Characterization and Control of Multi-Cylinder Partially Premixed Combustion

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Abstract

In the last decade diesel combustion has developed in a new direction. Research has been carried out trying to prolong the ignition delay and enhance fuel/air premixing to avoid diffusion combustion as well as lowering the combustion temperature through use of EGR. One of these new combustion concepts is Partially Premixed Combustion (PPC). PPC is aimed to provide combustion with low smoke and NOx without sacrificing fuel consumption.

This thesis presents the development of a multi-cylinder PPC concept. It reaches from the basic characterization of this new combustion strategy to the demands on hardware, control and fuels for a realizable PPC solution. In summary it contains a thorough PPC characterization where the results suggest that high EGR, early injection PPC strategies are to prefer over late injection approaches or smokeless rich diesel combustion. Further, a strong connection between mixing period, defined as the period between end of injection and start of combustion, and PPC has been ascertained. Based on this knowledge a combustion controller with feedback control of mixing period was derived. The operating range of multi-cylinder diesel PPC was then evaluated. The study showed that the PPC load range was limited covering only 25% of the operating region for conventional combustion. In order to reach higher loads for PPC the EGR system was rebuilt to a low pressure system. This system improves EGR/air mixing and cooling and enables high EGR and boost pressure simultaneously. Additionally, gasoline fuels were introduced to extend the ignition delay and mitigate soot formation. An extensive fuel comparison was carried out to find the most suitable fuel for PPC operation.

With the improved set-up the operating range was reevaluated. By combining the use of a low pressure EGR system and standard gasoline the operating region of PPC has been extended to cover 50% of the engine nominal operating region.

The final part of this thesis is dedicated to a novel method of cylinder individual efficiency estimation based on the cylinder pressure trace. With this method, control strategies aiming directly at fuel consumption optimization can be developed. An extremum seeking control algorithm was applied. The results show that the controller manages to find the maximum brake torque region both with and without external excitation. Finally, the estimation error in accumulated fuel consumption from the experiments is around 1% which shows the potential of using the absolute value of the efficiency estimation in other control concepts.

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Nomenclature

ATDC:	After Top Dead Center
BTDC:	Before Top Dead Center
BMEP:	Brake Mean Effective Pressure
CA50:	Crank angle 50% burned
CAD:	Crank Angle Degree
EGR:	Exhaust Gas Recirculation
HCCI:	Homogeneous Charge Compression Ignition
HRR:	Heat Release Rate
IMEP:	Indicated Mean Effective Pressure
LTC:	Low Temperature Combustion
MBT:	Maximum Brake Torque
MP:	Mixing Period
MPC:	Model Predictive Control
PPC:	Partially Premixed Combustion
PR:	Premix Ratio
SOI:	Start of Injection
λ :	Relative air/fuel ratio
ϕ :	Fuel/air Equivalence Ratio

Contents

Chapter 1

Introduction

1.1 Background

The automotive industry is in the middle of a very turbulent and challenging period. The demand for new energy efficient powertrains has become significantly stronger in the last couple of years. It started with the rapid increase in oil price and grows even stronger during the green house gas debate where carbon dioxide, one of the major combustion products, is in focus. During the second half of 2008 the price of crude oil went down to be not even a quarter of the all time high value recorded in July 2008. However, the oil price is increasing again and the attitude towards global warming stays unchanged, hence decreased energy consumption through higher energy efficiency is a research topic that will last.

Further, the global financial crisis and the economic depression that became evident during 2008 affected the automotive industry's sales negatively. The economic future was uncertain which made both companies and private persons hesitant with their purchases. However, this was temporary and the sales are now up again when the world economy has stabilized. It is of high interest for the automotive companies to maintain research and development during such periods since the ones with the most attractive products have a very good opportunity to cut some new market shares now.

One major part of automotive research concerns new powertrains with focus on high energy efficiency but also on low emissions of toxic compounds and particulates that are hazardous for climate and people. The automotive market of today is dominated by diesel and SI-engines (gasoline engines) which are internal combustion engines (ICEs). The ICE has been used to power vehicles for more than one hundred years. It has the benefit of being a robust and affordable power source with a high power to weight ratio making it very suitable for mobile applications. Due to the widespread use, the infrastructure supporting ICEs is very extensive. ICE research is carried out both on engine design, control and fuels. These research disciplines need to be combined in order to optimize new strategies. Since the introduction the ICE has been constantly improved and it has high potential to be an important part of environmentally friendly, energy efficient powertrain solutions for the future. However, there are alternatives to ICEs such as fuel cell and electric powertrains.

Fuel cell vehicles have been thought of as the next generation of environmentally friendly vehicles. The most common concept uses hydrogen as fuel to produce electricity which then powers an electric engine. The biggest advantage with hydrogen fuel cells is that they produce no local emissions and in that aspect they are environmentally friendly. However, hydrogen gas does not appear in nature but has to be produced in some way. One way is the conversion of fossil fuels, such as natural gas, into hydrogen. This process produces carbon dioxide and cannot improve overall efficiency or reduce the emission of GHGs [18]. Another way of producing hydrogen gas is to use electricity to split water by electrolysis. This process is clean but only if it utilizes renewable electricity. The efficiency from electricity at the power-station to vehicle wheel using this method is just at around 20% [19] which is low when considering that the total efficiency must contain additional losses from electric power production which in the US is only about 35% efficient [20]. In addition to this, the hydrogen molecule is so small that it is practically impossible to store hydrogen gas a longer period of time without significant leakage and, to wrap this up, the interest for this powertrain solution has decreased.

The term electric vehicle (EV) refers to vehicles powered by electric motors that use batteries as their energy source. The powertrain solution has no exhaust and the efficiency from electricity at the power station to vehicle wheel is at around 66% [19]. As with the hydrogen fuel cell powertrain the total efficiency and cleanness depend on how the electricity is produced. In the US, coal power production is dominant and responsible for almost 50% of the electricity produced [21, 22] while the Swedish electricity demand is covered mainly by hydroelectric and nuclear power which suggests that an EV is more environmentally friendly in Sweden than in the US. The major challenge for the vehicle concept itself concerns the batteries that need higher energy storage capacity and weight reduction which then can improve the vehicle range. A long term objective is to make sure that the increased electricity consumption is compensated for by renewable electricity.

Combinations of the above powertrains are called hybrids. There are different kinds of hybrid vehicles, some of them are already on the market. Most hybrids use a combustion engine in combination with an electric motor. In the parallel hybrid both the electric and the combustion engines can power the vehicle individually or together. The electric motor is used at lower speeds while both electric motor and combustion engine work together at higher speeds. The battery is recharged by recovering the braking energy. The electric part of the powertrain alone can not be used to cover normal driving. Plug-in hybrid electrical vehicle (PHEV) is another concept which is similar to the parallel hybrid but has increased battery capacity and the possibility to charge directly from the electricity grid.

A series hybrid uses an electric powertrain driven by an ICE which operates at its most efficient operating points. The ICE can be used to charge the battery, charge a capacitor and directly power the electric motor. When large amounts of power are needed the electric motor can draw electricity from a combination of batteries, capacitors and the generator. Most hybrids uses SI engines but it is likely that the drivetrain would be even more efficient if it used a diesel engine instead.

1.2 Diesel Engines

The diesel engine was invented by Rudolf Diesel in the 1890's [68]. Diesel had one major goal with his engine: it must burn less fuel per unit of power than any other engine yet devised.

Diesel succeeded and low fuel consumption is what the diesel engine has become most famous for. However, diesel engines also have a reputation to be dirty in terms of smoke and nitrogen oxide emissions where the former might be carcinogenic and the latter cause photochemical smog and acid rain when reacting with air [69]. Development, especially regarding fuel injection equipment and super charging, has made it possible to reduce the levels significantly but they are still a major issue.

Diesel engines have low fuel consumption due to a couple of reasons. At part load the pumping losses are significantly lower than with SI engines since the intake air does not need to be throttled as the diesel engine works with excess air. The excess air is furthermore a reason why a diesel engine has higher combustion efficiency which also lowers the fuel consumption. Finally the compression ratio is higher in a diesel engine which generally yields higher thermodynamic efficiency. The low fuel consumption is the reason why diesel engines are dominant in the commercial transport sector where just a slight decrease in fuel consumption means significantly lower costs. It is also one of the reasons why the diesel passenger car sector is rapidly increasing in the aftermath of the global warming debate where buyers have become more concerned about CO2 emissions[1]. For the consumer it is easy to overlook the higher smoke and NOx emissions that a diesel engine has compared to SI-engines when just looking at the CO2 level. However, both particulates and NOx are regulated by international emission legislations which are becoming more and more stringent with time.

1.3 Conventional Diesel Combustion

Diesel engines use the high air charge temperature that results after the compression stroke to ignite the air/fuel mixture. Fuel is injected directly into the cylinder at the end of the compression stroke near Top Dead Center (TDC). The temperature and the amount of fuel in relation to air are the most important parameters for combustion theory. The latter is often described by a normalized fuel/air mass ratio, called equivalence ratio, ϕ, defined by Equation 1.1.

$$\phi = \frac{(F/A)_{actual}}{(F/A)_{stoichiometric}} \tag{1.1}$$

When $\phi = 1$ the combustion is stoichiometric meaning that there is just enough oxygen to oxidize all fuel. Conditions where $\phi < 1$ are called lean and conditions with $\phi > 1$ are called rich. Since the fuel injection is so close to TDC there is not enough time for fuel and air to mix to a homogenous mixture before combustion starts. This means that the fuel elements will burn at a variety of equivalence ratios and in different ways depending on the history of fuel injection and combustion. In order to structure a conventional diesel combustion event, it can be divided into four distinct parts.

The first part of the combustion event is the ignition delay which is the time from start of injection to start of combustion. In order for the combustion to start there needs to be a burnable mixture of fuel and air which has to be exposed to high temperatures long enough for the charge to ignite. This is what causes the ignition delay. The ignition delay is influenced by both physical and chemical factors and can be described by the expression in Equation 1.2 presented in [50].

$$\tau_{id} = A \cdot \bar{p}^{-n} \cdot X_{O_2}^{-m} \cdot e^{E_a/R\bar{T}_a} \tag{1.2}$$

A, n and m are constants dependent on fuel and airflow characteristics, p is the ambient pressure, X_{O_2} is the oxygen concentration, R is the universal

[1] According to [23] a majority of the people living in EU believe that global warming is a very serious issue and that the impact of CO2 emissions is significant. Over 60% claims that they have taken personal action against climate change and out of those almost 20% did this by "purchasing a car that consumes less fuel, or is more environmentally friendly."

gas constant, E_a/R is an apparent activation energy and $\bar{T}_a$ is the estimated average in-cylinder temperature.

When the preconditions are right the combustion starts spontaneously in the air/fuel mixture. The energy released increases pressure and temperature in the cylinder which makes other parts of the charge ignite. This part is called premixed combustion. If a large quantity of fuel has been injected during the ignition delay there will be a high amount of energy released during this phase which makes the cylinder pressure increase rapidly. This results in the characteristic diesel engine sound.

The next phase of the combustion event consists of spray-driven combustion which is quasi-stationary. The fuel is injected, mixed with air, vaporized and combusted in a continuous way where the combustion occurs in zones where a burnable mixture can be found. A conceptual model of the combustion jet is presented in Figure 1.1 published in [3]. It consists of a liquid region close to the fuel injector, a fuel-rich premixed reaction zone in the central region and a diffusion flame at the periphery where fuel burns at high temperature near stoichiometric conditions.

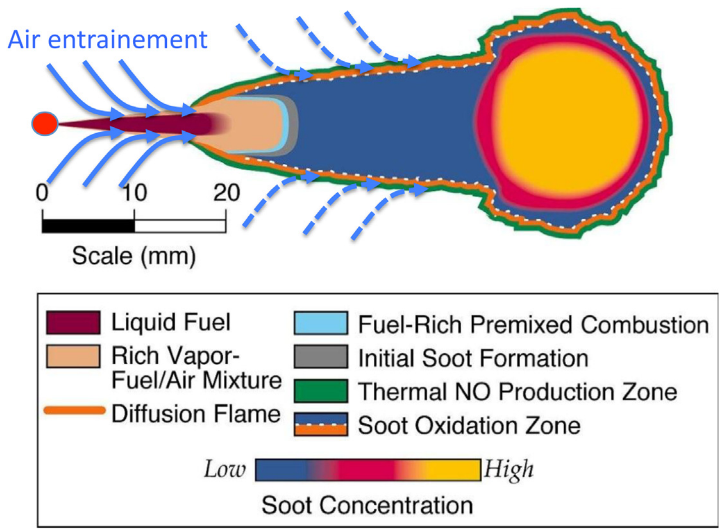

Figure 1.1: Conceptual model of diesel combustion. [3]

The final part is simply called the late combustion phase. There are several reasons for the late heat release. A small fraction of the fuel may not yet have burned, a fraction of the fuel energy is present in soot and intermediate combustion products and can still react.

Emissions

Conventional diesel engines produce very little carbon monoxide and have low hydrocarbon emissions as they burn the fuel in excess air even at full load. Instead the emission issue concerns NOx and smoke as mentioned above.

NOx is primarily formed when a mix of nitrogen and oxygen is subjected to high temperatures. A diesel engine has high local combustion temperature, especially around the diffusion flame described above. This combined with excess oxygen provides favourable conditions for NOx formation.

Smoke consists mainly of soot particles that are produced in locally rich zones where the amount of air available is not sufficient for complete combustion. The fuel-rich premixed reaction zone in Figure 1.1 is where soot starts to form. The rate of soot formation is coupled to the equivalence ratio in this region. The equivalence ratio is determined by lift-off length, which is the distance between the injector and the most upstream part of the burning flame, and the air entrainment rate. The reason why ϕ at lift-off is so important for soot formation is that down stream of the lift-off, air entrainment cease since the oxygen is consumed in the diffusion flame at the jet periphery meaning that the equivalence ratio at lift-off is approximately the equivalence ratio of the premixed reaction zone. The emperically compiled expression in Equation 1.3 shows the governing parameters affecting the lift-off length, H [49], while Equation 1.4, based on physical models, describes the equivalence ratio at a given position in the spray [46, 47]. Combined these equations give an estimate of the actual ϕ at lift-off.

$$H \propto T_a^{-3.74} \rho_a^{-0.85} d^{0.34} U Z^{-1} \tag{1.3}$$

T_a is the gas temperature, ρ_a is the gas density, d is the injector hole diamter, U is the fuel velocity and Z_{st} is the stoichiometric mixture fraction, which is the ratio of the fuel mass and the total mass at stoichiometric conditions.

$$\phi(x) = \frac{2(A/F)_{st}}{\sqrt{1+16(\frac{x}{x^+})^2} - 1} \tag{1.4}$$

$(A/F)_{st}$ is the stoichiometric air-fuel ratio, x is the axial location and x^+ is the characteristic length defined by Equation 1.5

$$x^+ = \sqrt{\frac{\rho_f}{\rho_a}} \frac{d\sqrt{C_a}}{tan(\frac{\alpha}{2})} \tag{1.5}$$

where ρ_f is the fuel density, C_a is the contraction coefficient and α is the jet spreading angle which depends on fuel and ambient gas densities. According to [6], soot formation can be suppressed if ϕ can be kept below 2.

The soot particles that have formed in the fuel rich region can be oxidized later in the cycle if the temperature is high and they are exposed to oxygen. When optimizing combustion with respect to emissions, the trade-off between smoke and NOx becomes clear. If the combustion temperature is lowered to reduce NOx formation the soot oxidation decreases, hence the exhaust gas contains more smoke given a certain soot formation. Figure 1.2 shows how NOx and soot formation and soot oxidation depend on temperature and fuel/air equivalence ratio. In addition, generic trajectories for the main CI combustion strategies are shown giving a conceptual picture of the phases each fuel element undergoes during combustion.

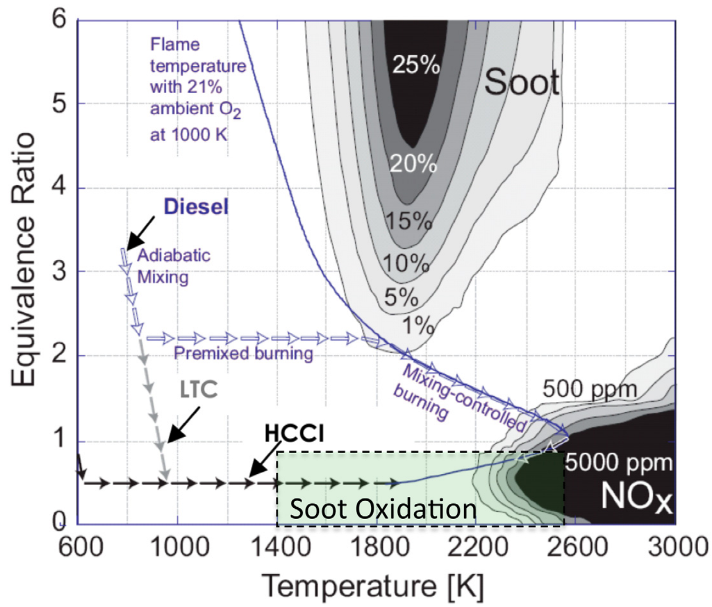

Figure 1.2: Soot and NOx formation zones as function of temperature and equivalence ratio. [6]

Another potential problem when reducing combustion temperature to suppress NOx are increased carbon monoxide (CO) and unburned hydrocarbon (HC) emissions. CO and HC can originate from both lean and rich mixtures and are highly temperature dependent as can be seen in Figure 1.3, originally published in [53]. For lean conditions, high temperature enables complete oxidation of both CO and HC. At intermediate temperature HC and CO will be present in the exhaust while there will be only HC in the exhuast if the temperature is so low that the fuel does not ignite. At rich conditions, a high temperature will result in partial oxidation of HC to CO giving low engine out HC and high CO while both HC and CO are present at lower combustion temperatures.

