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Capabilities & Challenges

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Capabilities & Challenges

by Nikolaos Dimitrakopoulos

Thesis for the degree of Doctor of Philosophy
Thesis advisors: Prof. Martin Tunér
Faculty opponent: Prof. Martti Larmi, Aalto University

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Abstract
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Gasoline Partially Premixed Combustion (PPC) is an alternative combustion concept that can offer both high indicated efficiency and low exhaust emissions in terms of NO_x and soot, compared to conventional diesel combustion (CDC). Previous research has shown that this concept can work well with gasoline fuels of different octane ratings and can be used both in light and heavy duty engines. Although PPC has been tested substantially in research engines, results from engine designs that are closer to production are limited.

In this thesis, PPC is evaluated in a multi-cylinder light duty diesel engine. Results show that while it can perform well with both low octane RON75 and higher octane RON90 gasoline, the available load range is limited compared to similar diesel operation. Despite that, efficiency is high, with gross indicated numbers of around 48 %, while brake efficiency reaches up to 41 %. Soot emissions are improved compared to diesel while NO_x emissions are in similar numbers. A reason for that is the limited use of EGR compared to previous studies. This was deemed necessary to improve the upper achievable load of the engine.

As the use of EGR was a limiting factor, an evaluation of the two possible EGR routes was performed, to investigate the possible gains compared to single route operation. Results show that by combining routes possibility for a 4 % gain in efficiency could be found. Also, low load operation is limited due to the type of combustion and the fuel that is used. A minimum amount of temperature is necessary to promote combustion at lower loads and that is not always possible. By utilizing the glowplugs of a diesel engine, combustion stability can be improved, helping reducing the low load limit.

Finally these engine results are evaluated on a simulation model of a hybrid PPC powertrain. Even with an unoptimized diesel engine under PPC conditions, fuel consumption and emissions are comparable to a similar diesel powertrain from a production vehicle, showing that with further development, PPC can become a future possibility that can replace the less efficient SI engine.

Key words
Partially Premixed Combustion, PPC, Gasoline, Renewable gasoline, Efficiency, Light-duty engine, Hybrid powertrain, Gas exchange

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Lund University

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

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

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Life is fair and full of constant good luck

Contents

List of publications

This thesis is based on the following publications, referred to by their Roman numerals:

I **PPC Operation with Low RON Gasoline Fuel. A Study on Load Range on a Euro 6 Light Duty Diesel Engine**
N. Dimitrakopoulos, G. Belgiorno, M. Tunér, P. Tunestål, G. Di Blasio, C. Beatrice
9th International Conference on Modeling and Diagnostics for Advanced Engine Systems, COMODIA, 2017

II **Effect of EGR routing on efficiency and emissions of a PPC engine**

N. Dimitrakopoulos, G. Belgiorno, M. Tunér, P. Tunestål, G. Di Blasio
Applied Thermal Engineering, Vol. 152, pp. 742–750, 2019

III **Evaluation of engine efficiency, emissions and load range of a PPC concept engine, with higher octane and alkylate gasoline**
N. Dimitrakopoulos, M. Tunér
Submitted to journal: Fuel

IV **Performance and emissions of a series hybrid vehicle powered by a gasoline partially premixed combustion engine**
Antonio García, Javier Monsalve-Serrano, Rafael Sari, **N. Dimitrakopoulos**, M. Tunér, P. Tunestål
Applied Thermal Engineering, Vol. 150, pp. 564–575, 2019

V **Investigation of the effect of glowplugs on low load PPC**

N. Dimitrakopoulos, M. Tunér
Submitted to SAE Powertrains, Fuels & Lubricants Meeting 2020

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Publications not included in this thesis:

I **Parametric Analysis of the Effect of Pilot Quantity, Combustion Phasing and EGR on Efficiencies of a Gasoline PPC Light-Duty Engine**
G. Belgiorno, **N. Dimitrakopoulos**, G. Di Blasio, C. Beatrice, P. Tunestål, M. Tunér
SAE Technical Paper 2017-24-84, 2017

II **Effect of the engine calibration parameters on gasoline partially premixed combustion performance and emissions compared to conventional diesel combustion in a light-duty Euro 6 engine**
G. Belgiorno, **N. Dimitrakopoulos**, G. Di Blasio, C. Beatrice, P. Tunestål, M. Tunér
Applied Energy, Vol. 228, pp. 2221–2234, 2018

Nomenclature

γ	Specific heat ratio, (C_p/C_v)
λ	Air/Fuel ratio (in relation to stoichiometric)
BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
CAD	Crank Angle Degree
CCS	Carbon Capture and Storage
CDC	Conventional Diesel Combustion
CN	Cetane Number
CO	Carbon monoxide
CO_2	Carbon dioxide
EGR	Exhaust Gas Recirculation
FMEP	Friction Mean Effective Pressure
FuelMEP	Fuel Mean Effective Pressure
GCI	Gasoline Compression Ignition
HCCI	Homogeneous Charge Compression Ignition
IEA	International Energy Agency
$IMEP_g$	Gross Indicated Mean Effective Pressure
$IMEP_n$	Net Indicated Mean Effective Pressure
IPCC	Intergovernmental Panel on Climate Change

LHV	Lower Heating Value
LTC	Low Temperature Combustion
MK	Modulated Kinetics
NO_x	Nitrogen Oxides
p.p.	Percentage point
PCCI	Premixed Charge Compression Ignition
PMEP	Pumping Mean Effective Pressure
PPC	Partially Premixed Combustion
PPCI	Partially Premixed Compression Ignition
PPRR	Peak Pressure Rise Rate
r_c	Compression ratio
RoHR	Rate of Heat Release
RON	Research Octane Number
SACI	Spark Assisted Compression Ignition
SI	Spark Ignition
TDC	Top Dead Center
UHC	Unburned Hydrocarbons
UNIBUS	Uniform Bulky Combustion System
VGT	Variable Geometry Turbine

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Reaching the end of a long journey in the academic and research field, it is time to look back on how all these started and thank all the people who were involved in that.

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Thank you
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Abstract

Internal combustion engines have been the most used engine design when it comes to vehicle propulsion and transportation. But as the number of vehicles increase, new problems arise as well. Engine emissions such as carbon dioxide that has an effect on a global scale and other harmful emissions that affect on a local scale such as soot and nitrogen oxides are on the rise, forcing the countries to take measures on controlling and reducing them.

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In this thesis, PPC is evaluated in a multi-cylinder light duty diesel engine. Results show that while it can perform well with both low octane RON75 and higher octane RON90 gasoline, the available load range is limited compared to similar diesel operation. Despite that, efficiency is high, with gross indicated numbers of around 48 %, while brake efficiency reaches up to 41 %. Soot emissions are improved compared to diesel while NO_x emissions are in similar numbers. A reason for that is the limited use of EGR compared to previous studies. This was deemed necessary to improve the upper achievable load of the engine.

As the use of EGR was a limiting factor, to be able to extend on that, an evaluation of the two possible EGR routes was performed, to investigate the possible gains compared to single route operation. Results show that by combining routes possibility for a 4 % gain in efficiency could be found. Also, low load operation is limited due to the type of combustion and the fuel that is used. A minimum amount of temperature is necessary to promote combustion at lower loads and that is not always possible. By utilizing the glowplugs of a diesel engine, combustion stability can be improved, helping reducing the low load limit.

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Popular summary in English

Internal combustion engines for the last 100 years, have been the best choice for vehicle propulsion due to their simplicity, durability and low cost of operation. More specifically, diesel engines are the predominant option in vehicles for land transportation due to their low fuel consumption. But as the amount of vehicles start increasing globally, new problems arise from their use.

The fuel consumption of vehicles has lead to unsustainable increase of the CO_2 in the atmosphere, while the rest of the exhaust emissions cause local emission problems, especially from nitrogen oxides (NO_x) and soot, which are the main emissions coming out of diesel engines. Due to this, legislation becomes more strict every time, forcing manufacturers to invest into new technologies that could potentially improve the fuel consumption, while keeping the emissions low at the same time.

Low temperature combustion (LTC) or gasoline compression ignition (GCI), is a group of combustion concepts that could potentially provide these two possibilities at the same time. The basis of the low temperature combustion is that a highly diluted mixture of air and fuel when compressed at high pressure, will combust at temperature that is low enough to keep the NO_x emissions low, while the combustion will be fast, keeping the fuel efficiency high. Another positive of low temperature combustion is that it can be combined with fuels that can be produced in a renewable way and can be gasoline-like fuels or alcohols. One major issue though with the low temperature combustion concepts is that it is difficult to control, compared to the conventional combustion as it is used today.

Gasoline partially premixed combustion or PPC is one of these newer concepts which appeared the last 15 years. The aim is to improve over the older concepts, providing better control on the combustion process while maintain the positive aspects from them. A substantial amount of research on PPC was performed on single cylinder research engines, giving positive results for the future. In this thesis, the same PPC concept is evaluated in a complete production multi-cylinder engine. The aim is to see how well the previous findings can be applied in an engine with minimum hardware modifications.

Results show that when PPC is used in a multi-cylinder diesel engine, fuel consumption is comparable to regular operation with diesel fuel while soot emissions are much lower. NO_x emissions seem to be at a similar level. The engine also cannot operate within similar power range compared to diesel operation. The reasons behind these results comes down to the unoptimized design of the engine for this type of operation. By investing into further research of a PPC type engine, it can evolve into a possible replacement for the less efficient SI engine.

Evaluation of Gasoline PPC in a Multi-cylinder Engine: Capabilities & Challenges

Chapter 1

Introduction

Since their invention in 1876 (gasoline) and 1896 (diesel) [1], internal combustion engines have become the most prominent prime movers for light duty and heavy duty vehicles around the world. Reasons for that are their simplicity, reliability and adaptability to different kind of fuels, resulting in over 1 billion vehicles (as of 2017) [2] around the world. Combined with low cost of usage, vehicles helped improve the economic growth of countries leading to better living standards.

While the internal combustion engine revolutionized the modern society it also created new problems. Carbon dioxide (CO_2) which is also produced from the combustion of fossil fuels has been constantly increasing in the atmosphere for more that 100 years, while we have detailed measurements for the last 60 years (figure 1.1). As carbon dioxide is a strong greenhouse gas, it leads to anthropogenic global warming, meaning that the increase of the average atmospheric temperature is an effect of the human activity, industry and agriculture.

In order to avoid severe effects from the rise of the global temperature, the Intergovernmental Panel on Climate Change (IPCC) says that it should be limited to less that 2 °C or even better a maximum of 1.5 °C compared to pre-industrial levels. They also state that in order to achieve that, a 25 % reduction in CO_2 is needed by 2030 and net zero CO_2 emissions should be produced by 2070 to 2080 [4].

The transportation sector contributes a large amount of the global CO_2 production, with the International Energy Agency (IEA) stating that for 2017 around 25 %, (8000 million tonnes) of the CO_2 that is produced from fuel combustion, comes from the transportation sector and 74 % of that CO_2 is from road transport [5]. In European Union, the transport sector accounts for more than 29 % of all CO_2 emissions. Passenger cars produce the majority of transportation CO_2 (50 %), while heavy duty vehicles come second (21 %). Aviation, light duty trucks and non-road applications cover the rest 29 % [6].

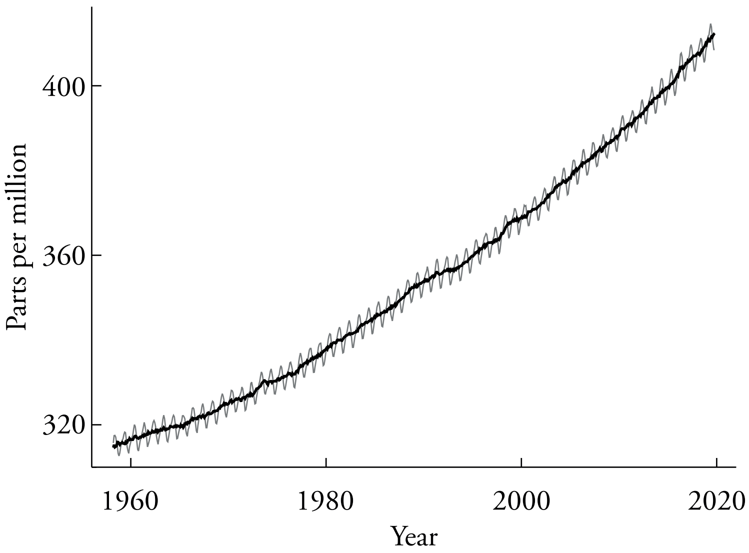

Figure 1.1: Carbon dioxide concentration [ppm] in atmosphere for the last 60 years, solid line represents average data, while jagged line is seasonal variation, data reproduced from [3].

One "simple" way to reduce the CO_2 emissions from the transportation sector is to improve overall engine efficiency, since the lower amount of fuel is used per distance or time unit, the less CO_2 will be emitted. In the case of heavy duty engines, that was the norm, as it can be seen in figure 1.2, brake efficiency was increasing steadily through the years, until year 2002 when there was a step decline that took eight years to recover. The reason for that was that more strict emission regulations took place at that time and had a negative impact on engine efficiency, showing that the manufacturers had to find compromise between these two requirements, good fuel economy and exhaust emissions within legislation limits.

But as the effects of climate change become more apparent [4] the countries push towards lower overall CO_2 emissions forcing the automotive sector to drastically improve on that. As an example, the European Union demands that by 2021 the fleet average CO_2 emissions for passenger cars will be lower or equal to 95 g/km [8]. Although it has become clear that improving the engine efficiency is a straightforward way to reduce fuel consumption and therefore CO_2 emissions, it might not be enough only by itself to reach the future CO_2 targets and will need to be accompanied together with other new technologies such as biofuels, hybridization, electrification and Carbon Capture and Storage (CCS) [9].

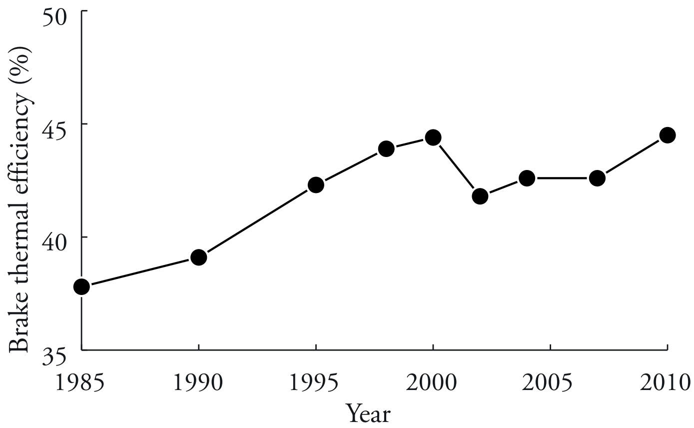

Figure 1.2: Long haul truck engine brake thermal efficiency at cruising condition, reproduced from [7].

1.1 Motivation & scope

As it was mentioned before, engines tend to produce a significant amount of the total CO_2 that is produced from fossil fuel combustion as well as other harmful exhaust emissions. Therefore the evaluation of the different possibilities that partially premixed combustion has on reducing these two is a strong motivation for this thesis.

Although there are many results with PPC, most of them have this concept evaluated in single cylinder engines. It is therefore interesting to apply it in a multi-cylinder engine which has a production intake and exhaust system, to quantify and evaluate its performance (in terms of efficiency and emissions) with stricter boundary conditions and see how close it is, compared to the older studies. After all these are evaluated, recommendations on how this concept can be improved in the future can be proposed.

1.2 Contributions

The main contributions of the thesis is the complimentary knowledge of using gasoline PPC on a multi-cylinder diesel engine. The possibilities and the drawbacks are discussed, on the efficiency, emissions and performance. Ways to circumvent these are presented as well as possibilities to expand more on PPC with altered hardware decisions. These can be summarized as:

- PPC operation is possible in a diesel engine with minimal modifications, resulting in efficiency comparable to diesel operation with improved soot emissions, but limited in the available load range.

