift, in combination with the TPZ profile, results in low levels of turbulence generation over the valves. However, small scale turbulence generated during the intake event is quickly dissipated and does not survive until the combustion event. Hence the turbulence generated should not affect combustion; it will however affect mixing. If stratification exists in the intake runners, the turbulence generated over the intake valves will improve mixing and thus combustion quality. In order to investigate this hypothesis, a sweep in CH$_4$ injection timing is carried out for STD and TPZ valve profiles and high valve lift. If the mixing is indeed poor, then the timing of the CH$_4$ injection, whether injection takes place during the intake stroke or not, should affect combustion quality. The results of these tests are shown in Figure 47.
It is clearly seen that the injection timing of CH₄ has a strong effect on HC emissions when TPZ profile is used, but no effect when STD valve profile is used. These results show that a large reduction of HC emissions can be achieved, for TPZ profile, by timing the gas injection properly. The best timing to start the injection is at approximately 300° BTDC. Using this timing, the bulk of the injection takes place during the time when the piston speed is highest and when gas velocities and turbulence across the intake valves is maximized. In Figure 47, HC emissions are used as a measure of combustion quality, but the trend remains the same if CoV or maximum heat release is used instead.

For λ = 1.7 and TPZ profile in combination with high valve lifts, an identical effect was seen on combustion speed; combustion was much slower compared to STD profile. However, since the mixture was richer, and most of the HC was derived from crevices, the lower combustion speed did not affect the emissions significantly.

To summarize, combustion at λ = 1.9 is very sensitive to small changes in local λ and hence very sensitive to mixing quality. For this reason, it was not possible to evaluate the effect of large scale flows properly, since the method used to generate these flows also affected mixing.

6.2.3 Conclusions regarding throttling and incomplete combustion

Two main pathways to improve the performance at lean conditions were evaluated: advanced injection strategies for the diesel pilot and varying in-cylinder air motion. The main conclusions are as follows:

- Split pilot injection provides 10-20% emissions reduction without any drawbacks and should be used at light load if the hardware permits this.
- RCCI extends the lean limit greatly, with maintained combustion efficiency, but is very challenging from a controls perspective and is for this reason not considered viable in the
The feasibility of successful methane aftertreatment, at such high \( \lambda \) and corresponding low exhaust temperatures, is also doubtful. Moderate levels of swirl, SN between 2.5 to 3, should be used to promote oxidation of crevice hydrocarbons. This can reduce HC emissions by approximately 20% compared to quiescent operation and the penalty with regards to heat transfer is not very large. Air motion affects the flame propagation speed through the \( \text{CH}_4 \)-air mixture, both directly and indirectly, through the effect it has on distribution of the diesel pilot. Dual fuel combustion is insensitive to tumble.

For additional information regarding unconventional combustion modes and the effects of in-cylinder air motion, refer to Papers I and V.

### 6.3 Coking & nozzle tip temperature

Nozzle tip temperatures and the related issue of nozzle hole coking during dual fuel operation were investigated in Papers III and VI. A summary of the findings is presented in this section.

#### 6.3.1 Nozzle coking during DDF operation

A common issue with diesel injectors is nozzle coking. A review of the current understanding of nozzle hole coking was published in 2008 by [71]. Three factors affecting the fouling process: fuel composition, nozzle tip temperature and hole-geometry, were highlighted in [72]. The accelerating effect of high nozzle tip temperature on nozzle hole coking is also demonstrated in [73]. Experiences from earlier engine tests, carried out within this project, indicate that very high nozzle tip temperatures occur during dual fuel combustion. This is evident by material changes occurring in used injectors. High temperatures are to be expected due to the reduction in diesel flow through the injector, while heat input from the combustion chamber remains high. In the DDF engine, as opposed to the diesel engine, high load does not necessarily imply a high flow of diesel through the injector nozzle. Since high nozzle tip temperatures are known to accelerate nozzle hole coking, this mechanism can be expected to pose a challenge for DDF operation. Nozzle coking results in spray deterioration, which could increase emissions, worsen fuel efficiency and ultimately cause misfire.

To investigate this issue, engine tests using an instrumented diesel injector, equipped with a K-type thermocouple mounted close to the tip, have been carried out. Details on the injector and the thermocouple placement can be found in chapter 5, Experimental setup. To find whether different parameter settings affect coking, beyond the effect they have on the nozzle tip temperature, it is desirable to find parameter settings which result in similar nozzle tip temperatures. For this reason, sweeps of \( \lambda \), E\% \( \text{CH}_4 \), start of injection (SOI) and common rail pressure were performed and the nozzle tip temperatures recorded. Based on these sweeps, operating points with similar nozzle temperatures were selected for extended tests with zinc contaminated fuel. The baseline case had a nozzle tip temperature of approximately 325°C. Based on this and the results from the different parameter sweeps, a low temperature range was defined around 300°C, and a high temperature range around 350°C.

Points from each parameter sweep, with temperatures which matched the high and low temperature range as closely as possible, were selected for further study. These load points are shown in Figure
48. For reference, the nozzle tip temperature from maximum load diesel operation, on the same engine, is also included in Figure 48 [74].

![Nozzle tip temperature for different load points for DDF-runs and full load diesel operation.](image)

Figure 48. Nozzle tip temperature for different load points for DDF-runs and full load diesel operation.

It is noteworthy that the temperatures during DDF operation at 7.5 bar IMEP, approximately 30% load, are 50-100°C higher than during full load diesel operation. The resulting parameter variations are shown in Table 9.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Low T</th>
<th>High T</th>
</tr>
</thead>
<tbody>
<tr>
<td>Start of injection</td>
<td>5° BTDC</td>
<td>14° BTDC</td>
</tr>
<tr>
<td>λ</td>
<td>1.7</td>
<td>1.3</td>
</tr>
<tr>
<td>E% CH₄</td>
<td>75 %</td>
<td>98 %</td>
</tr>
<tr>
<td>Common rail pressure</td>
<td>1000 bar</td>
<td>2000 bar</td>
</tr>
</tbody>
</table>

The common rail pressure does not influence the nozzle tip temperature and is therefore excluded from Figure 48, but it is included in Table 9 for convenience. The results of the steady state tests with zinc contaminated fuel, with regards to variations in E%, SOI and λ are seen in Figure 49. The flow loss was acquired from flow bench testing of the used nozzles.
3.5 hours of DDF operation at the baseline settings resulted in a flow loss of 12 and 15 percent for the two repeats. Except for the operating point at high temperature, using early SOI, a weak trend is seen that higher nozzle tip temperature result in larger flow loss. After continuous full load diesel operation, using the same zinc contaminated diesel fuel, on the same engine, a maximum flow loss of 5.5% was recorded from 6 hours of operation [74]. The average flow loss from all DDF operating points was approximately 12%. This is more than twice the flow loss which occurred from full load diesel operation at 30 bar IMEP, in spite of the fact that the diesel case was run for 6 hours, compared to 3.5 hours for the DDF cases. The data for λ and E% both follow the expected trends; the coking, which occurs from the high temperature settings, is greater than that which occurs at the low temperature settings. Considering the repeatability of the baseline test, no conclusion can be made as to whether E% and λ has any effect on coking, beyond the temperature effect. For SOI, the results are the opposite; the high temperature setting has a lower amount of flow loss compared to the low temperature setting. Currently, no clear explanation for this discrepancy has been found.