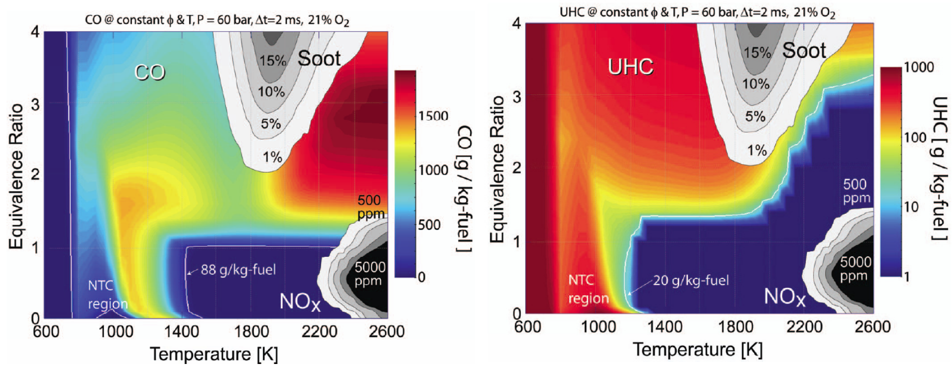

Figure 1.3: Carbon monoxide and unburned hydrocarbons as function of temperature and equivalence ratio.

Emission Aftertreatment

The upcoming emission legislations in Europe and North America are very stringent. In 2007 it was proposed that the EU emission standard Euro VI would limit NOx emissions to 0.4 g/kWh and particulate matter (PM) to 0.01 g/kWh. This standard will be implemented in 2013. The corresponding emission standard in North America is US10. US10 limits NOx to 0.27 g/kWh and PM to 0.0135 g/kWh and will be fully implemented already in 2010.

With conventional diesel combustion Euro VI and US10 can only be met by installing after treatment systems for NOx and smoke. NOx can be reduced by Selective Catalytic Reduction (SCR) [69, 70] which is a means of converting NOx, with the aid of a catalyst into diatomic nitrogen, N2, and water, H2O. A gaseous reductant, typically ammonia produced from urea injection, is added to a stream of flue or exhaust gas and is absorbed onto a catalyst. Carbon dioxide, CO2 is a reaction product when urea is used as the reductant. In order to achieve complete reduction of NOx the amount of the reductant will be on the border of leakage where just a slight decrease in NOx will make some of the reductant slip through the system without any chemical reaction.

Smoke levels can be lowered by a Diesel Particulate Filter (DPF) [69, 70]. The DPF is basically a filter where particles are removed from the exhaust gas. One major problem with this method is that the filter gets saturated. That increases the exhaust back pressure which deteriorates the engine performance. The solution is to change or clean the filter. Some filters are single use, while others are designed to burn off the accumulated particles which requires engine modifications. It would be of interest to be able to meet future emission legislations without relying on expensive after treatment systems.

Therefore new combustion concepts are under investigation.

1.4 Low Temperature Combustion

In the last decade diesel combustion has developed in a new direction. Research has been carried out trying to prolong the ignition delay and enhance fuel/air premixing to avoid diffusion combustion as well as lowering the combustion temperature. A common name for this kind of combustion is Low Temperature Combustion (LTC) [11, 24, 25].

The LTC research is inspired by the Homogeneous Charge Compression Ignition (HCCI) concept. HCCI has proved to combine low emissions and high efficiency under certain conditions at low load. HCCI engines traditionally use port injection, i.e. the fuel is injected outside the cylinder in the inlet manifold, to assure a homogeneous charge. This method causes severe issues regarding the combustion phasing control since there is no connection between injection timing and start of combustion as in traditional CI (diesel) engines. The most effective solution to this problem is to use Direct Injection (DI) with fuel injection sufficiently close to TDC to yield high combustion timing controllability but still early enough to provide extensive fuel air premixing. It is also in this region the LTC concepts are found.

As mentioned above, one obvious method to increase the ignition delay in a DI engine is to inject fuel earlier in the compression stroke when the compressed air is cooler. This was done by Takeda et al. [2] where the authors call the strategy PREmixed lean DIesel Combustion (PREDIC). In order to maintain combustion around TDC a lower cetane fuel i.e. a fuel that ignites at higher temperature, was used. With this concept low NOx and smoke were obtained at a cost of higher CO and HC emissions and deteriorated efficiency. This very early stage injection also makes it hard to control combustion phasing since the connection between injection timing and start of combustion is weaker. This is a major issue since over-advanced combustion can destroy the engine when running on higher loads and over-retarded combustion might result in misfire the following engine cycle. Other experiments of this kind have been carried out by [9, 10] among others. It is also possible to use standard diesel fuel for this kind of combustion if the compression ratio is lowered which can be seen in e. g. [12].

EGR

Exhaust Gas Recirculation (EGR) is another way to extend the ignition delay. EGR works by recirculating a portion of the engine's exhaust gas back into the cylinders. The exhaust gas dilutes the intake mix with inert gas, lowering the adiabatic flame temperature and reducing the amount of excess oxygen. The exhaust gas also increases the specific heat capacity of the mix, lowering the peak combustion temperature.

Akihama et al. [6] showed a concept called Smokeless Rich Diesel Combustion in 2001 where very high EGR levels and advanced injection timing were combined yielding a combustion with very low smoke and NOx emissions. With this method NOx is suppressed by the reduced combustion temperature while smoke is reduced by a combination of enhanced mixing and a combustion temperature which is below the reaction temperature for soot formation. Additionally, injection timings close to TDC assures good combustion controllability. However, also here the improvements come with increased HC, CO and fuel consumption.

There is also a possibility to increase the ignition delay by injecting fuel early in the expansion stroke i.e. after TDC. With this retarded fuel injection the cylinder temperature is already decreasing when the fuel is injected into the cylinder. Smoke reduction follows directly from the enhanced fuel/air premixing while NOx formation is decreased due to the lower combustion temperature caused by placing the combustion later in the expansion stroke. Kimura et al. show this type of combustion called Modulated Kinetics combustion in [4, 5].

PREDIC, Smokeless Rich Diesel Combustion and Modulated Kinetics Combustion show strategies to avoid the smoke/NOx trade off but when doing so a trade-off between simultaneous reduction of smoke/NOx and CO, HC and fuel consumption occurs instead. Even though HC and CO might be taken care of by an oxidizing after treatment system, increased fuel consumption is not defendable neither from an economical nor environmental perspective.

Another challenge is to extend the operating region since many LTC concepts only work at low load and low speed operation. In more recent studies multiple injection experiments at high EGR levels have been able to increase load and speed. In [14] this concept is able to operate at medium speed and load with low emissions and maintained engine efficiency. In other studies it is shown that higher octane fuels combined with cooled EGR can extend the load range above medium load at low speed [67]. These experiments have been carried out on single cylinder engines with external boost and controlled inlet temperatures and it is hard to directly translate the results to a production engine.

1.4.1 Partially Premixed Combustion

The combustion concepts above can be rated as Partially Premixed Combustion (PPC) concepts meaning that fuel and air mix before combustion but the charge is not homogeneous. The PPC research at Lund University aims to provide combustion with low smoke and NOx without sacrificing fuel consumption. This thesis concerns PPC in multi-cylinder heavy duty engines resembling standard build production engines. More specifically, the research contribution regards characterization of PPC and development of combustion controllers to aid the optimization and adaptation of PPC to multi-cylinder engines.

Chapter 2

Combustion Engine Analysis, Modeling and Control

There are different approaches for combustion engine analysis depending on the purpose of the research. Engines with optical access are used for studies of fuel sprays and detailed studies of emission formation and combustion characteristics. These engines are often of single cylinder type in order to exclude cylinder to cylinder variations which are of high interest mainly in engine control research. The experiments in this thesis are control oriented and have been carried out on multi cylinder full metal engines resembling production engines.

The research engine set-ups use many different sensors in order to measure pressures, temperatures, fuel consumption and emissions. The demand on accuracy and sample rate of a sensor signal depends on what it is used for. In-cylinder pressure measurements are the most important for combustion analysis and require a very fast sampling rate in order to be useful.

2.1 Heat Release

Cylinder pressure versus crank angle data is used to obtain quantitative information about the progress of combustion. By analyzing the cylinder pressure trace using thermodynamics the combustion energy release can be calculated. The energy release is often referred to as Heat Release (HR) [1].

There are different methods to calculate heat release. However, they all start with the first law of thermodynamics for an open system that is uniform in pressure and temperature. This law can be seen in (2.1)

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt} \tag{2.1}$$

where dQ/dt is the heat-transfer rate across the system boundary into the system, $p(dV/dt)$ is the rate of work transfer done by the system due to system boundary displacement. $\dot{m}_i$ is the mass flow rate into the system across the system boundary at location i, h_i is the specific enthalpy of mass entering or leaving the system, and U is the energy of the material contained inside the system boundary.

Under some assumptions, including that the contents of the cylinder can be modeled as an ideal gas, (2.1) can be rewritten using the ideal gas law, $pV = mRT$, to become

$$\frac{dQ}{d\theta} = \frac{\gamma}{1-\gamma}p(\theta)\frac{dV}{d\theta} + \frac{1}{\gamma-1}V(\theta)\frac{dp}{d\theta} \tag{2.2}$$

where θ is crank angle degrees CAD. Equation 2.2 is called the apparent heat release equation with a fixed ratio of specific heats and it is the basis for the heat release calculations in this thesis.

Figure 2.1 is taken from [71] and shows a typical Heat Release Rate (HRR) for conventional diesel combustion. The different combustion phases presented in the previous chapter can be seen.

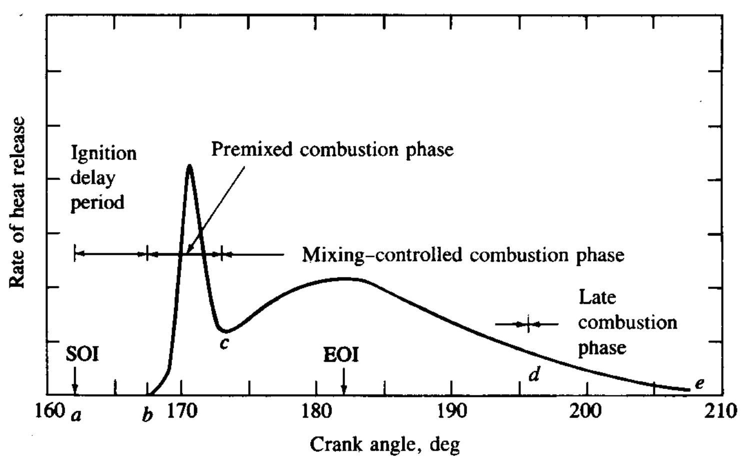

Figure 2.1: Typical HRR for diesel combustion [71].

The HRR for a Partially Premixed Combustion event differs from the above combustion event since it consists mainly of premixed and late combustion. PPC HRRs are presented later in this thesis.

Heat Release Parameters

From the Heat Release it is possible to extract different parameters. One of the most important parameters is called CA50 and is often used to measure combustion phasing i.e. when in the cycle the combustion takes place. Figure 2.2 which was published in [15] shows the definition of the most common parameters. CA50 is the crank angle where 50% of the maximum energy has been released. Other parameters of interest are CA10 and CA90 which are used to calculate ignition delay and combustion duration. CA1 and CA99 appears to be a better choice for these calculations but due to measurement and model uncertainties these are considered less robust which in turn makes them less suitable for control purposes.

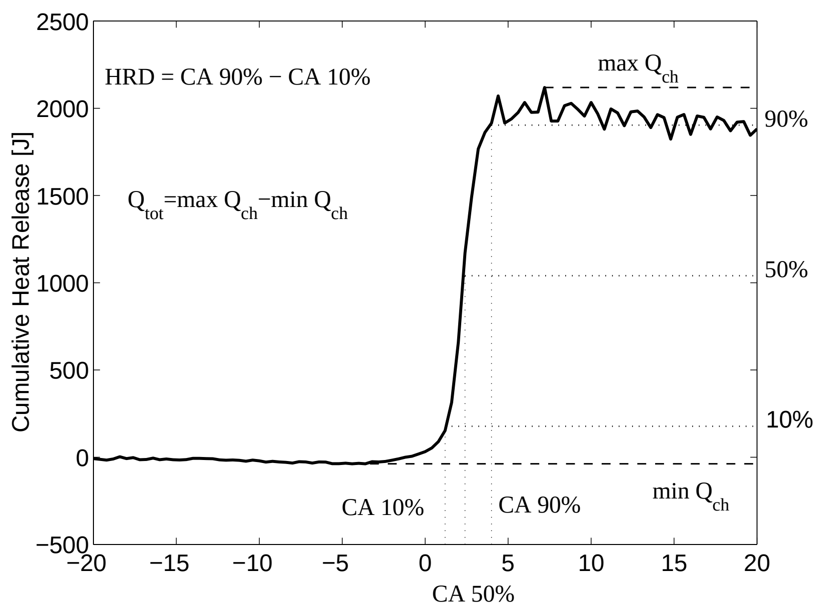

Figure 2.2: Heat release parameter defintions [15].

2.2 Engine Load

The engine-out torque is measured by a dynamometer. Although torque can be seen as a measure of load it is often normalized by the engine displacement volume in order to be able to compare small and large engines on the same scale. The resulting term is called Brake Mean Effective Pressure (BMEP). Another way of measuring load is to calculate the Indicated Mean Effective Pressure (IMEP) which is calculated from the cylinder pressure trace. IMEP does not take friction losses into account.

2.3 Modeling and Control

Physical modeling of a combustion engine is a very complex matter. It involves many scientific disciplines such as thermodynamics, fluid dynamics, chemistry and mechanics. The modeling methods differ from each other depending on their purpose. There are models for analysis, optimization and design which often are detailed models with high resolution such as Computational Fluid Dynamics (CFD) models of combustion. Another kind of models are those derived for control purposes. These models have higher requirements on calculation speed in order to be able to run in real time. The structure of these models is often simpler and the number of inputs and outputs is lower compared to the previous model group. Combustion models for control can be developed on different time scales depending on the control objective. Cycle resolved models are the most common. These models predict the output, for example combustion phasing, based on the present state and control signals, such as injection timing, once for each cylinder and cycle. Although cycle based models often are sufficient for combustion control, additional optimization is enabled with time or crank angle based models that can be used for rate shaping where the HRR is controlled in real time during combustion. This is beyond the state of the art of combustion control and it requires very fast actuators to be realized.

An alternative to physical modeling is called black box modeling. Black box models are derived using system identification which is a process of deriving a mathematical system model from observed data with some predetermined criterion [72]. The models used for the experiments in this thesis are based on system identification.

All modern truck and passenger car engines are equipped with Electronic Control Units (ECUs) which monitor engine behavior and control fuel injection, intake pressure, EGR etc [73]. A typical diesel engine ECU relies mainly on a control system where certain conditions, such as the engine operating point, determine the action taken using a map containing predefined control strategies. This type of control can be classified as feed-forward control. However, ECUs are becoming more and more powerful and sophisticated and it is expected that the engine performance could be improved significantly if more advanced control methods were utilized. One of the most important features of the next generation of ECUs would be cycle to cycle based, closed loop control of combustion. This kind of control enables further optimization of efficiency and emissions. The research engines used in the experiments for this thesis have the possibility to run cycle to cycle based combustion control.

Chapter 3

Experimental Set-up

The experiments in this project have been carried out on heavy-duty multi-cylinder diesel engines. Two full-size truck engines from Volvo have been used, see Figure 3.1 and 3.2. There are many similarities between the engines and the specifications can be found in Table 3.1. The experiments for Paper I were made on the Volvo D12 while experiments for the following papers were made on the Volvo MD13.

Table 3.1: Engine specifications.

Engine	D12	Engine	MD13
Operated cylinders	6	Operated cylinders	6
Displaced Volume	2 l	Displaced Volume	2.13 l
Bore	131 mm	Bore	131 mm
Stroke	150 mm	Stroke	158 mm
Connecting Rod Length	260 mm	Connecting Rod Length	267.5 mm
Number of Valves	4	Number of Valves	4
Compression Ratio	18.5:1	Compression Ratio	16:1
Fuel Supply	DI	Fuel Supply	DI

Both engines are equipped with an injection system that consists of the Delphi E3 unit injector which has injection pressure capability up to 2500 bar. The injection pressure, start of injection (SOI) and injection duration (ID) can be changed on a cycle to cycle basis. The SOI and ID can be specified directly in crank angle degrees (CAD) while injection pressure is controlled by an angle relative to SOI. This angle is referred to as Needle Opening Pressure (NOP). The resulting fuel pressure in the unit injector is determined by NOP in combination with several other factors including the fuel cam rate, cam phasing relative to the engine and fuel characteristics. The engines have production nozzles with six holes and an included spray angle of 140 degrees. The flow is 1.85 liter/min at 100 bar.

