# Parametric Study to understand Ethanol Partially Premixed Combustion

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**Abstract**

As global concern increases regarding pollution does the demand to develop more efficient and environmental friendlier propulsion system come in high focus. The main systems for transportation have been dominated by the Otto and Diesel engine for a long time, but both are trade-off in local and global emissions, i.e. $NO_X$, soot, HC and CO versus $CO_2$. This has driven the evolution to concepts as homogeneous charge compression ignition, HCCI and later partially premixed combustion, PPC. Although HCCI has been shown to have ultra-low engine-out emissions in soot and $NO_X$, has it been proven to be unreliable in terms of combustion control and lack in efficiency at higher loads. PPC on the other hand is proven to be a good contester in both fields. This takes us to where we are now, where the PPC analyzing and testing phase is ongoing. PPC has been run with diesel as fuel, but with only a 25% of the conventional diesel combustion range, due to self-ignition [6]. This problem has been solved by running high octane fuels with high resistance to ignite. These have shown to reduce emissions in soot and NOx as a result of longer air and fuel mixing period and high EGR values for a low combustion temperature. Ethanol has been tested with good results in efficiency, combustion control and emissions. The growing market for renewable fuel in Sweden and globally [1] indicates that this fuel is worth our effort for further investigation.

This thesis attempts to characterize PPC with ethanol as fuel. It has been shown, based, and confirmed, that earlier testing by Manente [3] with double injection, EGR=50% and $\lambda$=1.5 are suitable for PPC operation. New results shows that an injection strategy with a pilot injection around -35 CAD ATDC, -5 CAD ATDC for second injection and an injection ratio around 52% give good efficiency, low emissions and low acoustic noise at 12 bar $IMEP_g$. In this operation does the second injection provide with control over the combustion duration. Earlier it has been stated that start of combustion (SOC) was triggered by second injection, but results show no connection. Trends show that a combination of, inlet temperature, lambda, and start of first injection are key factors, as they all coincide in creating a fuel and air mixture which is igniting at its certain temperature. One can divide the combustion sequence in two parts. The pilot injection which provides with a sufficient fuel and air mixture which kinetically ignites due to the high temperature just before TDC, causing a high rate of heat release. Followed by the second injection, which is partially kinetically ignited and partially stratified that burns in a diffusion flame.

Concerning combustion sensitivity to parameter variation, inlet temperature has shown to be an important factor, as variation of $\pm$ 5-10 $^o$C clearly affects combustion in all aspects. With too high temperature, SOC and phasing are advanced resulting in loud acoustic noise and high combustion temperature, with too low inlet temperature SOC and phasing are retarded which also leads to high noise, as well as higher emissions with decreasing combustion efficiency. Small variations of rail pressure, EGR and lambda does not affect combustion at the same proportion, but even so they are as important to be combined properly to gain a sufficient ignition delay and combustion temperature.

## Acknowledgement

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# Nomenclature

| | |
|---|---|
| ASI | -After Start of Injection |
| ATDC | -After Top Dead Center |
| BDC | -Bottom Dead Center |
| CAD | -Crank Angle Degree |
| CO | -Carbon Monoxide |
| $CO_2$ | -Carbon Dioxide |
| EGR | -Exhaust Gas Recirculation |
| FSN | -Filtered Smoke Number |
| ID | -Ignition Delay |
| IMEP | -Indicated Mean Effective Pressure |
| $IMEP_g$ | -Gross Indicated Mean Effective Pressure |
| $NO_x$ | -Nitrogen Oxides |
| PM | -Particulate Matter |
| PPC | -Partially Premixed Combustion |
| SI | -Spark Ignition |
| SOC | -Start of Combustion |
| SOI | -Start of Injection |
| TDC | -Top Dead Center |
| UHC | -Unburned HydroCarbons |

# Contents

# 1 Introduction

To understand Ethanol Partially Premixed Combustion, one must first have knowledge about the origin and the reason to why there is need for a new combustion concepts.

## 1.1 Diesel

### 1.1.1 Basic principles

Diesel engines for automotive applications works with the four stroke principle, this mean that one cycle occurs during two revolutions of the crank. (Mechanically is the principle more or less the same for SI engines.) The process is easiest explained in four steps:

**Intake stroke (A):** As the name indicates this is the stroke where as the inlet valve opens and air is aspirated in to the cylinder by the downwards movement of the piston. On a diesel engine the air is not limited by a throttle so there is less pumping losses. Normally is the diesel engine equipped with turbo to supercharge the inlet air into the cylinder to get a higher fill rate.

**Compression stroke (B):** The inlet valve closes as the piston reaches BDC and starts its upward stroke. This stroke compresses the air hence the temperature rises.

**Power stroke (C):** As the piston reaches TDC, fuel is injected with high pressure (500-3000 bar[10]). The high pressure and temperature ignites the fuel shortly after injection and the power stroke begins. The diesel engine doesn't use a premixed fuel and air mixture, it controls the load with the amount of injected fuel and the combustion continues as long as fuel is injected.

**Exhaust stroke (D):** With all fuel combusted there will be exhaust gases to be evacuated. When the piston reaches BDC the exhaust valve opens and gases are released and pushed out by the remaining high pressure from the combustion and the upwards moving piston.

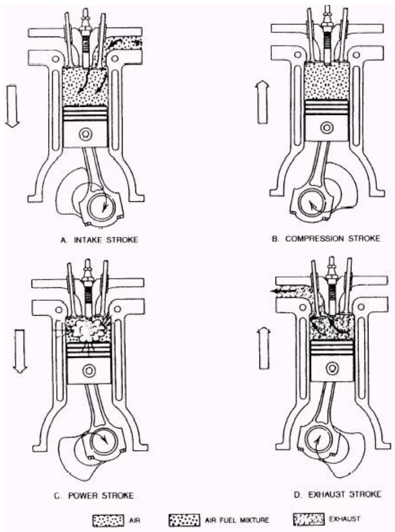



Figure 1: Four stroke Diesel principles

### 1.1.2 Characteristics

The Diesel engine has a higher efficiency than the Otto engine, which is a result of various factors; higher compression, lower pumping losses and the engine is running with lean combustion. With higher compression do we get a higher in cylinder pressure, a consequence is that the engine have to be more robust hence heavier and more expensive. Lower pumping losses and the lean mixture are correlated and are possible since combustion in a Diesel engine is controlled by the injection of fuel. To get a proper mixture in this short amount of time, the excess air is necessary for the injected fuel to be mixed and combusted properly. Therefore, there is no need to throttling the inlet air in Diesel engines. The lean mixture and slow burning diesel fuel leads to lower torque than a corresponding Otto engine. The combustion process in a diesel engine can be divided into four stages:

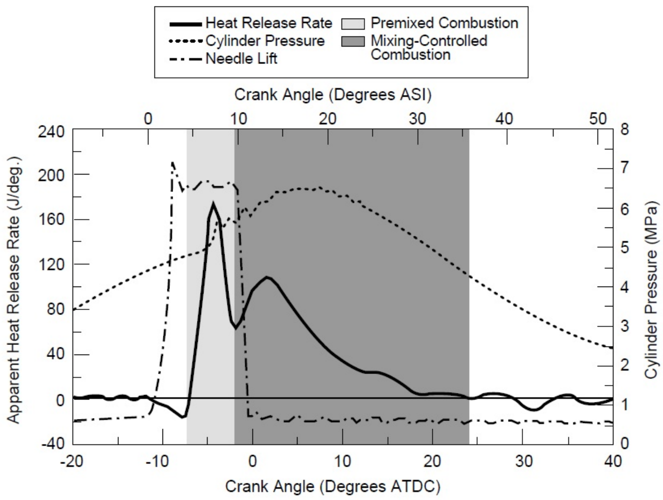



Figure 2: Typical diesel Combustion model "Dec-model" [8]

**Ignition Delay (ID)**
The ID is defined as the crank angle degrees (CAD) between start of injection (SOI), to the start of combustion (SOC). This is seen in figure 2 as a gap between the start of injection and start of the heat release. The small negative heat release is due to that the energy needed to evaporate the fuel, and is taken from the in-cylinder walls. To get a proper fuel and air mixture during the ID the fuel has to go through a mixing process. As the fuel is injected the fuel droplets are atomized, heated and vaporized. The vaporized fuel mixes with air to a combustible fuel and air mixture. How long the ID becomes depends on fuel properties, how fast the fuel disperse with the air, the in-cylinder temperature at SOI and injection duration[5]. A too short ID can result in a violent combustion.

**Cetane number**
Diesel fuels major property is given with a cetane number, the higher the number describes the ease of self igniting. This number is set by comparing with reference fuels in a standardized test engine[5].

**Premixed Combustion**
As the injected fuel and air reaches a combustible mixture and the temperature is high enough, it ignites. This occurs under approximately 5-9 degrees after the start of injection [5], and is seen as the first sharp peak in heat release in figure 2. The self-igniting mixture results in a high energy release which in its turn ignites the remaining fuel mixture. Too much fuel ignited initially can lead to acoustic noise, due to the high rate of heat release and pressure rise rate. This is not preferable and to avoid this are methods used to either mix the fuel

faster by modifying the combustion chamber hence a shorter ID and to distribute the fuel injection over a longer period of time.

**Mixing-Controlled Combustion**

After the first peak in heat release, starts the mixing-controlled combustion or also called spray-driven combustion, which is a quasi-stationary process. The combustion slows down compared to the premixed combustion and burns continuously in a diffusion flame as the fuel is injected, mixed and vaporized. Figure 3 shows a schematic model of the injection and combustion process.

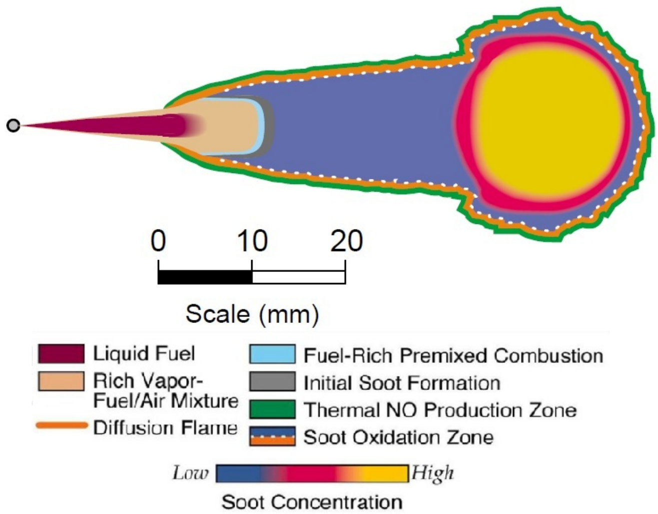



Figure 3: Model of spray-driven combustion [8].

**Late Combustion**

After premixed combustions ended there may be fractions of unburned fuel left in products from the main combustion that can still react and provide with a late heat release [6].

### 1.1.3 Emissions

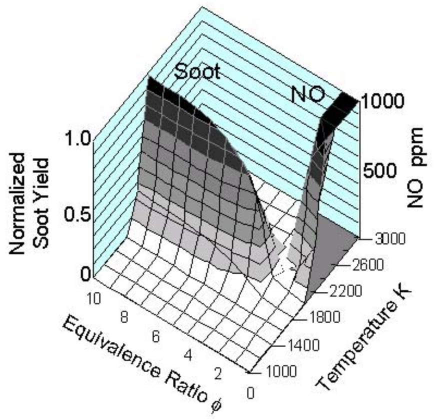
The main emissions that is a problem from the diesel engine are $NO_X$ and PM. This problem is directly connected to the combustion. As seen in figure 4 there is always a compromise in diesel engines, if you lower the combustion temperature you decrease $NO_X$ but soot increases, and vice versa. Other emissions are hydrocarbon(HC) and carbon monoxide(CO), but they are in very small fractions in a diesel process due to the high temperature and lean combustion.