The effect of common rail pressure on nozzle flow loss is seen in Figure 50.
As previously mentioned, the nozzle tip temperature was the same across all common rail pressures, and it should therefore not be a factor in explaining the flow loss. It appears that injection pressure is the parameter that has the largest potential to influence nozzle hole coking. A difference in flow loss by a factor of two is seen as the injection pressure is increased from 1000 bar to 2000 bar. This trend has previously been shown for diesel combustion in [75]. The explanation suggested in that publication is that a higher injection pressure results in increased cavitation and shear stress in the nozzle holes, which in turn results in improved removal of the nozzle hole deposits. Since the injected amount was kept constant, the injection duration has increased as the injection pressure is reduced. This means that the contact time between the fuel and the inside of the nozzle holes has increased, and it is possible that this can cause an increase in the coking rate.

Comparing SEM imaging from DDF operation at 1000 bar injection pressure, Figure 51, with imaging from high load diesel operation with a similar injection pressure of 1250 bar, Figure 52, some differences can be seen.
Figure 51. SEM image showing the structure of deposits in the nozzle hole from DDF operation.

From looking at the deposits on the right side in Figure 51, it appears that the deposits from DDF operation are smoother, and the structures are of a larger scale compared those that have resulted from diesel operation. The rate of formation has been shown to affect the structure of deposits [76]. Also shown was that the surface of the deposits was smoother when a higher rate of formation had occurred. This observation appears to be in agreement with the appearance of the deposits in Figure 51 and Figure 52.

Figure 52. SEM image showing the structure of deposits in the nozzle hole from diesel operation.

Spectrum analysis using energy dispersive X-ray spectroscopy, EDX, of the deposit composition was made at locations marked in Figure 51 and Figure 52. The results from the spectrum analysis are shown in Table 10.
The composition of the deposits from diesel and DDF combustion with a zinc doped diesel fuel are fairly similar. The concentration of the main constituents, carbon, zinc and oxygen rank in the same order with similar concentrations. These are the most commonly found elements in injector deposits [73], [77]. There are also indications that elements from the lubrication oil: calcium, sulfur, phosphorous and potassium, find their way into the deposits. The DDF deposits show a higher concentration of these elements, which could be explained by the significantly shorter injection duration, and therefore the much longer time available for back flow of combustion gases. It is possible that the higher concentration of zinc, in the diesel case, can be explained considering the higher flow of zinc doped fuel that has passed through the nozzle. A higher carbon content, on the expense of Zn, can also be expected in the DDF deposits, since they have been formed at higher temperatures. Precipitation of Zn-salts occurs at lower temperatures, while higher temperatures of approximately 300°C are required for coking of hydrocarbons [78].

In addition to the complex formation process, there is also the competing removal process due to the shear forces and cavitation in the nozzle holes. These factors make the nozzle fouling process very complicated and may have influenced the sometimes poor repeatability of the coking experiments performed in this work. Due to the variation between the repeats, it is difficult to draw statistically significant conclusions from the factor tests. However, the large difference in flow loss between DDF and diesel combustion is clearly seen, and the strong influence of the injection pressure is also significant. These factors highlight the challenge that nozzle hole coking poses for DDF. Additional discussion regarding nozzle hole coking is found in Paper VI.

### 6.3.2 Controlling the nozzle tip temperature

Considering the effect of high nozzle tip temperature on coking, efforts aimed at reducing the nozzle tip temperature is a logical countermeasure. Paper III details an investigation into the parameters affecting the nozzle tip temperature. The parameter which influences the nozzle tip temperature most strongly is the E% CH₄; it is therefore the most efficient parameter available for controlling it. 300°C was identified by [72] as a critical temperature before serious coking problems were encountered. For this reason it was considered an acceptable limit for injector operation. Figure 53 shows the maximum diesel substitution rate over the engine operating range, limited by a maximum tip temperature of 300°C.

---

**Table 10. Composition of nozzle hole deposits from diesel and DDF operation.**

<table>
<thead>
<tr>
<th>Elements [Weight %]</th>
<th>Diesel</th>
<th>DDF</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>43.8</td>
<td>49.7</td>
</tr>
<tr>
<td>Zn</td>
<td>26.5</td>
<td>18.4</td>
</tr>
<tr>
<td>O</td>
<td>21.9</td>
<td>17.7</td>
</tr>
<tr>
<td>Ca</td>
<td>4.4</td>
<td>9.3</td>
</tr>
<tr>
<td>Cr</td>
<td>2.0</td>
<td>0.3</td>
</tr>
<tr>
<td>S</td>
<td>0.8</td>
<td>1.6</td>
</tr>
<tr>
<td>P</td>
<td>0.4</td>
<td>1.2</td>
</tr>
<tr>
<td>K</td>
<td></td>
<td>1.7</td>
</tr>
<tr>
<td>Al</td>
<td>0.2</td>
<td></td>
</tr>
</tbody>
</table>
From Figure 53 it is evident that injector tip temperature does not limit diesel substitution rate at light load. As the power output of the engine increases with load and speed, more diesel is needed to cool the injector and to remain below the temperature limit. At loads above 13 bar BMEP, combustion phasing is retarded due to knock. Because of late combustion phasing, less heat transfer from the gases to the injector occurs, and for this reason the diesel substitution rate can be maintained and even increased slightly. The most critical operating point, from the perspective of injector tip temperature, is therefore the maximum speed and load where MBT combustion phasing is maintained. Maintaining a temperature below 300°C in this operating point requires a reduction in diesel substitution to 84%. This behavior is undesired since maximizing \( E\% CH_4 \) is necessary to obtain maximum benefits with regards to fuel economy and to the environment.

In order to reduce the temperature, without imposing unfavorable limits on engine operation, a copper sleeve was installed around the injector tip, as shown in Figure 54.
The copper was expected to provide a path of reduced thermal resistance and enable higher diesel substitution rates. This was confirmed by modeling of the injector tip heat balance. Temperature data from test runs, with the standard injector configuration and with the copper sleeve, were then fed into the model. It was measured that the sleeve reduces the thermal resistance by almost 80%. An example of actual performance, with regards to nozzle tip temperature, is shown in Figure 55.