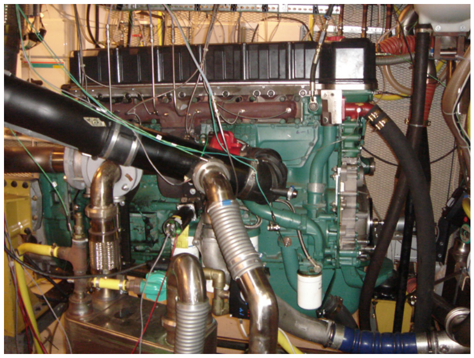

Figure 3.1: Volvo D12.

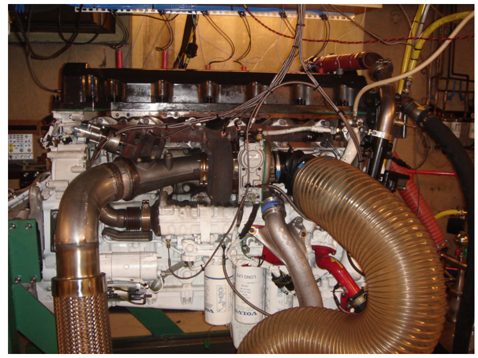

Figure 3.2: Volvo MD13.

The D12 has an in-house made long route EGR system and Variable Geometry Turbine (VGT) that made it possible to adjust the inlet pressure and EGR level independently. The MD13 was originally equipped with a production, short route EGR system, i.e. a high pressure EGR system with the EGR source up stream of the turbine. However, for Paper IV a long route system, similar to the D12's was installed. The MD13 has a production VGT which is used to get the desired inlet pressure.

The control software for the engines was developed in C/C++ and is described in detail in [27]. It is an open system which runs on a Linux PC.

All cylinders were equipped with pressure transducers to measure cylinder pressure. The D12 uses Kistler 7061B transducers while the MD13 has AVL QC43D pressure transducers. The rocker arms have force transducers whose signals were used to calculate the injection pressure. The cylinder and injection pressure signals were sampled every 0.2 CAD by a Microstar DAP 5400a/627 data acquisition card. The emissions were measured by a Horiba Mexa 7500 while smoke was measured by an AVL 415S. Fuel consumption was calculated based on measurements from a Sartorius CPA 10001 scale. The MD13 engine setup has been validated against input data from Volvo powertrain at 13 standard operating points reaching from idle to full power operation.

Chapter 4

Results and Discussion

4.1 PPC Characterization

As was previously described in the introduction there are several premixed combustion strategies that are aimed at simultaneous low NOx and smoke. However, the definitions of these combustion concepts are vague. In order to understand the foundation of PPC operation and to determine which of the strategies that offers the best performance an investigation was carried out.

The intention was to show where and how PPC occurs for different combinations of injection timing and EGR. In particular it was studied how the heat release was affected. The investigation focused on changes in mixing period, defined as the period between end of injection and start of combustion, but other parameters were also considered and together they form a map that shows the development from conventional diesel combustion to partially premixed combustion.

The results presented below are from experiments at 1200 rpm and 5 bar BMEP corresponding to approximately 25% of the rated engine load. The injection duration and injection pressure were adjusted and fixed for conventional diesel operation without EGR. The inlet pressure was set to 1.3 bar which provided high brake efficiency and enabled lean operation up to 60% EGR.

Combustion Map

The period between End of Injection (EOI) and Start of Combustion, here defined by CA10, is called Mixing Period (MP). MP can be used to determine the fuel air mixing prior combustion. Figure 4.1 shows MP as function of EGR and injection timing. In this study, the region with a positive mixing period is considered a Partially Premixed Combustion region. The black iso curve shows where this period is zero. This means that this curve defines the PPC border. In order to verify this definition the connection to combustion character was investigated. The Heat Release Rates (HRR) for various conditions are shown in the lower part of the figure. The typical shape of a premixed combustion event can be seen for early injection at 50% EGR. There is also a large part of premixed combustion at 35% EGR although some spray-driven combustion seems to occur. The difference in combustion timing is distinct while the maximum in HRR is similar. Both combustion conditions show low temperature reactions but they are more pronounced at high EGR operation. In the other end of the spectrum, late injection at 50%

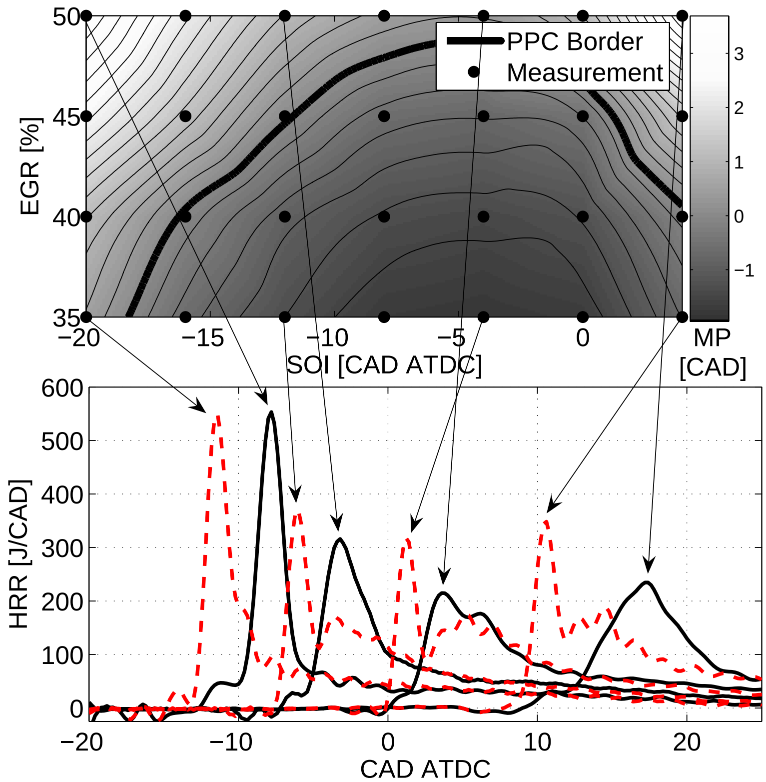

Figure 4.1: Mixing period and the corresponding HRR at A25 with 1000 bar injection pressure.

EGR results in a MK combustion event described in [4, 5]. When looking at points in the SOI interval [-12,0] conventional diesel HRR is observed for 35% EGR whereas the same interval shows borderline PPC combustion for 50 % EGR operation. It is also noted that the combustion duration increases for retarded injection.

Emissions Maps

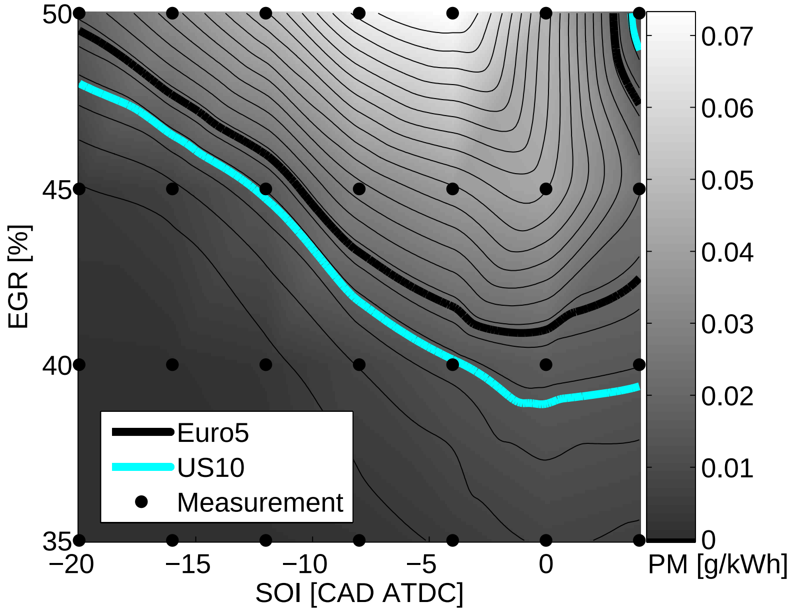

Figure 4.2: PM emissions in the critical region with level curves indicating Euro V and US10 Emission Standards.

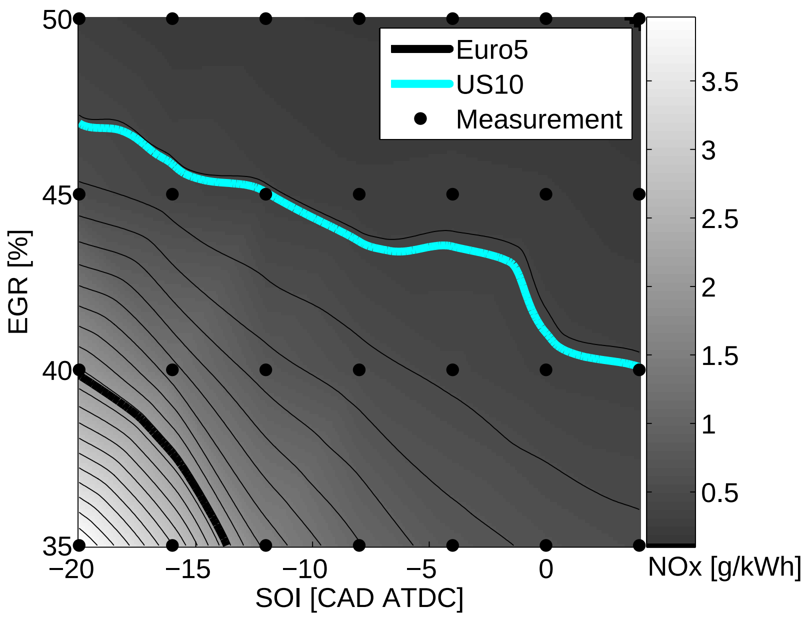

Figure 4.3: NOx emissions in the critical region with level curves indicating Euro V and US10 Emission Standards.

Figure 4.2 shows how the Particulate Matter (PM) level varies with injection timing and EGR. The PM level starts to increase rapidly with increased EGR level. At 50% EGR, PM emissions are very high which is a similar trend to that reported in [6]. The increase in PM is probably due to a temperature reduction in combination with less excess oxygen that follows an EGR increase when the inlet manifold pressure is kept constant.The SOI dependency is connected to changes in mixing period. A long mixing period reduces PM formation. The level curves corresponding to emission legislations show that early injection is to prefer in order to keep PM at a low level at these EGR rates. The peninsula shape can be recognized as a combination of the smoke development in [6] when sweeping EGR and the smoke development that [7] shows when sweeping SOI at 60% EGR.

In the region between 35 and 50% EGR NOx emissions reach low levels. The major reason for the decrease in NOx is the reduced combustion temperature. When studying Figure 4.3 it can be seen that the dependency on injection timing is weaker at higher EGR levels.

The development in efficiency can be studied in Figure 4.4. Combustion efficiency is fairly constant and independent of SOI up to 40 % EGR where it starts to decrease. The most noticeable change in thermodynamic efficiency is that the maximum tends to be found for earlier injection timings when increasing EGR. This is due to a longer ignition delay which follows increased EGR ratio where Maximum Brake Torque (MBT) timing is achieved for earlier SOI.

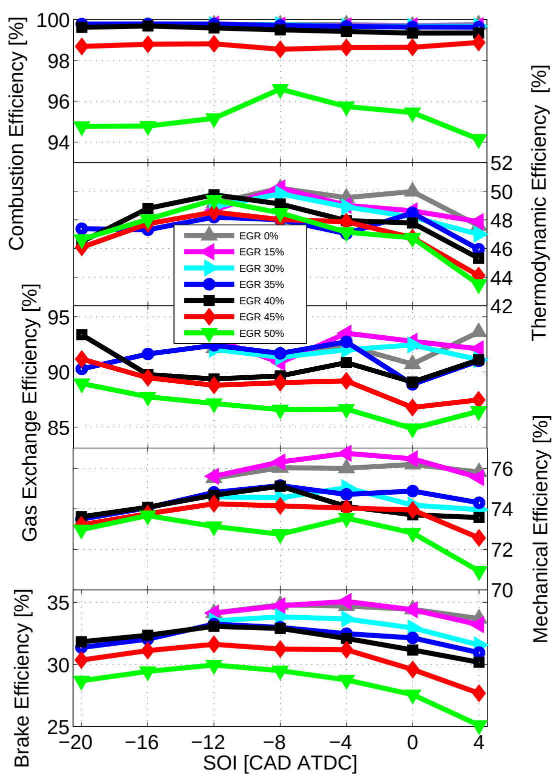

Figure 4.4: Efficiencies vs. SOI for different EGR rates.

The gas exchange efficiency deteriorates with increased EGR due to higher exhaust back pressure needed to raise the EGR level. It is also seen that the mechanical efficiency decreases which follows directly from the reduction in IMEP. The result is a decreased brake efficiency with increasing EGR, a behavior which is very distinct for EGR levels over 40 %.

In summary, the overall engine performance, described by the results presented above, suggests that high EGR, early injection PPC strategies are to prefer. This PPC region is at the low EGR side of the PM maximum where sufficient mixing and soot oxidation prevent high engine out smoke whereas NOx formation is suppressed by reduced temperature. Late injection timing also has the potential of simultaneous low NOx and PM operation with reduced efficiency. The mixing period has proven to be very closely connected to the combustion behavior where a positive mixing period results in a premixed combustion event.

4.2 PPC Control

Inspired by the results from the PPC characteristics investigation a control strategy that assures that the engine operates in early injection PPC mode was sought after.

This section describes the development of a cylinder individual, cycle to cycle based early injection PPC controller that uses the fuel injection system as actuator. It was desired to find a concept that controls PPC at different operating points under varying conditions. Since load is highly dependent on the amount of fuel injected, load control needs to be integrated in the concept. Additionally, the combustion timing should be kept within a certain region to prevent high pressure rise rates and maintain engine efficiency. Finally, there are constraints in the injection equipment that need to be considered. Given these requirements a suitable control method is Model Predictive Control (MPC) which has good MIMO properties and takes constraints into account explicitly [74]. MPC is a model based control method that uses online calculations to predict the state trajectories that start in the current state, and find a cost-minimizing control signal until time $t + Hp$ by solving a finite horizon optimal control problem over the prediction horizon, Hp. Only the first step of the control signal is implemented, then the plant state is sampled again and the calculations are repeated.

Concept Design

To make sure that the engine operates in PPC mode a measurable system output, closely related to the combustion characteristics was needed. In the previous section, Figure 4.1 shows that the mixing period provides such information. Additionally it is measured every cycle and suitable for feedback control.

The possible injection control signals are SOI, ID and injection pressure. Since high injection pressure generally yields lower smoke levels it was decided that constant, high pressure should be used for all injection timings. This means that SOI and ID are the remaining control signals. The outputs for the system are the crank angle of 50% heat released (CA50) to describe combustion phasing, IMEP to measure load and MP to control fuel/air premixing. These parameters are all calculated from the cylinder pressure trace. The model used for control design was derived through system identification [72]. The validation of the model can be found in Paper II.

Since an operating point is defined by engine speed and load, error free set-point tracking of IMEP is crucial. The engine shall run in PPC mode when possible so control of MP is also very important. MP and CA50 are highly dependent on both SOI and ID while IMEP depends almost solely on ID. It is possible to control MP or CA50 to arbitrary set-points given a specific ID but not simultaneously. Hence, a mixing period that assures PPC should be prioritized before set-point tracking of CA50. The resulting strategy was that CA50 was kept in a predefined region by constraints that allowed early and conventional combustion timings while IMEP and MP were controlled to set-point values. Regarding mixing period it was kept at zero which should give sufficient air/fuel premixing if maintained. The devised strategy ensures sufficient but not excessive MP to ensure premixed combustion without sacrificing efficiency.

Control Results

Set-Point Changes

Figure 4.5 shows the results for changes in load at 1500 rpm. The controller follows the set-point in IMEP while mixing period is slightly negative for the highest load at all cylinders except 4 and 5. The reason for this is found in the saturation of SOI and in the shorter injection duration needed at cylinder 4 and 5. When load is reduced the controller manages to keep MP at its set-point again. CA50 is close to the constraint and exceeds it slightly for cylinders 5 and 6 during the load change.

Figure 4.6 illustrates how the controller achieves PPC operation in different ways for three cylinders in cycle 50 from the experiments presented above. A comparison is made with combustion obtained with conventional timings at the same load. Studying the shape of the HRR it seems like cylinder 1 operates with less EGR than cylinder 4 which has longer combustion duration and later timing.

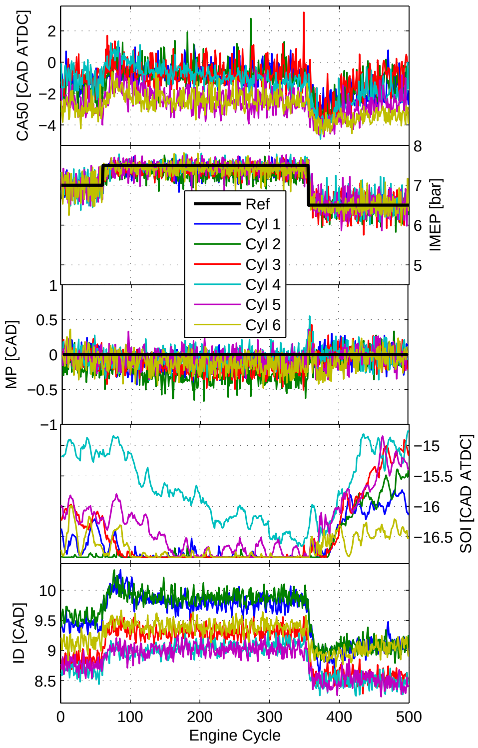

Figure 4.5: Step response for changes in load at 1500 rpm.