**$NO_X$ - Nitrogen oxide**

Formation of NO and $NO_2$, which is commonly summed under $NO_X$, occurs in high temperature and where there is access of oxygen and nitrogen. Most of the $NO_X$ is formed in front of the diffusion flame where the temperature is high, see figure 3. Highest temperatures are reached when the fuel mixture is rich, but the highest $NO_X$ is produced when the mixture is somewhat lean due to the excess of oxygen, which is called thermal $NO_X$. Nitrogen oxide can also be formed as Prompt $NO_X$, which is a advance chemical reaction and most present in the flame front of rich mixtures. Also Fuel $NO_X$ contributes as fuel bound nitrogen which reacts with oxygen and forms $NO_X$ [9]. The $NO_X$-formation zone can be seen in figure 4.

**PM - soot**

Particulate matter contains most carbon or as its normally called soot. It is formed in fuel rich zones as seen in figure 3. Later in the cycle, if the combustion temperature reaches high enough and oxygen is present, a large part of the formed soot particles oxidized [6]. Another



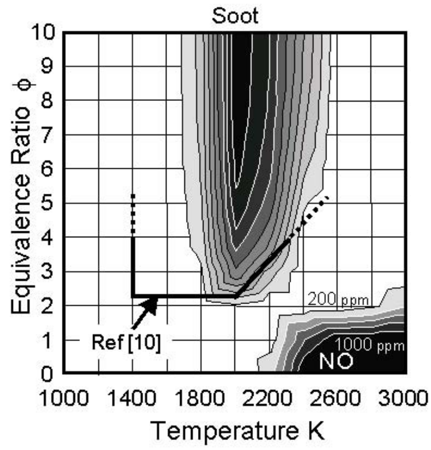




Figure 4: Soot and $NO_x$ formation zones as function of temperature and equivalence ratio. [7]

contributor to the amount of produced soot is the choice of fuel, depending on chemical composition and cetane number(CN). With a higher CN will the premixed phase be shorter and therefore will soot formation also be smaller [5].

**Possible solutions**

As mentioned $NO_X$ and PM are highly linked to combustion temperature, high temperature generates $NO_X$ but oxidise produced soot and low temperature decrease $NO_X$ but increase soot formation. So there is always a trade-off. One solution to decrease $NO_X$ and soot formation is to lower combustion temperature below formation zones, but it will lead to decrease in efficiency and an increase in formation of CO and unburned HC [6]. Low Temperature Combustion (LTC) experiments have been done and by using EGR (Exhaust Gas Recirculation) and early injection, resulting in lower combustion temperature which lowers $NO_X$ formation respective a homogeneous premixing which results in low soot. A concept that evolved from LTC is Homogeneous Charge Compression Ignition (HCCI). The procedure is to inject the fuel during the inlet stroke, which leads to a homogeneous charge. The fuel mixture is then ignited by the temperature caused by compression. To get desired ID and SOC, high levels of EGR is used to lower the temperature, the cost is higher values in CO and HC (which can be after-treated by oxidation in the exhaust system) and the fuel consumption increased. There is shown problems in control of the combustion because of the long ID, e.g. there is low connection between start of injection and start of combustion. With these facts was Partially Premixed Combustion (PPC) developed, which concept is similar to strategies stated above. The difference is that injection is closer to TDC which leads to a partially homogeneous fuel mixture at the SOC. It can simply be described as in [2] as: "If standard diesel combustion is black and HCCI is white, PPC is some shade of gray".

## 1.2 Partially Premixed Combustion (PPC)

The concept of Partially Premixed Combustion has its roots in both Diesel and HCCI. The diesel engine was invented by Rudolf Diesel in 1890's and has until the 21th century been constantly improved. It has gone from mechanical control too electric ECU for higher accuracy and improvement on fuel consumption as well as lower emissions. Even though has increasing legislations and demands on emissions and fuel consumption pushed for new combustion concepts, in late 20th century was the HCCI concept developed, as described in 1.1.3. As this candidate been shown to be unreliable for further experiment did the focus turn to PPC. The general concept is as the name indicates, Partially Premixed Combustion, and to get the partially premixed or in other words non homogeneous fuel and air mixture is fuel injected during the compression stroke. As in HCCI is EGR used as an inert gas to cool down the combustion and control ID, as the mixture self ignites under the temperature created by compression. With late injection the fuel does not have time to completely mix with the air, and creates a mix of lean and homogeneous zones which leads to a more controllable combustion. PPC has been shown to reduce $NO_X$ and soot levels without increasing fuel consumption [6].

### 1.2.1 Characteristics

How a PPC engine behave is highly dependent on SOI, EGR level and fuel composition. These parameters controls premixing phase and in its turn start of combustion. To locate the optimal combination or in some cases compromises to get low values of local emissions, PM and $NO_X$, is it needed to understand what and how big influence each parameter have.

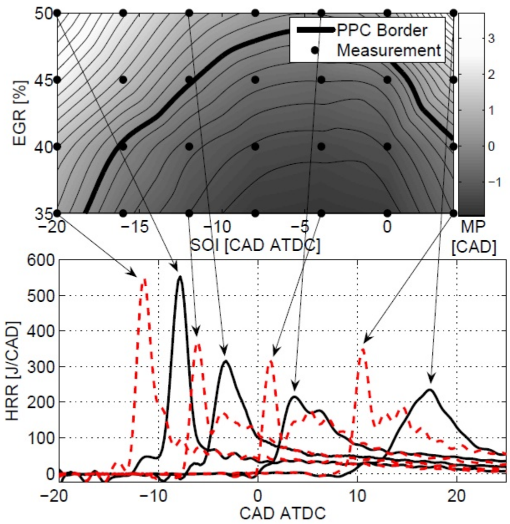
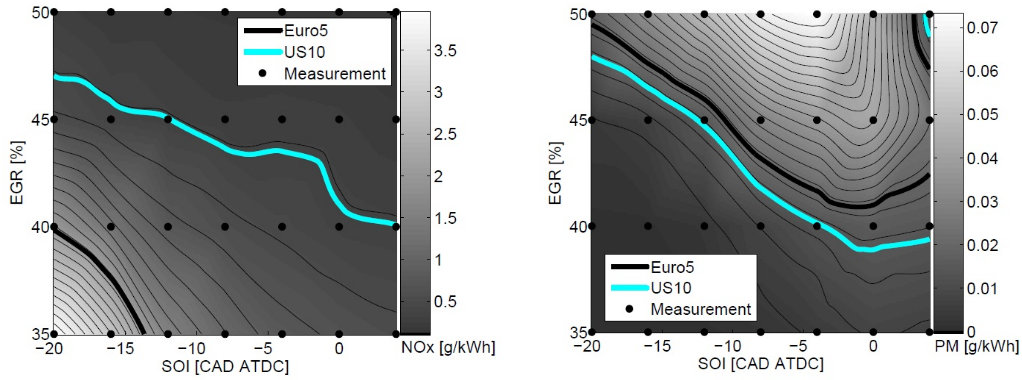



Figure 5: Mixing period and corresponding Heat release rate [6].

**SOI and EGR**

Start of injection and exhaust gas recirculation determines the rate of premixing of fuel and air. In figure 5 from [6] have experiments been done with running PPC on diesel, this gives a general idea of relation between SOI, EGR and the Mixing Period (MP). The PPC border describes the area of where it's able to run PPC, in other words it is the area outside the border where premixing is possible before ignition. By setting SOI early provides that you get a more mixing, but at the same time you get a more volatile mix. To prevent unwanted ignition is the EGR level increased to cool down the mix, see figure 5. As seen in figure 6 is it desirable to run with with early SOI, -20 to -15 CAD ATDC and EGR at 35-40%, which gives low emissions and the earlier stated trade-off(see section 1.1.3) can be avoided. This area also provides with good combustion efficiency and somewhat worse but still comparable values in brake efficiency. Although the good results is the operating range that fulfills the requirements very small. This requires a more suitable approach and as diesel, which is originally designed for immediate combustion, can the conclusion be drawn that it may not be the optimal fuel for PPC.




(a) $NO_X$ (b) PM

Figure 6: Mixing period and corresponding local emission rate and curves indicating Euro V and US10 emission standards [6].

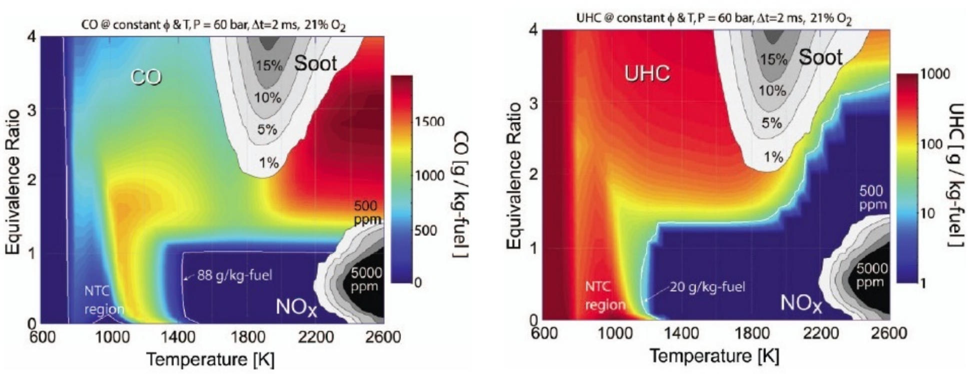



Figure 7: CO and UHC as function of temperature and fuel equivalence ratio [6].

**Fuels**

Since emission have been shown to have dependency of SOI and ID fuel is desired that fulfils the requirements to achieve a longer premixing period. In [3] are comparisons done between Diesel, Gasoline and Ethanol, and by the results it is seen that local emissions of soot is significantly lower for Gasoline and Ethanol. This is because of the resistant of self ignition in high octane fuel. The higher octane number, the higher is the resistance for self-ignition. With the longer premixing does the fuel have time to mix sufficiently which decrease soot formation, and with EGR as inert gas is the combustion temperature kept low, which reduces $NO_X$. Unfortunately does the longer ID and lowered temperature produce higher values of HC and CO than with diesel. HC and CO can both originate from lean and rich mixtures and are highly dependent of temperature, lower temperature will lead to reduced oxidization of these emissions [6]. These emissions can be oxidized in a after treatment system [6]. It is preferred to have EOI before SOC to minimize soot production, this is possible with high levels of EGR when running with diesel, the same thing can be archived with gasoline and lower EGR levels. On the downside has the separation with gasoline been shown to increase fuel consumption as high pressure oscillations is created which enhance heat transfer and in its turn higher fuel consumption [3]. By compromising and let SOC begin before EOI is the fuel consumption lowered for a cost of higher level of soot. It should be mentioned that these levels are still lower compared to diesel. Another problem with high octane fuels is that it is not possible to run on low load as the fuel won't ignite, this can be solved with a inlet heater.