A dramatic reduction in injector tip temperature is achieved with the introduction of the copper sleeve. The difference in temperature is approximately 50°C at late combustion phasing and more than 100°C at early combustion phasing. The improved thermal resistance, provided by the copper sleeve, effectively removes injector tip temperature as a limiting factor. In the most critical operating point in Figure 53, where the E% CH₄ was previously 84%, it is now possible to reach 99%. For additional information regarding nozzle tip temperature during dual fuel operation, refer to Paper III.
6.3.3 Conclusions regarding nozzle coking and nozzle tip temperature

Investigations were made into nozzle hole coking during DDF operation, with focus on the effect of high injector tip temperature. The main conclusions from this section are:

- The rate of injector nozzle hole coking in DDF mode can be several times higher than during diesel operation. A maximum flow loss of 18%, after 3.5 hours of DDF operation at medium load, was recorded.
- The highest rate of nozzle coking occurred at the lowest injection pressure. This shows that overly low injection pressure can be critical during DDF operation since the decreased shear forces and cavitation in the nozzle holes limit the deposit removal process.
- Parameters settings that result in high injector tip temperatures weakly increase the rate of nozzle coking.
- With regards to $\lambda$, E% and SOI, no effect on coking, beyond the temperature effect, could be distinguished.
- Deposits from DDF operation had a similar composition compared to deposits from diesel operation. In the DDF deposits, more elements from the lubrication oil were found.
- Decreasing the thermal resistance between the injector tip and the cooling water, by inserting a copper sleeve around the injector tip, has the potential to greatly reduce the injector tip temperature and should be beneficial in reducing coking.

6.4 Knock & pre-ignitions

Knock and pre-ignitions are recognized as major limitations to engine operation and were therefore investigated. The results presented in this section have not been previously published.

6.4.1 Limits to engine operation

Knock, or autoignition of the end gas, is a phenomenon which has been known since the very first Otto-engines. The autoignition temperature of the fuel is exceeded during the compression stroke, and if enough time is provided, autoignition, or engine knock will occur. Hence, to avoid knock, the fuel must be burned in a controlled manner during this autoignition delay time. In modern Otto-engines, measures are taken both to increase the autoignition delay time, and to increase the controlled burn rate of the fuel, so that knock can be avoided. The autoignition delay time is increased through the use of high octane number fuels, cooling of the charge air and efficient scavenging. The burn rate of the fuel can be increased through in-cylinder air motion, combustion chamber design and engine speed. During engine operation, knock mitigation in Otto-engines is achieved through adjustment of the spark timing. Delaying the spark timing reduces maximum temperature but also provides more time for the autoignition reactions, since the controlled burning of the charge is delayed. However, since autoignition delay time increases exponentially with decreasing temperature, the net result is that knock can be avoided in this manner. The same option is available in the dual fuel engine; the injection of the diesel pilot can be delayed to avoid knock, and in this way the load can be increased further. However, above a certain load, some engine cycles exhibit start of combustion before the injection of the pilot fuel. This is known as pre-ignitions and is illustrated in Figure 56.
Figure 56. Cylinder pressure versus crank angle for five consecutive engine cycles.

Cycles one and two illustrate normal combustion; the combustion starts with the diesel pilot and slight knock is seen at 10-15° ATDC. In cycles three to five, however, combustion starts earlier and results in heavy knocking for all engine cycles. In this case an accelerating behavior was seen, but this appears to be an exception, the behavior is mostly sporadic. When accelerating pre-ignition occurs, control over combustion phasing is lost, and other means such as torque reduction or fuel cut must be implemented quickly to avoid engine damage. In Figure 57, a load sweep is shown for three different gas qualities. The gas qualities are detailed in chapter 5, Experimental setup.
In the left graph, it is seen that the onset of pre-ignitions occur at approximately the same load independently of gas quality, around 15 bar IMEP. In the middle graph, where knock amplitude is plotted as function of load, it can be seen that the consequence of these pre-ignitions is much more severe for lower methane numbers. The knock amplitude for MN70 and MN80 is much greater than that of MN100. In the right graph the knock amplitude is also shown, but the pre-ignition-cycles have been removed. It is seen that without pre-ignitions, the knock amplitude would stay below 10 bar for loads below 20 bar IMEP. It is also possible to retard the combustion further, to enable higher loads without causing unacceptable knock amplitudes. For this reason, pre-ignition, not knock, is the mechanism which limits the maximum load. The maximum allowable knock amplitude varies slightly between manufacturers, but for SI engines it is usually in the range of 2-6 bar. During dual fuel operation, the premixed combustion of the diesel pilot also causes oscillations in the combustion chamber. The magnitude of these oscillations depends on the several factors, but can exceed 6 bar, for this reason a more generous limit of 10 bar is assumed for the knock amplitude during DDF operation. Using this limit, the operating limits shown in Figure 58 are obtained.
The target level is based on pure diesel operation from a Scania production diesel engine. For maximum market acceptance it is highly desirable to reach the same torque levels as the corresponding diesel engines. While higher gas qualities are definitely desirable, it is likely that the vehicle needs to function with methane numbers as low as 70. Using MN70 fuel, due to preignitions, the load is limited to approximately 65% of the target level. It is clear that the issue of pre-ignitions is something which needs to be overcome, if development of attractive dual fuel engines is to be possible.

6.4.2 Phenomenology

A simple model for autoignition is the well-known Livengood-Wu-integral [79]:

\[
\int_{n_C}^{SOC} \frac{1}{\tau(CA)} d(CA) = 1
\]  

(1)

Where the ignition delay, \( \tau \), is given as:

\[
\tau = A \cdot p^{-n} \cdot e^B
\]  

(2)

When the integral reaches unity autoignition occurs. If the zone of autoignition is large enough, a propagating flame can originate from the autoignition zone, this would constitute a pre-ignition. In this expression, A, B and n are parameters related to the chemistry of the mixture while pressure, p, and temperature, T, are related to engine operation. Different occurrences during engine operation, which influence these parameters and allow the integral to reach unity before the desired start of combustion, can therefore cause pre-ignitions. The following root causes are documented in literature:
- Hot spot ignition
- Lubrication oil ignition
- Ignition from deposits
- Ignition from soot
- Ignition from residual gases
- LSPI; low speed pre-ignition, HCCI-type autoignition of main fuel.

These mechanisms are also possible during dual fuel operation:

- DI injector leakage or dribble.

Lubrication oil ignition and the possible DI injector leakage, affect the parameters related to chemistry since a higher cetane number component is introduced into the main fuel. The other root causes mostly influence the temperature. Figure 59 shows pre-ignition frequency as a function of load for three different engine speeds.