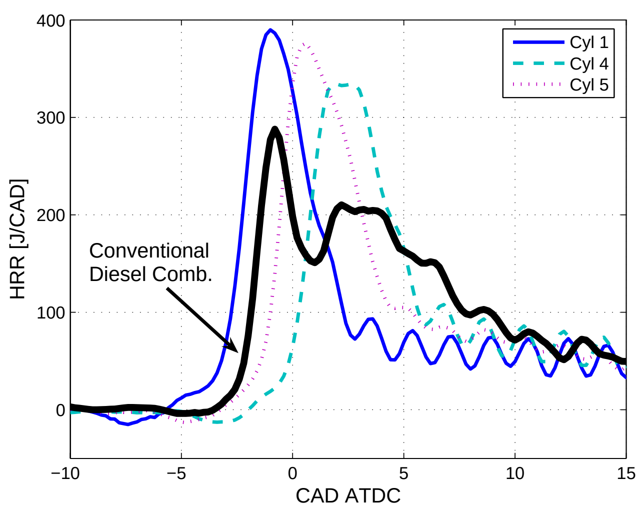

Figure 4.6: PPC for different cylinders versus conventional diesel combustion.

EGR Disturbance

In Figure 4.7 a rapid decrease in EGR level is applied to the controlled system. Since the EGR level can not be correctly measured during this change the EGR level is presented as a step change from the initial value to the final value. For this case the controller manages to keep both IMEP and MP close to their set-points. It is observed that the difference in CA50 between the 40% EGR case and the 30% EGR case is large both in terms of the mean value and the spread between the cylinders.

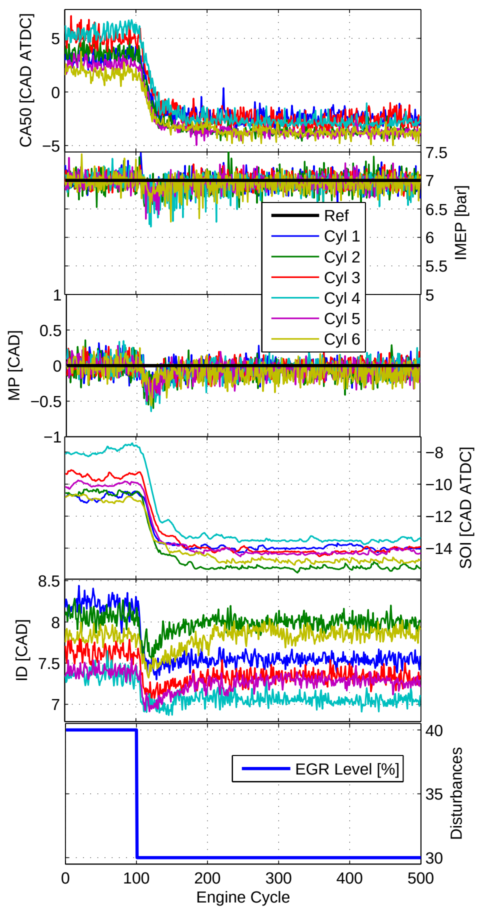

Figure 4.7: Response for a rapid change in EGR at 1200 rpm.

The HRR for cylinder 1 during the EGR change is presented in Figure 4.8. The change in combustion timing is very clear as well as the change in shape. The figure shows that the controller manages to make the engine operate in PPC mode during the decrease in EGR since all HRR show premixed combustion. In Paper II additional results on the response to changes in engine speed are shown.

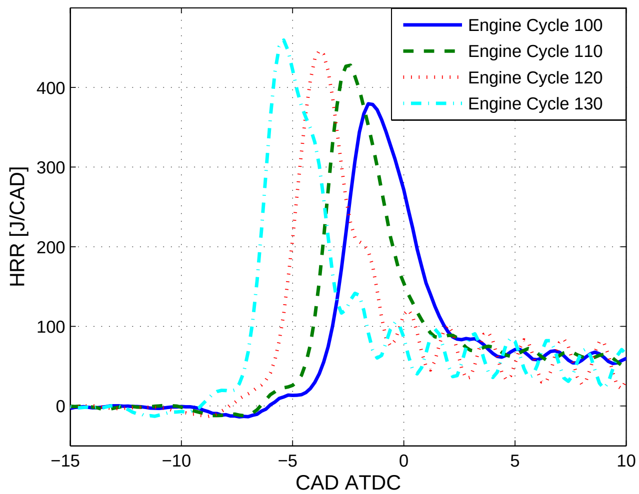

Figure 4.8: PPC development in one cylinder for a rapid change in EGR level.

The conclusion is that the controller manages to assure PPC operation until the physical PPC limit is reached. It handles both load changes and disturbances well by cylinder individual adjustments in injection timing. An issue regarding uneven distribution of EGR between the cylinders has also been addressed.

4.3 PPC Operating Region

After the characterization of PPC and the development of a PPC controller the next phase was to investigate the operating region of low NOx multi-cylinder diesel PPC. Experiments were carried out at low, medium and high speed where the load was increased until predefined limits in efficiency and emissions were reached. The PPC controller derived in the previous chapter was applied to assure PPC operation in order to investigate if the load range could be expanded even further using this strategy. The impact of reduced EGR-temperature was also studied.

PPC Criteria

Although the aim of PPC is to have a fully premixed combustion event the PPC boundaries for this study were based on emission and fuel consumption limitations. The objective was to determine the highest load that could be achieved for PPC in a standard build production engine fulfilling:

- $sNOx \leq 0.3\, g/kWh$
- $Smoke \leq 2\, FSN$
- $bsfc < bsfcBaseline \cdot 1.05$

The criteria is based on a concept of an engine, equipped with a particulate filter, operating with low NOx emissions and maintained efficiency.

Manually Controlled PPC

The first approach was to use manual control of fuel injection. Advanced injection timings were chosen since they generally yield higher brake efficiency than retarded timings based on conclusions from the PPC characteristics investigation. The injection pressure was kept at a constant high pressure while the injection duration was increased gradually and EGR and injection timing were tuned until the emission limits were reached. High EGR levels are needed which requires a fully open EGR throttle and an almost closed VGT setting.

Closed-Loop Controlled PPC

The second approach was to use the previously derived PPC controller. As mentioned above, it is sufficient that MP is positive to assure PPC but in order to suppress smoke the mixing period was kept around 1 CAD.

Reduced EGR Temperature

The third approach was to reduce the EGR temperature by introducing an additional heat exchanger between the engine and the EGR-cooler. The temperature of the coolant going into the EGR-cooler was decreased by 20 degrees which lowered the EGR-temperature compared to the nominal value.

Baseline for Comparison

A baseline for comparing fuel consumption was established in the test cell with control input data originating from the Volvo MD13 map.

Experiments

All experiments presented have a NOx level of 0.3 g/kWh and a smoke level of 2 FSN based on data from 2000 consecutive cycles. The different methods are named as follows:

- **MNom** = Manually controlled PPC, nominal coolant temperature
- **CLNom** = Closed-Loop controlled PPC, nominal coolant temperature
- **MCold** = Manually controlled PPC, reduced coolant temperature

In Figure 4.9, combustion timing, load and mixing period for the MNom case are shown. The combustion timing is early, compared to conventional diesel combustion, and the mixing period positive for all cylinders which is typical for advanced injection PPC. It is observed that Cylinder 3 has the latest timing and the highest load while Cylinder 6 has the earliest combustion timing and lower load. Cylinder 3 is quite stable in terms of load while it has larger variations in CA50. Cylinder 6 on the other hand shows variations in both combustion timing and load.

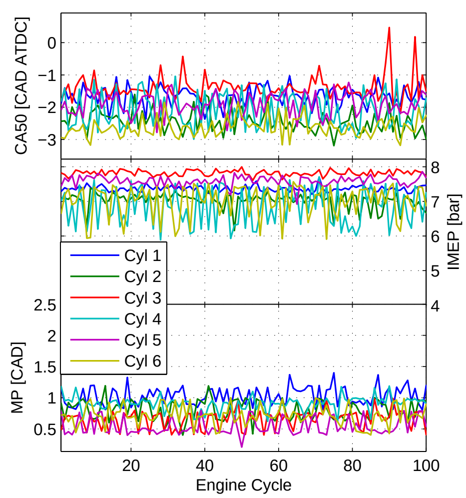

Figure 4.9: Combustion timing, load and mixing period for MNom.

Since premixed combustion is rapid, IMEP is sensitive to changes in combustion timing. However, the sensitivity decreases when CA50 approaches MBT. This is probably one of the reasons for the difference in behavior between Cylinder 3 and 6. The difference in timing might be due to an uneven distribution of EGR. With the short route system the EGR mixes with air very close to the inlet manifold which probably does not provide enough time for the gases to form a homogeneous mixture.

By lowering the EGR temperature the load could be increased. Figure 4.10 shows the results from the MCold case and it can be seen that the combustion timing is close to the MNom values. The combination of a reduced EGR level, counteracting lower inlet temperature and the increased injection duration, are probable reasons for this outcome. The spread in load between the cylinders is smaller than with MNom and the ignition delay is slightly short meaning that MP is reduced by more than the 0.4 degrees that is a direct result of longer injection duration.

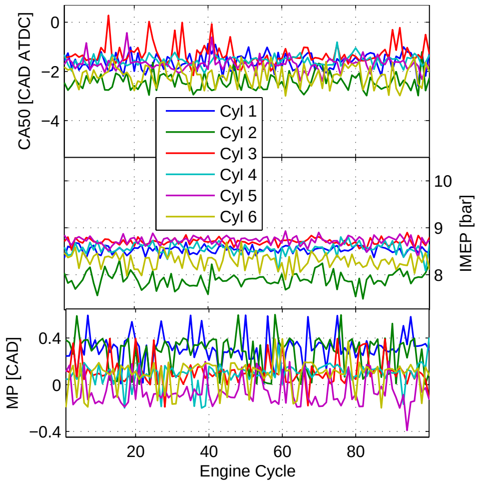

Figure 4.10: Combustion timing, load and mixing period for MCold.

Figure 4.3 shows results from the CLNom approach. It is observed that the difference in load between the cylinders is much less although there are some dips on cylinder 4, 5 and 6 due to the constraint violation on combustion timing (CA50 < -4). The average mixing period is longer than for MNom. The spread in combustion timing is somewhat higher than in the previous cases and the injection timing differs between the cylinders. This follows from the control design which only includes set-point control of IMEP and MP. The mixing period needed to be extended compared to MNom in order to fulfill the emission criteria. This yields earlier combustion timing for most cylinders which probably is the main reason why the fuel consumption increases for the closed loop controlled cases.

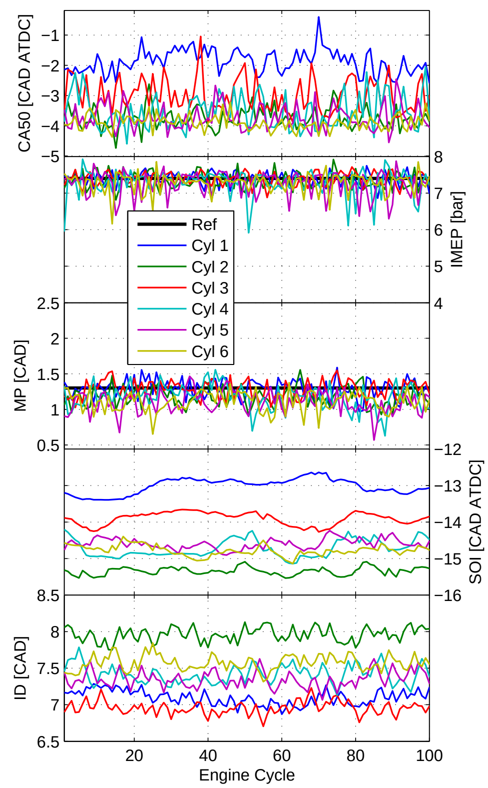

Figure 4.11: Combustion timing, load, mixing period and injection timing for CLNom.

When operating the engine at high EGR levels a small increase in EGR could result in a severe increase in soot formation while it lowers NOx significantly and vice versa. This means that Cylinder 1 could produce high smoke while Cylinder 6 produces high NOx simultaneously.

Table 4.1: Load, relative fuel consumption and EGR level from experiments at 1200rpm.

	MNom	CLNom	MCold
BMEP [bar]	5.8	5.8	7
Δbsfc [%]	-2.4	-1.1	-3.5
EGR Level [%]	41.5	42	40
T Inlet[° C]	61.8	65.5	56.6

Table 4.1 shows the maximum load achieved at 1200 rpm for the different methods together with the corresponding EGR level and the relative fuel consumption. Load is increased by 1.2 bar with a reduced coolant temperature. The fuel consumption is slightly lower compared to baseline in all cases where MCold shows the largest improvement.The reader is referred to Paper III for results from operation at higher engine speeds.

Operating Range

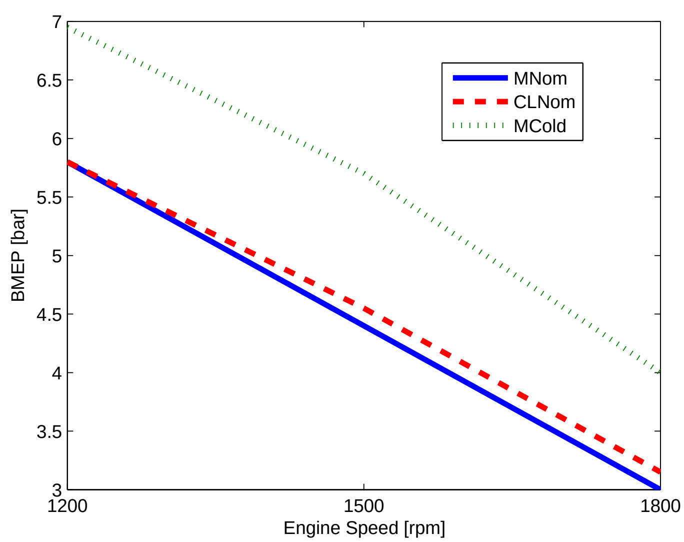

Figure 4.12: Operating range for MNom, CLNom and MCold.

Figure 4.12 shows the maximum load that can be achieved for the different methods at 1200, 1500 and 1800 rpm not exceeding the limits in emissions or fuel consumption. The difference when introducing closed-loop control is small while reduced coolant temperature yields a significant increase in load. This outcome goes for all engine speeds. It is noticeable that the decrease in load with increased engine speed is close to linear. The maximum load for PPC at 1200 rpm corresponds to approximately 30% of maximum load with conventional combustion while the maximum load for PPC at 1800 rpm corresponds only to 21% of maximum load with conventional combustion.

The factor which prevents higher load operation is primarily smoke. By advancing the combustion phasing smoke is reduced but this measure can only be taken to some extent since too early combustion phasing results in deteriorated efficiency. In addition, the EGR level needs to be increased for advanced combustion phasing which increase the stress on cooling and boosting. The suspected uneven distribution of EGR is also an issue which should be taken care of.

By balancing the cylinders to give similar simultaneous combustion characteristics it is thought that the cylinder individual differences regarding emissions will be reduced which would give room for further load increases without violating the criteria. Finally, means of extending the ignition delay without compromising combustion phasing should be investigated.

4.4 Improving Multi-Cylinder PPC

To improve the engine for PPC operation several measures were taken. These were based on conclusions from previous investigation carried out by the author but also on results from other PPC research projects. The most important changes to the MPPC concept are presented below.

Long route EGR

The engine was equipped with a long route, low pressure EGR system [26]. This system was installed since it has features that handle many of the issues addressed in the operating range investigation above. The key features of the new system are;

- Increased EGR cooling capacity → reduced inlet manifold temperature
- Increased EGR/air-mixing → improved EGR distribution
- Enables simultaneous high levels of EGR and λ

Cylinder balancing

As discussed in the previous section there was a suspicion that some cylinders operate at high NOx conditions while others operate at high smoke conditions simultaneously. The improved EGR system will solve part of this issue but as a complement, a cylinder balancing controller was introduced. The controller uses cycle to cycle based feedback control of CA50 and IMEP to assure equal combustion phasing and load for all six cylinders.

High Octane Fuels

Results from single cylinder experiments, [60] to [67], and numerical simulations [42] suggest that there is potential in using high octane number fuel to improve Multi-Cylinder PPC. The increased ignition delay achieved for fuels with high resistance to auto-ignition will promote fuel air premixing which suppresses smoke. This is the major benefit, although there is potential for additional advantages regarding fuel consumption et cetera since the fuel characteristics might enable new strategies for low NOx/smoke operation.

4.5 PPC Fuel Characterization

In the search for the most suitable Multi-Cylinder PPC fuel, a thorough fuel analysis was carried out. The objective was to map the connection between different fuels and the important factors of PPC combustion and its emission yield. The design of experiments aimed to capture and compare the sensitivity to combustion phasing, changes in load and speed, and changes in λ and EGR conditions. Furthermore, the baseline was concentrated around low NOx operation with Maximum Brake Torque (MBT) combustion phasing.