- The brake efficiency can be improved if a split route EGR configuration is used. Depending on the speed and load, the more optimal route can be used, leading to improved gas exchange efficiency.
- Low load operation can be improved if the start of combustion is supported with added heat from a glowplug, leading to improved stability.
- A PPC operated engine can be combined with a hybrid drivetrain to give improved results in terms of efficiency an emissions, compared to a similar vehicle with a diesel engine.

Chapter 2

Literature

Here a description of the different combustion concepts which were developed over the years, will be presented. The conventional diesel combustion is the predominant engine operation that can provide high fuel efficiency, while the other concepts aim to improve both on efficiency and emissions over that.

2.1 Conventional diesel combustion

Modern diesel engines operate in a way that is referred in literature as Conventional Diesel Combustion (CDC). Under CDC, fuel is injected in the cylinder close to top dead center, in high pressure and temperature conditions and after a short delay, it is ignited. This implies that a fuel with low autoignition resistance is necessary, meaning a high cetane number (CN). This procedure ensures excellent combustion control since the combustion event, combustion timing and engine load can be controlled by the start of the injection event. Also, because the fuel injection happens moments before the actual combustion, high fuel combustion efficiency can be achieved, that can reach over 99 %.

As the fuel is injected just before the combustion, there is no possibility for uncontrolled combustion as in gasoline engines (knocking and preignition) making this engine design ideal for high compression ratios (r_c). As the fuel is ensured that it will always ignite due to high temperature, it is not necessary to control the air to fuel ratio as tightly as in a gasoline engine therefore intake throttling is not used and lean mixtures can be used through the whole load range. All these things help improving fuel conversion efficiency even further. Literature shows that light duty diesel engines can reach up to 44 % brake efficiency while heavy duty engines can go up to 48 % efficiency. As a comparison, SI engines only recently (2018) reached the 40 % mark. Because of these reasons diesel engines are the predominant engines in heavy duty applications.

Although CDC has the advantage of high efficiency, due to the way the combustion happens, there is a disadvantage in terms of emissions, more specifically soot and NO_x emissions. The reason for the high amount of these two emissions was explained in Dec's work from 1997 [10]. From his conceptual model of a quasi-steady diesel fuel jet (figure 2.1) it is explained that NO_x emissions are created at a thin layer in the diffusion flame, where local conditions are close to stoichiometric ($\lambda \approx 1$) and temperature is at its highest level. The same model explains also how soot is produced. In the early parts of the combustion,

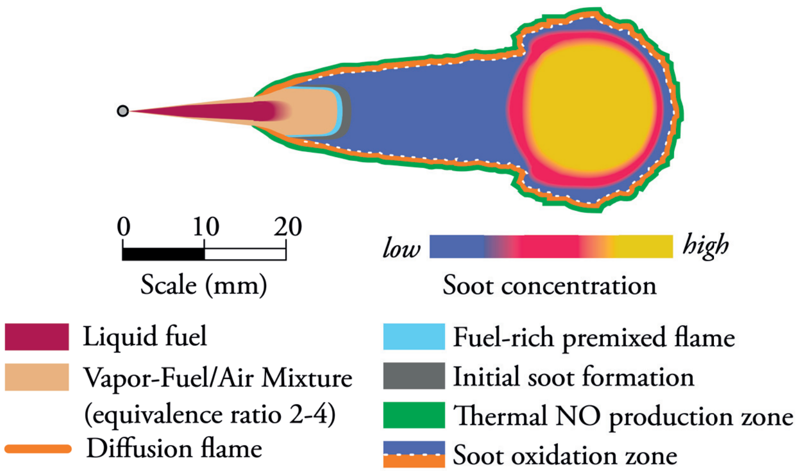

Figure 2.1: Conceptual model of a diesel spray combustion, reproduced from [10].

a fuel rich (ϕ between 2 and 4) premixed combustion takes place. In that region soot starts forming and while a significant amount is oxidized by the end of the combustion event, it is not enough to meet the regulatory levels.

2.2 Low temperature combustion

HCCI

As the emission regulations started becoming more stringent, the cost of developing newer diesel engines increased, also due to the higher cost of the aftertreatment system needed to reduce NO_x and soot to harmless emissions. As an example, the aftertreatment system of a Euro 6 diesel engine was estimated around \$1600 [11]. It was therefore beneficial for the automotive sector to look into alternative combustion concepts which could offer high fuel efficiency as well as low emissions at the same time, or emissions that would be easier to control without the need of sophisticated aftertreatment systems. A promising alternative to CDC is a type of combustion that is now known by the name of low temperature combustion (LTC).

Probably the most known low temperature combustion concept is the Homogeneous Charge Compression Ignition (HCCI). It is based on kinetic combustion. Under HCCI, a fully premixed (homogeneous) air/fuel mixture auto-ignites after being compressed to high temperature and pressure. As the mixture is homogeneous, soot emissions are minimal and since the mixture is also very lean, combustion temperature is low enough to keep the NO_x production very low. HCCI was proposed as an alternative concept for two stroke engines in 1979 [12] and in 1983 as a possibility for four stroke engines [13]. Moreover, HCCI can be used with a wide variety of fuels, of different octane ratings, as has been shown in [14, 15]. Although the high efficiency and low emission claims are valid, HCCI never made it into production and is still an interesting research topic. The reason for that is the same reason that gives HCCI its high efficiency potential. The combustion is chemical kinetics controlled, meaning that the ignition timing is affected by the in-cylinder temperature, pressure and mixture equivalence ratio, which cannot be easily controlled as in regular SI or CI engines. Another drawback of HCCI is also the limited load range. At low loads, due to high dilution the combustion temperature is low enough that fuel does not burn properly leading to high carbon monoxide and unburned hydrocarbon emissions. At high loads, the increased cylinder pressure and the higher peak pressure rise rate limit the achievable load to values lower than a naturally aspirated gasoline engine. Efforts were done trying to increase the load with the use of boosting, with positive results, but the control issues remained [16].

Diesel PCCI

Although HCCI, as developed in the 1990s never went into production, the results helped some manufacturers to develop their own ideas based on these findings.

Nissan motor company developed a concept named Modulated Kinetics, (MK) [17, 18] to reduce soot and NO_x emissions in a diesel engine. The concept is based around the understanding that use of EGR helps with the control of NO_x production, while diffusion combustion leads to high soot emissions. In order to control both emissions at the same time a low-temperature, premixed combustion is needed. With this concept high amounts of EGR and lower than usual compression ratio were used to control the NO_x emissions, while a late direct injection of fuel prolonged the mixing time before the combustion leading to premixed combustion and creating minimal soot emissions. The company claimed an reduction of NO_x up to 98 % compared to their older diesel engines. Although this concept worked well at low loads, the company concluded that at higher loads there is not enough time to premix the fuel and CDC has to be used.

Around the same time, Toyota Motor Corporation was investigating a similar concept aiming at the same outcomes, reduced soot and NO_x emissions while maintaining high efficiency in diesel engines. They named their design Uniform Bulky Combustion System, (UNIBUS) [19] describing it as HCCI in direct injection diesel engine. Similar to MK, they used an engine with reduced compression ratio and high amounts of EGR for the NO_x control, but they opted for a double injection strategy to control the premixing of the fuel. A low duration early injection was used to start low temperature reactions, while a late main injection used to control the start of combustion and the load. Although a similar strategy, with a pilot and a main injection is used in many modern diesel engines, it is not the same as this concept because in the pilot main strategy the injections are placed close together, while in UNIBUS the injection separation could be as long as 67 crank angle degrees. Toyota came also to the same conclusion regarding high loads, that is not possible to premix the fuel in the available time for combustion and used a CDC strategy. According to Toyota, they released a diesel engine to the market in early 2000s using the UNIBUS low load strategy.

As this combustion strategy showed positive results in terms of emission control, it was further investigated in the academic field, verifying that high amount of EGR and an amount of fuel premixing could reduce NO_x and soot simultaneously [20–23].

Gasoline PCCI, PPC

As discussed before, diesel premixed charge compression ignition (PCCI), or partially premixed compression ignition (PPCI), despite its positive results in terms of emissions at low loads, had a limited operating range. This was due to the fuel used. Diesel fuel is blended in a fuel refinery in such a way that is a highly reactive fuel with low auto-ignition limit. It would be logical, if the aim is to get long ignition delay, to use a fuel which has the opposite behavior of diesel. The most common fuel for that use would be gasoline which had been used before on the early HCCI research. Early results [24–26] with gasoline PCCI/PPC showed that gasoline gives increased ignition delay which as a result helped improved the achievable load while maintained low emissions up to 15 bar $IMEP_g$ with a maximum indicated efficiency of around 47 %.

After these early positive results, further research continued on the gasoline PPC both in heavy duty [27, 28] and light duty engines [29–31]. Because PPC benefits from high octane fuels, a lot of research has been done also on different fuels, gasoline with different octane rating, ethanol and methanol [32–36]. But higher octane fuels created a new problem similar to what had happened with HCCI. While they can operate well at higher loads, they don't perform as good at lower loads. To overcome this limitation some researchers suggest that a lower octane gasoline, with a RON number around 70 would be beneficial because it would give wider load range possibilities [37–44]. Although a drawback of the lower octane fuel would be that at higher loads the combustion would shift more towards mixing control, increasing the emissions.

While research on this type of combustion is still going on, the results so far are very promising in terms of engine efficiency. Results from light duty engines converted for PPC operation show that gross indicated efficiency ranges from 47 to 50 % [39, 45–47] while it has been demostrated a brake efficiency number of 43 % from a new engine design, built around PPC type of combustion [48]. Similarly from the heavy duty engine side, all of the research so far has been performed on modified diesel engines, probably reducing the maximum efficiency compared to a design focused entirely on PPC. Still, in heavy duty engines, gross indicated efficiency numbers between 47 to 53 % [33, 38, 49–51] have be published. These higher efficiency numbers have been reached with the use of methanol and high compression ratio (r_c=27) [36] while simulations on an optimized engine operating with methanol and PPC show that a brake efficiency of around 48 % can be achievable [52, 53].

SACI

At the same time, during the development of the PPC concepts, other ideas were tested, trying to overcome the limitations of HCCI and make LTC a viable combustion strategy for future engines. One of them is Spark Assisted Compression Ignition or SACI. Reasoning behind this development is that the combustion control can be done much easier when using a spark plug. The spark is used to initiate combustion as in a regular SI engine. The expanding flame front, flame propagation as in a normal SI engine, increases the temperature and the pressure of the remaining gas, causing it to go through autoignition, rapidly combusting the rest of the fuel air mixture.

Due to possibilities for improved control of the autoignition process during steady state as well transient operation, a substantial amount of research has also been done on this specific concept both on heavy [54, 55] and light duty engines [56, 57]. Optical studies verify that the early combustion behaves as a regular SI combustion with flame propagation which, as the pressure and temperature increase triggers the autoignition of the remaining unburned gas [58–60].

As the spark plug is used for combustion control it is much easier to combust different types of fuels that have wide octane rating numbers, similarly to PPC. Under SACI, combustion was possible with RON91 gasoline, E85 [61] and even RON104 gasoline [62].

Although SACI has potential for better combustion control, it is still limited by the same reasons as HCCI in terms of maximum load. Because of that, in naturally aspirated form, the load limit is around 8 to 10 bar $IMEP_n$, [63–66] although results with boosted operation could reach up to 27 bar $IMEP_g$ [62].

Future of LTC

Based on all the previous studies, at least three companies saw a potential for implementing a premixed type compression ignition concept with gasoline fuel in their engine development. One of them is Delphi Technologies which started developing a light duty engine, operating with partially premixed combustion through the whole load range [67]. Through yearly developments [46, 68, 69], the last iteration of the engine [48] can reach 43 % brake efficiency and can operate up to 25 bar BMEP while maintaining the exhaust emissions within the legal limits.

Achates Power Inc., another US company, started developing a medium duty six cylinder engine, operating under GCI conditions. The unique feature of this engine is that is using an opposed piston design, claiming that with this layout they are able to minimize heat transfer losses. Having developed a 4.9 L research engine, they claim that it could achieve up to 44 % brake efficiency [70]. Using the research engine results they simulated a higher displacement heavy duty engine of 9.8 L claiming up to 55 % brake efficiency at part load conditions [71, 72]. They also developed a light duty 2.7 L engine with the same configuration, that could achieve 31.7 % average efficiency when evaluated on different driving cycles, claiming an 11 % higher efficiency compared to a competitive gasoline engine [73].

Finally, while these two manufacturers continue developing engines based on the LTC combustion concepts, Mazda Motor Corporation released in the end of 2019 an engine that they claim that can operate both in LTC and regular SI mode (mode switching). In order to be able to control the combustion they opted for a combustion process that has similarities to SACI, that was mentioned previously. Information released by the company claims a 20 % better efficiency compared to their best gasoline engine at the moment, suggesting that this new engine would be at around 44 % brake efficiency.

Chapter 3

Methodology

All of the experimental work on this thesis was based on a Volvo VED-D4 light duty diesel engine. Engine operation was supported with an computer based operating environment, emission measurement system and data post processing software. Further information for these components will be presented here.

3.1 Engine setup

The engine that was used, was a Volvo VED-D4 light duty diesel engine. It is a four cylinder engine with a total displacement of 1969 cm^3 and at the time of its introduction to the market it complied with the Euro 6 emission limits. The engine was kept as much as possible to its original configuration. This includes the gas exchange system which consists of two turbochargers in serial configuration. The reasoning behind this design is that a small turbo performs well at low engine speeds, while a big turbo operates better at high engine speeds. At low speed, all the exhaust gases pass through the high pressure turbine, increasing the intake pressure fast, which leads to improved transient responce and higher low-end torque. As the engine speed increases, a part of the exhaust gases is routed to the low pressure turbine. This helps keeping the high pressure turbo within its operating speed range and getting more boost from the low pressure turbo at intermediate speeds. Finally at higher engine speeds, the high pressure turbine is bypassed completely and all the exhaust flow goes through the low pressure turbine which is designed for high speed and high power operation. Because of these reason the high pressure turbocharger has Variable Geometry Turbine (VGT) technology for better boost control, while the low pressure turbocharger uses a simple wastegate setup (figure 3.1).

Although most of the original hardware remained on the engine, some modifications were

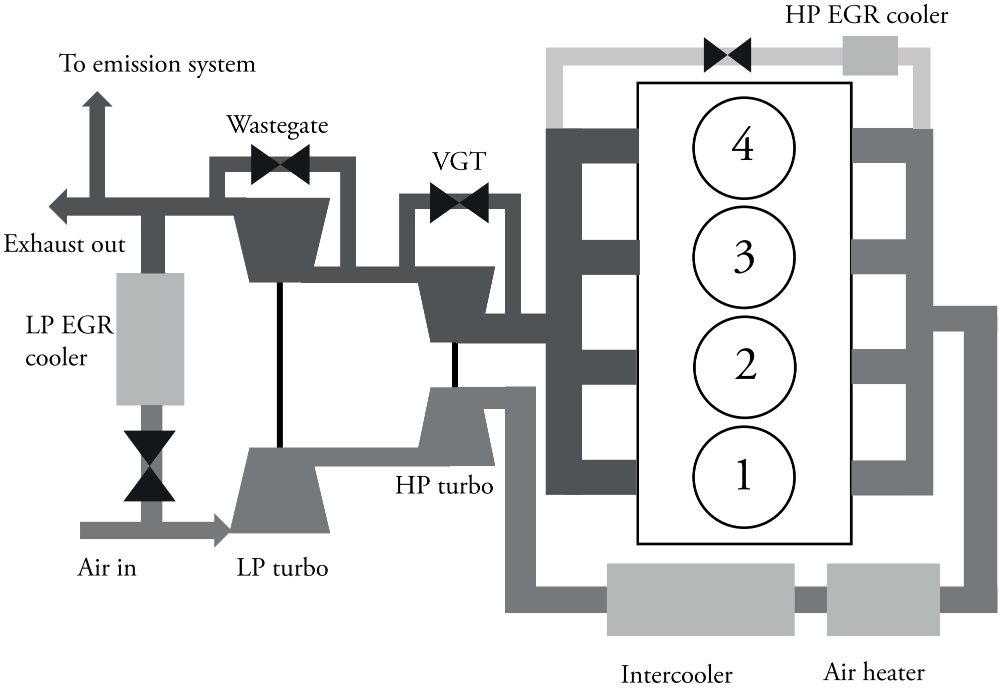

Figure 3.1: Engine overview

needed to be able to operate the engine under low temperature combustion conditions. The original air to air intercooler was replaced with a water to air intercooler, which provides better control of the intake air temperature and higher level of cooling compared to the alternative. After the intercooler, a 6 kW air heater was placed, which was necessary in order to reach intake temperatures that are generally not possible to get with regular engine operation. In the exhaust side, the engine was fitted with a cooled long route EGR system loop. Using a system like that, can provide higher EGR rates compared to the existing short route system and can also get the EGR gas temperature as low as the ambient temperature, if it is needed. The intake of the long route EGR was placed downstream the position of the after-treatment system, which in this case was removed. The output of the long route path merges with the intake pipes just before the inlet of the first compressor. A plate heat exchanger provided the necessary cooling. After the long route bypass, a backpressure valve was added, to provide control of the amount of the exhaust gases that go through the long route. Finally, the exhaust after-treatment system was removed, to be able to measure the raw emission output of the engine.