![Figure 59. Pre-ignition frequency versus IMEP for three different engine speeds. Each measurement consists of 100 cycles; one percent of the cycles therefore corresponds to one pre-ignition event.](image)

It is seen that the issue is greater at low engine speeds; 800 RPM and 1225 RPM exhibit similar behavior, while at 1600 RPM no pre-ignitions are detected. The change in engine speed may affect the temperature of the mixture slightly but mostly affects the time available for autoignition. Since gas engines derived from heavy duty diesel engines operate at very low engine speeds, this behavior aggravates the issue greatly. Hence, regardless of which of the root causes is responsible for the pre-ignitions, the issue can be expected to improve with higher engine speed. The exception to this is hot spot ignition; the temperature of the hot spot can be expected to increase significantly with engine speed, as more power is fed into the system. Figure 60 shows how the propensity towards pre-ignitions is affected by five different parameter settings.
The largest difference from the baseline case was achieved by heating of the intake air. At 80°C inlet temperature, the onset of pre-ignitions starts at approximately 12.5 bar IMEP. Naturally, raising the inlet temperature affects $T$ in equation (2); the compression temperature at TDC is approximately 100 °C higher than the baseline case. The addition of 20% EGR proved to be most effective in preventing pre-ignitions. EGR has a higher ratio of specific heats compared to air, which leads to a lowering of the compression temperature. However, this is compensated for since the addition of EGR affected the intake temperature slightly; it increased from 30°C in the baseline case to approximately 40°C when 20% EGR was added. The net result was a similar compression temperature to the baseline case. It is therefore deemed that the suppressing effect of EGR on pre-ignitions is mostly a result of its effect on the chemical parameters in equation (2). A reduction in the $O_2$ concentration is known have a strong increasing effect on the autoignition delay. Late CA50 reduces $p$ and $T$ due to lower combustion chamber surface temperature and increases the ignition delay but late combustion also impacts thermal efficiency negatively. This means that the charge mass into the cylinder must be increased to maintain the same IMEP; the net result is a similar behavior to the baseline case. For the low $\lambda$ case, $\lambda = 1.1$, the ignitability of the mixture is increased compared to the baseline case, but lower air mass is required for the same IMEP, and the net result with regards to pre-ignitions versus load is similar to the baseline case. These data are in agreement with the widely accepted Livengood-Wu model for autoignition but do not reveal the root cause of the pre-ignitions in the dual fuel engine.

In recent development of SI engines, sporadic pre-ignitions, caused by fuel-lubrication oil interaction, have received a lot of attention. However, since the dual fuel engine has a much higher compression ratio, runs lean and uses methane as fuel, it is not necessarily the same mechanism that is dominating. A high output SI engine with pre-ignition issues may exhibit pre-ignitions in 1/5000 engine cycles, but during dual fuel operation, the pre-ignition frequency may exceed 50% under certain operating conditions, an increase by a factor of 2000 compared to the SI engine. The high
frequency of pre-ignitions during dual fuel operation, and the fact that an accelerating behavior is sometimes seen, could indicate that more traditional pre-ignition from hot spots is occurring.

6.4.3 Hot spot ignition
Since the DDF engine lacks spark plug, it is not clear which component would be responsible for the hot spot ignition. The critical spark plug temperature before onset of pre-ignition versus compression ratio, for a naturally aspirated SI engine at full load, was investigated in [56]. The critical temperature was in the range of 970-1070°C and decreased by approximately 50°C per unit of increasing compression ratio. Other authors report critical temperatures in the range of 800-900°C [80]. It is unlikely that any component in the lean burning DDF engine reaches temperatures in this range. However, since the critical temperature is a function of pressure and the DDF engine has a much higher compression ratio than the gasoline engine, it is probable that lower component temperatures are needed to trigger pre-ignitions. Since a spark plug was lacking, and the temperature of the nozzle tip is well documented in chapter 6.3 and significantly below this temperature range, it was considered that the hottest surface in contact with the gas most likely belonged to the exhaust valves, and that excessive exhaust valve temperature might be the root cause for the pre-ignitions. Simulations of full load diesel operation indicate that exhaust valve temperature approaches 750° in the most demanding operating point. This temperature is also below the critical temperature range mentioned above but since the effect of pressure on hot spot ignition is not documented for the relevant pressure range [80], ignition from the exhaust valves could not be ruled out and was considered a possible root cause. In order to test this hypothesis, the standard valves were replaced with sodium cooled exhaust valves, and a load sweep was performed. Empirically, sodium cooled valves will run approximately 100° cooler than the corresponding standard valves, and the tendency towards pre-ignitions should be significantly reduced. The results from this load sweep are seen in Figure 61.
No discernible difference can be seen between the standard and the sodium cooled exhaust valves with regards to onset of pre-ignitions. Sporadic pre-ignitions start to occur at approximately 16 bar IMEP for both valve types. At loads above 21 bar IMEP, the frequency of pre-ignitions increases for the standard valves but not for the sodium cooled ones. Since the occurrence of pre-ignitions is a function of time, temperature and chemistry, removing heat from the combustion chamber through better valve cooling can be expected to influence the pre-ignition frequency slightly, regardless of the main root cause. It is also possible that the perceived difference between the valve types is the result of stochastic variations which would disappear if the sample size was larger. For these reasons, and since the onset of pre-ignition occurs at similar loads, these results do not support the hypothesis that hot spot ignition from the exhaust valves is the dominating root cause for the pre-ignition issues.

In order to enable a more fundamental investigation of hot spot ignition at lean methane mixtures and high pressures, a set of glow plugs with thermocouples in the tip were commissioned. Closed loop feedback from the thermocouple together with a PWM-controlled glow plug driver enabled precise control of the hot spot temperature, independent of engine operating conditions. Calibration sheets from the manufacturer of the glow plugs enabled the translation of the measured thermocouple temperature into the actual surface temperature of the glow plug. The pre-ignition frequency as a function of glow plug surface temperature, for three different operating conditions, is seen in Figure 62.

Figure 61. Pre-ignition frequency versus IMEP for two different types of exhaust valves at 1225 RPM. Each measurement consists of 100 cycles; one percent of the cycles therefore corresponds to one pre-ignition event.
6.4.4 Lubrication oil ignition

Lubrication oil ignition is well documented from downsized SI engines [81], [82], and it has been shown to occur for gas engines as well [83]. In [82] it is shown that oil additives containing Ca can increase the frequency of pre-ignitions while additives containing Zn have a preventative effect. It was also concluded that the autoignition temperature of the oil at high pressure had an influence on pre-ignitions. The temperature and pressure occurring during dual fuel operation are greater than during SI operation, and no publications were found which evaluate different oil formulations with regards to pre-ignitions during lean gas operation. For these reasons a set of tests were performed using six different oil formulations. These are detailed in chapter 5, Experimental setup. The goal of the oil matrix was to test oils with high and low viscosity and high and low Ca content. The motivation being that, if the mechanism is identical to that which is occurring in SI engines, then the Ca content should prove most influential. However, if other mechanisms are dominating, for instance due to the high compression temperature, the viscosity might be of greater importance, since it should influence the probability of oil droplets being ejected into the bulk gas. The results from a load sweep using these 6 different oils are shown in Figure 63.
It is seen that oil #3 performs best and enables a load of 21 bar IMEP before onset of pre-ignitions. Oil #1 and oil #2 show the biggest propensity towards pre-ignitions, the onset of pre-ignitions occurs at approximately 20 bar IMEP. Since there is some difference between the first and second run of oil #2, these results need to be interpreted with care. Oil #2 is one of the oils with moderate Ca content; however, oil #4 has zero Ca content and still performs worse. For this reason, no conclusion regarding the influence of Ca content can be made. Oil #3 which performs well in the load sweep and oil #5 which performs well in the constant load test shown in Figure 64, have the lowest viscosity. Based on these tests, viscosity appears to be more important than Ca content and low viscosity appears to be beneficial. The viscosity of the oil should affect the probability for oil droplets to be ejected into the bulk gas, it is therefore reasonable that it has an effect on the pre-ignition frequency. It is also clear that none of the oils which were tested solves the issue by themselves. In addition to the load sweep, constant load testing of each oil was performed at 800 RPM, 1.5 bar relative inlet pressure, $\lambda = 1.5$ and very late combustion phasing. These settings were selected to provoke maximum response with regards to pre-ignitions. The results are seen in Figure 64.
Figure 64. Pre-ignition frequency versus measurement number for six different oil formulations at 800 RPM, constant load operation.