Table 1: General data on fuel properties

| **Fuel** | Octane No.(RON) | Cetane No. | LHV [MJ/kg] |
|---|---|---|---|
| Diesel | - | 44-55 | 43.8 |
| Gasoline | 91-99 | - | 44 |
| Ethanol | 107 | - | 26.9 |

## 1.3 Ethanol Partially Premixed Combustion (EPPC)

Earlier experiments have shown that Ethanol is proven to be a promising fuel to use in PPC operation mode. With its high ON number and with high EGR it is able to run with a longer ignition delay providing with better mixing, which has resulted in lower readings in soot and NOx compared with Diesel PPC. The reason to this features is for starters, the good homogeneity in the mixture as well as Ethanol molecular structure that is only containing a two carbon chain, as well as bound oxygen. A complete combustion of Ethanol results in $CO_2$ and water. Ethanol has also shown to be burning violent, leaving pressure oscillations which resulted in higher heat transfer and as a consequence higher fuel consumption. This was solved with a second injection causing a partially stratified injection which in its turn lowered combustion temperature as well as oscillations. This proved to increase thermodynamic efficiency and lower fuel consumption, as heat is transformed into mechanical work instead of leaking trough the in-cylinder walls. As earlier stated and on the downside does EPPC produce high values of HC and CO due to wetting respectively over mixing between fuel and air in the combustion chamber. Based on these observations found by Manente [3], [2], this thesis have been produced with its main purpose to analyse and learn more about EPPC characteristics.

# 2 Diagnostic method

To analyze the combustion is the pressure trace and volume used. With these parameter can you calculate the energy released during the cycle, or more commonly referred to as Heat Release. This parameter gives an understanding of the combustion event and its different zones, as earlier been described in section 1.1.2. It should be noted that the heat release is based on simplifications and assumptions and is not to be viewed as a exact value. This means that e.g. variations in volume due to material stress [4] is neglected as well as assumptions been made for a constant cylinder wall temperature etc. Even so does this calculation give us a fair picture of the combustion event. Below is the heat release model for this thesis described .

## 2.1 Heat Release

All heat release calculations origin from the first law in thermodynamics as seen in equation 1.

$$\frac{\partial Q}{\partial t} = \frac{\partial U}{\partial t} + \frac{\partial W}{\partial t} + \sum_i m_i h_i \tag{1}$$

| | |
|---|---|
| $Q$ | -heat release |
| $U$ | -internal energy |
| $W$ | -work |
| $m_i$ | -mass for one element, $i$ |
| $h_i$ | -enthalpy for one element, $i$ |

By assuming that the cylinder contains an ideal gas ($pV = mRT$) and neglecting heat losses due to blow by (i.e assuming that the piston rings and valves are completely sealed) can we rewrite equation 1 into equation 2. The heat transfer parameter $\frac{\partial Q_{HT}}{\partial \theta}$ is added to equation 1 as well. The data is collected with the CAD as reference, and therefore will the derivative of collected data be with respect to CAD.

$$\frac{\partial Q}{\partial \theta} = \frac{\gamma}{\gamma - 1} p \frac{\partial V}{\partial \theta} + \frac{1}{\gamma - 1} V \frac{\partial p}{\partial \theta} + \frac{\partial Q_{HT}}{\partial \theta} \tag{2}$$

| | |
|---|---|
| $Q$ | -heat release [J] |
| $\gamma$ | -ratio of specific heats [-] |
| $p$ | -pressure [Pa] |
| $V$ | -volume [m$^3$] |
| $Q_{ht}$ | -wall heat transfer [J] |

The specific heats $\gamma = C_p/C_v$ varies throughout the whole combustion with changes in gas composition and temperature [4]. There is no exact method to calculate $\gamma$, hence is approximations made to predict the parameter. In this thesis is equation 3 used, which is published in [11]. The constants and reference value has to be tuned for each specific engine.

$$\gamma = \gamma_0 - k_1 exp(-k_2/T) \tag{3}$$

| | |
|---|---|
| $\gamma_0$ | -reference value [-] |
| $k_1$ | -constant [-] |
| $k_2$ | -constant [-] |
| $T$ | - temperature [K/CAD] |

The last part of the heat release $\frac{\partial Q_{HT}}{\partial \theta}$ is the heat transferred by wall convection and is stated below in equation 4. As the temperature varies with the flame propagation in the cylinder is a non-dimensional model used to simplify the calculations. This means that the cylinder wall temperature $T_{wall}$ is assumed to be homogeneous in the whole in-cylinder area and constant during the combustion cycle [5]. The gas is assumed to be ideal is the temperature $T_{gas}$ is calculated with respect to pressure and volume per CAD.

$$\frac{\partial Q_{HT}}{\partial t} = h_c Area_{wall}(T_{gas} - T_{wall}) \tag{4}$$

| | |
|---|---|
| $Q_{HT}$ | -heat transfer [J] |
| $h_c$ | -heat transfer coefficient [W/K m$^2$] |
| $Area_{wall}$ | -area of cylinder wall [m$^3$] |
| $T_{gas}$ | -temperature of the gas [K] |
| $T_{wall}$ | - temperature of the cylinder wall [K] |

The heat transfer coefficient $h_c$ is calculated with the Woschni correlation (equation 5), presented in [12].

$$h_c = 3.26B^{-0.2}p^{0.8}T_{gas}^{-0.55}w^{0.8} \tag{5}$$

| | |
|---|---|
| $B$ | -cylinder bore [m] |
| $p$ | -cylinder pressure [kPa] |
| $T_{gas}$ | -temperature of the gas [K] |
| $w$ | -average gas velocity[m/s] |

Whereas the average gas velocity $w$ is calculated with the equation 6, presented in [12]. The $C_1$ and $C_2$ is constants that are tuned for the specific engine used in this thesis.

$$w = C_1 S_p + C_2 \frac{V_d T_{ivc}}{p_{ivc} V_{ivc}}(p - p_m) \tag{6}$$

| | |
|---|---|
| $C_1$ and $C_2$ | -adjustable constant for specific engine [-] |
| $S_p$ | -mean piston speed [m/s] |
| $V_d$ | -displacement volume [m$^3$] |
| $T_{ivc}$ | -temperature when inlet valve closing [K] |
| $T_{evo}$ | -temperature when exhaust valve opens [K] |
| $V_{ivc}$ | -volume at inlet valve closing [m$^3$] |
| $p_{ivc}$ | -pressure at inlet valve closing [Pa] |
| $p$ | -cylinder pressure [Pa] |
| $p_m$ | -motored cylinder pressure [Pa] |

### 2.1.1 Heat Release Analysis

The Heat Release from equation 2 is accumulated over the cycle to give the total amount of released heat from the combustion, as seen in figure 8. With this information is it possible to determine CA50, this is a very important parameter in combustion control as it describes the phasing of the combustion event. The definition is where 50% of the released heat takes place in CAD. CA10 and CA90, where 10% respectively 90% are released in CAD, are also important as they describe where the combustion begins and ends. One would think that 1% and 99% of heat released would give a more accurate value but due to noise in readings and variations in combustion does CA10 and CA90 give a more stable value. These values gives us ignition delay ($SOI - CA10$) and combustion duration ($CA90 - CA10$), which was earlier described in section 1.1.2.

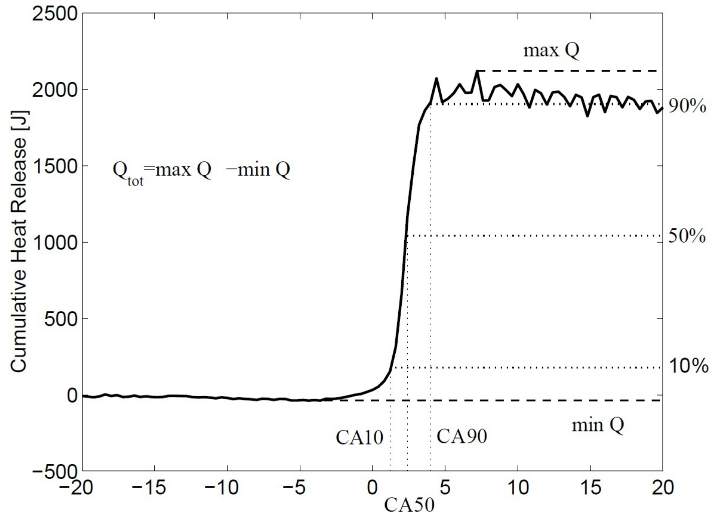



Figure 8: Example of Accumulated heat released during a combustion and CA10, CA50 and CA90 [6].

# 3 Experimental Set-Up

## 3.1 Engine System

The engine used for the experiments is a modified Scania D13 (see Table 2 for engine specifications) where only one of six cylinders is engaged, the remaining cylinders are motored by an ABB electric machine, see figure 12 with specifications on page 19. The ABB electric machine does also work as a dynamometer as torque is measured by a meter which in its turn prevents the engine from revolving in its axial bearings. The reason by just running on one cylinder is the convenience and less cost to control and measure the phenomenon on that one, consequently are sources of error reduced. To supercharge are two Lysholm Superchargers externally installed in series to reach desired inlet pressure. As the D13 normally is fitted with a turbocharger has a restriction been installed in the exhaust to simulate the turbo and EGR-valve. EGR is produced in the combustion chamber and recirculated to the inlet where its cooled and mixed with air. The compound is adjusted by motored valves on both air and EGR inlet, down the inlet line is the compound measured with a flow meter (see Figure 12 for layout). Since the engine mainly will run in PPC mode is an inlet air heater installed to facilitate ignition of the fuel and air mixture. The fuel injection is of a modern XPI type and is connected to the engines standard fuel pump. With only one cylinder in use is the fuel rail needed to be modified to even out pulsations in the system, these are normally evened out with pulses from the six original injectors. To overcome this problem are two rails installed, coupled in a parallel fashion, see figure 13. There is also a fuel leakage installed to simulate the missing injectors, this gives normal operating conditions for the fuel pump. For cooling is an heat exchanger installed to the engines standard cooling system, which also is equipped with a water heater which allows the engine to be preheated before running campaigns. In table 3is the Ethanol fuel specifications presented.