The engine is connected to a 300 kW electric motor intended to be able to operate the engine without combustion and provide the necessary load for the engine to function. Engine specifications can be summarized in the following table.

Table 3.1: Engine properties

Displacement	1969 cm^3
Number of cylinders	4
Bore × Stroke	82 mm × 93.2 mm
Compression ratio	15.8[–]
# of valves per cylinder	4 [–]
Diesel Injection System	Common rail (maximum injection pressure: 2500 bar)
Injector	Solenoid, 8-hole, 155° umbrella angle

3.2 Measurement systems

In order to be able to aquire usefull data from the engine, an array of measurement equipment are connected to it. These can be divided into fast sampling and slow sampling.

Fast sampling sensors

The fast sampling system is synchronized with the use of a Leine-Linde crank angle encoder. It can provide both angular speed/position with a resolution of 0.2 CAD as well as TDC signal. The crank angle encoder is used to provide signals for the following fast sampling sensors: The high sampling frequency pressure transducers which are AVL GH14D with a maximum pressure rating of 250 bar, the HBM T40B torque transducer, the four proprietary injection pressure sensors which are placed on each fuel injector and the injection signal sensor. All these fast sampling sensors are calibrated from the factory.

Slow sampling sensors

The rest of the sensors that are connected on the engine are referred as slow speed sensors, meaning that the sampling frequency is much lower compared to the fast sensors and is generally between 1 to 5 Hz. These sensors include the temperature probes, which are K-type thermocouples, the Keller PAA-23SY pressure sensors with a range of 0 to 5 bar or 0 to 10 bar depending on their placement, an ETAS ES635.1 exhaust lambda sensor and the emission systems. These sensors are also factory calibrated.

The emission systems include a Horiba Mexa 7100DEGR which measures the following emissions: carbon dioxide (CO_2) in exhaust and intake, carbon monoxide (CO) (non-dispersive infrared measurement technique both for CO_2 and CO), unburned hydrocarbons (UHC, flame ionization technique), nitrogen oxides (NO_x, chemiluminescence technique), exhaust oxygen (O_2, paramagnetic technique) and a AVL Micro Soot Sensor that measures particulate matter (PM) or soot. The two emission measurement systems were calibrated before each use with the use of specific calibration gases.

Finally the fuel flow is calculated with a Sartorius CPA-10001 fuel scale. Fuel weight is sampled every 0.2 s and then the fuel flow is calculated with a linear regression model. The sensors which were used are summarised in table 3.2.

Table 3.2: Sensor information

Measured value	Sensor	Range	Accuracy
Crank angle	Leine & Linde RSI503	0–6000 rpm	$\pm$ 0.02 CA
Cylinder Pressure	AVL GH14D	0–250 bar	$\pm$ 0.3 % FS
Injection Pressure	OEM sensor	0–2500 bar	$\pm$ 1 % FS
Pressure (other)	Keller 23SY	0–5 & 0–10 bar	$\pm$ 0.2 % FS
Torque	HBM T40B	0–5000 N m	$\pm$ 0.05 % FS
Temperature	Pentronic 8105000	0–1100 °C	$\pm$ 1 °C
Fuel Flow	Sartorius CPA-10001	0–10 kg	$\pm$ 0.1 g
CO	Horiba Mexa 7100DEGR	0–1500 ppm	1% FS
CO_2 / CO_2 EGR		0–16 % / 0–5 %	
NO_x		0–2000 ppm	
UHC		0–1500 ppm	
O_2		0–25 %	

3.3 Post processing

The post processing involves analyzing the saved raw engine data. Generally 300 engine cycles are saved for each measurement point and then analysed with the use of in house developed Matlab© program. Data processing revolves around three different calculations, heat release calculation from in-cylinder pressure data, emissions and engine efficiencies.

However, the in-cylinder pressure sensor can only provide relative pressure values and is necessary to match this relative pressure to a known absolute value, a process that is refered as pressure pegging. In this case, the in-cylinder pressure at intake BDC is considered to be equal to the pressure in the intake manifold. To reduce the effect of pressure fluctuations, the average pressure within $\pm$ 5 CAD around BDC is used for the actual pegging.

Heat release calculation

Although the energy input to the engine can be easily calculated from the fuel flow, it is also important to know the rate of the release of that energy because it helps with the study of the combustion process. In a metal engine that can be calculated from the in-cylinder pressure. To simplify the heat release analysis, some assumptions must be made. The combustion chamber is considered a closed system with uniform pressure and temperature and the first law of thermodynamics is used to calculate the rate of energy addition to it. With a procedure that is described in [74] the rate of heat released (RoHR) can be calculated with the following equation.

$$\frac{\partial Q}{\partial \theta} = \frac{\gamma}{\gamma - 1} P \frac{\partial V}{\partial \theta} + \frac{\partial V}{\gamma - 1} V \frac{\partial P}{\partial \theta} \tag{3.1}$$

where V is the engine volume at each crank angle degree, calculated from the engine geometry, P is the in-cylinder pressure per crank angle degree, and γ is the specific heat ratio (C_p/C_v) calculated from the intake gas mixture composition and temperature. It should be noted that from 3.1 the heat transfer is omitted, therefore the net or apparent heat release is calculated.

When the rate of heat release is known, the total or accumulated heat release can be calculated and from that, three useful values can be extracted. These are the CA10, CA50 and CA90 which represent the crank angle degree that the 10, 50 and 90 % of the total fuel energy has been released, with the most crucial of these three being the CA50 because it has a important effect on the engine efficiency.

Emissions

The exhaust emission analysers provide the different emission numbers in molar mass. These numbers then are calculated into specific emissions, indicated and brake, based on the procedure described in [75]. EGR_{rate} is calculated from the molar CO_2 concentration in the intake and in the exhaust as

$$EGR_{\mathrm{rate}} = \frac{xCO_{2intake}}{xCO_{2exhaust}} \tag{3.2}$$

Efficiency and mean effective pressure

The energy flow in an engine, from fuel energy to useful mechanical work (crankshaft output, known also as brake work) are illustrated in figure 3.2. Because engines come with a wide range of displacements, is much easier if these energy flows are expressed in a normalised scale. Mean effective pressure (MEP) is way to express that, because each energy flow is divided by the engine displacement V_d. Knowing that, each MEP can then be defined as,

Fuel Mean Effective Pressure, FuelMEP Fuel Mean Effective Pressure is defined as fuel energy per cycle divided with the engine displacement. Fuel energy is the fuel mass per

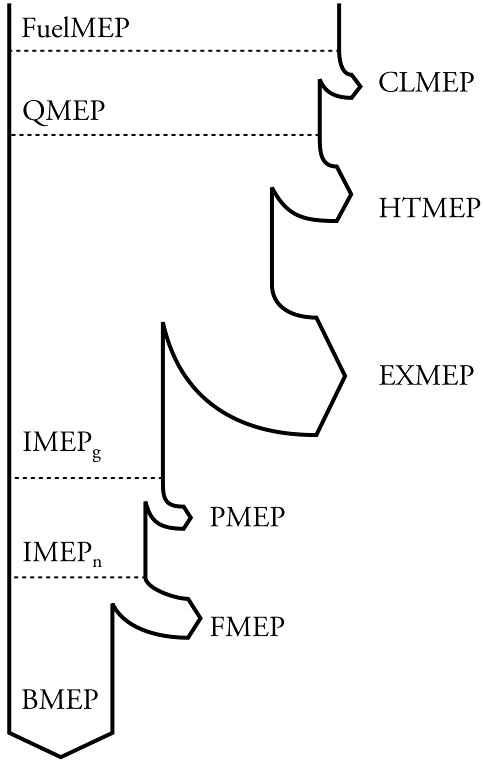

Figure 3.2: Energy flow in an engine, from fuel input to work output

cycle multiplied by the fuel lower heating value.

$$FuelMEP = \frac{m_f \cdot Q_{LHV}}{V_d} \tag{3.3}$$

Gross Indicated Mean Effective Pressure, $IMEP_g$ Gross Indicated Mean Effective Pressure, $IMEP_g$, is defined as the work to the piston during the compression and the expansion stroke divided with the engine displacement.

$$IMEP_g = \frac{1}{V_d}\int_{-180}^{180} pdV \tag{3.4}$$

Net Indicated Mean Effective Pressure, $IMEP_n$ Net Indicated Mean Effective Pressure, $IMEP_n$, is defined as the work to the piston during the entire four stroke cycle divided with the engine displacement.

$$IMEP_n = \frac{1}{V_d} \int_{-360}^{360} pdV \tag{3.5}$$

In both cases, zero degrees is defined at the Top dead center during the combustion part.

Pumping Mean Effective Pressure, BMEP Pumping Mean Effective Pressure is defined as the difference between $\mathrm{IMEP_n} - \mathrm{IMEP_g}$.

$$PMEP = IMEP_g - IMEP_n \tag{3.6}$$

Friction Mean Effective Pressure, BMEP Friction Mean Effective Pressure is defined as the difference between $\mathrm{IMEP_n} - \mathrm{BMEP}$.

$$FMEP = IMEP_n - BMEP \tag{3.7}$$

Brake Mean Effective Pressure, BMEP Brake Mean Effective Pressure is defined as the useful work per cycle divided by the displacement.

$$BMEP = \frac{P \cdot n_R}{V_d \cdot N} \tag{3.8}$$

where P is brake power [W], V_d is engine displacement [m^3], N is engine speed in [1/s] or in [Hz] and n_R is the number of crank revolutions per power stroke. With these definitions the different efficiencies can be defined as well.

Gross Indicated Efficiency

$$\eta_{gross} = \frac{IMEP_g}{FuelMEP} \tag{3.9}$$

Net Indicated Efficiency

$$\eta_{net} = \frac{IMEP_n}{FuelMEP} \tag{3.10}$$

Gas Exchange Efficiency

$$\eta_{ge} = \frac{IMEP_n}{IMEP_g} \tag{3.11}$$

Brake Efficiency

$$\eta_b = \frac{BMEP}{FuelMEP} \tag{3.12}$$

3.4 Energy balance

The calculations are performed using the following equation.

$$\dot{m}_f \cdot LHV_f = P_{brake} + P_{friction} + P_{pump} + \dot{Q}_{ht} + \dot{Q}_{EGR} + \dot{Q}_{exhaust} + \dot{Q}_{comb\ loss} \tag{3.13}$$

$\dot{m}_f \cdot LHV_f$ represents the power input to the engine from the fuel used ($\dot{m}_f$) with a lower heating value (LHV). P_{brake} is the brake power, measured out the crankshaft of the engine with the use of a torque transducer, $P_{friction}$ represents the total mechanical friction losses, calculated as the difference between $\text{IMEP}_\text{n} - \text{BMEP}$ and P_{pump} represents the total pumping losses, as the difference between $\text{IMEP}_\text{g} - \text{IMEP}_\text{n}$ calculated from the pressure trace. $Q_{comb\ loss}$ represents the combustion inefficiency that is calculated from the carbon monoxide and unburned hydrocarbons in the exhaust emissions. $\dot{Q}_{EGR}$, or EGR loss is the energy that has to be removed in order the exhaust gases to reach a lower temperature and is calculated as:

$$\dot{Q}_{EGR} = \dot{m}_{EGR} \cdot \frac{C_{p,high} + C_{p,low}}{2} \cdot (T_{high} - T_{low}) \tag{3.14}$$

Exhaust loss ($\dot{Q}_{exhaust}$) is calculated in a similar way to EGR loss. The total exhaust loss is the sum of two parts, the loss from the exhaust manifold to the LR EGR split and from the EGR split to the environment. This is done due to different mass flows before and after the LR EGR connection.

$$\dot{Q}_{exhaust} = \dot{Q}_{exhaust,1} + \dot{Q}_{exhaust,2} \tag{3.15}$$

$$\dot{Q}_{exhaust,1} = (\dot{m}_{EGR} + \dot{m}_{air} + \dot{m}_{fuel}) \cdot \frac{C_{p,exhaust} + C_{p,split}}{2} \cdot (T_{exhaust} - T_{split}) \tag{3.16}$$

$$\dot{Q}_{exhaust,2} = (\dot{m}_{air} + \dot{m}_{fuel}) \cdot \frac{C_{p,split} + C_{p,ambient}}{2} \cdot (T_{split} - T_{ambient}) \tag{3.17}$$

Chapter 4

Results

In this chapter, results from PPC operation on a light duty diesel engine will be presented and discussed. These results are divided into three parts. First, the performance of an engine under PPC conditions and different gasoline fuels will be presented. Afterwards, based on these results, two different ways to improve on the efficiency and low load stability will be presented and finally an investigation on how a PPC engine can be used together with a hybrid system in order to combine the best possible aspects of each system.

4.1 Partially premixed combustion with low octane gasoline

Load & efficiency As it was mentioned before, previous studies suggest that low octane gasoline can be an ideal fuel for PPC because it can provide wide engine load operation with improved efficiency and emissions when evaluated in gross engine parameters. This was evaluated in the multi-cylinder engine with the use of RON 75 gasoline fuel, (MON 68, LHV: 43.5 MJ/kg). The load range was evaluated at three different engine speeds, 1200, 1800 and 2400 rpm. A double injection strategy was used, to keep peak pressure rise rate (PPRR) under control, while EGR percentage was kept at 30 % to control both NO_x production and PPRR. An upper limit on PPRR was placed at 10 bar/CAD. The highest load would be the one where the exhaust lambda reaches a value of 1.2, while the lowest limit would be limited by the combustion stability, coefficient of variation (COV) of $IMEP_g$, which was placed at 3 %. These values are based on numbers that are used in the industry on production engines and represent realistic operating limitations.

It can be seen in figure 4.1 that the maximum achievable load is around 16 bar $IMEP_g$ at medium and high engine speeds but only around 14 bar $IMEP_g$ at the low speed. For all the engine speeds, the low load can go as low as 5 bar $IMEP_g$.

The limits on upper and lower load can be summarized as follows. At low speeds, the high load is dictated by the exhaust oxygen content. Due to high amount of EGR throughout the whole load range, exhaust lambda reaches the limit of 1.2 at around 10 bar $IMEP_g$ and it is necessary to decrease the EGR down to 20 % to increase the load up to 14 bar $IMEP_g$ before reaching the same lambda limit (figure 4.1). For the medium speed (1800 rpm), this limit is reached at about 16 bar $IMEP_g$. At the high speed of 2400 rpm maximum achievable limit is also around 16 bar $IMEP_g$. Unlike the other two speeds where the maximum load is limited by the exhaust lambda, at 2400 rpm other limitations are reached. If these could be removed, extrapolation shows that the maximum limit at this speed could be up to 18 bar $IMEP_g$. Nevertheless, the highest load at that speed was limited by the PPRR and the limitation of the fueling system to provide sufficient fuel flow.