Table 4.2: Fuel specifications.

	Diesel Class 1	Gasoline 70	Gasoline 87
Cetane No.	51	N/A	N/A
Octane No.	N/A	70	87
LHV [MJ/kg]	42.9	43.8	43.5
H/C	1.85	1.98	1.92

Specifications of the fuels included in this study can be seen in Table 4.2. Diesel was used as baseline since the engine is designed for conventional diesel operation and there is a solid database with results from studies carried out with diesel to compare with. Gasoline 70, also known as naphtha [59], is chosen since its characteristics are close to diesel and therefore interesting for comparison. Gasoline 87 is chosen for the opposite reason being a standard fuel for SI engines with a long ignition delay as the most distinct and desired feature.

The comparison was carried out at three engine speeds; 1200 rpm, 1500 rpm and 1800 rpm . The engine loads tested range from low to medium load. At each operating point three combinations of EGR and λ-level were evaluated.

1. High EGR/High λ
2. High EGR/Reduced λ
3. Reduced EGR/High λ

The first condition was chosen to suppress both NOx and smoke levels giving the best performance from an emission perspective. The second and third conditions are to study the effect of a slight decrease in λ and EGR respectively. Further, for all conditions, four combustion timings were tested reaching from advanced to retarded combustion. To assure that all six cylinders operate with equal indicated load and combustion timing, they were controlled cylinder individually by the cylinder balancing controller.

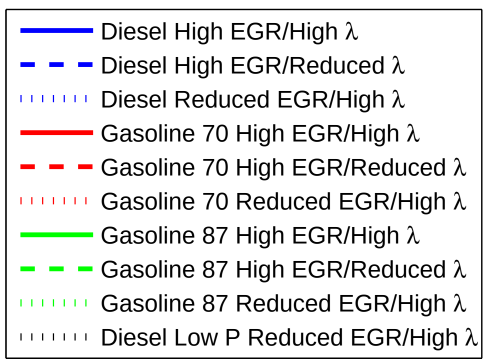

Figure 4.13: Legend for Figure 4.15 to 4.19.

The legend for Figure 4.15 to 4.19 is shown in Figure 4.13 while the λ and EGR conditions for each operating point can be found in Table 4.3.

Operating Conditions

Except EGR-ratio and λ-level there are other conditions that affect the combustion, the most important of these being inlet manifold pressure and temperature and injection pressure. Changes in EGR and λ-level will result in changes in inlet manifold pressure. Higher levels yield higher pressure. The largest difference is observed when changing EGR level while a change in λ, given the limited step size, has a more subtle effect on inlet pressure.

Table 4.3: EGR and λ at different operating points.

Load/Speed	1200 rpm	1500 rpm	1800 rpm
6 bar EGR [%]	55, 50	55, 50	55, 50
λ [-]	1.65, 1.6	1.7, 1.65	1.85, 1.8
8 bar EGR [%]	50, 45	50, 45	
λ [-]	1.7, 1.65	1.7, 1.65	
10 bar EGR [%]	55, 50	55, 50	
λ [-]	1.55, 1.5	1.6, 1.55	
12 bar EGR [%]	45, 40		
λ [-]	1.5, 1.45		

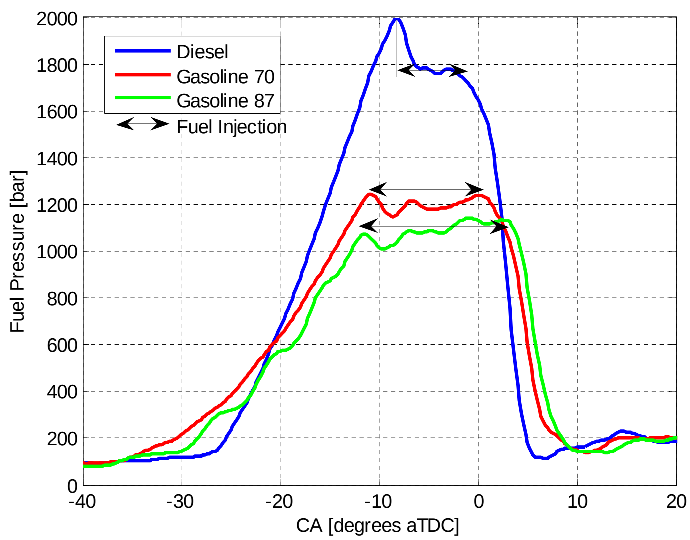

Figure 4.14: Fuel pressure at medium load with early combustion phasing.

The inlet temperature is controlled by an intercooler. The temperature changes follow the pressure changes quite closely. Higher pressure gives higher temperatures. The difference between highest and lowest temperature at a certain load is between 6 and 8 degrees. The difference at a certain EGR/ λ-combination is within 2, which is considered to have a very marginal effect on the outcome regarding ignition delay and flame lift-off [30].

The injection pressure at a medium load operating point with advanced combustion timing can be studied in Figure 4.14. For diesel, the injection system was tuned to give a pressure of 2000 bar at SOI, while for gasoline, the system is set to give highest possible injection pressure. The difference between diesel and gasoline, with an approximate pressure reduction of 40%, is due to the hydraulic behavior of the fuels. The unit injector has an internal leakage which affects gasoline more due to its lower viscosity.

For this thesis, additional measurements were carried out to study the effect of reducing the injection pressure of diesel to the level of gasoline. The complementary data was obtained under reduced EGR/High λ conditions and it is presented in the emissions and efficiency sections.

Combustion Characteristics

In previous investigations, mixing period has been used as an indicator of fuel air premixing. A positive MP gives a strong indication that the combustion event will be highly premixed. However, a negative MP does not contain information on whether the premixed part or the spray-driven part will be dominant since it has no connection to the injection duration. To solve this issue a new parameter is introduced, the Premix Ratio (PR) defined by 4.1

$$PR = \frac{Ignition\ Delay}{Injection\ Duration} \tag{4.1}$$

According to the definition a premix ratio above 1 means that fuel is injected before start of combustion while values below 1 show the proportion of the total fuel amount that has been injected before start of combustion. Hence, PR is an indicator of the amount of premixed combustion that takes place during the combustion event.

PR is based on the assumption of a constant injection pressure and an estimated ignition delay close to the actual delay between start of injection and start of combustion. Since injection duration is determined almost solely by fuel type and load the trends in PR follow the ignition delay. Figure 4.15 shows Premix Ratio and Heat Release Rates for CA50 at 3 and 9 CAD ATDC. It is observed that PR is only above 1 for diesel and Gasoline 70 at 6 bar while Gasoline 87 has a PR over 1 up to 10 bar IMEP for the low EGR case. It is also observed that Gasoline 70 has higher PR compared to diesel at 6 bar IMEP while the difference decrease to be practically equal at 12 bar.

When comparing the shape of the HRR to premix ratio it is clear that there is a strong connection between the amount of premixed combustion and PR. Operating points with PR above one show very low or no spray-driven combustion independent of fuel, load, combustion timing and conditions. The difference in HRR for different EGR/λ-combinations is most distinct for Gasoline 87 while the diesel HRR are almost identical especially at higher load.

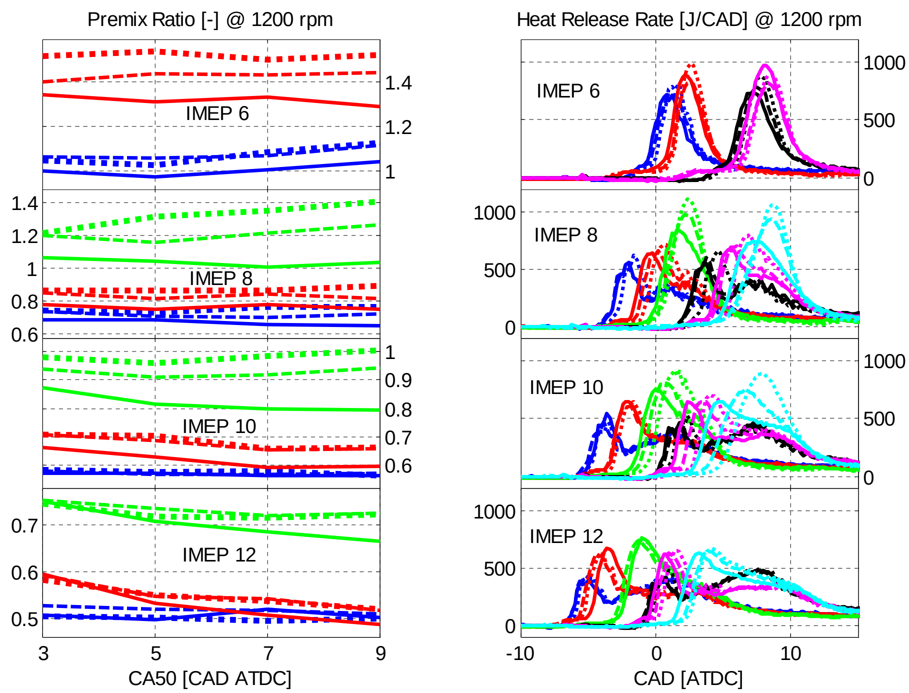

Figure 4.15: Premix Ratio and Heat Release Rate at 1200 rpm.

Emissions

Figure 4.16 shows the NOx emissions at 1200 rpm. The most distinct trend is that NOx decreases for later combustion phasing independent of fuel type and conditions. This is a well known behavior and follows the temperature reduction when combustion takes place further into the expansion stroke. It is observed that the NOx reduction with later combustion phasing is larger for high NOx conditions. However, a closer study reveals that the percentage drop is of similar magnitude for all conditions. Actually, the decrease is approximately 35 to 40 % at all conditions and loads when retarding the combustion phasing from 3 CAD ATDC to 9 CAD ATDC.

Regarding the three EGR/λ-cases it is clear that high EGR produces significantly less NOx. The connection between EGR and NOx is well proven and the major reason why EGR is used in modern CI-engines. Among the high EGR cases the difference is not large even though it was expected that the decreased intake oxygen concentration, for a reduced λ, would have a distinct effect. However, the increased premixed combustion for these conditions causes higher HRR and combustion temperatures that counteracts the expected outcome.

The difference between fuels is fairly subtle with Gasoline 70 yielding the lowest NOx values followed closely by diesel and Gasoline 87 showing slightly higher levels. The difference decreases at higher loads. A comparison of NOx to HRR suggests that there is a connection to maximum HRR which affects the cylinder temperature, although it can not be explained by that relation alone. Instead a combination of the maximum HRR and its timing are thought to give a more complete picture of the underlying mechanisms. Even though CA50 is equal, the location of the premixed HR varies. NOx formation is a function of both temperature and residence time meaning that if the temperature is increased earlier in the cycle NOx formation starts earlier.

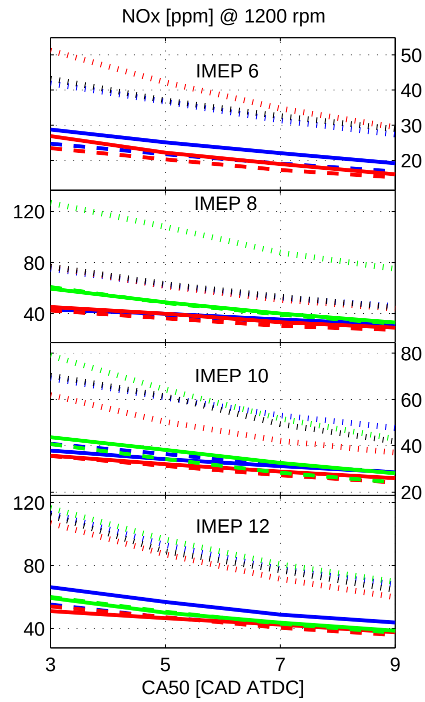

Figure 4.16: NOx emissions at 1200 rpm.

The NOx level obtained from the low injection pressure diesel experiments is practically identical to the high pressure injection diesel results. At the operating points tested and under these combustion conditions, injection pressure has no effect on the diesel NOx yield.

Figure 4.17 shows engine out smoke in FSN. Opposite to NOx, smoke increases for diesel at later combustion phasing. Since the premix ratio is practically independent of combustion phasing in the interval studied, the remaining differences is in combustion temperature which affects soot formation but also soot oxidation. A retarded combustion phasing gives less beneficial conditions for soot oxidation regarding both temperature and residence time and the hypothesis is that decreased soot oxidation is the major reason for increased smoke at later combustion timing. This deterioration in soot oxidation would apply for gasoline as well but it seems that the effect is small for Gasoline 70 while Gasoline 87 shows no dependency of combustion phasing at all. As oxidation is deteriorated a simultaneous decrease in soot formation of a similar magnitude could explain the trend. For Gasoline 87, part of this explanation can be found in slightly higher injection pressure for injection closer to TDC. Another theory is that the temperature increase, following the rapid combustion for early phasing, might affect the lift-off to a large extent. However, the underlying reasons need further investigation.

Smoke increases with increased load due to less fuel/air premixing and decreased air/fuel ratio which together provide larger zones with rich combustion where soot is formed. At low load Gasoline 70 has an advantage over diesel with very low smoke while at higher load this difference has disappeared. The Premix Ratio can be used as a tool to explain this behavior and it shows a higher ratio for Gasoline 70 compared to diesel at low load while at higher load the ratio is of the same magnitude. Gasoline 87 shows a similar trend but due to higher Premix Ratio the effect appears at higher loads.

Generally and independent of fuel, low λ gives highest engine out smoke. Less intake oxygen is the obvious explanation of this result where the fuel is burning richer and probably at lower temperature which affects soot oxidation. Among the higher λ cases, high EGR level gives more smoke. This is a combination of lower combustion temperature affecting soot oxidation and the lower intake oxygen concentration resulting for increased EGR at a constant λ.

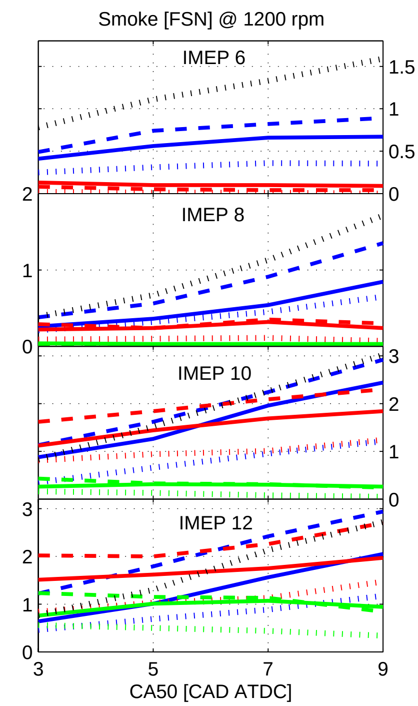

Figure 4.17: Engine out smoke at 1200 rpm.

Engine out smoke is drastically increased when reducing the injection pressure of diesel. The difference is most distinct at low load while it decreases as load increases. The dependency of combustion phasing is similar to the high pressure injection results. The most probable reason for the overall behavior is deteriorated fuel air mixing giving increased soot formation while soot oxidation might be fairly similar.

Unburned hydrocarbons and carbon monoxide emissions are presented in Figure 4.18. For combustion phasing a general trend of higher levels of HC for retarded combustion is observed. Although this trend is quite weak it should be connected to decreasing temperatures at later combustion phasing. The sensitivity is highest for high octane fuel at low load. The level decreases gradually with increased load due to increased in-cylinder temperatures. Diesel has the lowest emissions while both gasolines have similar levels except at late combustion phasing. This implies that the general dif-

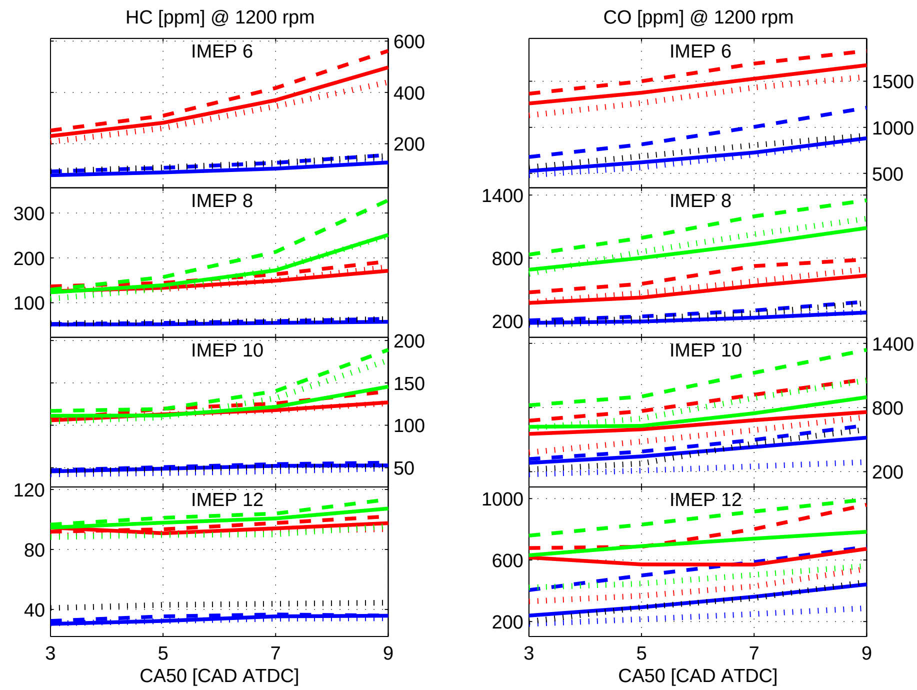

Figure 4.18: Unburned hydrocarbon and carbon monoxide emissions at 1200 rpm.

ference has to do with physical rather than chemical properties of the fuel. The low viscosity of gasoline is thought to increase injector dribble which is a known source of hydrocarbon emissions [32, 56]. Slightly higher levels of HC are observed for a reduced λ. This is probably due to the lower oxygen concentration. Reduced diesel injection pressure has no distinct effect on HC emissions except at 12 bar IMEP where a subtle increase, in the range of 10 ppm, is observed.