The engine was operated at 800 RPM and cylinder pressure was logged repeatedly in batches of 300 cycles, each batch of 300 cycles corresponds to a measurement point in the graph above. It is seen that after approximately 4 measurement points or roughly 4 minutes of operation, the pre-ignition frequency increases rapidly for oil #1, eventually reaching a pre-ignition frequency exceeding 50% of engine cycles. The other oils exhibit similar behavior. Since this behavior was not anticipated, only eight measurements were collected for some of the oils. However, the accelerating behavior for the repeat test of oil #2 starts after the eighth measurement so for this reason it is difficult to draw any definitive conclusions regarding differences between the oils from these tests. The reason for this behavior is currently not known. Attempts to reduce the pre-ignition frequency to the initial value, by operating the engine in diesel mode at medium load for five minutes, were not successful. If the cause for the accelerating pre-ignitions had been a hot spot, then the diesel operation should rather quickly result in a return to the initial state. Instead, the pre-ignition frequency remained high when attempting to return to dual fuel operation. Temperature related phenomena are also unlikely considering the previous investigations into the required hot spot temperature and the low heat input into the engine at these operating conditions of high \( \lambda \) and low engine speed. One remaining plausible reason would be high oil consumption at this specific operating point which would cause buildup of deposits which eventually reach a critical mass and cause extremely high rates of pre-ignitions. Further investigations, possibly using optical techniques, are needed to determine the root cause for this specific behavior and to be able to, with certainty, determine the root cause of pre-ignitions during dual fuel operation in general.

6.4.5 Conclusions regarding knock and pre-ignition

Experimental investigations into knock and pre-ignition have been made. The focus of these investigations was to attempt to identify the root cause for pre-ignitions and also find suitable countermeasures. The following conclusions are made:
- Pre-ignitions, rather than knock, limit the load. This holds for fuels with methane number in the range of 70-100. The behavior is most severe at low engine speeds in combination with retarded combustion phasing.
- EGR is the most effective countermeasure to pre-ignition.
- The temperature required for hot spot ignition was found to be approximately 1000°. This result, in combination with the lack of influence on the onset of pre-ignitions as the exhaust valves were replaced with sodium cooled valves, makes hot spot ignition an unlikely root cause for the pre-ignition issues.
- Influence of lubrication oil quality on pre-ignition frequency is seen. However, none of the tested oils had sufficient resistance towards pre-ignitions to solve the issue by itself. Further work is required. Oil originated ignition is currently the most probable root cause but other phenomena, such as DI injector dribble, cannot be ruled out.
- Accelerating pre-ignition frequency occurs during constant load operation at 800 RPM, the cause for this behavior is currently not known with certainty but a buildup of deposits is likely.
7  Discussion
The discussion will attempt to tie the results from the project together into suggestions for the design of a competitive dual fuel engine.

7.1  Hardware
These are the recommendations regarding the hardware configuration based on the results of the project and on available literature.

7.1.1  Aftertreatment
The methane aftertreatment is what sets the boundary conditions for engine operations with regards to HC emissions and exhaust temperature. For this reason it needs to be discussed first. The biggest challenge regarding aftertreatment is the high exhaust temperature required by the methan oxidation catalyst. Light-off temperature has historically been around 450°C. Development is ongoing, however, and current prototype catalysts feature light-off temperature as low as 300°C for a new catalyst and 350°C for an aged one. Another 50°C need to be added to the light-off temperature to reach maximum conversion. This would mean that, for an aged, modern catalyst, 400°C is the lowest tolerable exhaust temperature at the catalyst inlet.

As HC and CO are converted in the catalyst in an exothermic reaction, heat is produced. Different methods have been proposed to keep the heat from these reactions in the catalyst to enable operation at exhaust temperatures below the light-off temperature. If the flow direction of exhaust through the catalyst is cycled, the heat will remain in the catalyst to a much greater extent, resulting in high conversion efficiency at exhaust temperatures as low as 262°C [84]. These results are from the year 2000, so it is likely that the catalyst used in the study required at least 450°C for efficient methane conversion. This indicates that the cycling of the exhaust flow direction enables exhaust temperatures approximately 200° below what would otherwise be acceptable. Similarly, the catalyst can be arranged as a heat exchanger, where the flow is reversed inside the catalyst and the hot products will heat the incoming gases. No data is found regarding the performance of the latter configuration, but it can be assumed that it should be similar to the cycling-flow catalyst. If a heat exchanger configuration is applied to a state-of-the art catalyst with a minimum of 400° operating temperature, this translates into a lowest exhaust gas temperature of 200° at the catalyst inlet. If a 150° temperature drop across the turbocharger is assumed, this would mean a minimum allowable exhaust temperature of 350° in the exhaust manifold, 400° after the addition of a 50° safety margin. This translates into a $\lambda$ of approximately 1.6 at light load at a compression ratio of 17.3. Assuming a conservative conversion efficiency of 90% over a cycle and a legislated limit of 0.5 g/kWh for HC, then the maximum allowable engine-out HC emissions are 5 g/kWh. Based on this reasoning and with some additional safety margin, the target for engine-out HC emissions is set to 4 g/kWh. At 25% load, $\lambda = 1.6$ and high E% CH₄, engine-out HC emissions are approximately 6 g/kWh using the current engine configuration. Hence, changes to the engine which lower the engine-out HC emissions by approximately a third of the present level are needed.

The DPF and the NOₓ aftertreatment are carried over from the base diesel engine. Soot levels during DDF combustion are generally much lower than during diesel operation, but it is unlikely that the Euro 6 emissions level can be met without a DPF, especially with regards to particle number. Regardless of performance in DDF mode, the DPF is needed for diesel operation if fuel flexibility
should be maintained. The NOx levels are of the same order of magnitude as during diesel operation and should therefore be manageable by the SCR system.

7.1.2 Fuel system

The diesel injection system should be of the common rail type to enable maximum degrees of freedom. Multiple injections are a key tool to increase the combustion efficiency at light load. Capability for late injections to increase the exhaust temperature and enable quick light-off of the oxidation catalyst is another important application for a common rail system. The diesel injector could be fitted with a copper sleeve around the tip to lower the nozzle tip temperature and reduce the risk of rapid nozzle hole coking. The copper sleeve allowed for operation at high E% CH4 with tolerable nozzle tip temperatures.