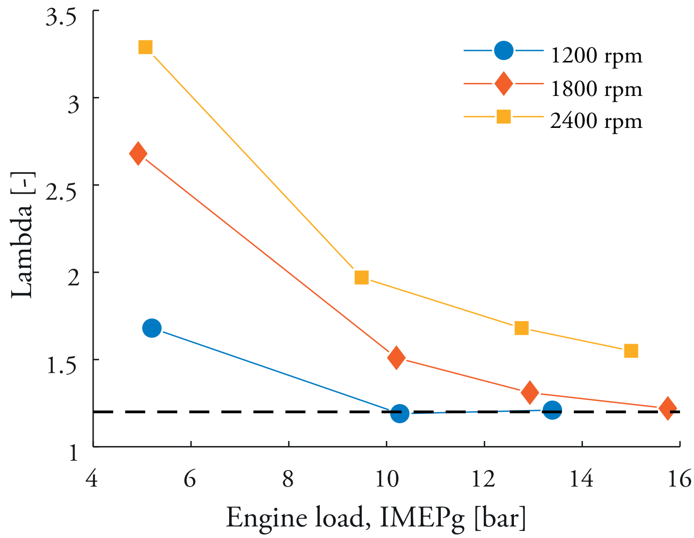

Figure 4.1: Exhaust lambda for the three different speeds

In terms of efficiency, as it can be seen in figure 4.2, gross indicated efficiency (a) is generally high, with numbers reaching up to 48 % and staying over 44 % through the whole load range. This changes when looking at the net indicated (b) and brake efficiency (c). Net indicated efficiency varies between 35 to 45 %, while brake efficiency is between 30 to 41 %. There are two main reasons for such a decrease from gross indicated to brake efficiency. Going from gross to net indicated efficiency, there are high pumping losses, associated with the need for air boosting even at low loads and the high demand for EGR with increases the throttling losses. Going from net to brake efficiency, there are more frictions losses compared to diesel operation, created from the higher demand that is put on pumping the fuel due to higher injections pressure that is required for PPC. Friction is increased also further, due to the different properties of the gasoline fuel compared to diesel.

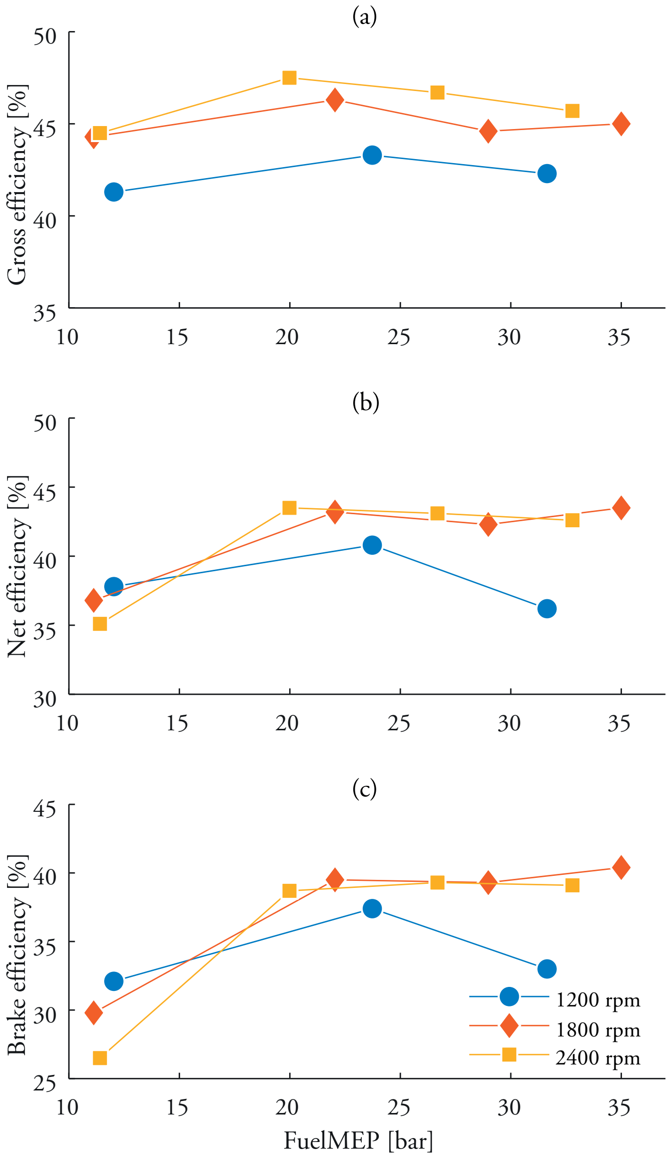

Figure 4.2: Gross, net and brake efficiency vs load for the three different speeds, with RON75 gasoline

Energy balance While PPC can provide high efficiency, it is also interesting to quantify how the fuel energy is divided in the engine. Therefore an energy balance was performed at the best efficiency points of each speed. The procedure to calculate the energy balance is described in chapter 3.4.

It can be seen (figure 4.3) that as the speed increases the efficiency improves due to lower heat transfer losses. This can be explained from the lower total cycle time, high amount of EGR and favorable combustion timing at higher speeds. Still, the exhaust losses increase at higher speeds due to higher combustion temperatures, reduced air dilution and lower HT loses. Combustion losses are negligible because of the late double injection strategy and pumping losses increase due to the higher air and EGR demand at higher loads.

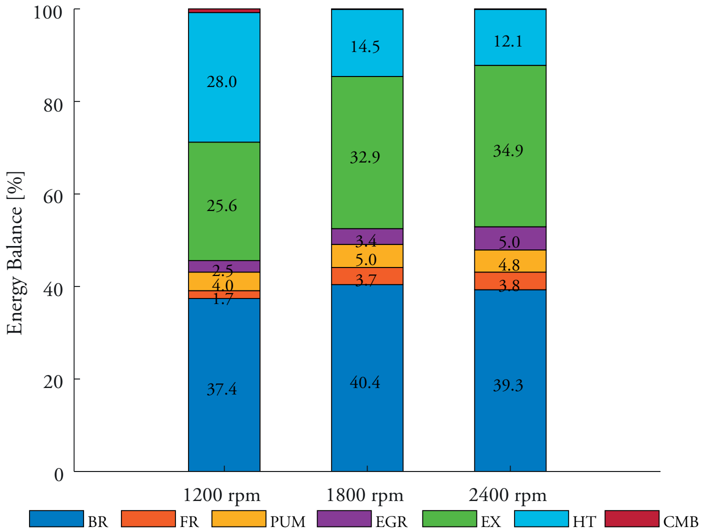

Figure 4.3: Energy balance for the best efficiency points, legend: Brake efficiency, friction, pumping, EGR cooling, exhaust, heat transfer, combustion losses

Emissions While efficiency and performance is an critical aspect of a combustion engine, emissions have also a great importance because of the health issues they can cause and the emission regulations that become more strict with every new iteration. With PPC, emissions such as carbon monoxide and unburned hydrocarbons are low from low load, reducing even further at higher loads due to higher combustion and exhaust temperatures in collaboration with late injections timings that reduce the premixed fraction of the combustion. Specific NO_x emissions (figure 4.4) have a decreasing trend as load increases, due to the constant fraction of EGR and the later combustion phasing, except of the highest possible load at low speed, where EGR has to be reduced down to 20 %. Soot emissions have the opposite trend (figure 4.5), they are low at low loads and increase as the load increases. More premixed combustion and higher lambda values can explain the low numbers at low loads, while as the load increases, the combustion turns more into mixing controlled, as there is limited time to inject the required amount of fuel, and the lambda decreases, which increases the soot substantially but still in terms of specific emissions, soot is still low in absolute numbers with a upper value reaching around 100 mg/kWh.

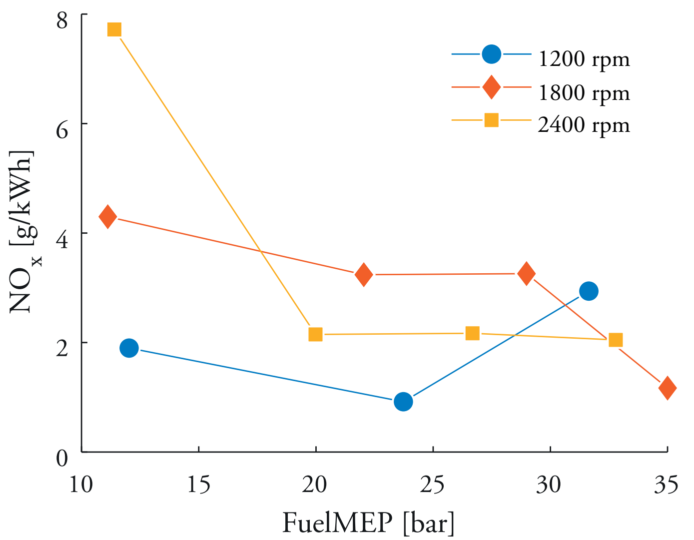

Figure 4.4: Brake specific NO_x emissions with engine load, for the three different speeds

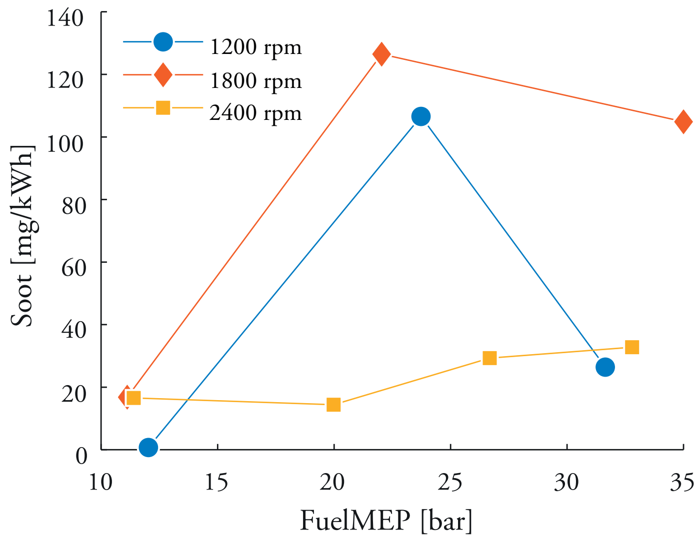

Figure 4.5: Brake specific soot emissions with engine load, for the three different speeds

Low load - "idle" An additional benefit of using low octane gasoline for PPC, is that it is possible to operate at loads that are generally not so easy with higher RON gasoline. Despite that, even with low octane gasoline combustion stability in terms of COV can be challenging. To evaluate the stability of PPC at low load conditions two different injection strategies were evaluated at 1200 rpm. Low load was defined at around 2 bar $IMEP_g$, while the injection strategy was either an early single injection or a double injection. A limited amount of EGR was used to control NO_x emissions, at around 10 %.

With these low load conditions, fuel is adequately premixed leading to very low soot emissions, while NO_x emissions are low but even lower with the single injection strategy. Due to low load conditions and premixed mixture, carbon monoxide and unburned hydrocarbon are increased. In terms of combustion stability, single injection is better, but both have high COV of around 9 %. The results from the low load conditions can be summarized in the following table.

Table 4.1: Summary of low load PPC at 2 bar $IMEP_g$ and 1200 rpm

Injection	Single	Double
COV [%]	8	10
Soot [FSN]	0.05	0.05
NO_x [ppm]	65	110
CO [g/kg fuel]	70	90
UHC [g/kg fuel]	170	130

4.2 Partially premixed combustion with high octane & "renewable" gasoline

Low octane gasoline is a promising future fuel that can be combined with PPC but it is necessary to be able to operate an engine with PPC using fuels that are commercially available at the current time. Therefore a similar evaluation on the load range, efficiency and emissions on the diesel engine operating under PPC was performed using RON90 gasoline. Gasoline of that RON it is available in USA but it is not common in Europe. To create this octane number, regular RON95E5 was mixed with 10 % n-heptane to reduce the octane number. Together with this gasoline a type of alkylate gasoline was evaluated as well. An alkylate gasoline consists mainly of alkanes (paraffins) which are generally saturated hydrocarbons. This type of gasoline has minimal aromatic content, with specifications stating less than 0.5 % aromatics and less than 0.1 % benzene. Commercially this type of gasoline is generally sold for use in small handheld equipment, to reduce the impact of the emissions on the operator's health. That fuel was chosen due to its fuel composition. A gasoline type fuel consisting of paraffins with minimal aromatics can be produced in a similar process to HVO diesel, which is produced from used oil or vegetable oils, making it a possible candidate for a renewable type of gasoline. As this fuel has low aromatics, it is also expected to reduce soot emissions for the same conditions, compared to the regular gasoline which has up to 35 % aromatics and maximum 1 % benzene (EN228 specification). This alkylate gasoline was also mixed with 10 % n-heptane to reduce the octane number. Finally, regular diesel fuel was used in order to compare the energy balance of the engine at the best brake efficiency point for the two gasolines at medium engine speed. The fuel properties can be found in table 4.2.

Table 4.2: Fuel properties

-	Regular Gasoline	Alkylate	Diesel
Original RON [-]	95	95	51 (Cetane)
Original MON [-]	85	92	-
LHV [MJ/kg]	43	44	43
AFR [-]	14.4	14.6	14.7
Density [kg/m^3]	750	700	810

As the engine was operated from minimum to maximum possible load, different conditions had to be met at each point and in a sense limiting the load range. These would be the PPRR with a maximum value of 10 bar/cad and COV with an upper value of 3 %. Also, the maximum load was defined at the load that lambda value reached 1.2. In all cases and loads EGR was used, at a constant rate between 30 to 31 %. That amount of EGR is a good compromise for low NO_x emissions and improved upper load. Also depending on the load, different intake temperature and injection timings were used to keep the combustion within the aforementioned limits. Three different engine speeds were used here as well, a low, a medium and a high engine speed, or 1200, 1800 and 2400 rpm. These conditions are summarized in table 4.3.

Table 4.3: Operating conditions

Load[bar $IMEP_g$]	Intake temp [°C]	Timings [CAD BTDC]
≤ 8	80	-25&-5
$8 \leq 12$	70	-25&-5 or 15&-3
≥ 12	60	-10&-3

Load, efficiency & emissions With the conditions that were mentioned before, the maximum achievable load at low speed was found at 14 bar $IMEP_g$ and at medium and high speed, at around 18 bar $IMEP_g$. Low load was identified at around 5 bar $IMEP_g$, which is around 3 bar BMEP. The explanation for these limits is similar to the previous results with the RON75 gasoline. Upper limit is limited at low speed by the exhaust lambda that reaches the value of 1.2 at 14 bar $IMEP_g$. At medium and high engine speeds the exhaust lambda limit is not reached but other limitations occur. For these two speeds these limitations are, efficiency deterioration after 18 bar $IMEP_g$, reaching the peak cylinder pressure limit of 160 bar and mechanical limitations of the fueling system which could not provide more fuel at this speed/load combination. Assuming that these limitations did not occur, by extrapolating from the lambda curve the possible engine load would be around 20 bar $IMEP_g$. At the low load, the limiting factor is the same for all three speeds and is the COV value which becomes greater than 3 % even when using the high intake temperature of 80 °C.

In figure 4.6 the gross, net and brake efficiency, when using regular gasoline, for the three different speeds are plotted. Gross indicated efficiency has a small variation throughout the load range and is between 40 to 47 %, with the higher numbers to appear at higher speeds. Net indicated and brake efficiency decrease significantly at lower loads, as it was observed also with the lower RON gasoline. The reasons are the same, high fraction of EGR and a need for higher boost compared to diesel combustion, put a higher demand on the gas exchange system leading to increased gas exchange losses. As for the total friction losses, higher injection pressure, different fuel properties and earlier combustion, lead to increased engine friction. Despite the fact the engine is not optimized to this type of combustion and fuel, it is still possible to get improved brake efficiency that can reach up to 41 % at medium engine speed.