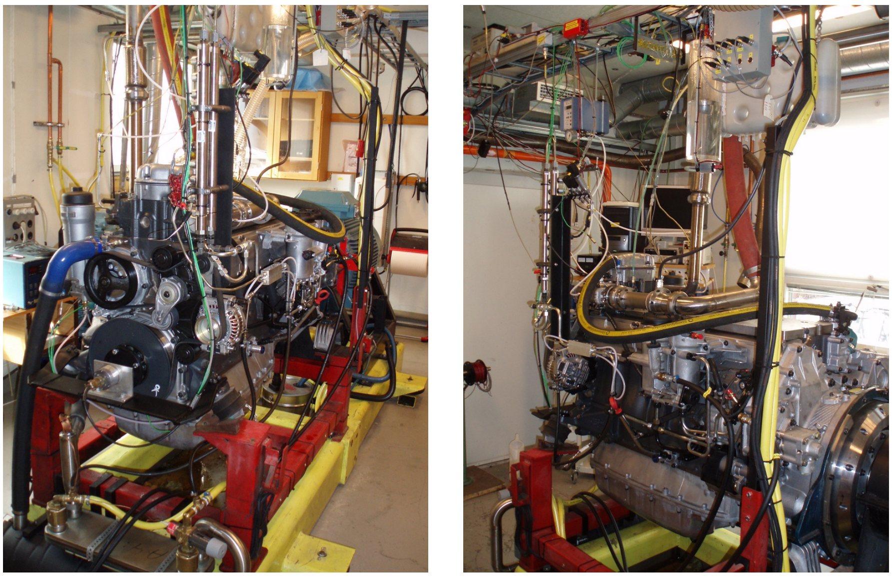


Figure 9: The single cylinder Scania D13 engine

Table 2: Engine specification

| | |
|---|---|
| **Type** | Scania D13 |
| Number of cylinders | 1 (plus 5 motored) |
| Displacement | 2123 $cm^3$ (×6=12.74l) |
| Bore x Stroke | 130 mm x 160 mm |
| Con Rod | 255 mm |
| Compression ratio | 17:1 |
| IVC | 40 ABDC |
| EVO | 50 BBDC |
| **Fuel system** | XPI Common Rail |
| Orifices | 8 |
| Orifice Diameter | 0.19 mm |
| Umbrella angle | 148 deg |

Table 3: Fuel used in experiments.

| Fuel | RON | C | H/C | O/C | LHV [MJ/kg] | A/F stoich |
|---|---|---|---|---|---|---|
| Ethanol 99.5 % | 107 | 2 | 3 | 0.5 | 26.9 | 9 |

## 3.2 Control and Measurement System

The control and measurement system is programmed in a LabView environment and together with a data acquisition unit are the sensor readings and control parameters managed. Emissions are measured in a Horiba Mexa-91000 emission unit and in a AVL 415 smoke meter. Pressure readings are high frequency signals and are located on the inlet, exhaust, oil pressure and most important the in-cylinder pressure (see figure 12). All the pressure sensors are of piezoelectric type. The in-cylinder pressure signal is amplified in a Kistler converter which passes on the reading to the control unit, and together with the CAD incremental encoder do these give real time readings at a resolution of 0.2 crank angle degrees. With this data is also heat release calculated and given in real time. Temperatures are read with thermocouples at the inlet, exhaust, engine oil and at the in- and outlet of the cooling water, figure 12.

Figure 10 shows a schematic layout of the system. The operator is located by the Host-PC (see figure 10) and in a LabView interface (Main_host.vi and Logger.vi) all necessary data is presented and control signals are managed. Here are all fast signals displayed, before displaying the signal is it processed in the Target-PC where in the RTLauncher.vi program calculates the raw sensor data to measurable values. This also works the other way around with control signals from the operator. The link between RTLauncher.vi and the engine measurement and control signals is the Field-programmable gate array card (FPGA-card). All slow signals such as temperatures and fuel consumption are handled by the Logger.vi program and are displayed in a separate window on the Host-PC. Temperature readings are linked by a data acquisition box to the Host-PC, the fuel consumption scale is directly serial connected, figure 10. All data collected during campaigns are processed in MATLAB. Here are the heat release (see section 2.1) calculated, as well as all parameters used to analyze combustion phasing, efficiencies and emissions.