The port injectors for the CH4 should be fitted with extensions to the nozzles extending into the port and enabling injection as close as possible to the back of the intake valves. The nozzles should be large enough so that the injection duration during full load operation does not exceed the opening duration of the intake valve. This configuration enables the entire injection to take place during the intake event, creating best possible mixing between the CH4 and the air. The second benefit of this arrangement is that very little CH4 resides in the intake runners between cycles, which means that large cycle-to-cycle changes in CH4 amount are possible with good control of in-cylinder λ. Firstly, this enables rapid reduction in E% CH4 as countermeasure when pre-ignition is detected. Compared to torque reduction, this has the benefit that engine output remains unaffected. If it is done correctly, the driver will not notice any reduction in performance. Injection on open intake valve also enables better λ-control during transients. This is important for reducing HC emissions; a state-of-the-art oxidation catalyst has close to 100% conversion efficiency for CH4 in steady state. Over a transient cycle, however, conversion efficiency for the same catalyst drops to just above 90%. A probable cause for this reduction in efficiency is surges in HC concentration resulting from poor λ-control during transients. Such surges are also likely to accelerate the ageing process of the catalyst since they can cause excessive local temperatures. For these reasons, precise cycle-to-cycle control of λ is crucial.

7.1.3 Compression ratio

The compression ratio in the dual fuel engine is a trade-off between two conflicting needs. It needs to be high enough to enable reliable autoignition of the high cetane number fuel, the diesel. It also needs to be low enough to enable high load operation without the occurrence of autoignition, knock, of the low cetane number fuel, the methane. Some guidance to this trade-off can be found by examining marine, purpose-built, dual fuel engines. These have a compression ratio of approximately 14. However, the large-bore marine engines use a small injector to inject the pilot fuel, and hence have a much longer path of flame propagation through the methane. This factor, in combination with the low engine speed, means that they are more susceptible to knock than transient truck engines. Marine engines are also not cold-started frequently, so the ignitability of the pilot fuel is of lesser concern. These two factors combined, mean that the optimal compression ratio for a transient heavy duty engine is likely to be greater than 14. The base diesel engine used in this study has a compression ratio of 17.3 and achieves cold starts without any starting aids such as glow plugs or intake air heating. This is therefore seen as the upper limit to the range of compression ratios. The optimal compression ratio for a heavy duty dual fuel engine is therefore found in the
range of 14-17. A reasonable best guess would be a compression ratio of 16, but since this parameter has not been specifically investigated in this project, a more precise recommendation cannot be given. A variable valve train which enables online variation of the effective compression ratio would appear ideal for the dual fuel application.

### 7.1.4 Piston & rings

If fuel flexibility is to be maintained, meaning that diesel operation should still be possible, modifying the shape of the combustion chamber would be costly since the diesel calibration would have to be repeated. From the tests with varying air motion it appears that dual fuel combustion is relatively insensitive to tumble. It is plausible that the turbulence generated by the diesel injection in combination with the large ignition source means that sufficient energy is available regardless of tumble. This in turn would imply that the design of the combustion chamber with regards to turbulence generation is of little importance. The shape of the piston bowl is therefore left intact. A redesign of the piston ring land is important, however. This redesign should serve two goals, reducing oil consumption and minimizing the topland volume.

Designing the ring-pack is a trade-off between wear, oil consumption and friction. Since pre-ignitions currently impose unacceptable limits to engine operation and the likely root cause is lubrication oil related ignition, minimizing oil consumption would have to take priority over wear and friction. The goal could be to reduce the pre-ignition frequency to a tenth of the current levels through reduced lubrication oil consumption.

Since the main source of HC emissions is the piston topland volume, a redesign of the piston ring land would aim to reduce this volume as far as possible; the target being a minimum of 50 percent reduction. One way of reducing the topland volume is to move the top compression ring upwards, thus reducing the topland height. The position of the top compression ring is limited by temperature; if the temperature is too high, coking of the lubrication oil will occur which cannot be allowed. However, if the maximum torque in diesel mode is reduced there would be less heat input into the inside of the piston bowl and the maximum piston temperatures would be lower. This, in turn would enable the top ring to be moved upwards. A 50% reduction in topland volume would translate into a reduction in HC emissions of approximately 25%, or 1.5 g/kWh at light load and λ = 1.6.

### 7.1.5 Lubrication oil

The lubrication oil is specified in the usual manner with the additional demand that it should also resist pre-ignitions to the greatest extent possible. Based on available literature, this translates into low Ca-content, high Zn-content and a base oil with high resistance against autoignition. From the results in this project, low viscosity also appears beneficial.

### 7.1.6 Air path

A throttle must be added in case this is lacking from the base diesel engine to enable λ control at light load.

The port design of the cylinder head should create a swirl number of approximately 3. The reason for this relatively high swirl is that it helps mix hydrocarbons returning from crevices and for this reason it has a reducing effect on the hydrocarbon emissions. It is assumed that choosing the
appropriate swirl can lower the HC emissions by 10% compared to the current configuration. This would translate into a reduction of approximately 0.5 g/kWh at light load and $\lambda = 1.6$.

Variable intake valve timing would be extremely valuable. This would enable variation of the effective compression ratio through Miller-timing and late closing of the intake valves. At light load, a high compression ratio leads to lower ignition delay of the diesel pilot which in turn enables lower loads before having to resort to diesel operation. Higher compression ratio also helps combustion efficiency in the lean CH$_4$-air mixture. At high load a reduced compression ratio helps greatly to reduce the issues with pre-ignition and knock. Variable valve timing is not yet implemented in heavy duty diesel engines and therefore the cost of adding it as part of the conversion to dual fuel operation would be substantial. For this reason, it is considered optional.

7.2 Software and calibration

One of the main tasks during calibration will be to provide sufficient thermal management so that the oxidation catalyst can work at high conversion efficiency without excessive ageing. This is mainly a matter of controlling $\lambda$ with high precision. In addition to $\lambda$-control, adjusting the combustion phasing or adding post injections of diesel are other tools for thermal management. At the lowest loads and idling where the throttling required would cause misfire of the diesel pilot, pure diesel operation is utilized.

An injection strategy using multiple injections should be used at light load. Splitting the pilot injection into two, with most of the fuel allocated to the first injection, improves combustion efficiency and reduces the amount of HC emissions by approximately 10%. At 25% load and $\lambda = 1.6$ this means a reduction in engine-out HC by 0.5 g/kWh.

In order to minimize the rate of nozzle hole coking in dual fuel mode, the common rail pressure should be kept as high as possible. It is also possible that intermittent operation using 100% diesel and high injection pressure can be used to clean the nozzles from deposits through cavitation. This effect is so far hypothetical since it has not been investigated but it is likely that this would occur to some extent.

The control system must be able to detect pre-ignitions. The simplest way to achieve this would be through the use of a knock sensor. The knock sensor would only detect pre-ignition cycles which lead to knock but this can be considered sufficient. If sudden, single knocking cycles are detected, a rapid switch to a lower E% CH$_4$ should be made to avoid repeated pre-ignitions. The switch is made while maintaining constant air mass and load. If the gas injection system is designed as suggested in section 7.1.2; such a change can be made from one cycle to the next without being noticeable by the driver. The E% can then gradually be increased again until pre-ignition occurs again or until the target E% level is reached.