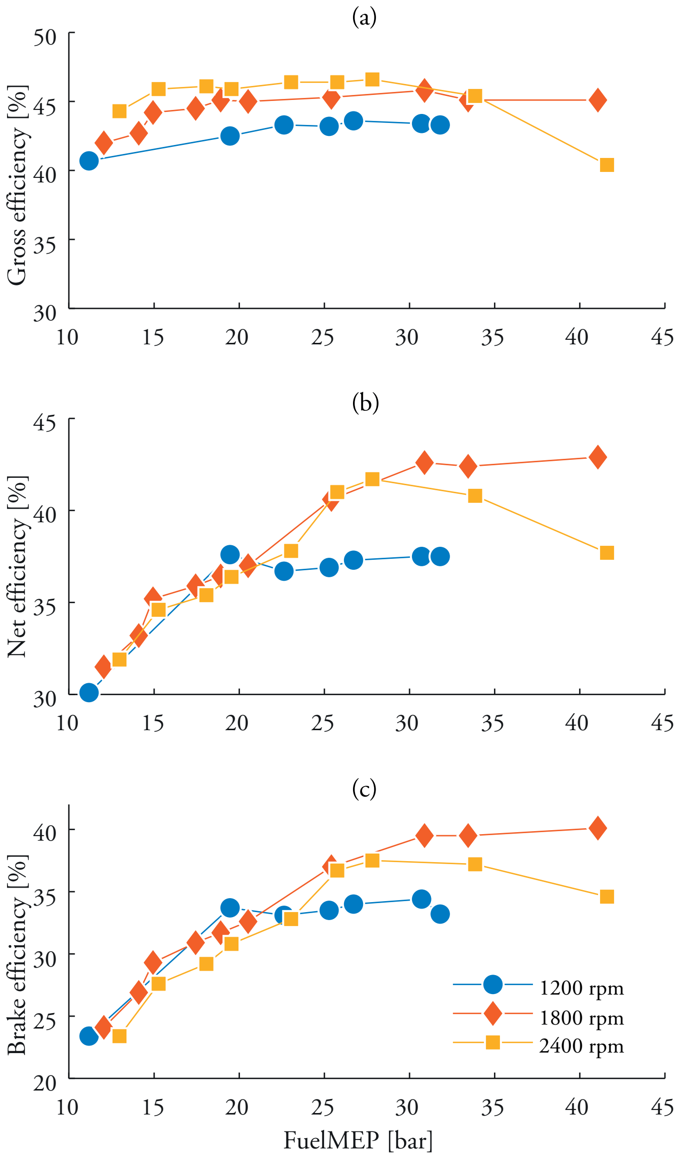

Figure 4.6: Gross, net and brake efficiency vs load for the three different speeds, with RON90 gasoline

As far as the emissions go, the two most important emissions and more difficult to control, soot and NO_x have an opposite, trend as before. As the load increases, for all speeds, soot goes up (figure 4.8) and NO_x go down (figure 4.9). Low and medium loads have leaner mixtures and more premixed combustion, leading to reduced soot emissions, but as load increases, lambda goes down and combustion turn more into mixing control (figure 4.7), soot increases but again is low in absolute numbers (mg/kWh). NO_x on the other hand, due to the constant fraction of EGR and the improvement of brake efficiency tends to decrease with the load, reaching a minimum value of about 1.5 g/kWh at the highest load.

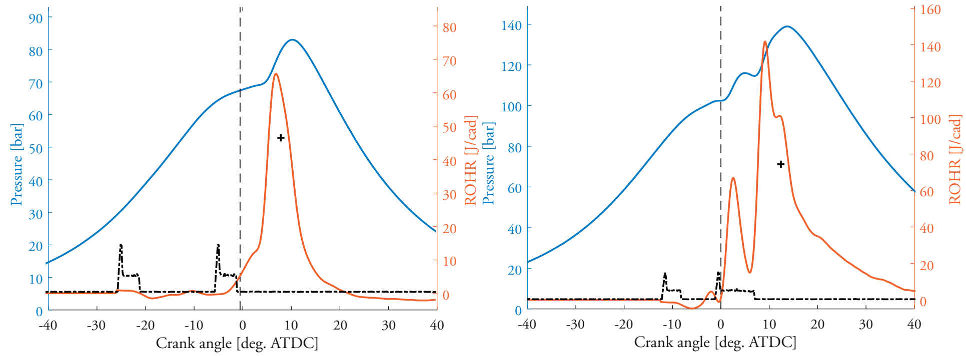

Figure 4.7: Pressure (blue line) and heat release rate (red line) for two different loads at 1800 rpm with gasoline. Left figure is at 5.5 bar $IMEP_g$ and right figure at 17.4 bar $IMEP_g$. Black line represents the injection signal. Black cross is CA50.

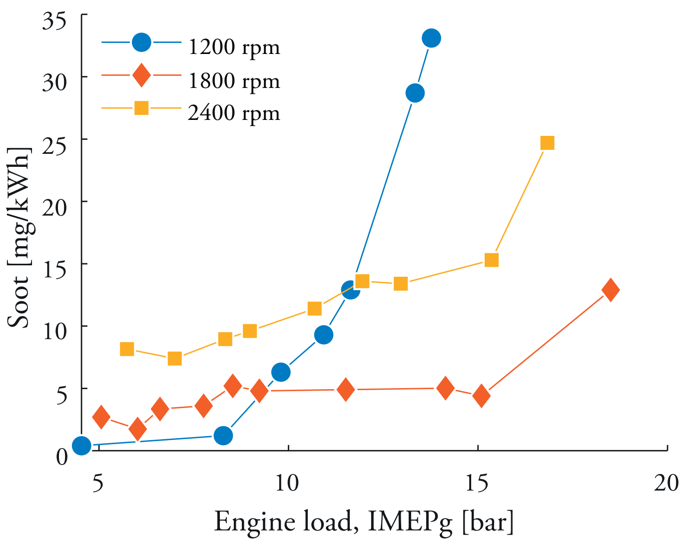

Figure 4.8: Brake specific soot for the three different speeds

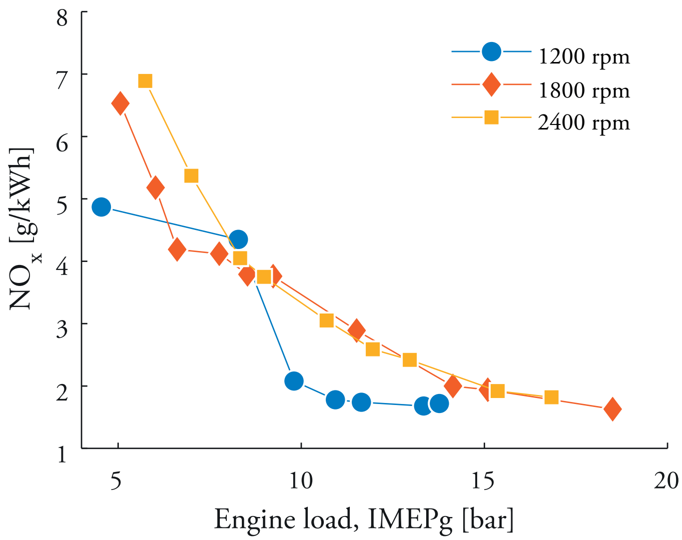

Figure 4.9: Brake specific NO_x for the three different speeds

Fuel comparison, efficiency, emissions & energy balance In terms of achievable load, the results don't differ between the two gasolines, maximum load is around 18 bar $IMEP_g$ and low load is close to 5 bar $IMEP_g$. Favorable combustion timing as well as fast combustion, lead to indicated efficiency of up to 47 % with the alkylate type of fuel and up to 46 % with the regular pump gasoline. Net and brake efficiency are affected heavily by pumping losses and friction, as was mentioned before, and brake efficiency ends to around 41 % for the alkylate fuel and 40 % for the regular gasoline (figure 4.10).

Regarding emissions, NO_x emissions are similar for both fuels, with same trend as before (figure 4.11), getting lower as the load increases due to the improved brake efficiency. Interestingly, soot emissions at low load have similar numbers, mainly due to premixed combustion and high mixture dilution but as the load increases and combustion turns more into mixing controlled, the alkylate gasoline gives higher soot emissions compared to the regular gasoline. At the maximum load, soot, in terms of mg/kWh is almost double for the alkylate fuel. An explanation for that can be that the ethanol content of the regular gasoline can help reducing soot emissions more than having a fuel with low aromatics but no ethanol. Still the absolute soot values are much lower when compared with similar conditions and diesel combustion and are four times lower than the legislation limit of 200 mg/kWh (heavy duty engines) (figure 4.12).

From the energy balance perspective (figure 4.13), it can be seen that all three fuels show similar brake efficiency ± 1 p.p., with the higher losses to be the exhaust losses. Heat transfer losses are similar to each fuel and that can be attributed to moderate amount of EGR and

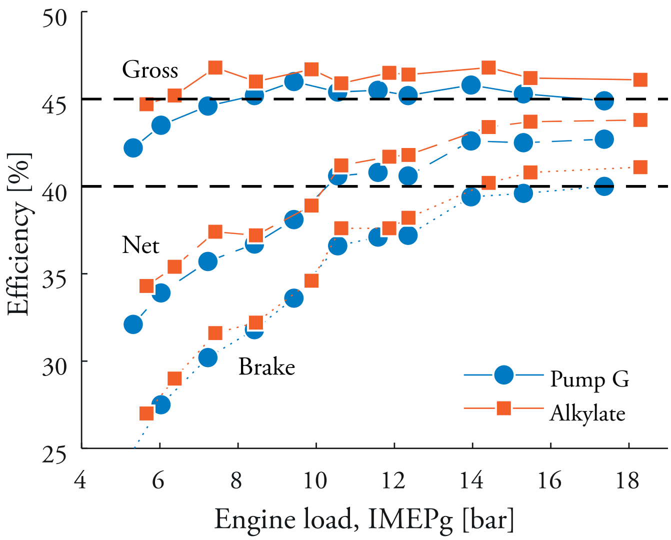

Figure 4.10: Gross, net and brake efficiency for the two different gasolines, at 1800 rpm

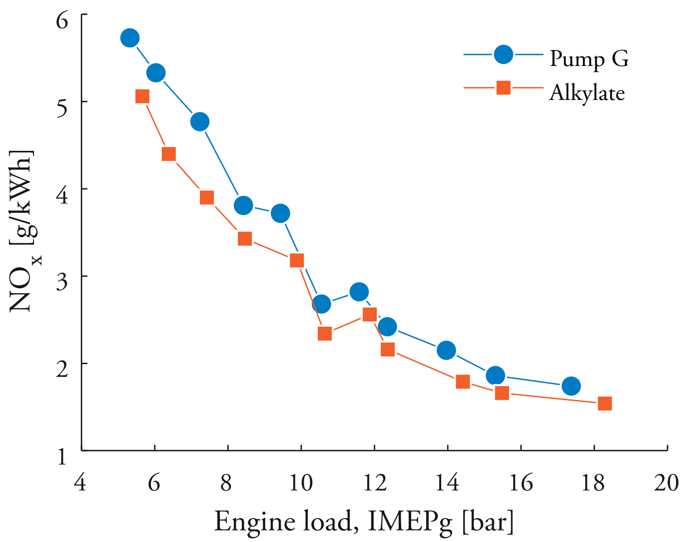

Figure 4.11: Brake specific NO_x for the two different gasolines, at 1800 rpm

late injection timings that premix the fuel less than optimal leading to heat transfer similar to the diesel case.

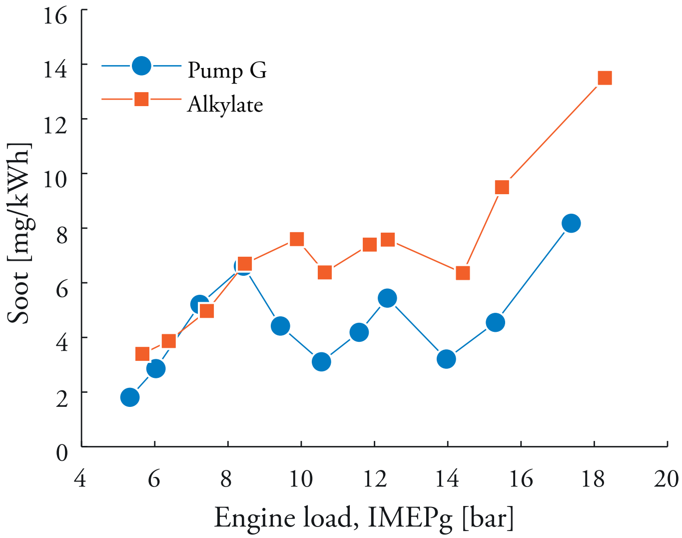

Figure 4.12: Brake specific soot for the two different gasolines, at 1800 rpm

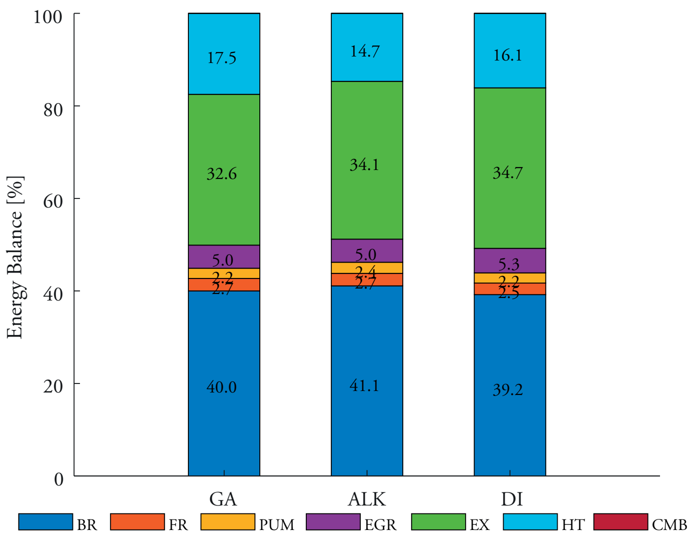

Figure 4.13: Energy balance for the three different fuels at 1800 rpm, 38 bar FuelMEP ($\approx$17 bar $IMEP_g$) , GA:gasoline, AL:alkylate, DI:diesel

4.3 EGR routing

From the previous results, it becomes clear that while PPC can provide high efficiency in gross indicated terms, but due to high friction and gas exchange losses, it is difficult to get improved brake efficiency. One reason for the high gas exchange losses is the demand for high EGR rates. With PPC, high amounts of EGR are necessary from low to high load and that increases the gas exchange losses and also affects the achievable load range. With CDC when an engine operates at high loads, EGR rate is reduced to a minimum to improve turbocharger performance and maintain a high enough oxygen content.

A way to improve on the gas exchange efficiency without losing on the EGR amount that is used, is to combine two different EGR routes. The first is the short route, or high pressure EGR, where exhaust gases are removed before the turbo and feed after the compressor. The second route is the long route or low pressure, where gases are removed generally after the aftertreatment system and mix with the intake air before the compressor. Each system has its own benefits and drawbacks and by combining the two systems the gas exchange efficiency can be optimized for the different load conditions, leading to improved brake efficiency.

The possibilities of combining the two routes were evaluated at different speeds and different loads keeping the total EGR amount at 20 %. These include three speed/load points of high efficiency and one high frequency speed/load point from the NEDC/WLTP drive cycle. These can be summarized in the following table.