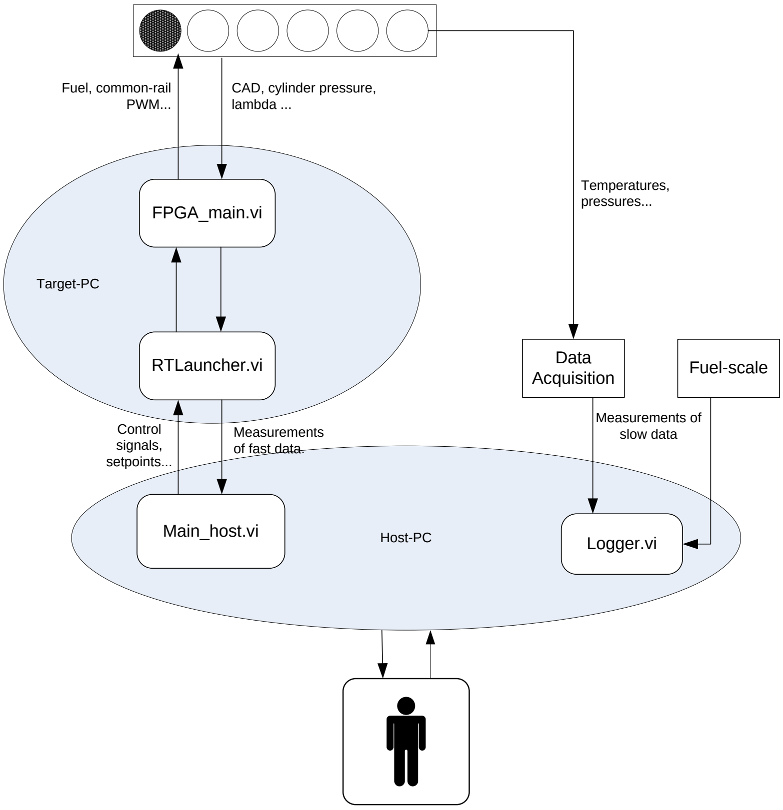



Figure 10: Control and measurement system schematic layout

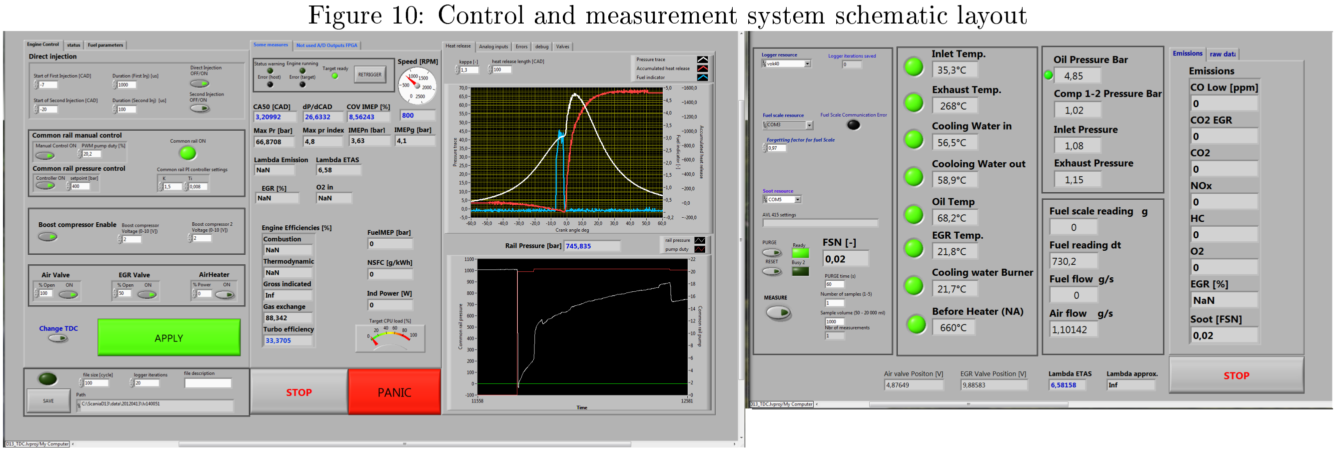



Figure 11: Main_Host.vi and Logger.vi interface layout

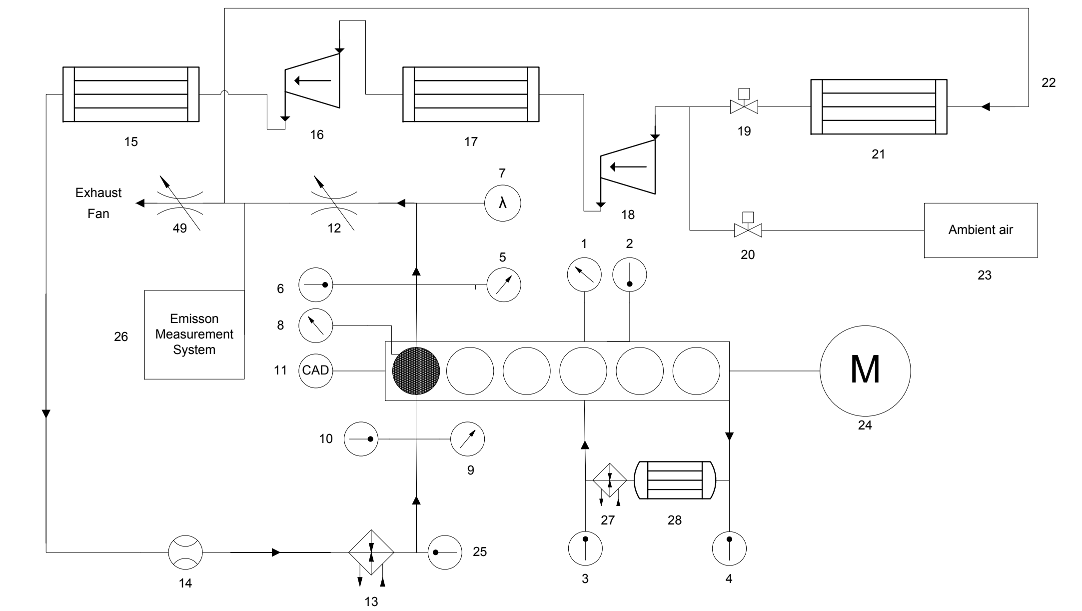



Figure 12: Engine schematic layout

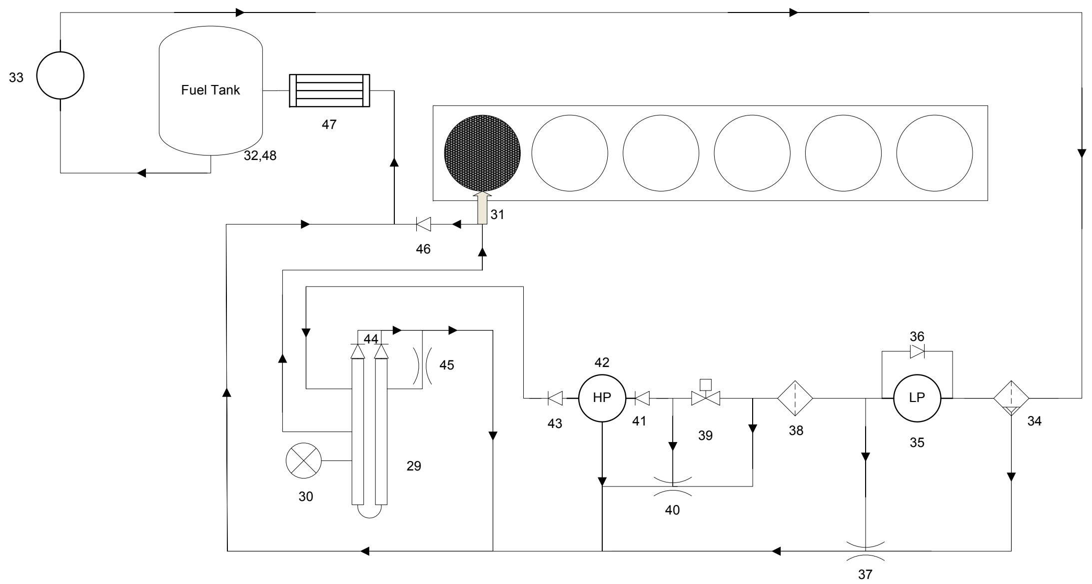



Figure 13: Fuel system schematic layout

| Nbr. | Name | Unit | Description | Specification | Output signal | Manufacturer | Model |
|---|---|---|---|---|---|---|---|
| **1** | Oil Pressure | [bar] | Engine operating condition | Range: 0-10bar Accuracy: ±0.1% | 4-20mA | GE http://www.ge-mcs.com/ | PTX 7517-3257 |
| **2** | Oil Temperature | [°C] | Thermocouple | Range: -40 - 375°C Tolerans: ±1.5°C | | Pentronic www.pentronic.se | Typ K Klass1 |
| **3** | Water temperature IN | [°C] | Thermocouple | Range: -40 - 375°C Tolerans: ±1.5°C | | Pentronic www.pentronic.se | Typ K Klass1 |
| **4** | Water temperature OUT | [°C] | Thermocouple | Range: -40 - 375°C Tolerans: ±1.5°C | | Pentronic www.pentronic.se | Typ K Klass1 |
| **5** | Exhaust Pressure | [bar] | | Absolute pressure Range: 0-5bar Error band ±0.4%FS@(-10...80°C) Frequency: 2kHz | 4-20mA | Keller http://www.keller-druck.com | PAA-23SY |
| **6** | Exhaust Temperature | [°C] | Thermocouple | Range: -375- 1000°C Tolerans: ±0.004°C | | Pentronic www.pentronic.se | Typ K Klass1 |
| **7** | Lambda | [Îż] | Air and fuel ratio | Range: 0.7-32.767 Interval: 2 ms | | ETA http://www.etas.com/en/ | LA4 |
| **8** | In-cylinder pressure | [bar] | Piezoelectric sensor. | Range: 0-250 bar. Acceleration sensitivity: <0.01 bar/g Linearity %FSO ≤ ±0.5 | | Kistler http://www.kistler.com | 7061B SN1195943 |
| **9** | Inlet Pressure | [bar] | | Absolute pressure. Range: 0-5bar. Error band ±0.4%FS@(-10...80°C). Frequency: 2kHz | 4-20mA | Keller www.keller-druck.com | PAA-23SY |
| **10** | Inlet Temperature | [°C] | Calculate heat release | Range: -40 - 375°C. Tolerans: ±1.5°C | | Pentronic www.pentronic.se | Typ K Klass1 |
| **11** | CAD | [°] | Crank Angle Degree, Incremental encoder. | Resolution: 1800 ppr 0.2° CAD/sample resolution | Uhigh>3.0V and Ulow<0.4V Load max ±20mA | Leine & Linde | 520026011, 19060191 |
| **12** | Exhaust restriction | [bar] | EGR and turbo dummy valve | | | | |
| **13** | Heater | [°C] | Inlet air heater | Min airflow: 850 cfm@7.5kW; Max outlet temp: 700 °C | | Leister www.leisterusa.com | Type 5000 DF |
| **14** | Mass Flow meter | [g/s] | Measure inlet mass flow | Accuracy: standard: ±0,8% RD plus ±0,2% FS; Optional: ±0,5% RD plus ±0,1% FS (±1% FS for ranges 3…5 ml/min; ±2%FS for ranges < 3 ml/min) Range: min 0,4…20 m3/h, max 4…200 m3/h | 0-20 mA Vsignal=K*cp*Îęm, K= constant factor, cp= specific heat, Îęm=mass flow | Brockhorst | F-106AI-AFD-02-V |
| **15** | Heat exchanger | | Inlet air/EGR cooler | | | | |
| **16** | Compressor | [rpm] | Stage 2 | See performance diagram: http://www.lysholm.us/pdf /diagram_lys1200ax.pdf | ABB control unit, controlled in LabVIEW environment | Lysholm www.lysholm.us/ | LYS 1200AX |
| **17** | Heat exchanger | | Inlet EGR/air cooler before stage 2 compressor | | | | |
| **18** | Compressor | [rpm] | Stage1 | See performance diagram: http://www.lysholm.us/pdf /diagram_lys1200ax.pdf | ABB control unit | Lysholm www.lysholm.us/ | LYS 1200AX |
| **19** | Mixing valve EGR | [%] | Determine EGR/Air ratio | | | ABB | |
| **20** | Mixing valve Air | [%] | Determine EGR/Air ratio | | | ABB | |
| **21** | Heat exchanger | | EGR cooler, condenser, now water in compressor allowed. | | | | |
| **22** | EGR furnace | | kW | | | | |
| **23** | Air inlet | | | | | | |
| **24** | EM | [kW] | Electric Dynamometer | 55kW@990rpm | | ABB | |
| **25** | Inlet temperature | [°C] | For heater (13) PID control | | | | |
| **26** | Emission measurement | | | | | Horiba | |
| **27** | Water heater | [°C] | Heater cooling water, to reach operating point quicker. | Output: 6kW Range: 30-85°C | | OEM | 6000 |
| **28** | Water heat exchanger | | | | | | |
| **29** | Fuel Rail | | Dual fuel rail to stabilize pressure due to lack of injectors. | | | | |
| **30** | Fuel Pressure | [bar] | Fuel Pressure | | | | |
| **31** | Injector | [ms] | Injector | Range: 350-2400 bar; Injector holes: 8; Umbrella angle: 148°; Mass flow: 207pph | | Scania | 1933613 |
| **32** | Tank | | | | | | |
| **33** | Electric supply pump | [bar] | Deliver fuel to low pressure pump | | | | |
| **34** | Fuel Filter/ water separator | | Fuel filter and water separator | | | | |
| **35** | Supply pump | [bar] | Low pressure pump | Deliver 9-12 bar | | | |
| **36** | Check valve | | | | | | |
| **37** | Venturi | | | | | | |
| **38** | Fuel Filter LP | | Low pressure fuel filter 9-12bar | | | | |
| **39** | Fuel pressure valve | | Regulator for fuel injector pressure | | | | |
| **40** | Venturi | | | | | | |
| **41** | Check valve | | | | | | |
| **42** | High Pressure Fuel pump | [bar] | | Max: 3000bar | | | |
| **43** | Check valve | | | | | | |
| **44** | Pressure relif valves | | Release pressure @3000bar, levels out @1000bar ±300bar | | | | |
| **45** | Pressure leakage | | To simulate absent injectors | | | | |
| **46** | Check valve | | | | | | |
| **47** | Fuel cooler | | | | | | |
| **48** | Fuel scale | [g] | Fuel consumption scale | Capacity: 6200g Readability: 0.01g Response: ≤ 1.5s | | Sartorius www.sartorius-echatronics.com | CPA6202s |
| **49** | Exhaust restriction | | For EGR recirculation | | | | |

# 4 Results and discussion

The results are presented in two sections, Injection strategy analysis 4.1 and Sensitivity Analysis 4.2. The Injection strategy analysis introduce the effect of injection control sweeps of double SOI, injection duration ratio, and single injection. The cause is to find a strategy where we can control the combustion, e.g that we can see a correlation between variation in injection control parameters and combustion phasing without compromising efficiency and low emissions. The Sensitivity analysis are divided into sweeps of inlet temperature, lambda, rail pressure and EGR. These sweeps will show the impact on combustion behaviour e.g how sensitive the combustion is to in parameter variations. These will also give us an idea of a suitable combination to gain an optimal combustion in terms of control, emissions and efficiency.

Each point presented in figures of results the mean value is for 400 cycles. For each 400 cycle has in general 2-3 soot measurements been done, the soot reading are presented in a smoke number [FSN], in a scale from 0-10. The soot measurement unit often displayed an error code telling that values of soot are to low to be measured, i.e undetectable. Other emissions are presented with Gross Indicated Specific Emissions [g/kWh], equation 7 and are compared with EURO VI emission standard which are presented in Brake Specific Emissions [g/kWh] in equation 8, thus the Gross Indicated Emissions will show a slightly lower value than if the emissions could be calculated on the power output on the flywheel,[5],[12]. This because of the unrealistic efficiency losses from the power stroke to the flywheel since the engine is modified and motored, see section 3.1. To display the stability of the combustion from cycle to cycle is Coefficient of Variation (COV) used. If any COV value aren't presented the combustion is stable and below a COV of 2%. The injection control strategy is developed in the opening section 4.1 and is used as a basis for all analyses, see table 4. The Gross Indicated Mean Efficiency Pressure, (IMEP$_g$), is the total work done during the compression and combustion stroke, normalized with the displacement volume.

$$\text{Gross Indicated Specific Emissions [g/kWh]} = \frac{EI_x}{1000} \times gisfc \tag{7}$$

$$\text{Brake Specific Emissions [g/kWh]} = \frac{EI_x}{1000} \times bsfc \tag{8}$$

$EI_x$ -Emission Index number,where x is the emission specimen [-]
*gisfc* -Gross Indicated Specific Fuel Consumption [g/kWh]
*bsfc* -Brake Specific Fuel Consumption [g/kWh]

When calculating Emission Index number in equation 9, fuel consumption $\dot{m}_f$ is recalculated with the total mass flow of emission products in the exhaust, [5].

$$EI_x = \frac{\dot{m}_x}{\dot{m}_f} \times 1000 \tag{9}$$

$EI_x$ -Emission Index number,where x is the emission specimen [-]
$\dot{m}_x$ - Emission mass flow, where x is emission specimen [g/s]
$\dot{m}_f$ - Fuel mass flow [kg/s]

All parameters are kept constant during each parameter sweep, unless other is described. For every displayed figure is the sweep-parameter presented on the x-axis.

Table 4: General operating conditions for experiments

| **Sweep:** | **12 bar IMEP$_g$** | **5 bar IMEP$_g$** |
|---|---|---|
| 1st/2nd SOI [CAD ATDC] | -35/-5 | -35/-8 |
| 1st/2nd Inj.dur [$\mu$s] | 1100/980 | 1100/980 |
| Rail pressure [bar] | 1500 | 715 |
| Inlet pressure [bar] | 2.5 | 1.5 |
| Inlet temperature [$\circ$C] | 100 | 140 |
| Speed [rpm] | 1200 | 1200 |
| EGR [%] | 50 | 50 |
| $\lambda$ [-] | 1.5 | 1.5 |

## 4.1 Injection strategy analysis

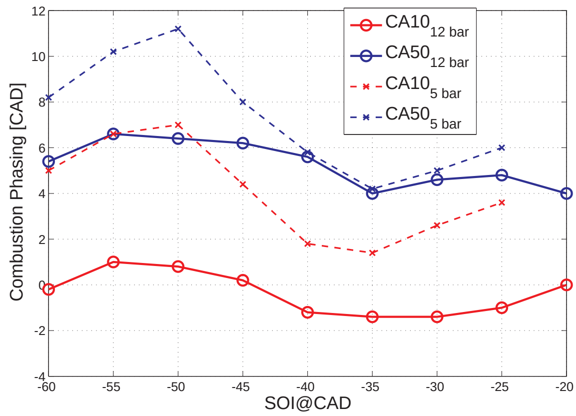
Analyses presented below are carried out to find a suitable operation strategy where the combination of; low emissions, low noise, and combustion control are obtained. The injection strategy experiment is divided into double and single injection sweeps, for double injections sweep has also an injection duration ratio sweep been done. An additional legend is presented in section 4.1.1 which describes an alternative fuel ratio, which is only used in that section, see table 5.