7.3 Summary of discussion

If these recommendations are implemented, the resulting engine reaches Euro 6 emissions standards while maintaining an average diesel substitution rate in excess of 90% over a drive cycle. Indicated efficiency is approximately 42% at 25% load and up to 46% at medium to high load. It may be necessary to limit full load torque in diesel mode in order to reduce the topland volume. The engine-
out HC emissions which originally were 6 g/kWh at 25% load and \( \lambda = 1.6 \), are reduced by the following measures:

- 50% reduction in topland volume, 1.5 g/kWh
- Optimized swirl level, 0.5 g/kWh
- Split diesel injection, 0.5 g/kWh

These changes accumulate to a total reduction in HC emissions by 2.5 g/kWh. This translates into engine-out emissions of 3.5 g/kWh which is below the 4 g/kWh limit imposed by the oxidation catalyst. The most critical assumptions that have been made are:

- The heat exchanger configuration for the methane oxidation catalyst lowers the minimum tolerable exhaust temperature to 200°C at the catalyst inlet.
- The frequency of pre-ignitions can be lowered to within acceptable limits through optimization of ring pack and lubrication oil. If pre-ignitions remain an issue, additional development efforts are needed to address this.

If these assumptions are realized an attractive engine concept is possible where high methane utilization is combined with fuel flexibility. The concept would benefit further from the addition of variable inlet valve timing which has the potential to greatly expand the load range.
8 Summary & Conclusions

This thesis summarizes a four year project executed by AVL, KTH and Scania and supported financially by the Swedish Energy Agency. The main findings and conclusions are organized based on the three topics that constitute the principal challenges: HC emissions, thermal issues and pre-ignitions:

- **HC emissions:**
  - The topland crevice is the largest contributor to the HC emissions, and efforts must be taken to reduce this volume in order to meet current emissions legislation. The topland crevice contributes between 50-70% of HC emissions at operating conditions relevant to lean burn gas engines. Swirl can be used to improve oxidation of HC returning from crevices, and the optimal swirl number for this purpose was found to be approximately 3. This can reduce HC emissions by approximately 20% compared to quiescent operation. At λ > 1.8, quenching becomes the main source of HC emissions.
  - Split pilot injection provides 10-20% emissions reduction without any drawbacks and should be used at light load if the hardware permits.
  - RCCI extends the lean limit greatly with maintained combustion efficiency, but is very challenging from a controls perspective, and is for this reason not considered viable in the short term. The feasibility of successful methane aftertreatment at the high λ and corresponding low exhaust temperatures associated with this combustion mode is also doubtful.
  - Air motion affects the flame propagation speed through the CH₄-air mixture both directly and indirectly through the effect it has on distribution of the diesel pilot. The effect is seen as a function of swirl; dual fuel combustion is insensitive to tumble.

- **Thermal issues:**
  - The rate of injector nozzle hole coking in DDF mode can be several times higher than during diesel operation. A maximum flow loss of 18% after 3.5 hours of DF operation at medium load was recorded. The highest rate of nozzle coking occurred at the lowest injection pressure. This shows that too low injection pressure can be critical in DDF operation since the decreased shear forces and cavitation in the nozzle holes limit the deposit removal process. Decreasing the thermal resistance between the injector tip and the cooling water by inserting a copper sleeve around the injector tip has the potential to greatly reduce the injector tip temperature and is assumed to be beneficial in reducing coking.

- **Pre-ignitions:**
  - Pre-ignitions, rather than knock, limit the load. This holds for fuels with methane number in the range of 70-100. EGR is the most effective countermeasure to pre-ignitions.
  - The temperature required for hot spot ignition is shown to be approximately 1000°C. This result in combination with the lack of influence seen as the exhaust valves were
replaced with sodium cooled valves, makes hot spot ignition an unlikely root cause for the pre-ignition issues.

- Influence of lubrication oil quality on pre-ignition frequency is seen. However, this effect is insufficient to solve the issue by itself. Further work is required. Oil ignition is currently the most probable root cause, but other phenomena such as DI injector dribble cannot be ruled out.

The results from the project are, in chapter 0, summarized into recommendations for the development of a Euro 6 compliant dual fuel engine. Two key challenges must be addressed for this development to succeed: an aftertreatment system which allows for low exhaust temperatures must be available, and the root cause of pre-ignitions should be determined and suitable countermeasures found.
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## Definitions/Abbreviations

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
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<tbody>
<tr>
<td>1D</td>
<td>One-dimensional</td>
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<tr>
<td>AFR</td>
<td>Air to fuel ratio</td>
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<td>AFRdiesel</td>
<td>Stoichiometric air to fuel ratio for diesel</td>
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<tr>
<td>AFRmethane</td>
<td>Stoichiometric air to fuel ratio for methane</td>
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<td>ATDC</td>
<td>After combustion top dead center</td>
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<td>AVT</td>
<td>Active valve train</td>
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<td>BMEP</td>
<td>Brake mean effective pressure</td>
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<td>BSCO</td>
<td>Brake specific emissions of carbon monoxide</td>
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<td>BSHC</td>
<td>Brake specific hydrocarbon emissions</td>
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<tr>
<td>BSNOX</td>
<td>Brake specific emissions of nitrogen oxides</td>
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<td>BTDC</td>
<td>Before combustion top dead center</td>
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<tr>
<td>CA</td>
<td>Crank angle with reference to combustion TDC</td>
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<tr>
<td>CA50</td>
<td>Crank angle of 50% heat release</td>
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<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
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<td>CI</td>
<td>Compression ignited</td>
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<tr>
<td>CNG</td>
<td>Compressed natural gas</td>
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<tr>
<td>CO</td>
<td>Carbon monoxide</td>
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<tr>
<td>CO₂</td>
<td>Carbon dioxide</td>
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<tr>
<td>CoV</td>
<td>Coefficient of variation of IMEP</td>
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<tr>
<td>DDF</td>
<td>Diesel dual fuel</td>
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<tr>
<td>DI</td>
<td>Direct injection</td>
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<tr>
<td>DME</td>
<td>Dimethyl ether</td>
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<td>DPF</td>
<td>Diesel particulate filter</td>
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<tr>
<td>ECM</td>
<td>Engine control module</td>
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<tr>
<td>EDX</td>
<td>Energy dispersive X-ray spectroscopy</td>
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<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
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<tr>
<td>EPA</td>
<td>Environmental protection agency</td>
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<tr>
<td>EVC</td>
<td>Exhaust valve closing</td>
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<tr>
<td>EVO</td>
<td>Exhaust valve opening</td>
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<tr>
<td>FID</td>
<td>Flame ionization detector</td>
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<tr>
<td>FT</td>
<td>Fischer-Tropsch</td>
</tr>
<tr>
<td>GHG</td>
<td>Greenhouse gas</td>
</tr>
<tr>
<td>GUI</td>
<td>Graphical user interface</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>HCCI</td>
<td>Homogenous charge compression ignition</td>
</tr>
<tr>
<td>HD</td>
<td>Heavy duty</td>
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<tr>
<td>IMEP</td>
<td>Indicated mean effective pressure</td>
</tr>
<tr>
<td>IVC</td>
<td>Intake valve closing</td>
</tr>
<tr>
<td>IVO</td>
<td>Intake valve opening</td>
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<tr>
<td>LSPI</td>
<td>Low speed pre-ignition</td>
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<tr>
<td>MBT</td>
<td>Combustion phasing of maximum brake torque</td>
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<tr>
<td>MN</td>
<td>Methane number</td>
</tr>
<tr>
<td>NG</td>
<td>Natural gas</td>
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<tr>
<td>NOₓ</td>
<td>Oxides of nitrogen; nitric oxide and nitrogen dioxide</td>
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<tr>
<td>OP</td>
<td>Operating point</td>
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<tr>
<td>PPCI</td>
<td>Partially premixed compression ignition</td>
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<tr>
<td>Abbreviation</td>
<td>Full Form</td>
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<tr>
<td>ppm</td>
<td>Parts per million</td>
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<tr>
<td>PWM</td>
<td>Pulse width modulation</td>
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<tr>
<td>RCCI</td>
<td>Fuel reactivity controlled compression ignition</td>
</tr>
<tr>
<td>RME</td>
<td>Rapeseed-oil methyl ester</td>
</tr>
<tr>
<td>ROHR</td>
<td>Rate of heat release</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions per minute</td>
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<tr>
<td>SACI</td>
<td>Spark assisted compression ignition</td>
</tr>
<tr>
<td>SCR</td>
<td>Selective catalytic reduction</td>
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<tr>
<td>SEM</td>
<td>Scanning electron microscope</td>
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<tr>
<td>SI</td>
<td>Spark ignited</td>
</tr>
<tr>
<td>SN</td>
<td>Swirl number</td>
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<tr>
<td>SOC</td>
<td>Start of combustion</td>
</tr>
<tr>
<td>SOE</td>
<td>Start of energizing of injector coil</td>
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<tr>
<td>STD</td>
<td>Standard</td>
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<tr>
<td>TDC</td>
<td>Top dead center</td>
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<tr>
<td>TN</td>
<td>Tumble number</td>
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<tr>
<td>TPZ</td>
<td>Trapezoid</td>
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<tr>
<td>WTW</td>
<td>Well to wheel</td>
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<tr>
<td>$\lambda$</td>
<td>Air excess ratio</td>
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11 Summary of publications/Contributions