In these experiments one part of CO is thought to originate from over-lean regions near the injector after end of injection [30, 52]. As with HC, CO increases for later combustion phasing but linearly. Also here deteriorated oxidation due to lower temperature is the probable reason. Diesel CO emissions are less dependent on load compared to gasoline although decreased levels are observed for all fuels at higher load. The differences in CO level between fuels are more pronounced at low load where high octane number yields high CO for all combustion conditions. In contrast to HC, CO emissions has a distinct connection to octane number which is supported by research presented in [57]. A long ignition delay creates locally lean regions where combustion is not completed. Further, in [35] it is reported that reduced injection pressure increases CO which would affect the gasoline fuels negatively. However, when studying the low injection pressure diesel results it seems that the deterioration obtained for reduced injection pressure is only present at higher loads for retarded combustion phasing.

At high load, levels are determined by EGR/ λ-conditions to a greater extent. Finally, reduced EGR decreases CO and reduced λ increases CO compared to baseline. The resulting changes in temperature and equivalence ratio are thought to be the reason for this behavior [53].

Efficiency

The combustion efficiency trends will be opposite to the HC and CO trends. Decreased efficiency for retarded combustion phasing and increased efficiency for higher loads are observed and in accordance with the emission levels. That applies for the differences between fuel type and conditions as well. Due to moderate HC and CO levels in this study, the combustion efficiency is constantly high, between 99% and 100%.

Since combustion phasing is around the Maximum Brake Torque (MBT) point, the gross indicated efficiency is fairly stable. Except some small deviations, probably caused by minor measurement errors on the fuel scale, there is a tendency that MBT combustion phasing is found at around 3 CAD at low load to around 5 CAD at mid load. This change is probably connected to increased combustion duration at higher load. Ideally, combustion phasing and duration should optimize the trade off between heat losses and combustion early enough to utilize the released energy in the expansion stroke. The efficiency is also dependent on the maximum HRR which in turn is connected to heat losses. At lower loads the maximum HRR is higher giving lower efficiency. This is also the case when comparing fuels where Gasoline 87 stands out giving slightly lower efficiency due to increased heat losses. When comparing the different EGR/λ-conditions the same trend is observed with lower efficiency for the reduced EGR case with highest HRR.

The gas exchange efficiency is not affected by changes in combustion phasing in this region. Instead, the dominant factors are the required inlet pressure and the exhaust temperature. The low EGR case requires the lowest boost and it has also the highest exhaust temperature giving it an advantage compared to high EGR operation where a similar, but not as distinct advantage can be found for the reduced λ case. The most noticeable development for increased load is that the spread between lowest and highest efficiency decreases. This follows directly from the increase in IMEP but is probably also a result of similar combustion characteristics at high load.

The mechanical efficiency depends on friction in the engine, the engine auxiliary systems but also fuel injection equipment and the pressure requirements.

The mechanical losses are fairly constant at a given engine speed and hence, do not depend on combustion phasing either. The increase in mechanical efficiency with increased load is due to the friction losses becoming smaller in relation to engine out torque. The differences between fuels and conditions are harder to explain. The losses in the injector are higher for gasoline due to internal leakage caused by the low viscosity. This will affect the mechanical efficiency to some extent. Difference in oil temperature caused by heat losses to the cylinder wall is another possible reason for changes in friction which affects the efficiency.

When multiplying all the part-efficiencies, brake efficiency is obtained. Figure 4.19 shows the brake efficiency results for these experiments. Since the underlying mechanisms have already been discussed, the final comments on efficiency will be to summarize the outcome. The brake efficiency is highest around 3 to 7 CAD ATDC and somewhat lower at 9 CAD. There is a constant increase for higher loads for all part-efficiencies which translates into brake efficiency. Low injection pressure diesel at low EGR operation has the overall highest efficiency followed by Gasoline 70 and then Gasoline 87. The gain in efficiency for low pressure injection diesel, compared to high injection pressure diesel, is solely due to the improved mechanical efficiency. Brake efficiency would be further increased at high load operation.

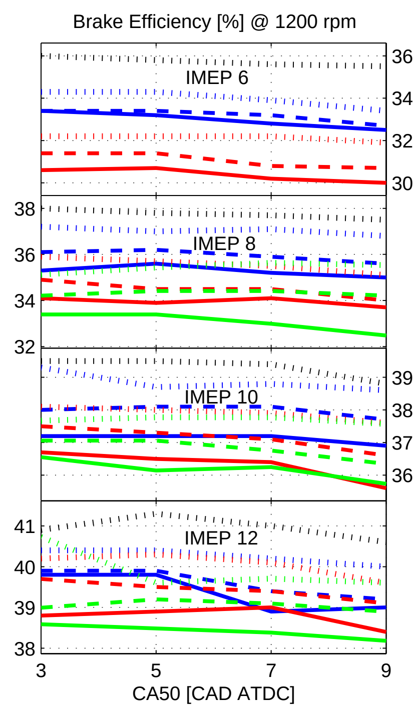

Figure 4.19: Brake Efficiency at 1200 rpm.

4.6 Extending the PPC Operating Region

NOx can be kept at a low level with sufficient EGR while smoke has a strong connection to fuel, EGR and λ. Gasoline 87 has a clear advantage both regarding maximum smoke level, but also regarding the relative independence from combustion phasing which enables low NOx phasing without sacrificing smoke. Based on this knowledge it was decided to continue the load increase with focus on Gasoline 87 since it was well below the emission limits.

Updated PPC Criteria

The PPC criteria to be fulfilled were slightly modified compared to the nominal ones presented in section 4.3. The NOx limit was increased from 0.3 g/kWh to 0.4 g/kWh which is the Euro VI emission legislation for specific NOx. Further, a combustion phasing interval was added to improve, rather than maintain fuel consumption compared to the nominal engine map. The resulting criteria are specified below:

- $sNOx \leq 0.4\, g/kWh$
- $Smoke \leq 2\, FSN$
- $Combustion\ phasing:\ 3 \leq CA50 \leq 9$

Figure 4.20 shows the maximum brake load that fulfills the PPC criteria for each fuel. It is observed that a 3 bar load increase is achieved for Gasoline 87 compared to diesel at 1200 rpm. At higher engine speed the difference decreases somewhat, still with an advantage for Gasoline 87.

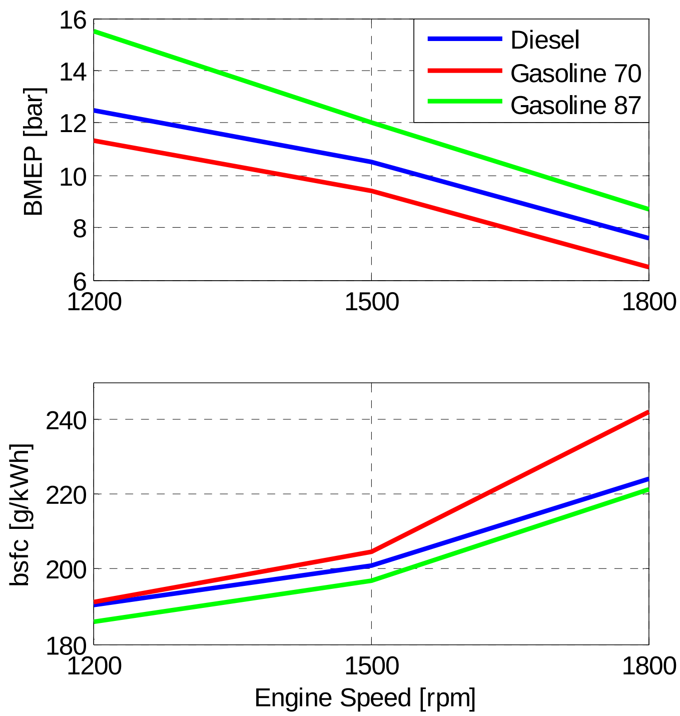

Figure 4.20: Maximum PPC load and corresponding fuel consumption for low, medium and high speed operation.

The operating point at 1200 rpm corresponds to approximately 70% of the engine rated load. Hence, it is low NOx operation with low fuel consumption (187 g/kWh) at a highly relevant heavy duty operating point.

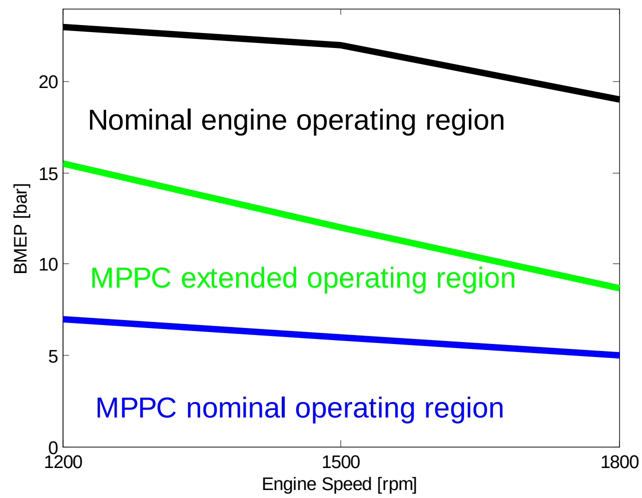

Figure 4.21: Extended MPPC operating region.

By comparing the results above to the nominal PPC operating region, achieved in Paper III, and the nominal engine operating region, it is clear that the PPC region has been significantly increased, see Figure 4.21. By combining the use of a low pressure EGR system and standard gasoline the operating region of Mutli-Cylinder PPC has been extended to cover 50% of the engine nominal operating region.

Idle operation

In the other end of the load range is idle operation. Idle operation for the MD13 is defined as zero engine out torque at 600 rpm. When using a higher octane fuel with increased auto-ignition resistance there is a concern that idle operation is hard to achieve. Initial experiments have been carried out to reach stable idle operation and, if necessary, identify what modifications need to be made to the engine to make idle operation feasible. In Figure 4.22 idle operation for Gasoline 70 is shown together with the lowest load achieved at 600 rpm for Gasoline 87.

With a warm engine Gasoline 70 reaches idle with stable combustion at an inlet manifold temperature of 25° C. Gasoline 87 is not able to operate at idle without an inlet air heater using the current setup. The heater at hand is a production heater which is already mounted on the engine in its standard

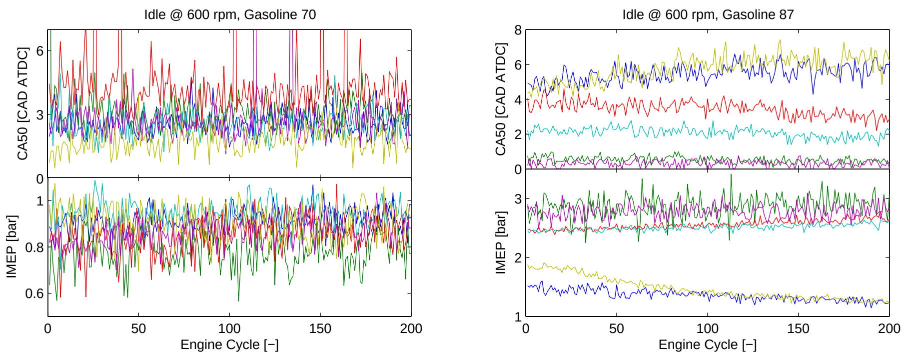

Figure 4.22: Idle operation @ 600 rpm for Gasoline 70 and Gasoline 87.

configuration. It is able to provide an inlet temperature of $70°C$ which is sufficient for Gasoline 87 to operate at 1.8 bar BMEP before misfiring. In order to reach lower load a more powerful heater would be needed. In addition, the exhaust back pressure was increased to 1.15 bar using the VGT in order increase the trapped hot residual gases. The hypothesis, based on these initial experiments, is that over-leaning is the most probable cause of the engine misfires for the high octane fuel.

4.7 Current PPC Concept

This section is a discussion which gives the author's view of the current PPC-concept. Possible explanations to the fundamental differences in combustion between PPC fuels are handled as well as the hardware limitations and the feasibility of putting a PPC engine into production. An attempt to combine knowledge from PPC investigations with conventional diesel theory is carried out. This is necessary since the similarities between PPC and conventional diesel combustion increase at higher load.

Combustion

The main objective when introducing gasoline fuels for PPC is to reduce smoke at low NOx (high EGR) operation. The extended ignition delay, achieved for high octane fuel, results in increased fuel/air mixing before combustion and yields lower soot formation. This is supported by numerous studies [60, 61, 62, 63, 40, 43, 44] and conceptually explained by the $\phi - T$ map in Figure 1.2. However, since there are further differences between the fuels than just resistance to auto-ignition, parallel theories for the smoke reduction exist. Effects of fuel volatility, fuel viscosity, average molecular length, fuel density and fuel oxygen content are also frequently investigated and discussed.

Fuel volatility might have an effect on smoke for very early staged fuel injection since it will reduce wall impingement [62]. However, for the injection timings used in the current PPC-concept, volatility does not affect emissions or the heat release rate and spatial location of the premixed burn according to [63, 55]. Fuel viscosity affects the spray momentum where a reduced viscosity gives increased jet velocity. The avarage fuel molecular length is used as an explaination for the low engine out smoke in [64, 65, 66, 67]. The study presented in Paper IV shows a large difference in engine out smoke between the gasoline fuels even though the avarage molecular length is similar, however, there might still be positive effects for a shorter molecular length, directly due to the chemical properties or due to reduced density. The lower fuel density of gasoline would affect the air entrainment rate in a positive way according to Equation 1.4, which helps reduce soot formation. Regarding fuel oxygen content, careful oxygenate selection is important to reduce smoke since there are additional characteristics that will affect the outcome and in some cases increase the soot yield [48] . The complete picture is probably a combination of several mechanisms concerning both physical and chemical properties of the fuels.

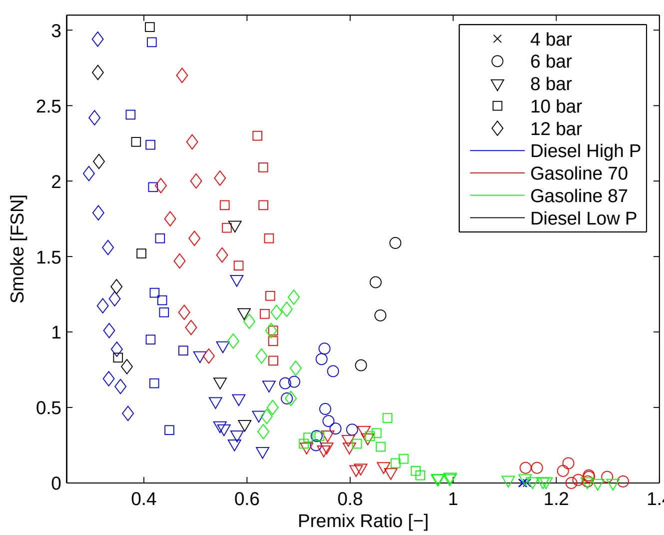

Figure 4.23: Smoke versus Premix Ratio at 1200 rpm.

Figure 4.23, based on data from the PPC fuel characterization, shows smoke as a function of premix ratio. Recall that a premix ratio above 1 means that the estimated start of combustion appears after the commanded end of injection. This indicates a total premixed combustion event while ratios below one indicate the amount of premixed combustion taking place in a more conventional diesel combustion event.

The author's general interpretation of the data is that soot formation is to a high extent governed by PR. Engine out smoke is very low for PR above one while a rapid increase regarding maximum engine out smoke is observed when spray-driven combustion is introduced. It is important to take into account that engine out smoke is the result of not only soot formation but also soot oxidation [31]. When using an engine without optical access it is hard to distinguish between formation and oxidation, however, the spread for a given PR is thought to be primarily caused by differences in soot oxidation while the maximum level is governed by PR.

The final observation regards the rate of increased engine out smoke for decreased PR. Although starting at a lower level, gasoline smoke increase slightly faster than high injection pressure diesel when spray-driven combustion is introduced. The lower injection pressure of gasoline compared to diesel is probably part of the explanation. Given a certain fuel, decreased injection pressure gives shorter lift-off according to [49]. ϕ at lift-off is, as discussed in the introduction, connected to soot formation during spray-driven combustion. Furthermore, the residence time for a fuel element in the soot forming region is controlled by the momentum of the fuel element and the size of the soot forming region [28, 31]. The high EGR level at PPC operation will increase the soot forming region [51, 33] while the momentum is controlled by injection pressure. Additionally, soot oxidation will be further deteriorated for a reduced injection pressure since the turbulence induced by the fuel jet decreases.