### 4.1.1 Double SOI sweep

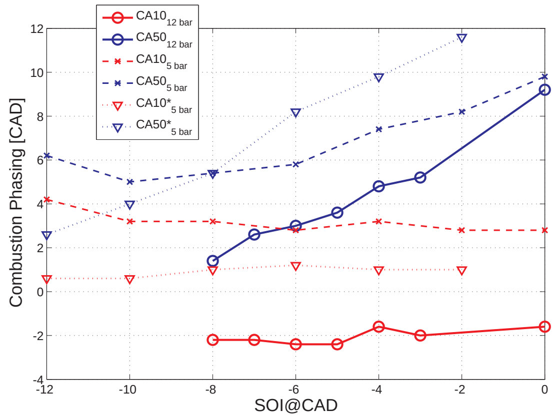
The first injection does not show any clear trend of phasing neither in low nor half load, see figure 14,(a). Even so there are similarities between the loads. The cause may be wall wetting. With early injection the fuel is sprayed on the cylinder walls and therefore not mixed and burnt properly. This can also be seen in combustion efficiency for both loads, figure 15(a). During the first injection sweep the second injection is held constant at -5 CAD ATDC, figure 14. Seen in the advanced injections CA10 is delayed past TDC and the operation is more like HCCI mode, which means that there is no connection between SOI for the injection and the SOC. With the first SOI delayed the end of duration of 2nd injection and SOC merges, hence PPC as result. A too late injection on generates loud noise (figure 15(a)) as CA50 closes in to TDC, therefore seems an operation points around -35 CAD ATDC suitable. High pressure rise rates [bar/CAD] causes acoustic noise. Acceptable levels of noise are equivalent with pressure rise around 20 bar/CAD at max.
In the second injection sweep, figure 14(b), we can see a linear correlation between SOI and CA50 at half load, which indicates an extension or a shortening of combustion duration. The CA10 is not affected by second injection and remains constant. This tells us that the second injection is not the trigger of SOC. Also here we are limited by noise, as with the earlier injections is the combustion duration shortened and moved closer to TDC and therefore the combustion is more aggressive and generating noise. Seen on the low load in figure 14(b) two injection duration strategies are used. It was shown that the injection duration strategy for half load was not compatible with low load since no control of combustion was achieved. To get control was injection duration of first and second injection reversed, see *Second injection** table 5.

Table 5: Alternative injection duration strategy with legend, for 2nd SOI sweep. (Only used in this section.)

|  | **5 bar IMEP$_g$*** | |
|---|---|---|
| | First injection* | Second injection* |
| Inj.dur [$\mu$s] | 860 | 1220 |

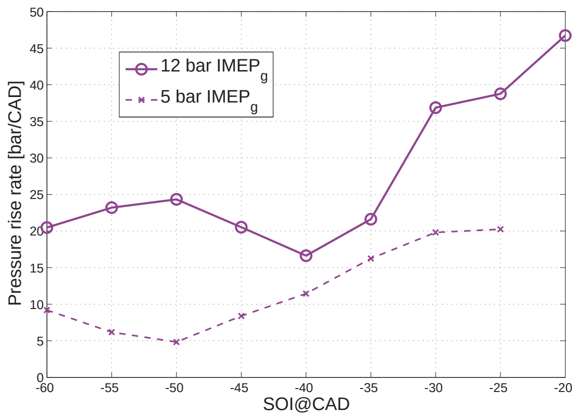



(a) 1st SOI sweep,(2nd SOI constant at -5 CAD ATDC)

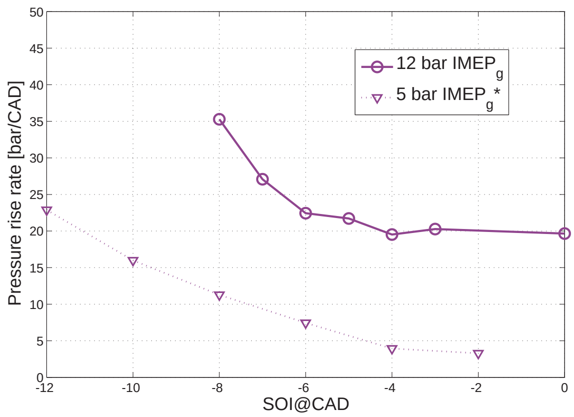



(b) 2nd SOI sweep,(1st SOI constant at -35 CAD ATDC)

Figure 14: Combustion phasing as a function of SOI.



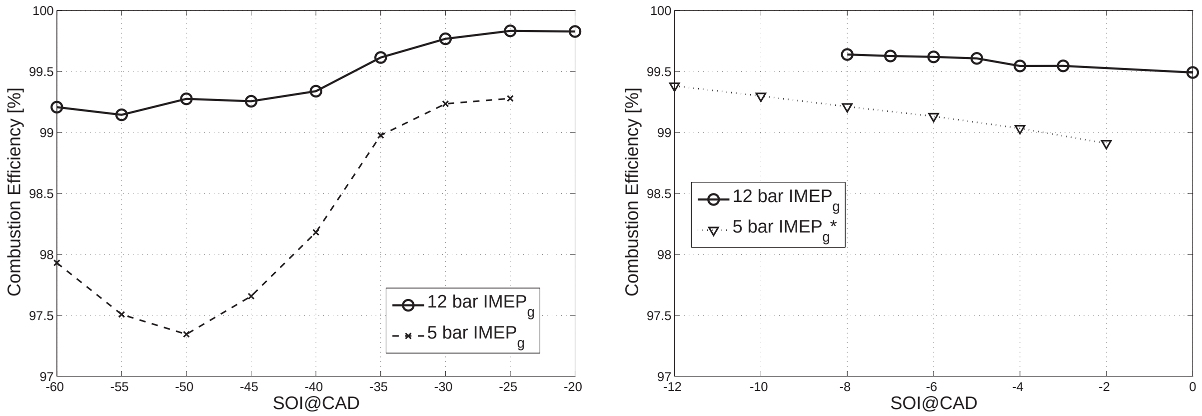

(a) 1st SOI sweep,(2nd SOI constant at -5 CAD ATDC)




(b) 2nd SOI sweep,(1st SOI constant at -35 CAD ATDC)

Figure 15: Pressure rise rate ($\delta P/\delta CAD$) as a function of SOI.

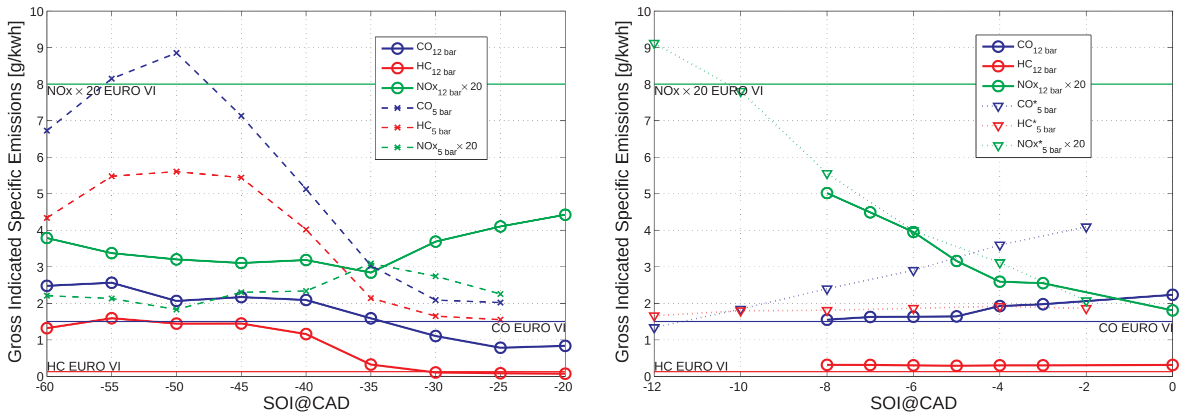
Efficiencies follow the trends discussed above. When running in the area around -35 CAD ATDC does efficiency increase but also here do we have to compromise efficiency to keep the noise level down , see figure 16. Thermodynamic efficiencies for 12 bar show good figures for both first and second injection, see figure 19. One can see the correlation between good thermodynamic efficiency and low pressure rise rate. With pressure oscillation caused by a violent combustion heat losses are increased and fuel consumption does likewise. Thermodynamic efficiency for 5 bar load is lower due higher heat losses as fuel and inlet pressure is lower, which in its turn causes larger droplets and requires longer mixing period.




(a) 1st SOI sweep,(2nd SOI constant at -5 CAD ATDC)   (b) 2nd SOI sweep,(1st SOI constant at -35 CAD ATDC)

Figure 16: Combustion efficiency as a function of SOI.

Emissions of first injection sweep in figure 17(a) shows a trend to be decreasing with a later injection. The early higher HC and CO readings can be caused by wall wetting and fuel that hides in between piston and cylinder wall. $NO_x$ readings are below EURO VI and SOOT is too low to measure, and not presented. This results are gained as we are running very lean and with high EGR to keep combustion temperature down (see figure 18), in figure 4 we can then tell that we are running in the desirable lean and low temperature zone. Moving forward to second injection 17(b), it is observed that with a later injection the $NO_x$ emissions is lowered, which is because of the reduction in combustion temperature 18. With the later injection the CA50 is phased away from TDC, i.e. the combustion duration is broaden as CA10 remains constant, see figure 14. This gives a slower combustion as well as lower dP/dCAD, see figure 15(b). Overall does low load give higher HC and CO emissions, which is because the of the lower rail pressure and in turn results in larger fuel droplets and poorer fuel and air mixture.




(a) 1st SOI sweep,(2nd SOI constant at -5 CAD ATDC) (b) 2nd SOI sweep,(1st SOI constant at -35 CAD ATDC)

Figure 17: NOx,HC, and CO emissions as a function of SOI. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on $NO_x$)

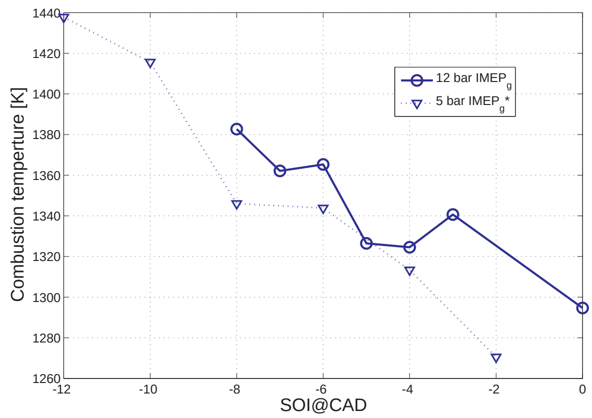
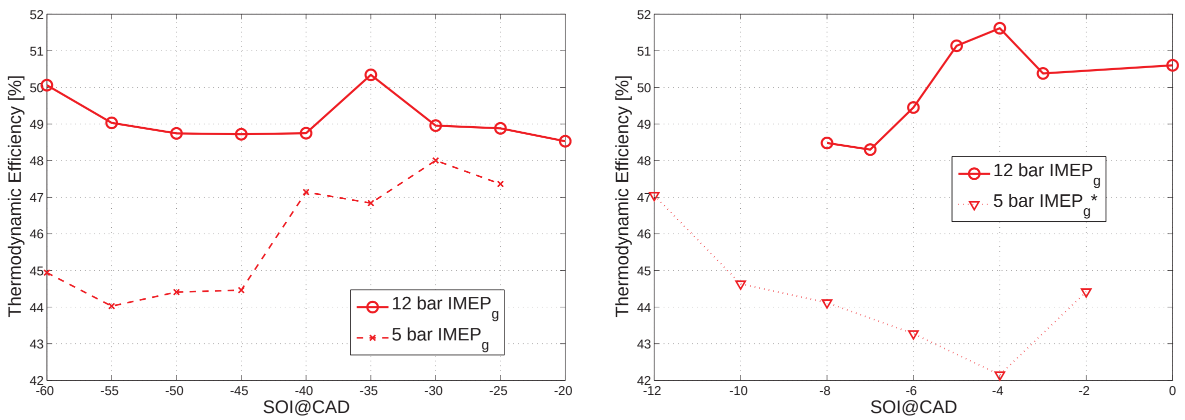



Figure 18: Combustion temperature as a function of **2nd SOI sweep**,(1st SOI constant at -35 CAD ATDC).