Table 4.4: Engine operating conditions

Speed [rpm]	Intake temp [°C]	FuelMEP [bar]	Rail Pressure [bar]
1200	77	23.5	1100
1800	80	21.5	1100
2400	75	21.5	1100
1500	75	13.5	800

Results show that routing the EGR from long route to short route, has an effect on gas exchange efficiency, as it can be seen in figure 4.14. In the high load cases the best efficiency is found when EGR is split 50 to 50 %, while at the low load case the best result comes when gases pass only through the short route. This behavior can be due to the required back-pressure needed to move the gases from the exhaust to the intake. This back-pressure is created by throttling the exhaust flow, forcing it to go through the different EGR routes. In the case of the long route, the back-pressure is created with the use of a throttle valve in the exhaust pipe, while for the short route it is created by the adjustment of the VGT vanes in the turbine. At higher loads, the total back-pressure that is created by combining both routes, is lower compared to each route by itself. The opposite is happening at low load

(1500 rpm) where the low exhaust flow requires a high amount of throttling in the exhaust, making the long route option the best option, to maximize the gas exchange efficiency.

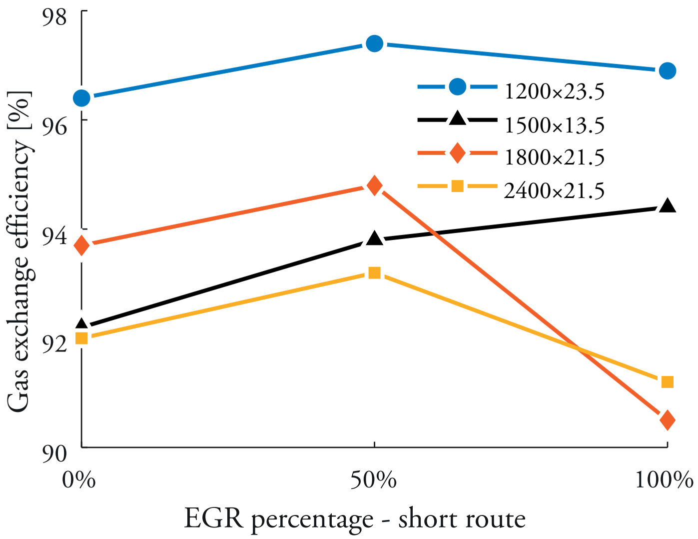

Figure 4.14: Gas exchange efficiency for different rpm – load

In terms of emissions the short route gives higher NO_x (figure 4.15) and lower soot (figure 4.16) emissions compared to the long route option.

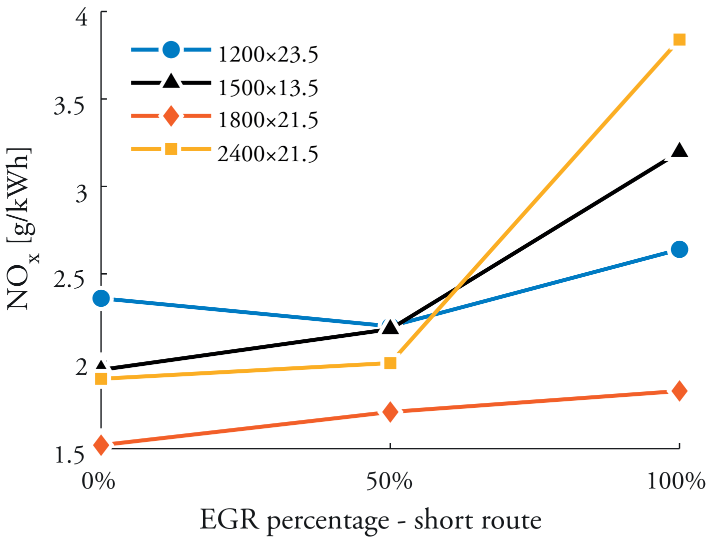

Figure 4.15: Brake specific NO_x for different rpm

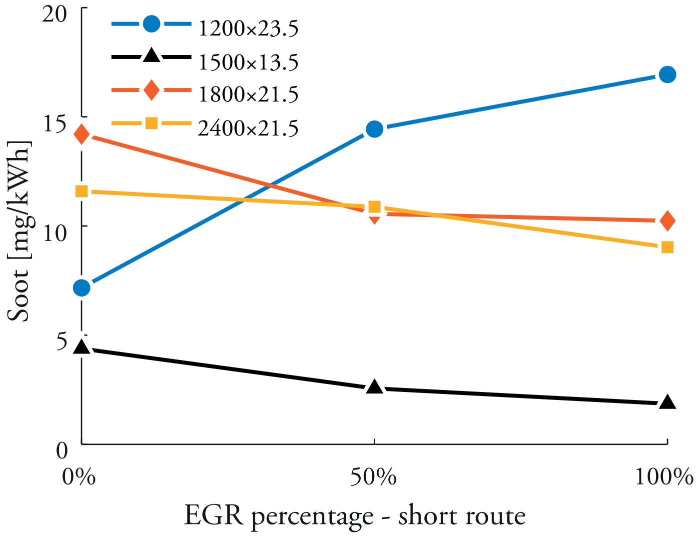

Figure 4.16: Brake specific soot for different rpm

This happens due to two different facts. One is that the exhaust lambda values are not constant when moving from long to short route EGR as it can be seen in figure 4.17. Long route has slightly higher oxygen concentration that can lead to increased NO_x and lower soot emissions. Another hypothesis that can influence the NO_x emissions is the possibility that EGR and air don't mix as well as when going through the long route, leading to different EGR percentage per cylinder, causing higher emissions for some of the cylinders, leading to higher total exhaust emissions. These results from this part can be summarized in table 4.5

Table 4.5: Percentage difference for NOx, soot and gas exchange efficiency between the best and the worst conditions. (NOx and soot are percentage reduction; gas exchange efficiency is percentage improvement)

rpm × FuelMEP	NO_x	Soot	Gas exchange efficiency
1200×23.5	16.7 %	56 %	1 %
1500×13.5	39 %	57 %	2.3 %
1800×21.5	17 %	28 %	4.5 %
2400×21.5	50.5 %	22 %	2.1 %

While these results show that generally, a combination of both EGR route is the best option in maximizing gas exchange efficiency, in the previous results, exhaust oxygen content was not constant when moving from long route to short route. This is due to the fact that as exhaust gases are removed before the turbocharger, less mass flow goes through, reducing the intake pressure. In many cases it would be necessary to keep similar intake oxygen content despite which EGR route is being used. This would probably have an effect on the

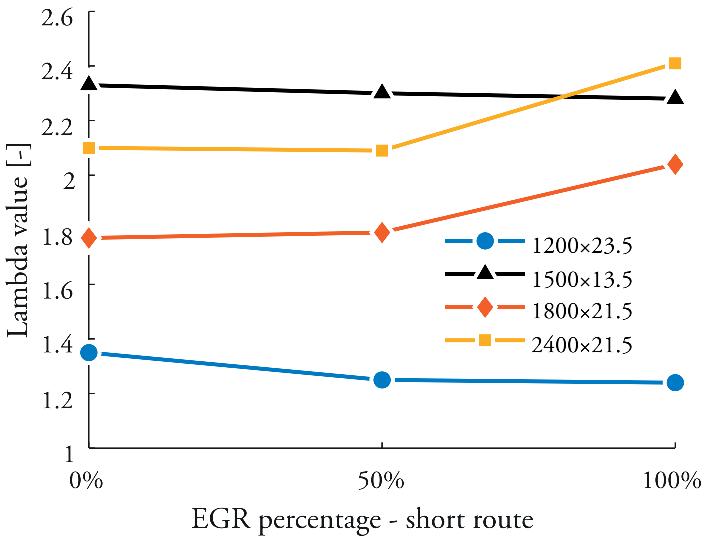

Figure 4.17: Exhaust lambda for different rpm

gas exchange efficiency, again due to the required adjustments on the back-pressure. This can be seen in table 4.6. Here the exhaust lambda value is kept at around 1.9 while the load is around 9.7 bar $IMEP_g$, (FuelMEP: 20.5 bar).

Table 4.6: Gas exchange efficiency (GEE) when keeping exhaust lambda constant and routing EGR from long to short route (0 % means all EGR goes through long route, while 100 % means all EGR goes through short route)

EGR %	0	50	100
GEE %	93.1	89.5	91.2

What can be seen, is that in this case, combining both routes gives the worst gas exchange efficiency, while the best efficiency is found when using the long route. This again can be explained in terms of reduced back-pressure needed to move the exhaust gases through the long route compared to the other two options.

Emissions (figure 4.18) follow similar trends as before, with increased NO_x when going towards to short route, strengthening the argument that the EGR mixing through the short route is not as complete as through the long route.

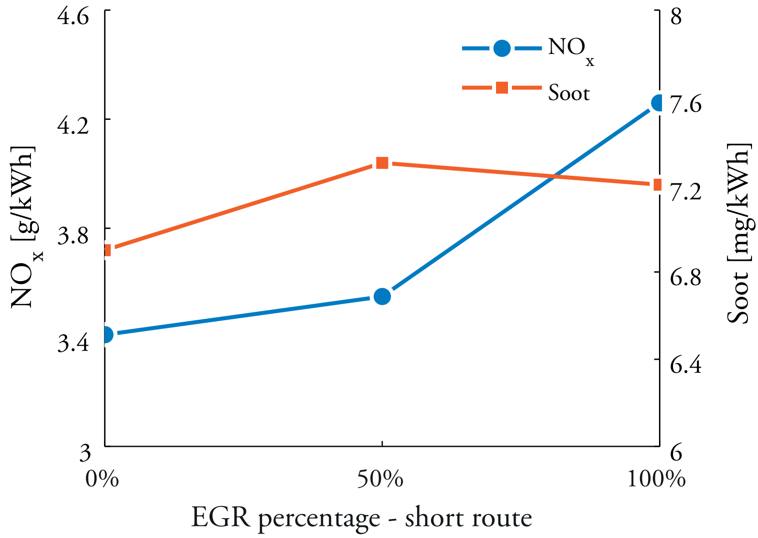

Figure 4.18: Brake specific soot and NO_x emissions when exhaust lambda is kept constant

4.4 Glowplug assistance

To keep the combustion within reasonable COV limits (<5 %) at lower loads (<5 bar $IMEP_g$), intake pressure and intake temperature have to be higher than what this specific engine can provide at that load. To overcome that limitation the glowplugs can be used to increase the in-cylinder temperature. Unlike normal CDC where glowplugs are only used during cold start, with PPC the glowplugs will be on when the engine load is lower than a limit.

Glowplugs can be an easy way to increase the in-cylinder temperature but unlike CDC, if the intake temperature gets low enough, even with the glowplug assistance the combustion will deteriorate and the COV will go over the limit that is considered stable. That can be seen in figures 7.1, 7.2, 7.3, in the appendix, where the pressure from each cylinder is plotted. The difference between these three figures is the intake temperature which is 65, 85 and 105 °C. The shaded area represents one standard deviation of the pressure and it can be seen that at the low temperature of 65 °C, even with the glowplugs on, the variation is much higher, leading to unstable combustion. Also the two outer cylinders are getting more affected, probably due to higher heat losses. The individual and average COV from these three different intake temperatures can be seen in figure 4.19. Despite the higher COV of the outer cylinders, on average, COV is lower than the limit of 5 % even with the lower intake temperature, indicating that glowplugs can have a positive effect. Engine operation with the glowplugs off, is only possible with the 85 °C and 105 °C intake temperature. COV difference between on and off is minimal at these temperatures. At the low temperature of 65 °C, operation without using the glowplugs is not possible (COV>25 %).

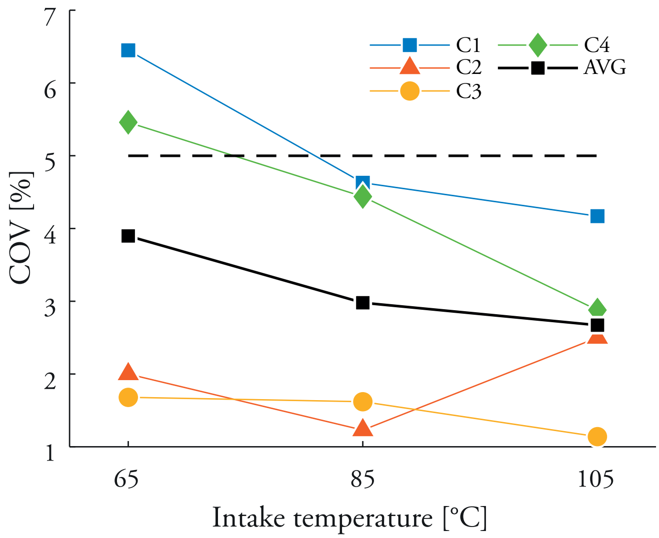

Figure 4.19: COV variation with different intake temperature, at low load

Injection timing at low temperature Although the combustion stability at low temperature is within limits, it is beneficial to keep it low as it can affect the emissions and the efficiency of the engine. Also PPC is affected from fuel stratification which is controlled from the injection timings and it is interesting to see how that can affect the combustion stability with the low intake temperature setting. Three different injection timings were evaluated. An early injection, an intermediate and a late injection (table 4.7). As with the previous section, plots from each cylinder for the three different conditions can be found in figures 7.4, 7.5, 7.6 (appendix).

Table 4.7: Injection timings, numbers are degrees ATDC

Name	early	interm	late
First timing	-24	-20	-18
Second timing	-12	-10	-8

It can be seen that earlier injections improve the combustion stability while later injections make it worse (figure 4.20). The reason for that is that early injections give enough time for the fuel to mix properly with the intake air and form an adequate level of stratification. Assisted from the glowplugs this setting gives the best stability at this low load. Based on these trends, it can be argued that even earlier injection timings can help towards improved combustion stability. Unfortunately the current engine design, more specifically the injector umbrella angle, limits that because earlier injections would result in fuel reaching the cylinder liner which would lead to poor combustion.

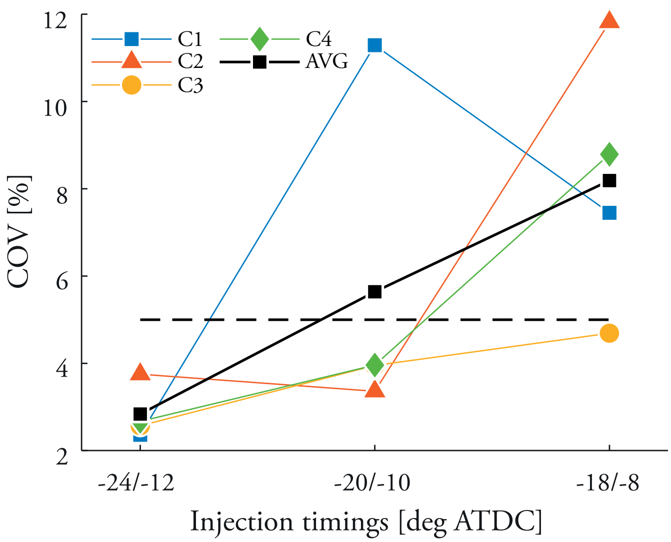

Figure 4.20: COV variation with different injection timings, at low load

Load range with glowplugs While glowplugs can help low load operation, as was demonstrated before, it is important to evaluate how much they can benefit at higher loads as well. Therefore, the effect of glowplug operation was evaluated on engine loads of up to 10 bar $IMEP_g$. In figure 4.21 it can be seen that glowplugs have a positive effect on combustion stability up until around 16 bar FuelMEP, which is around 6.6 bar $IMEP_g$. At higher loads the increased combustion temperature combined with the increased boost pressure, gives a good combustion without the need of the glowplug, as it can be seen again in figure 4.21, where there is no difference in the COV number between the two cases, glowplug on and glowplug off.