The conclusion is that the long ignition delay of gasoline enables low soot formation through a high degree of premixed combustion but the injection pressure is also important when increasing load since higher load operation will include spray-driven combustion even for gasoline. Soot oxidation is less fuel dependent but highly connected to temperature and equivalence ratio and hence controlled by turbulence, combustion phasing, EGR and global λ.

Hardware requirements

When taking the step from simulation and single cylinder experiments to a multi-cylinder engine with turbo and EGR-circuit there are additional challenges introduced. The cylinder to cylinder differences are, as reported above, in some cases substantial and demand cylinder individual feedback control to reach full potential. Additionally, the PPC concept requires high boost, high EGR and a moderate inlet manifold temperature. At the mid load experiments carried out for the PPC fuel characterization the turbo was working at 90 to 95% of maximum capacity to provide the engine the conditions required. In order to increase the operating region further a new turbo, designed for the increased mass flows, needs to be installed. A numerical study at the South West Research Institute has investigated the requirements for such a turbo charging system for PPC and other new combustion concepts [45]. The cooling capacity of EGR and air in the test cell is sufficient but there is a question whether cooling will become a problem in a vehicle due to lack of space to install larger heat exchangers.

Furthermore, to operate the engine on gasoline a system that can provide high injection pressure needs to be installed. This is not only to suppress smoke but also to enable full load operation since the injection duration saturates for gasoline at approximately 70% of full load due to the decreased fuel flow. Multiple injection strategies for multi-cylinder PPC should be investigated. Results indicate that both engine out smoke [58] and CO/HC emissions [29] can be reduced if a post injection is applied properly. By using pilot injection , the maximum pressure rise rate can be reduced and possibly also smoke, however, numerous studies suggest that single injection strategies give the lowest fuel consumption [67, 60, 38]. Finally, a more powerful inlet air heater is needed to assure idle operation for high octane fuel and the, even more challenging, engine cold start.

4.8 Efficiency Estimation

Engine efficiency is often controlled in an indirect way through combustion timing control. This requires a priori knowledge of where to phase the combustion for different operating points and conditions. With cylinder individual efficiency estimation, control strategies aiming directly at fuel consumption optimization can be developed.

There are several useful applications for cylinder individual, cycle resolved, efficiency estimation. The most obvious is to use it as a feedback variable in fuel consumption optimizing control strategies focusing primarily on steady state optimization but also on transients. The absolute value of the estimation is of importance if it is used in a control strategy together with emissions where, for example, the choice of controller weights depend on a trade-off between efficiency and emissions. Otherwise, if the control objective simply is to find maximum brake torque combustion timing it is only important that the maximum of estimated and actual efficiency coincides. In this work effort has been put into finding a method where both absolute values and trends can be used for controller development.

Method

Accurate estimation of the fuel energy going into the engine is the key when estimating efficiency. In this paper, fuel energy is estimated using the total heat release value, divided by combustion efficiency. The combustion efficiency is high and fairly constant for conventional diesel combustion [71] and therefore not considered an issue regarding estimation accuracy. The total heat release value may vary extensively for different heat release methods. In order to be accurate, the heat release calculation needs to take care of heat losses, blow by and other losses [34]. There are methods that include these effects, however, these methods need extensive tuning to be valid in the complete engine operating region. In order to avoid a large part of the necessary calibration, an alternative heat release method is used. This method is a self-tuning gross heat release computation which is described in detail in [16]. The method uses estimation of the polytropic exponent before and after combustion to implicitly include energy losses in the calculation. The estimation intervals need to be chosen properly.

Estimation Intervals

A correctly phased interval gives a straight tail of the heat release curve which implies that the heat losses are balanced. The late estimation interval gives a lower heat release and a slightly negative gradient towards the end of the calculation. This is due to an underestimation of the heat losses which decrease gradually in the expansion stroke. The result is an interpretation of the heat losses as a negative heat release. An early estimation interval yields the opposite behavior with overestimated heat losses which in turn results in increases in heat release well after the combustion has ended. It should be noted that the error in total heat release is much larger with early estimation compared to late estimation which leads to the conclusion that a later estimation is to prefer if there is an uncertainty of where to place the interval in order to maximize the accuracy.

In order to work for different injection timings and loads the intervals should be adaptive. A estimation interval before combustion can be controlled by the Start of Injection (SOI). An estimation that ends at SOI is sufficiently close to combustion but will never overlap with it. The placing of the estimation after combustion is a more complex matter. One option is to use feedback from the combustion event in the previous cycles. However, this would not be feasible for transient operation.

The apparent heat release rate presented in [71] is a fast heat release computation which gives a good indication of when combustion occurs. The calculated end of combustion from the apparent heat release in the current cycle is used to give input on when to start the post combustion gamma estimation.

The length of the estimation interval is also a parameter that needs attention. By extending the interval the statistics improve for the parameter estimation. However, the cost for this action is that the polytropic exponent varies more over the interval. Due to the robustness requirement a fairly long estimation interval was chosen.

TDC-Offset

The heat release computation is highly sensitive to changes in TDC-Offset. In a multi-cylinder engine the offsets might vary between the cylinders as well. In this investigation the TDC-offset was calculated using a method developed in [17] which determines the TDC offset based on motored cylinder pressure measurements.

Validation Method

The accuracy of the total engine efficiency can be validated by comparing the estimation with the value calculated using measurements from a fuel scale. The cylinder individual validation on the other hand is much harder since there is limited knowledge about how much fuel is injected in each cylinder. There is a slight difference in load between the cylinders but the combustion timings are close to each other. The total combustion efficiency is very high so it is reasonable to assume equal combustion efficiency for all cylinders. If the gas exchange process is excluded, the differences in efficiency between the cylinders should primarily be due to differences in heat losses. These differences may depend on cylinder position, EGR rate, load differences etcetera. It is assumed that the outer cylinders should have slightly lower efficiency than those in the middle.

Results

Table 4.4: Specifications of the four operating points tested in the experiments.

	A50conv	A50ret	A25conv	B50conv
Engine Speed	1200 rpm	1200 rpm	1200 rpm	1500 rpm
Load	50%	50%	25%	50%
CA50	Conventional	Retarded	Conventional	Conventional

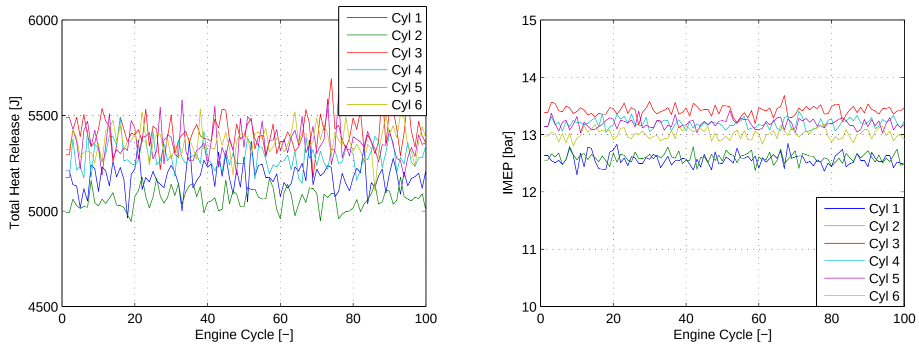

Figure 4.24: Total heat release and IMEP for A50Conv.

The method was applied to four operating points, specified in Table 4.4. Figure 4.24 shows the total heat release, calculated once for each cycle and cylinder following the method described above. As can be seen, the difference between the highest and the lowest values are about 500 J. In order to translate the total heat release to fuel energy it is divided by the combustion efficiency which is typically around 99 % for conventional diesel operation. The total fuel energy for all cylinders during 100 cycles can then be calculated as the sum of all total heat releases and then compared to the amount of diesel consumed according to the fuel scale. The cycle and cylinder resolved fuel energy is also converted to FuelMEP [75] in order to enable cylinder individual efficiency calculation on a cycle basis. FuelMEP is calculated according to (4.2).

$$FuelMEP = \frac{max(Q_{HR})}{V_D \cdot \eta_{comb}} \tag{4.2}$$

Figure 4.24 also shows IMEP for the same 100 cycles. As with the total heat release, there is a significant difference between the cylinders with Cylinder 1 and 2 showing lower values and Cylinder 3 the highest. By dividing IMEP with FuelMEP the indicated efficiency is obtained. The unfiltered indicated efficiency is displayed in Figure 4.25.

Table 4.5 shows a comparison between the estimated fuel energy (EE) and the measured fuel energy (EFS) for all four operating points. A50conv has a very low estimation error while A50ret shows that the energy consumption is underestimated at that operating point. The reason for this might be connected to the increased distance between the polytropic exponent estimation before combustion and the start of combustion. The fuel energy at A25conv is slightly overestimated while the opposite behavior is observed for B50conv.

The mean efficiency values (ME) generally show lower indicated efficiency at the outer cylinders which was expected. Regarding the standard deviation

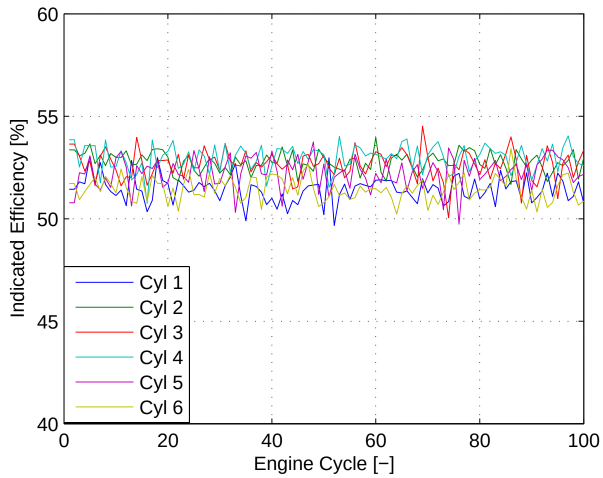

Figure 4.25: Indicated efficiency for A50conv.

Table 4.5: Energy consumption, cylinder individual mean efficiencies with standard deviation within parenthesis and total mean efficiencies.

	A50conv	A50ret	A25conv	B50conv
EE[MJ]	3.20	3.39	1.62	2.65
EFS [MJ]	3.18	3.55	1.56	2.75
Error [%]	0.46	-5.00	3.85	-3.64
1 ME [%]	51.4 (0.6)	47.0 (0.8)	50.6 (0.7)	50.2 (0.7)
2 ME [%]	52.8 (0.5)	49.6 (0.6)	52.4 (0.6)	52.8 (0.4)
3 ME [%]	52.7 (0.7)	49.0 (0.9)	52.3 (0.7)	52.5 (0.6)
4 ME [%]	53.0 (0.6)	48.4 (0.9)	51.4 (0.7)	52.9 (0.4)
5 ME [%]	52.2 (0.8)	48.0 (0.8)	51.3 (0.7)	53.0 (0.5)
6 ME [%]	51.5 (0.6)	46.2 (0.8)	51.0 (0.8)	52.9 (0.4)
TMEE [%]	52.3	48.0	51.5	52.4
TMEFS [%]	52.5	45.6	53.5	50.5

of the measurement it is lowest at Cylinder 2. It is also observed that the standard deviation decreases with increased engine speed for all cylinders except Cylinder 1. For retarded injection timings on the other hand, the standard deviation increases. These effects are probably connected to the location of the estimation interval. A50ret has the estimation interval after combustion approximately 5 CAD later than A50conv while B50conv has the estimation interval after combustion located 5 CAD earlier. The signal to noise ratio improves when the estimation interval moves towards TDC. An adaptive interval length could be applied to balance the different operating points.

The total mean efficiency for the estimated (TMEE) and for the measured energy consumption (TMEFS) shows that the total error in efficiency is between 0.2 and 2.5 units.

4.9 Steady-State Fuel Consumption Optimization

There are various control strategies that could utilize the cylinder individual efficiency estimation. As mentioned above, the most obvious would probably be to optimize fuel consumption during steady state. A practical application for such optimization could be to, with a certain interval, search for optimal injection settings for the engine of a truck driving at constant speed and load on a highway.

Efficiency has a non-linear behavior with respect to combustion phasing. The curve is a typical concave function with its maximum in a region fairly close to TDC. Even though the basic shape will be similar, the combustion phasing that maximizes efficiency will vary with engine speed, load and operating conditions and the control method should be able to handle this. Given these requirements extremum seeking control is a suitable method. The extremum seeking algorithm used for this study is presented in Paper VII.

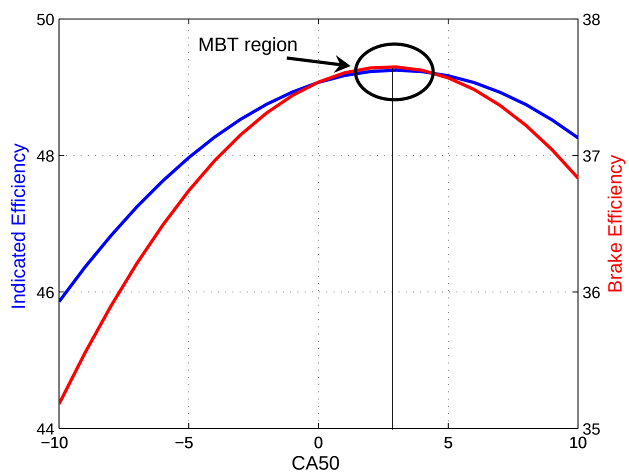

Figure 4.26: Maximum Brake Torque combustion region at 8 bar IMEP.

The results presented are from steady state operation at 1200 rpm with a load of 8 bar IMEP. Figure 4.26 shows the global indicated and brake efficiency curves at this operating point. It can be seen that the curves coincide well

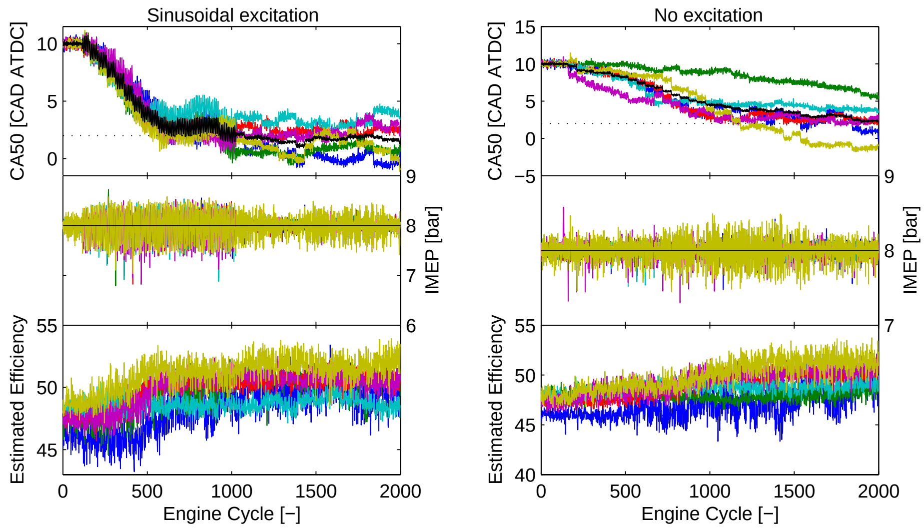

Figure 4.27: Optimizer performance when excited with a sinusoid and without external excitation.

at MBT and that the maximum is found for combustion phasings between 2 and 4 CAD ATDC at this particular operating point.

To show the performance of the efficiency optimizing controller the starting point for combustion was retarded towards late phasing at 10 CAD ATDC. After about 100 engine cycles with a constant set-point the optimizer was activated. The results can be studied in Figure 4.27. Since efficiency estimation and optimization is cylinder individual, the results could be treated as six parallel experiments. Starting at a combustion phasing of 10 CAD ATDC all six cylinders move towards the global MBT combustion phasing in a fairly linear and consistent manner. The MBT region is reached after about 500 engine cycles corresponding to 50 seconds at this particular engine speed. At cycle 1000 the sinusoid is removed from the set-point but the extremum seeking algorithm is still active without external excitation. As can be seen the combustion phasing remains in a region around the global MBT. The inner cylinders show a somewhat later timing compared to the outer cylinders. The main differences are the cylinder wall temperature which is lower for the outer cylinders. IMEP is fairly stable around 8 bar although the variance decreases when removing the sinusoid. The trends of the estimated efficiency are consistent comparing the cylinders while the individual values differ somewhat. However, the total fuel energy consumed during these 2000 engine cycles is 44.2 MJ and accumulated estimated fuel energy is 44.2 MJ which indicate that the values are very accurate.

Although diesel engines have a very stable combustion behavior compared to SI engines there are some cycle to cycle variations. It is of interest to study whether these natural variations can be utilized to give the extremum seeking algorithm sufficient information to perform the optimization without external excitation. Figure 4.27 also shows an experiment, similar to the previous, but this time extremum seeking is activated at cycle 100 without the sinusoid. As observed, four out of six cylinders reach the region around MBT within 1000 cycles just based on the cycle to cycle variations of the engine. The remaining two cylinders show a somewhat different behavior, however, cylinder 2 (green) is approaching global MBT and cylinder 6 (yellow) stabilizes after 1500 cycles.