(a) 1st SOI sweep,(2nd SOI constant at -5 CAD ATDC) (b) 2nd SOI sweep,(1st SOI constant at -35 CAD ATDC)

Figure 19: Thermal efficiency as a function of SOI.

### 4.1.2 Injection duration sweep

The injection duration ratio is calculated in equation 10 below, note the reversed scale on the x-axis in this section.

Table 6: Operating conditions for Injection duration sweep

| **Injection duartions** | | | |
|---|---|---|---|
| **12 bar IMEP$_g$** | | **5 bar IMEP$_g$** | |
| 1st | 2nd | 1st | 2nd |
| 1260 | 820 | 1100 | 980 |
| 1220 | 860 | 1040 | 1040 |
| 1180 | 900 | 980 | 1100 |
| 1100 | 980 | 920 | 1160 |
| 1040 | 1040 | 860 | 1220 |
| 980 | 1100 | 800 | 1280 |
| 880 | 1200 | 740 | 1340 |
| 780 | 1300 | 680 | 1400 |

$$\frac{1^{st}\text{ injection}}{1^{st}\text{ injection} + 2^{nd}\text{ injection}} \times 100 = \text{Injection duration ratio, \% of fuel in first inj.} \quad (10)$$

The sweep has been made with CA50 constant, controlled by the second SOI which are continuously delayed, see figure 20. This tells us that the injection ratio is a way to control the combustion phasing. As earlier been stated the second injection is the control parameter for phasing of CA50. Seen in the pressure rise rate in figure 22, the injection strategies with ratio in the region of 60% and 35% for half load are generating high noise, i.e. one injection is larger than the other. With a major fuel in the first injection is the combustion duration shortened, hence the pressure rise rate. A major amount of fuel in the second injection on the other hand does not have time to mix properly, hence it results in a locally slight richer mixture that burn rapidly. With the fuel more evenly distributed leads to a mixture with areas of richer zones in an otherwise lean mixture and as a result we have a more moderate but acceptable noise level, see pressure rise rate in figure 20. Low load behaves in most aspects the same as half load. It has been observed that an opposite injection ratio for low load is required to gain combustion control, i.e. the ratio is for low and half load approximately 41% and 53% respectively. Low load runs really smooth in this ratio area and compared to pressure rise rate of half load in figure 20 it also shows a different trend as the pressure rise rate levels out with a lower ratio. A too high ratio tends to generate high noise, which indicates that there is low control over the combustion and can also be seen in the phasing diagram, figure 20. The effect of the mixture getting to homogeneous, and due to low rail pressure the ID is extended causing the whole mixture to ignite simultaneously.
Emissions show overall small variations. CO varies as a result of access to air, with an high ratio you can get a locally leaner mixture that helps CO to oxidize, figure 22. This also results in higher combustion efficiency (figure 21) but at the cost of control. HC for low load shows a higher value and is related to the combustion efficiency that can be a result of the lower rail pressure that creates larger fuel droplet.

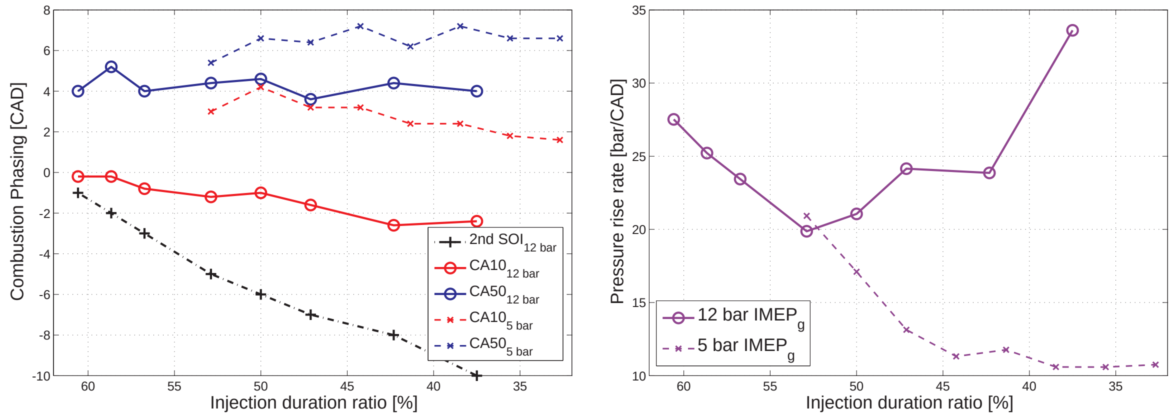



Figure 20: Phasing and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of injection duration sweep. SOI for second injection, half load, is presented to show correlation between SOI and CA50.

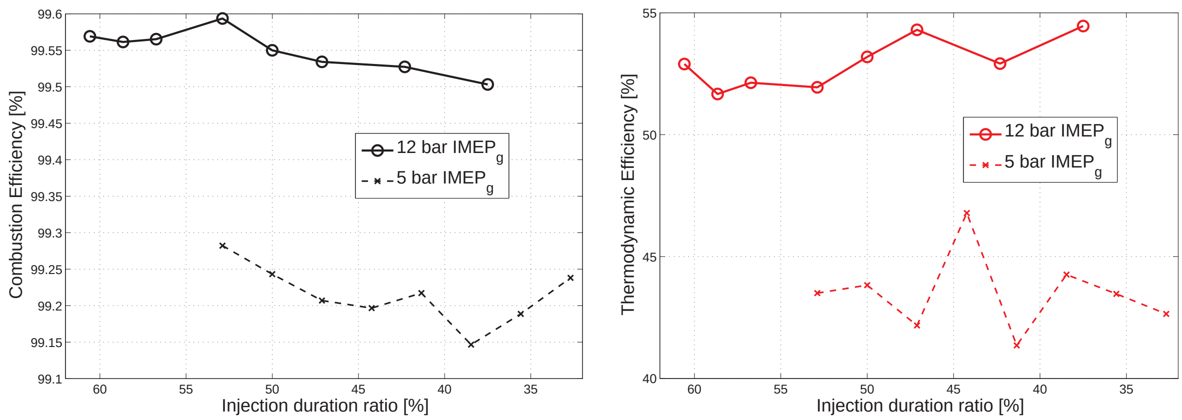



Figure 21: Efficiencies as a function of for injection duration sweep

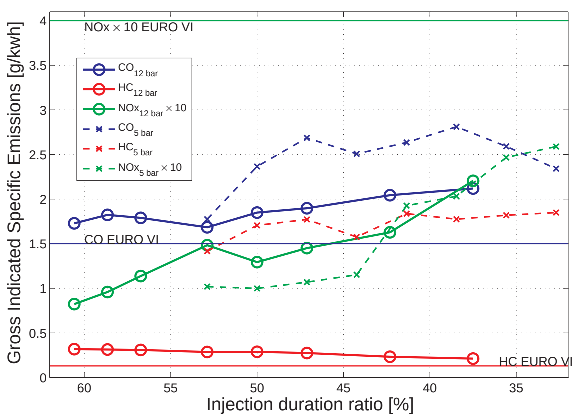



Figure 22: Emissions as a function of for injection duration sweep. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on $NO_x$)

### 4.1.3 Single Injection SOI sweep

Although single injection gives good efficiency and low emissions (figure 25) the operating window is way too short. For half load the window is approximately 3-4 CAD around -10 CAD ATDC, since the combustion is creating high noise(see figure 23) at early injection and almost dying at later injection (see efficiencies 24 as well as COV in figure 25). This results can also be connected to injection duration in section 4.1.2. The trend in phasing and noise with a major amount of fuel in one or the other injection can be compared with the single SOI sweep. With a too homogeneous mixture does the combustion duration become very short and with high noise. A too homogeneous mixture injected late on the other hand has to short mixing period and does barely ignite.

Table 7: Operating conditions for single SOI sweep

| **Sweep:** | **12 bar IMEP$_g$** | **5 bar IMEP$_g$** |
|---|---|---|
| SOI [CAD ATDC] | x | x |
| Inj.dur [$\mu$] | 2080 | 1680 |
| Rail pressure [bar] | 1500 | 715 |
| Inlet pressure [bar] | 2.55 | 1.4 |
| Inlet temperature [$\circ$C] | 100 | 140 |
| RPM | 1200 | 1200 |
| EGR [%] | 47 | 49 |
| $\lambda$ | 1.46 | 1.46 |
| CA50 [CAD ATDC] | x | x |
| IMEP$_g$ | 11.4 | 5.5 |
| COV [%] | 0.6 | 1.8 |

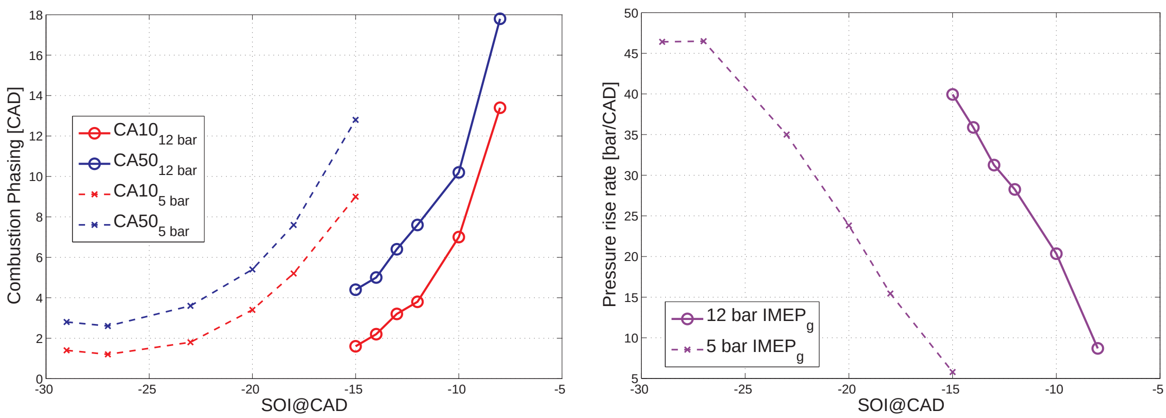



Figure 23: Phasing and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of single injection sweep.

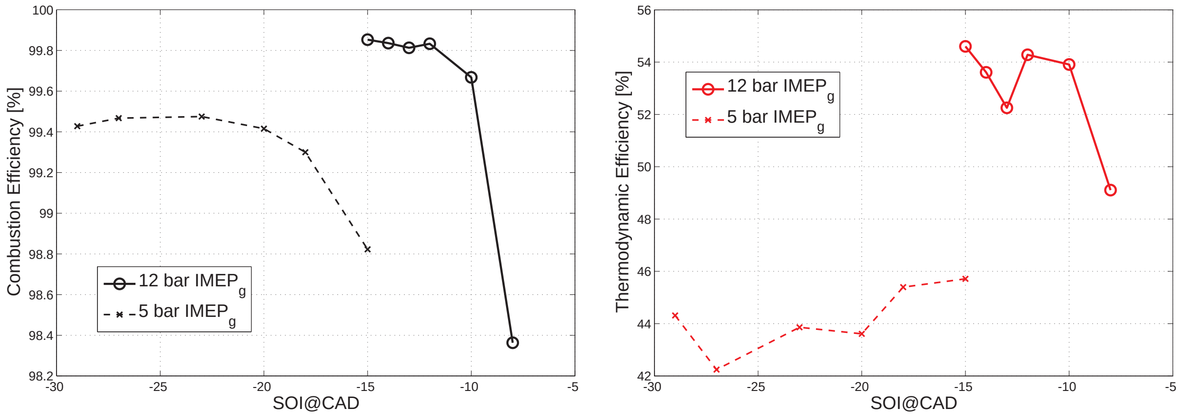



Figure 24: Efficiencies as a function of for single injection sweep.