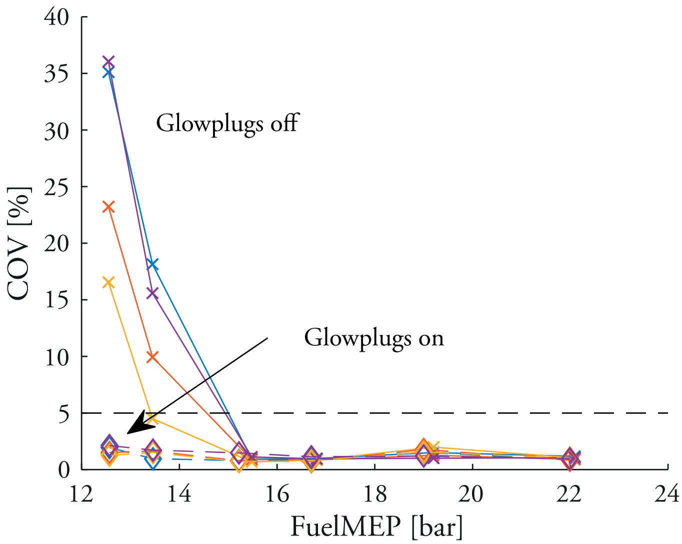

Figure 4.21: COV variation with engine load, with and without glowplugs

The effect of glowplug on the combustion stability at low loads can be seen in figure 4.22, where the transient behavior of the combustion of cylinder one is plotted 1, 7 and 14 s after the glowplug is turned off. What can be seen is that after the glowplug stops (solid line), the combustion starts dwindling (dashed line) and finally stabilizes at a state of almost no combustion (dotted line) after around 14 s. If the glowplug would be on all the time, the combustion would be as in the solid line indefinitely.

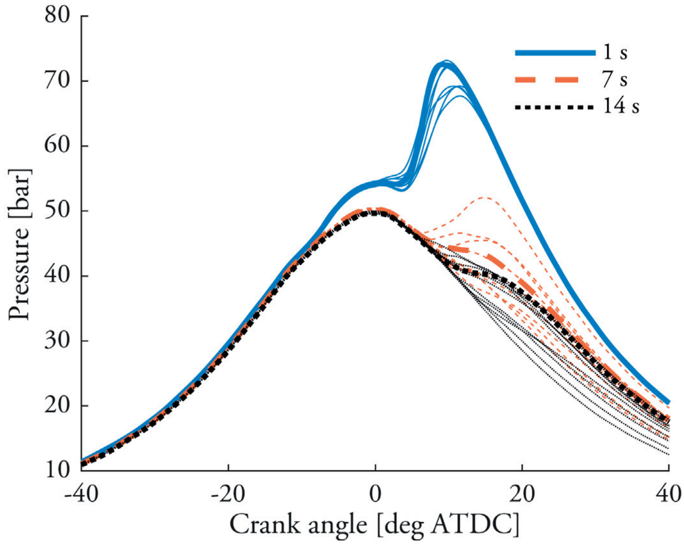

Figure 4.22: Transient operation with glowplug off at low load

Efficiency Although glowplugs seem to be beneficial to low load operation, they consume energy that has to come from the engine operation. Is this case the power demand of the glowplugs was around 210 W (8 V·25 A) or around 0.1 bar MEP (although it can argued that if an engine alternator is used to produce the power for the glowplugs, the actual power demand can be close to double, because generally alternators are around 60 to 70 % efficient). Despite that, as it can be seen in figure 4.23 glowplugs can be operated at low load (<8 bar $IMEP_g$, 15 bar FuelMEP) with an small efficiency penalty but still provide better results than without and switched off above that point.

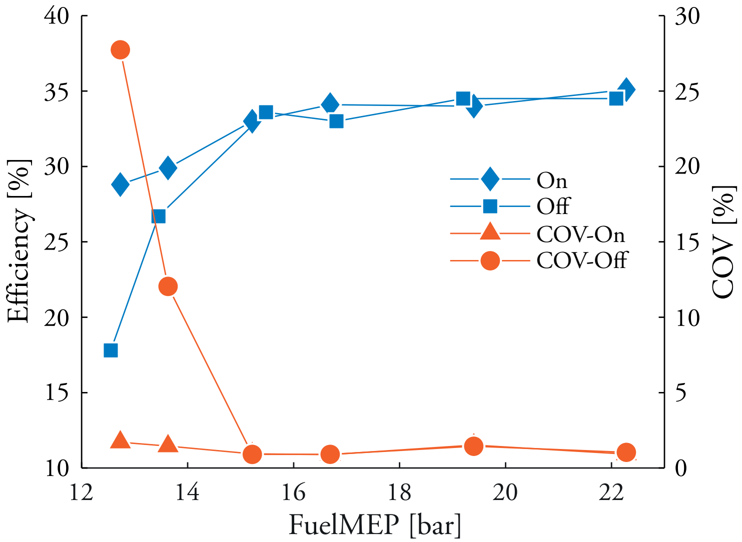

Figure 4.23: Brake efficiency (blue line) with the glowplug on (diamond) and off (square) and COV (red line) for the same conditions (on, triange), (off, circle)

4.5 Hybridization

Although partially premixed combustion is a promising future technology, there are always going to be situations when the engine will be operated at low efficiency conditions. A way to counteract this situation is to combine PPC with some form of hybridization in order to keep the engine shut off at the low efficiency conditions and use it when power demand brings the engine operating in high efficiency operating points. To investigate this possibility, a simulation model of a hybrid vehicle was built and evaluated with the use of GT-Power software.

Input data to the model were the engine results that were presented in the first section of this chapter, with the RON75 gasoline. Based on these results, maps of specific fuel consumption and emissions were created. It was decided to evaluate the hybrid powertrain on two different types of vehicles. A mid size car and a large SUV. The vehicles that were evaluated were a Volvo V60 and a Volvo XC90 with the following properties.

Table 4.8: Vehicles specifications

Vehicle [-]	Mid-size	SUV
Mass [kg]	1824	2434
Drag coefficient [-]	0.29	0.33
Frontal area [m^2]	2.23	2.78
Tire size [mm/%/inch]	215/55/R16	235/55/R19
Electric motor [kW]	50	60
Generator [kW]	30	35
Battery size [A h]	32	45

To reduce the computational load, it was decided to model the hybrid system as a series hybrid, meaning that the engine is only connected to the generator when it is necessary to charge the battery at a predetermined point, decided by the battery controller in order to keep the battery within an optimal state of charge (SOC). Base on this, different engine power states are defined to charge the battery as needed (figure 4.24). To evaluate the performance of the hybrid vehicle two common certification driving cycles were used, the older NEDC and the newer WLTP, also due to the fact that the manufacturers have to publish fuel consumption and emission data based on these cycles.

The results show that the heavier SUV has higher energy demands in the driving cycle and needs to charge the batter more often and for longer duration compared to the smaller car, figure 4.25. Both of the two vehicles exhibit fuel consumption that is close to the lower limit of the real vehicle when driven with diesel fuel, figure 4.26. The results of the engine running PPC are not fully optimized for efficiency or reduced emissions and that can be seen in the NO_x emissions, that require the use of a SCR aftertreatment system to be within the limits. But even if the added urea consumption is accounted for, the total consumption, fuel and urea, are still within the fuel consumption range of the fully optimized diesel engine (figure 4.26, third column), which points towards better overall consumption when more thorough optimization will be performed on the PPC engine. These results can be summarized in table 4.9.

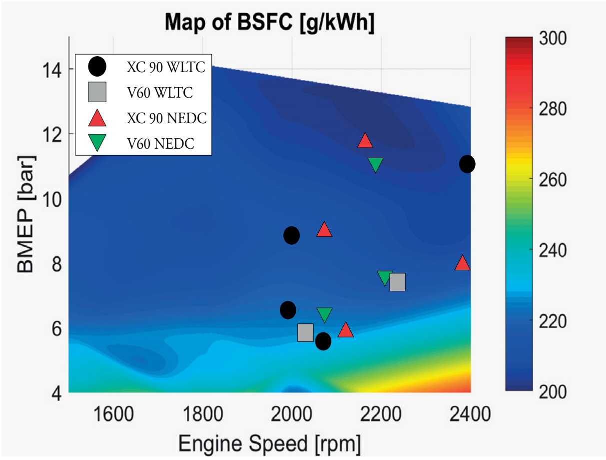

Figure 4.24: Optimized engine charging points

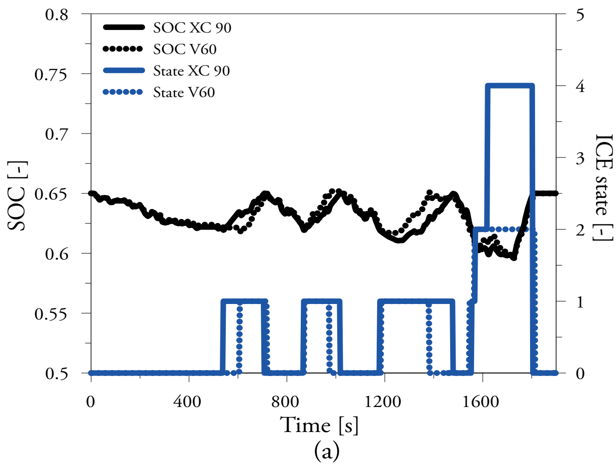

Figure 4.25: State of charge and engine state for the two vehicles during the WLTC driving cycle.

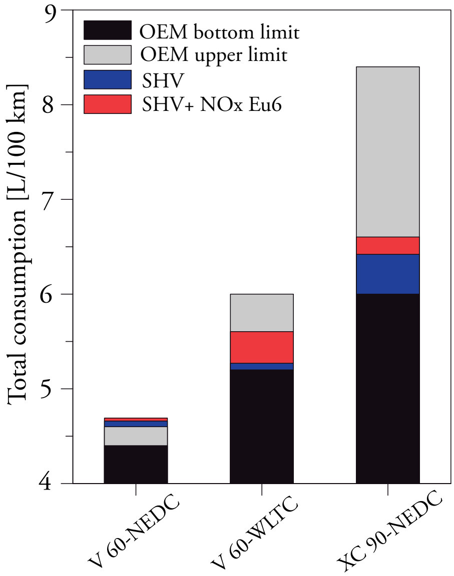

Figure 4.26: Total fuel consumption of tested vehicles

Table 4.9: Fuel consumption and CO2 emissions for the two different vehicles when operated with PPC and diesel

Vehicle	Cycle	CO_2 [g/km]	fuel use [L/100 km]
Car SHV	NEDC	118.7	4.7
Car OEM	"	117-126	4.4-4.6
SUV SHV	"	163.7	6.4
SUV OEM	"	158-221	6.0-8.4
Car SHV	WLTP	134.4	5.3
Car OEM	"	136-157	5.2-6.0
SUV SHV	"	187.6	7.3

Chapter 5

Conclusions

In the constant search for improvement in the field of internal combustion engines, low temperature combustion concepts have been a promising development that can provide simultaneously diesel-like efficiency and reduced emissions, especially in terms of soot and nitrogen oxides. At the same time, an engine that is using LTC, has the ability to operate with fuels of different octane numbers, making it a fuel flexible combustion concept for the future.

This thesis aimed to evaluate the performance, efficiency and emissions of a multi-cylinder diesel engine under PPC conditions, with limited modifications. Two different types of gasoline were used and based on these results further investigations were performed to improve supported from these conclusions.

The first fuel that was used was a gasoline with an octane rating of 75. The aim was to evaluate the load range, the efficiency and the emissions with a low octane gasoline. The reason behind that is that previous research has shown that gasoline with an octane rating close to 70 can be an ideal future fuel for these applications, because it can provide good low load stability and operate at higher engine loads as well. Results show that the load range can be between 5 to 16 bar $IMEP_g$ and can operate stable at an idle load of 2 bar $IMEP_g$.

In terms of efficiency, gross indicated efficiency can be between 44 to 49 %.The lower efficiency numbers appear at low engine speeds and low engine loads, while as the load and the speed increases, the efficiency improves. While gross indicated efficiency is generally high, it can be seen that net indicated and brake efficiency suffer a lot, especially at lower loads. This is due to high gas exchange and high friction losses. Gas exchange losses are affected by the need to keep a high EGR percentage and the required intake boost while friction losses are also affected by the increased amount of energy loss in the fueling system due to different characteristics of the gasoline compared to diesel. Despite these shortcomings, the highest brake efficiency can reach up to 41 % at higher loads.

Regarding emissions, carbon monoxide and unburned hydrocarbons are generally low throughout the load range. Soot is low but increases with load due to the reduction of the oxygen content, but is much lower compared to diesel. Nitrogen oxides are relatively high, due to relatively low EGR percentage and not optimized combustion behaviour.

While a low octane gasoline can be a promising future fuel, it is not available outside the fuel refineries. A solution to that, is to be able to operate with PPC with commercially available fuels, therefore a similar investigation as before was performed with a RON90 gasoline. In terms of load range, the low limit is around 5 bar $IMEP_g$ and the high limit at around 14 bar $IMEP_g$ at low engine speed and 18 bar $IMEP_g$ at higher engine speeds. Gross indicated efficiency reaches up to 47 % but similarly as before, due to high friction and pumping losses in the end brake efficiency is around 41 %. Emissions also follow a similar trend as before. Soot is affected by the combustion, with low soot at lower loads which increases as the combustion goes towards mixing controlled. Nitrogen oxides tend to decrease with load, due to high amount of EGR and improved brake efficiency.

Based on the results of these two investigations it became clear that two important factors that affect the PPC operation are gas exchange losses and low load stability. Therefore further work was performed on these two topics to investigate the benefits that they can provide.

PPC requires increased amounts of EGR compared to CDC and as a result puts a higher demand on the gas exchange system. One way to improve on that is to combine two different EGR routes, the long route and the short route. Based on the different speed/load combinations, either one of the routes can be used, or both routes can be active at the same time. Results show that in terms of gas exchange efficiency, is generally beneficial to use both routes at higher loads, as this option gives the best trade-off between efficiency and emissions. But this approach lets the exhaust oxygen content vary which might not be ideal in some cases. If EGR is routed from long to short route, with a constant lambda value, at medium engine speed and medium load, the best gas exchange efficiency and emissions appear when using only long route EGR.

Another issue that appears with gasoline operation and mainly with higher octane gasoline is the reduced combustion stability (COV) at low loads, less than 5 bar $IMEP_g$. This combustion instability is due to reduced intake pressure and temperature, that are necessary to start the early low temperature reactions that will lead to full combustion. Increasing the intake temperature is not a viable option if the load is low so another option has to be evaluated in order to increase the intake temperature. Glowplugs are a convenient choice for that, with low energy requirements. Results show that glowplug assisted operation is useful for engine loads of up to 5 bar $IMEP_g$. Having the glowplugs on at this low load stabilizes the combustion, with COV values of around 3 % compared to 20 % with the glowplugs off. At higher loads there is only a negligible effect on the combustion. Having the glowplug on during the whole engine operation results is less than 1 p.p. reduction in brake efficiency which can be a negligible amount for improved low load stability.

Finally in an effort to evaluate the performance of a gasoline PPC engine when used in a vehicle, a simulation model was designed around a series-hybrid operation of a PPC engine in a D-segment and J-segment car. The outcomes then were compared with real data of the same vehicles with diesel engines. Results show that an unoptimized PPC engine when used in a hybrid application can provide better fuel consumption, or similar total fluid consumption to the equivalent diesel engine. Fluid consumption was defined as the total amount of fuel and urea needed to meet the nitrogen oxide legislation limits.

With these promising results further research and development of a PPC type engine can be proven a viable alternative to the conventional types of internal combustion engines, with the possibility of replacing the less efficient SI engines.

Chapter 6

Future Work

In this thesis it has been demonstrated that a light duty diesel engine has the ability to operate under partially premixed combustion mode with minimal modifications. Despite that fact and although the efficiency and emission numbers are good, the engine is not designed around this type of combustion, therefore more improvements can be made to maximize the potential of low temperature combustion in the future.

Combustion chamber geometry The engine that was used, is designed around conventional diesel combustion. This can be seen in the piston shape geometry and injector umbrella angle which are adapted to late injections strategies. The umbrella angle is wide, meaning that if early injections with high pressure are used, a lot of fuel will end up on the cylinder wall. Also the closed piston shape means that even with later injections, some amount of fuel will end up outside the piston bowl. To further improve on the potential of PPC, piston shape and injector spray angle have to be adapted to the combustion characteristics of PPC. This would probably involve a wider bowl shape while keeping a high compression ratio and shorter injection angle to keep the fuel away of the cylinder walls even with multiple early injections. These two modifications can be evaluated both with numerical and experimental work on their effect on the efficiency and the emissions.