Six papers are appended. Unless noted otherwise, the writing of the publications along with all measurements, simulations and evaluation of data were performed by the author of this thesis, Fredrik Königsson, and reviewed by Professor Hans-Erik Ångström. Per Stålhammar contributed to the planning of the investigations in the cases where he is listed. All conference papers have been presented by Fredrik Königsson.


This paper characterizes the unconventional combustion modes available in the port injected Diesel Dual Fuel, DDF, engine; PPCI and HCCI/RCCI. They are compared to diesel and conventional DDF operation with respect to emissions and efficiency. Very little work was previously presented on the topic of diesel-methane HCCI/RCCI. To the author's knowledge, the application of a late pilot injection to trigger the combustion, in combination with an early diesel injection which conditions the bulk gas and enables PPCI combustion, is new. The comparison between PPCI, RCCI, DDF combustion and diesel operation is novel.

SAE Technical Paper 2011-01-2223, presented at the 2011 SAE Commercial Vehicle Engineering Congress in Chicago, USA

The objective of this paper was to characterize and investigate the potential for DDF combustion, utilizing all degrees of freedom available in a modern diesel engine. The paper discusses the effects of different parameters: combustion phasing, inlet temperature, EGR and diesel substitution rate. The main contribution from this paper is the presentation of the parameter variations in a clear and consistent way which has previously been lacking. The paper’s focus on the effect of each parameter on the tolerance for lean operation is new.

SAE Technical Paper 2012-01-0826, presented at the 2012 SAE World Congress in Detroit, USA

This paper investigated the factors causing high injector tip temperatures in a DDF engine and the underlying mechanisms which transfer heat to and from the injector tip. Parameter sweeps of each influential parameter were carried out and evaluated. In addition to this, a simple and useful model was constructed based on the heat balance of the injector tip. No previous work addresses the issue of nozzle tip temperatures in DDF engines. The entire content of the paper is therefore new.

SAE Technical Paper 2013-01-0848, presented at the 2013 SAE World Congress in Detroit, USA
This paper investigates how the sources of hydrocarbon emissions vary as engine operating conditions change. Five pistons with known increases to the topland volume were manufactured. For each of these pistons, parameter sweeps were carried out. Extrapolation of the data to a hypothetical topland volume of zero enabled the breakdown of hydrocarbon emissions into two parts; one part which originates from the topland crevice and one part which originates from other sources. The measurements presented in this publication were carried out by a Master’s student, Johannes Kuyper, under the supervision of the present author, Fredrik Königsson, who also performed the evaluation of the data and the writing of the publication. Previous work exists which investigates the sources of HC emissions for SI engines, mainly for gasoline fuel and stoichiometric operation. The breakdown of HC emissions into different categories for the combination of lean operation and CNG has not been previously shown. Additionally, the effect of parameter settings on the contributions to total HC emissions from quenching and crevices is new.


A fully variable valve train was used in combination with a high-swirl cylinder head to create different in-cylinder air flows. The effect of these air flows, swirl and tumble, on dual fuel combustion at high \( \lambda \) was investigated with regards to combustion, emissions and heat transfer. The measurements in the steady-state flow rig were carried out at Scania CV under the supervision of the co-author of the publication, Henrik Dembinski who also contributed with the GT-power simulations and supported the experimental work. Some of the measurements during this investigation were carried out by a Master’s student, Hans-Christian Nilsen-Vinnars under the supervision of the present author. Most of the measurements presented in the publication and in this thesis were carried out by the present author, Fredrik Königsson, along with the evaluation of the data and the writing of the publication. The combination of a fully variable valve train with DDF combustion has not been subject to prior investigations. The behavior of DDF combustion over a comprehensive swirl-tumble-range has, for this reason, not been previously documented. The new findings include the observation that swirl increases the oxidation of HC returning from crevices and that DDF combustion is insensitive to tumble.

**Paper VI: Nozzle Coking in CNG-Diesel Dual Fuel Engines. Fredrik Königsson, Per Risberg, Hans-Erik Ångström.**


The paper investigates injector nozzle coking during dual fuel operation as a function of different parameter settings and at different nozzle tip temperatures. Diesel fuel contaminated by zinc neodecanoate was used during the tests and nozzle flow rate was measured in a steady state test bench before and after the engine tests were performed. The single cylinder measurements were performed by two Master’s students, Martin Thyrestam and Marcus Svensson, under the supervision of Per Risberg. The publication was written in collaboration with Per Risberg who contributed to an equal degree as the present author, Fredrik Königsson, to the evaluation of the data and the writing of the paper. Since no prior publications have been found which addresses the
topic of injector nozzle coking during dual fuel operation, the entire content of the publication is new.