Outlook

This work is focused on demonstrating the concept of online steady state fuel consumption optimization. Hence, no extensive tuning of the controller and excitation signal has been carried out. The forced excitation using a sinusoid gives an improved signal to noise ratio which gathers the cylinders and gives faster response. This comes at a cost of a slight increase in COV of IMEP which probably can be compensated for in the combustion feedback controller if the excitation signal can be treated as a known disturbance. Since efficiency is fairly constant in a region around its maximum further development to suppress estimation noise would be required for the optimization to be even more precise. Due to the high accuracy in absolute value of the estimation other new control concepts are possible to realize. Efficiency could for example be included in a complete engine control strategy including emissions as well. An interesting application would be to use it together with soot and NOx estimates derived from the cylinder pressure trace [41].

Chapter 5

Conclusions

This thesis presents the development of a multi-cylinder PPC concept. It reaches from the basic characterization of this new combustion concept to the demands on hardware, control and fuels for a realizable PPC solution. Further, a novel method of cylinder individual estimation and control of engine efficiency, applicable for both conventional diesel combustion and PPC, has been presented and discussed.

Multi-Cylinder PPC

In order to understand the foundation of PPC operation and to determine which of the various strategies that offers the best performance an investigation was carried out. The intention was to show where and how PPC occurs for different combinations of injection timing and EGR. In particular it was studied how the heat release was affected. The investigation focused on changes in mixing period, defined as the period between end of injection and start of combustion, but other parameters were also considered and together they form a map that shows the development from conventional diesel combustion to partially premixed combustion.

The results suggest that high EGR, early injection PPC strategies are to prefer among the different PPC concepts. This type of PPC is at the low EGR side of the PM maximum where sufficient mixing and soot oxidation prevent high engine out smoke whereas NOx formation is suppressed by reduced temperature. Late injection timing also has the potential of simultaneous low NOx and PM operation but with reduced efficiency which is not defendable neither from an economical nor environmental perspective.

A strong connection between mixing period and PPC has been ascertained. Based on this knowledge a combustion controller with feedback control of mixing period was derived. By keeping MP at zero the air/fuel premixing was sufficient to yield PPC. The controller proved to be able to assure PPC operation even under influence of disturbances such as rapid changes in engine speed and EGR level. The control method used in this work was Model Predictive Control (MPC). MPC was found to be a suitable control method since it has good Multiple Input Multiple Output (MIMO) properties and takes constraints into account explicitly. This makes it possible to keep CA50 in a region of permitted combustion timings, without controlling it to a set-point which is necessary in order to be able to control MP and IMEP freely.

The operating range of single injection diesel PPC was then investigated to evaluate whether this combustion concept is applicable for production. The study showed that the PPC load range is limited. The highest loads that fulfilled the emission criteria correspond to 30% of maximum load for conventional combustion at low speed and 21% of maximum load for conventional combustion at high speed. Additional cooling of the EGR had a significant effect on the performance. The load range was increased by approximately 1 bar BMEP with this method at all engine speeds tested.

The factor which prevents higher load operation is primarily smoke. By advancing the combustion phasing smoke is reduced but this measure can only be taken to some extent since too early combustion phasing results in deteriorated efficiency. In addition, the EGR level needs to be increased for advanced combustion phasing to suppress NOx. This increases the stress on cooling and boosting. The suspected uneven distribution of EGR is also an issue with the standard EGR system. It aggravates the optimization since one cylinder might have high NOx formation while another has high smoke formation simultaneously. The load may therefore be further increased if the inlet system could provide more extensive mixing of EGR and air.

In order to reach higher loads for the PPC concept the EGR system was rebuilt to a low pressure system with the EGR source down stream the turbine. This improves EGR/air-mixing and cooling and enables high EGR and boost pressure simultaneously. Further, gasoline fuels were introduced to extend the ignition delay, which would mitigate soot formation, and an extensive fuel comparison was carried out. In summary, independent of fuel, NOx can be kept at a low level with sufficient EGR while the difference between the fuels can be seen in smoke level where Gasoline 87 has a clear advantage both regarding actual smoke level, but also regarding the relative independence from combustion phasing which enables low NOx combustion phasing without sacrificing smoke. The long ignition delay of gasoline enables low soot formation through a high degree of premixed combustion but the injection pressure is also important when increasing load since higher load operation will include spray-driven combustion even for gasoline. Soot oxidation is less fuel dependent but highly connected to temperature and equivalence ratio and hence controlled by turbulence, combustion phasing, EGR and global λ.

With the new system the operating range was reevaluated. By combining the use of a low pressure EGR system and standard gasoline the operating region of MPPC has been extended to cover 50% of the engine nominal operating region.

Efficiency Estimation and Control

A method to estimate cylinder individual indicated efficiency has been developed and presented. The intention is to use the estimation as a feedback variable in combustion control strategies. The method only requires the cylinder pressure trace as input. The pressure trace is used to calculate the total heat released which then is scaled by combustion efficiency to give an estimate on the amount of fuel injected into the cylinder. This information together with the indicated load is sufficient to calculate a cylinder individual efficiency estimate.

A novel control concept, which uses the efficiency estimation as a feedback variable in an extremum seeking strategy for online fuel consumption optimization has also been presented and discussed. It has been shown that the proposed method is able to find the maximum brake torque region during steady state operation both with and without external excitation. The benefits of using external excitation are a faster response and an improved signal to noise ratio while the drawback is increased COV of IMEP induced by the excitation signal. With proper compensation in the combustion controller, it is believed that this behavior can be suppressed. Further, it is possible to reach the MBT region with a controller excited solely by the cycle to cycle variations of the engine. This strategy does not seem to affect COV of IMEP but has a slower response and shows an increased spread in combustion phasing between the cylinders caused by a deteriorated signal to noise ratio. Finally, the estimation error in accumulated fuel consumption from the experiments is around 1% which shows the potential of using the absolute value of the efficiency estimation in other control or diagnostic concepts.

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Popular Summary in Swedish

Partiellt förblandad förbränning i en flercylindrig dieselmotor

Kraven på bränslesnåla och rena motorer har ökat under de senaste åren. Det började med att priset på olja steg kraftigt i början av 2000-talet. Kraven växte sedan ytterligare i kölvattnet av debatten kring växthuseffekten där koldioxid, en av de dominerande produkterna vid förbränning, spelar huvudrollen. Med tanke på att sänkt bränsleförbrukning rör både ekonomiska och miljömässiga intressen så kommer forskning kring ämnet att fortsätta vara högintressant inom överskådlig framtid.

Bilindustrin har traditionellt varit dominerad av ottomotorn som i kombination med trevägskatalysatorn har ansetts vara ett miljövänligt motoralternativ med mycket låga utsläpp av giftiga förbränningsprodukter. Lastbilsindustrin å andra sidan har sedan länge dominerats av dieselmotorer av en andledning, den lägre bränsleförbrukningen. Den spelar naturligtvis stor roll inom transportindustrin där små sänkningar i bränsleförbrukning betyder stora besparingar över tiden. På grund av den lägre förbrukningen har nu även andelen personbilar utrustade med dieselmotorer ökat kraftigt, framförallt i Europa.

Det finns dock ett problem med dieselmotorn, utsläppen av kväveoxider och rök. På grund av att dieselmotorn kräver ett överskott av luft i cylindern vid förbränningen så kan en trevägskatalysator inte användas då den inte fungerar med luftöverskott. För att sänka dieselmotorns utsläpp används idag istället mer komplicerade efterbehandlingssystem som SCR för att sänka kväveoxider och partikelfilter för att minska röken. Dessa system är relativt dyra och det finns ett stort intresse för att kunna klara sig utan åtminstone ett av efterbehandlingssystemen utan att kompromissa med utsläppsnivåerna.

För att lösa detta problem gäller det att angripa själva kärnan, förbränningen i cylindern. Hela tiden studeras och testas framtida förbränningskoncept världen över på universitet, forskningsinstitut och inom industrin. Ett av de mest lovande och uppmärksammade förbränningskoncepten kallas partiellt förblandad förbränning eller på engelska: Partially Premixed Combustion (PPC). Det är forskning kring detta förbränningskoncept som den här avhandlingen fokuserar på.

PPC bygger på att man, genom att återcirkulera kylda avgaser (EGR) till motorn, kan sänka förbränningstemperaturen och på så vis också bildningen av kväveoxider. Röken undertrycks genom en förlängning av tändfördröjningen så att diesel och luft blandas innan förbränningen startar. Det är denna princip som gett förbränningskonceptet dess namn.

Den första delen av avhandlingen handlar om att kartlägga när och hur diesel-PPC uppkommer. Kopplingar mellan dieselinsprutningens tidpunkt i kombination med EGR-halten görs till tändfördröjning och förbränningens karaktär. Genom detta fastställs vilken typ av förbränning och förbränningsstyrning som är mest lämplig ur både verkningsgrads- och utsläppsperspektiv. Kunskapen användes för att utveckla en reglerstrategi som säkerställer PPC-förbränning när det är möjligt. Denna styrning åstadkoms genom återkoppling av en parameter kopplad till tiden mellan insprutning och förbränningsstart vilken i sin tur används för att automatiskt styra insprutningstidpunkten. Den framtagna regulatorn upprätthåller PPC-förbränning under både last och varvtalsändringar oberoende av yttre störningar.

Följande fas är en undersökning av det praktiska driftssområdet för PPC i en flercylindrig motor. Det visar sig att PPC bara fungerar på låga laster och att driftsområdet endast täcker 25% av motorns nominella driftsområde. Genom undersökningen kan ett antal begränsande faktorer identifieras. Fördelningen av EGR skiljer mellan cylindrarna vilket gör att en cylinder riskerar hög rökbildning samtidigt som en annan har hög kväveoxidbildning. Detta komplicerar optimeringen. Temperaturen i insugssystemet är betydligt högre än önskat på grund av underdimensionerad EGR-kylare. Ytterligare förlängning av tändfördröjningen är önskvärd då rökbildningen är den begränsande faktorn för PPC i denna studie.

För att lösa ovanstående problem görs följande; Motorns EGR-system modifieras för att förbättra EGR/luft-blandningen samt öka kylkapaciteten. Vidare introduceras nya bränslen med högre oktantal vilket förlänger tändfördröjningen. För att konstatera vilken typ av bränsle som är mest lämpat för PPC görs en omfattande undersökning mellan diesel, låg- och högoktanig bensin där verkningsgrad, utsläpp och förbränningsstabilitet jämförs. Resultatet är att den ökade tändfördröjningen med högoktanig bensin minskar rökutsläppen drastiskt utan att kompromissa med kväveoxidutsläppen. Denna minskning kommer dock på bekostnad av försämrad förbränningsstabilitet på lägre laster. Baserat på resultatet i denna undersökning görs sedan en ny utvärdering av PPCs driftsområde där den maximala lasten nu ökas till 70% av dieselmotorns maxlast på lägre varvtal. Sammantaget så täcker PPC-konceptet nu 50% av motorns nominella driftsområde.

Den avslutande delen av avhandlingen behandlar en metod att skatta verkningsgraden i motorn cylinderindividuellt för att kunna använda reglerstrategier som optimerar förbränningen och sänker bränsleförbrukningen under drift. Metoden, som baseras på tryckmätningar från cylindrarna, är den första i sitt slag och den öppnar upp för många nya typer av motorreglering både gällande optimering under stationär drift och under varierande last och varvtal. Metoden är användbar för både konventionell dieselförbränning och PPC.

Summary of Papers

Paper I

Investigation of the Combustion Characteristics with Focus on Partially Premixed Combustion in a Heavy Duty Engine

By Magnus Lewander, Kent Ekholm, Bengt Johansson, Per Tunestål, Nathan Keeler, Nebojsa Milovanovic, Tony Harcombe, and Pär Bergstrand.

SAE 2008-01-1658, presented by the author at the 2008 SAE Powertrains, Fuels and Lubricants Congress in Shanghai, China, 2008.

The objective of this study was to investigate the changes in combustion characteristics with injection timing and EGR. The investigation focused on changes in emissions, efficiency and mixing period, in this work defined as the difference between the end of injection and the crank angle of 10% heat released. Together they created a map that shows the development from conventional diesel combustion to partially premixed combustion.

The experiments and initial data analysis were carried out by the author and K. Ekholm. The final data analysis was made by the author who also wrote the paper.

Paper II

Closed Loop Control of a Partially Premixed Combustion Engine using Model Predictive Control Strategies

By Magnus Lewander, Bengt Johansson, Per Tunestål, Nathan Keeler, Nebojsa Milovanovic, and Pär Bergstrand.

AVEC'08 Proceeding 006, presented by the author at the 9th International Symposium on Advanced Vehicle Control, Kobe, Japan, 2008.

This work focused on how to assure Partially Premixed Combustion in a multi cylinder engine by applying cylinder individual, cycle to cycle based control with the fuel injection system as actuator. The objective was to control PPC at different engine speeds, EGR levels, and loads to make the combustion system versatile. Model Predictive Control was used since it was found the most suitable control method.

The author wrote the paper and was also responsible for control design, experiments and data analysis.

Paper III

Evaluation of the Operating Range of Partially Premixed Combustion in a Multi Cylinder Heavy Duty Engine with Extensive EGR

By Magnus Lewander, Bengt Johansson, Per Tunestål, Nathan Keeler, Simon Tullis, Nebojsa Milovanovic, and Pär Bergstrand.

SAE 2009-01-1127, presented by the author at the SAE World Congress, Detroit, Michigan, 2009.

This paper investigated the operating region of early injection low NOx diesel PPC in a six cylinder heavy duty engine with a short route EGR system. The experiments were carried out on different engine speeds where the load was increased until predefined limits in efficiency and emissions were reached. Closed loop combustion control was applied to balance the cylinders in order to investigate if the PPC load range could be improved. Finally, the EGR temperature was reduced to see if it was possible to expand the operating region even further.

The author carried out the experiments, did the data analysis, and wrote the paper.

Paper IV

Investigation and Comparison of Multi Cylinder Partially Premixed Combustion Characteristics for Diesel and Gasoline Fuels

By Magnus Lewander, Bengt Johansson, and Per Tunestål.

SAE 2011-01-9341, accepted for publication at SAE Powertrains, Fuels and Lubricants, Kyoto, Japan, 2011.

This paper investigates the differences in PPC characteristics for three fuels; Diesel Swedish Mk 1, Low Octane Gasoline (70 Octane) and US Standard Gasoline (87 Octane). Engine operating conditions, combustion characteristics, emissions and efficiency are in focus. The experiments were carried out at a range of operating points on a Volvo MD13 which is a six cylinder heavy duty engine. At each operating point three combinations of EGR level and λ-value were evaluated. Further, for these three conditions, four combustion timings were tested reaching from advanced combustion timing at 3 CAD ATDC to retarded combustion timing at 9 CAD ATDC. The indicated load and the combustion timing were controlled cylinder individually by a feedback controller.

The method for the fuel comparison was developed by the authors, within a reference group also containing Volvo Powertrain and Chevron. The experiments were carried out by the author who also did the data analysis and wrote the paper.

Paper V

Extending the Operating Region of Multi-Cylinder Partially Premixed Combustion using High Octane Number Fuel

By Magnus Lewander, Bengt Johansson, and Per Tunestål.

SAE 2011-01-1394, presented by the author at the SAE World Congress, Detroit, Michigan, 2011.

This paper investigates the operating region of single injection PPC for three different fuels; Diesel, low octane gasoline with similar characteristics as diesel and higher octane standard gasoline. Limits in emissions are defined and the highest load that fulfills these requirements is determined. The investigation shows the benefits of using high octane number fuel for Multi-Cylinder PPC. With high octane fuel the ignition delay is made longer and the operating region of single injection PPC can be extended significantly. Experiments are carried out on a multi-cylinder heavy-duty engine at low, medium and high speed.

The method to determine the load range was developed by the authors, within a reference group also containing Volvo Powertrain and Chevron. The experiments were carried out by the author who also wrote the paper.

Paper VI

Cylinder Individual Efficiency Estimation for Online Fuel Consumption Optimization

By Magnus Lewander, Bengt Johansson, and Per Tunestål.

ASME ICEF2010-35113, presented by the author at the ASME 2010 Internal Combustion Engine Division Fall Technical Conference ICEF2010, San Antonio, Texas, 2010.

This paper presents a method to estimate indicated efficiency using the cylinder pressure trace as input. The proposed method is based on a heat release calculation that takes heat losses into account implicitly using an estimated, CAD resolved polytropic exponent. There are several useful applications for the cylinder individual, cycle resolved, efficiency estimation. The most obvious is to use it as a feedback variable in fuel consumption optimizing control strategies focusing primarily on steady state optimization but also on transients.

The design and development of the method to estimate efficiency was carried out by the author. The efficiency estimation is based on a heat release calculation developed by Per Tunestål who also acted as a disussion partner during the development of the method. The experiments were carried out by the author who also wrote the paper.

Paper VII

Steady State Fuel Consumption Optimization through Feedback Control of Estimated Cylinder Individual Efficiency

By Magnus Lewander, Anders Widd, Bengt Johansson, and Per Tunestål.

To be submitted

This paper presents a method to use estimated efficiency as a feedback variable in an extremum seeking control strategy for online steady state fuel consumption optimization. The results show that the controller manages to find the maximum brake torque region at the given operating point both with and without external excitation.

The design of this control concept was made by the author. Anders Widd helped with the implementation of the extremum seeking algorithm. The experiments were carried out by the author together with Anders Widd. The author wrote the paper except the description of the extremum seeking algorithm.