Gas exchange system For PPC to operate optimally, is necessary to have a high amount of dilution, either with a high amount of air or with high amount of EGR that can reach more that 50 %. It is also necessary for the engine to be able to maintain high dilution from low to high load. These requirements add more complexity to the gas exchange system which has to be adapted around these demands. A more optimally matched turbocharger system designed around these needs can help reduce the high gas exchange losses, improve on the engine brake efficiency and posibly extend the high load region of the engine.

Low load Operating the engine under PPC conditions at low load is a challenging process, leading to reduced combustion stability. While a investigation on the effects off glowplugs on the combustion stability was performed, more thorough investigations can be performed. An extension on that would be also the replacement of the glowplugs with sparkplugs, which would provide more precise control on the start of the combustion and improve stability even further.

Hybridization This first evaluation of a hybrid-PPC powertrain was performed as a series hybrid system and has positive results as it was shown through simulations. This powertrain can reach similar fuel consumption to a real production vehicle. The simulation model was built as a series hybrid system to reduce the complexity and the required computational time while getting useful results. As these results have been promising, an improved model with a parallel hybrid powertrain can ge created and further evaluated. Results are likely to be better compared to the series hybrid as each part of a parallel hybrid model can be better optimized compared to the series hybrid.

Chapter 7

Appendix

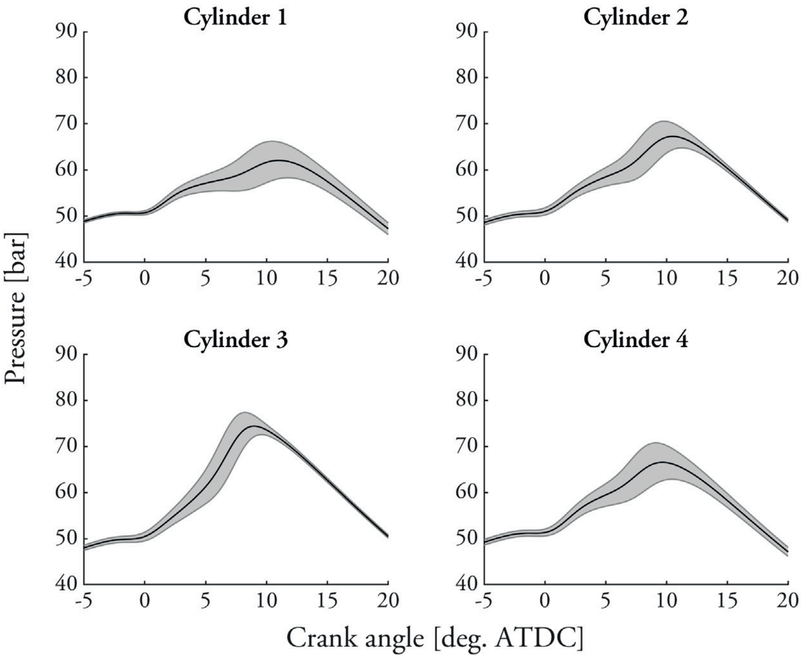

Figure 7.1: Cylinder pressure and one standard deviation for each cylinder at 65 °C with glowplugs on

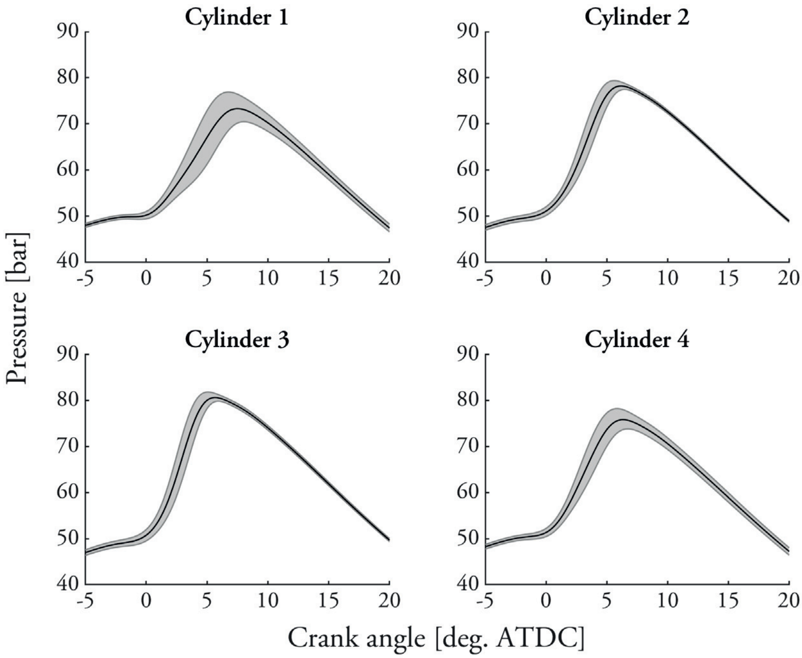

Figure 7.2: Cylinder pressure and one standard deviation for each cylinder at 85 °C with glowplugs on

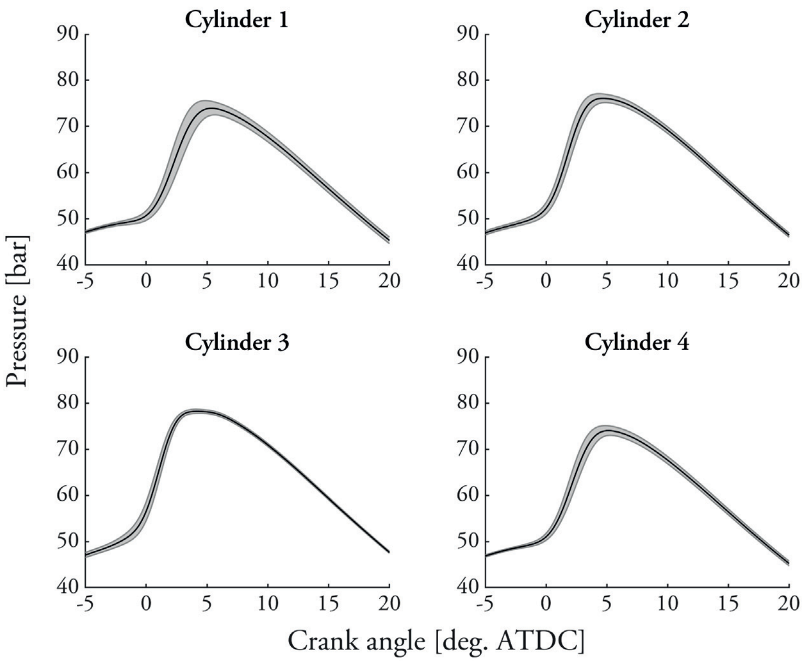

Figure 7.3: Cylinder pressure and one standard deviation for each cylinder at 105 °C with glowplugs on

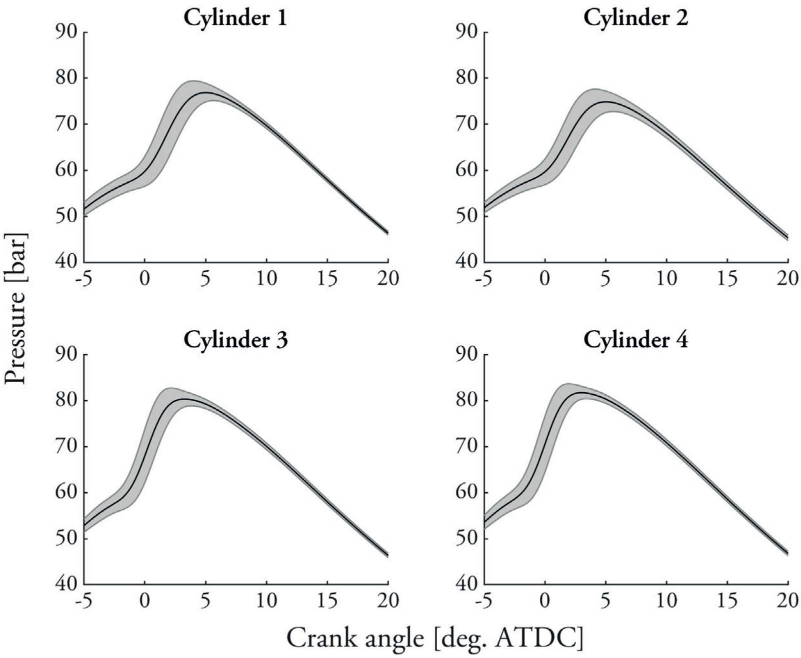

Figure 7.4: Cylinder pressure and one standard deviation for each cylinder with early injection timings (−24° & −12° ATDC) and glowplugs on

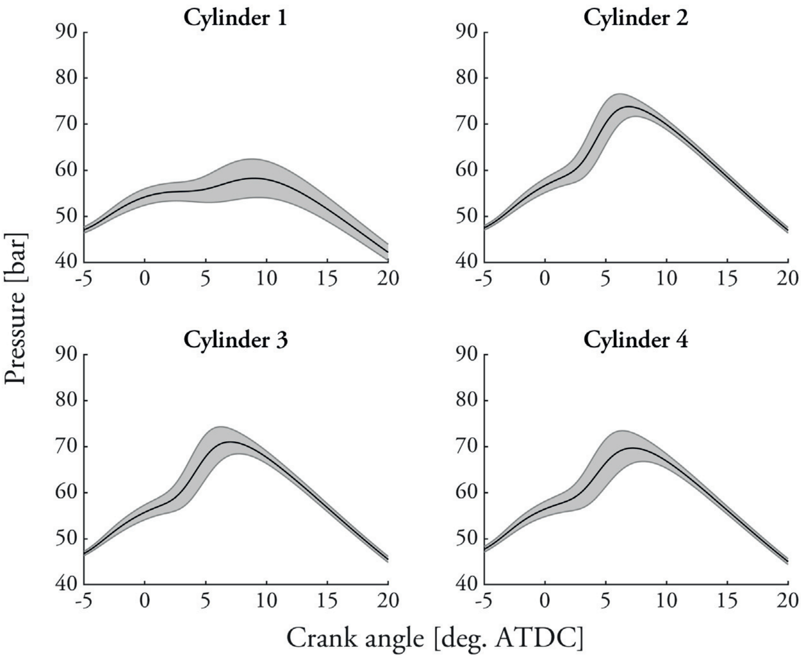

Figure 7.5: Cylinder pressure and one standard deviation for each cylinder with intermediate injection timings (–20° & –10° ATDC) and glowplugs on

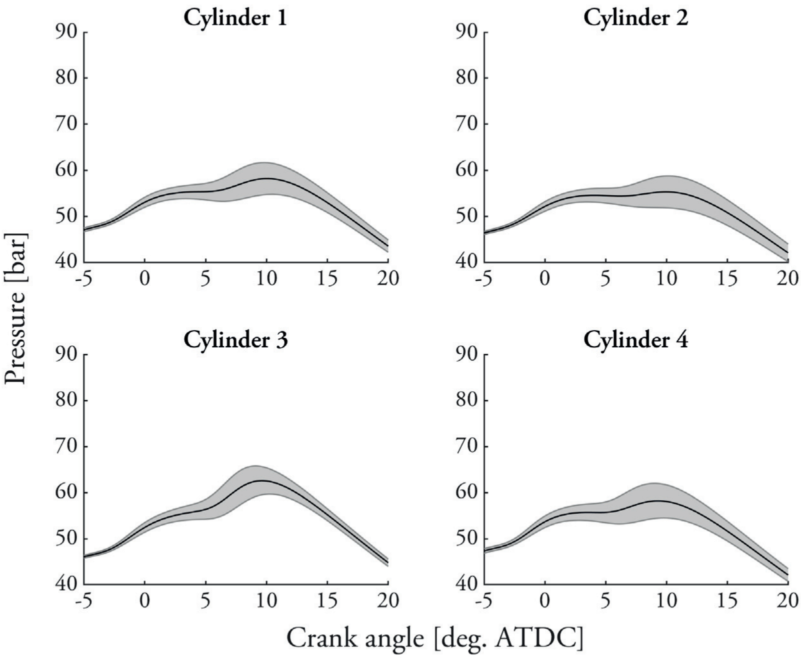

Figure 7.6: Cylinder pressure and one standard deviation for each cylinder with late injection timings (–18° & –8° ATDC) and glowplugs on

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Scientific publications

Author contributions

Paper I: PPC Operation with Low RON Gasoline Fuel. A Study on Load Range on a Euro 6 Light Duty Diesel Engine

N. Dimitrakopoulos, G. Belgiorno, M. Tunér, P. Tunestål, G. Di Blasio, C. Beatrice

9th International Conference on Modeling and Diagnostics for Advanced Engine Systems, COMODIA, 2017

I designed the experimental matrix, operated the engine and analysed the results. Giacomo Belgiorno assisted with the experimental matrix and the analysis. The other authors contributed with feedback and proofreading of the paper. I presented the paper at the 9th International Conference on Modeling and Diagnostics for Advanced Engine Systems. Okayama, Japan in August 2017

Paper II: Effect of EGR routing on efficiency and emissions of a PPC engine

N. Dimitrakopoulos, G. Belgiorno, M. Tunér, P. Tunestål, G. Di Blasio

Applied Thermal Engineering, Vol. 152, pp. 742–750, 2019

I designed the experimental matrix, operated the engine and analysed the results. The other authors contributed with feedback and proofreading of the paper. The paper was published in Applied Thermal Engineering, April 2019

Paper iii: Evaluation of engine efficiency, emissions and load range of a PPC concept engine, with higher octane and alkylate gasoline

N. Dimitrakopoulos, M. Tunér

Submitted to journal: Fuel

I designed the experimental matrix, operated the engine and analysed the results. The other authors contributed with feedback and proofreading of the paper. The paper is submitted to Fuel.

Paper iv: Performance and emissions of a series hybrid vehicle powered by a gasoline partially premixed combustion engine

Antonio García, Javier Monsalve-Serrano, Rafael Sari, **N. Dimitrakopoulos**, M. Tunér, P. Tunestål

Applied Thermal Engineering, Vol. 150, pp. 564–575, 2019

I provided the processed experimental data that were used to model the engine. The simulations for the hybrid powertrain were performed by Antonio García, Javier Monsalve-Serrano & Rafael Sari. The other authors contributed with feedback and proofreading of the paper. The paper was published in Applied Thermal Engineering, March 2019.

Paper v: Investigation of the effect of glowplugs on low load PPC

N. Dimitrakopoulos, M. Tunér

Submitted to SAE Powertrains, Fuels & Lubricants Meeting 2020

I designed the experimental matrix, operated the engine and analysed the results. The other authors contributed with feedback and proofreading of the paper. The paper is submitted to SAE Powertrains, Fuels & Lubricants Meeting 2020.

Paper vi: Parametric Analysis of the Effect of Pilot Quantity, Combustion Phasing and EGR on Efficiencies of a Gasoline PPC Light-Duty Engine

G. Belgiorno, **N. Dimitrakopoulos**, G. Di Blasio, C. Beatrice, P. Tunestål, M. Tunér

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I helped Giacomo Belgiorno with the design of the experimental matrix and data post-processing. The other authors contributed with feedback and proofreading of the paper. The paper was presented at SAE World Congress, April 2017.

Paper VII: Effect of the engine calibration parameters on gasoline partially premixed combustion performance and emissions compared to conventional diesel combustion in a light-duty Euro 6 engine

G. Belgiorno, **N. Dimitrakopoulos**, G. Di Blasio, C. Beatrice, P. Tunestål, M. Tunér

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I helped Giacomo Belgiorno with the design of the experimental matrix and data post-processing. The other authors contributed with feedback and proofreading of the paper. The paper was published in Energy, October 2018.