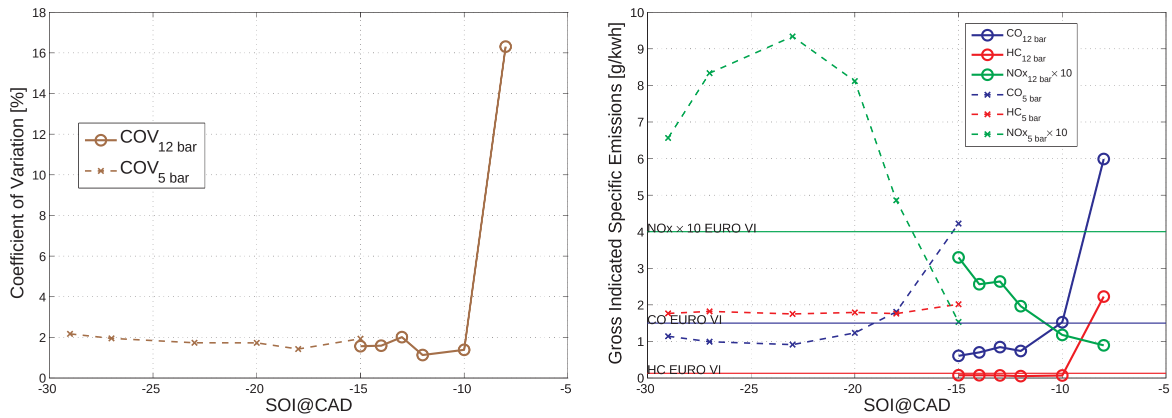



Figure 25: Coefficient of Variation and emissions as a function of single injection sweep. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on $NO_x$)

## 4.2 Sensitivity analysis

This analyse is done to test how sensitive the combustion is to variations of in-parameters. When thinking of terms of a production ready engine does this give an indication of which tolerances that are allowed to have a stable combustion.

Lambda, rail pressure, and EGR sweeps are only done 12 bar IMEP$_g$ load since effects on combustion is assumed to show the same behaviour for all loads.

### 4.2.1 Inlet temperature sweep

Inlet temperature was early noticed to be a important control factor, especially in low load. This may not be easy to see in figures below but observations as campaigns where run show that there was a clear trend as combustion went from almost dying to be knocking in a temperature range of 5-10 degrees Celsius. Otherwise does both loads shows similar trends as functions of the inlet temperature, figure 26. The high inlet temperature is needed to ignite the fuel from first injection, which is later phased with the second injection creating a controllable combustion. With too low temperature doesn't the first injection ignite resulting in both injections to ignite simultaneously, which in its turn create the higher pressure rise rate at lower temperatures, see figure 26. To high inlet temperature tends to ignite first injection to early hence creating noise when CA50 is closing in to TDC.

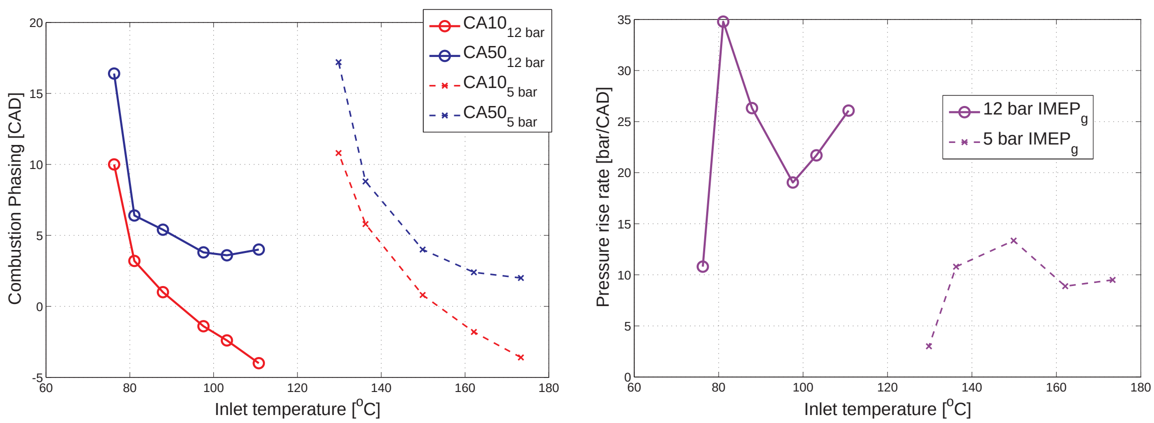



Figure 26: Phasing and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of inlet temperature sweep.

Efficiencies also shows that the combustion tends to die with low inlet temperature, figure 27. Thermodynamic efficiency is neglected as it tend to be misleading as fuel consumption measurement is unstable. Emissions are behaving as expected with higher HC and CO reading at low temperature, i.e low combustion efficiency, and as the temperature increases NO$_x$ is doing likewise. Even so are NO$_x$ readings very low. Soot measurement show no reading and are neglected.

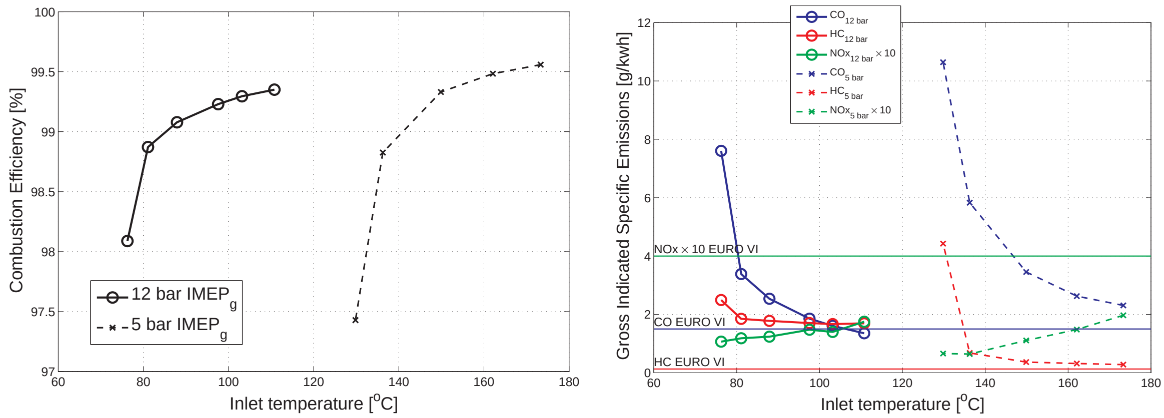



Figure 27: Combustion efficiency and emissions as a function of for inlet temperature sweep. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on $NO_x$)

### 4.2.2 Lambda sweep

This lambda sweep is done by adjusting inlet pressure, hence can this result also be seen as inlet pressure sweep.
When looking at phasing and pressure rise rate in figure 28 do we see a recurrent trend. As the fuel mixture gets richer does the fuel lack the excess of air and the mixture has a longer ignition delay, and as it ignites all ignites at once creating a uncontrollable and noisy combustion. As earlier stated does the leaner mixture ignite earlier and are smoothed and controlled by the second injection.

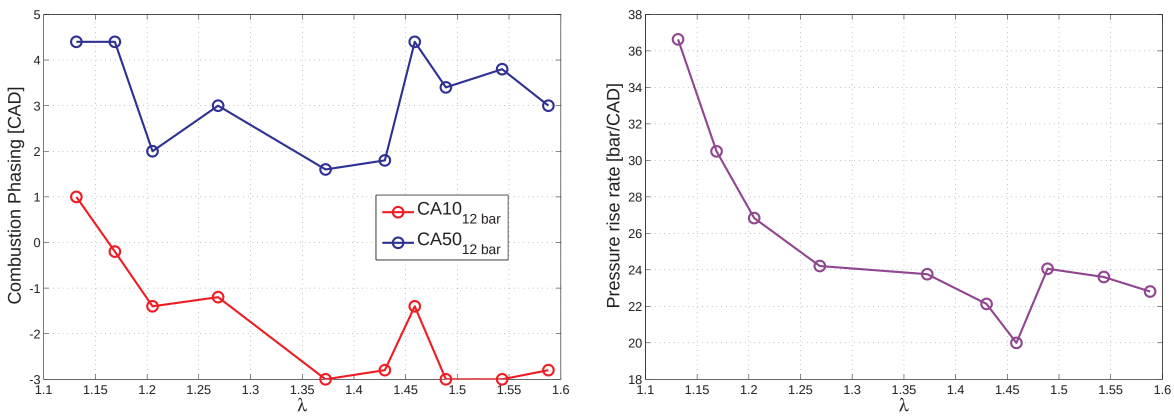



Figure 28: Phasing and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of $\lambda$.(Load = 12 bar IMEP$_g$)

Emissions in figure 29 does show the first readings of noticeable soot, the high reading could be soot particles that got caught and loosen but readings show a clear trend that soot is present at richer mixtures. As lambda is increased is emissions of CO and HC accordingly declining but $NO_X$ is increasing, due to higher combustion temperature and excess of oxygen for CO to be oxidized. Efficiencies in figure 30 also show that a higher lambda gives a better combustion up to lambda 1.5, thermodynamic efficiency is increasing indicates that fuel consumption is decreasing. An observation is that thermodynamic efficiency continue to increase even when combustion efficiency is declining, which can be due to low accuracy in fuel consumption measurement, or can this be the effect of over mixing of air and fuel, causing a increase of CO emissions seen in emissions figure 29 and lowering combustion efficiency.

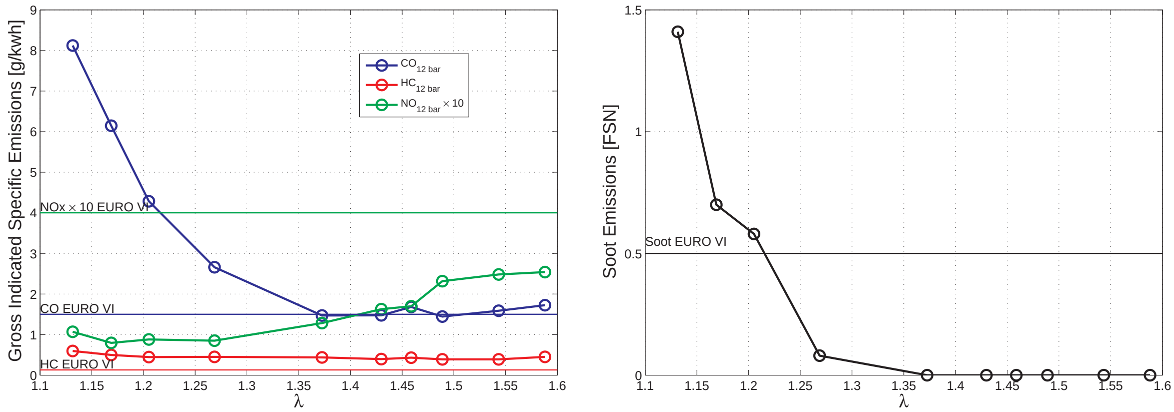



Figure 29: Emissions as a function of $\lambda$. **Euro VI emissions are given in Brake Specific Emissions [g/kWh], soot is given in [FSN].**(Note multipliers on $NO_x$)(Load = 12 bar IMEP$_g$)

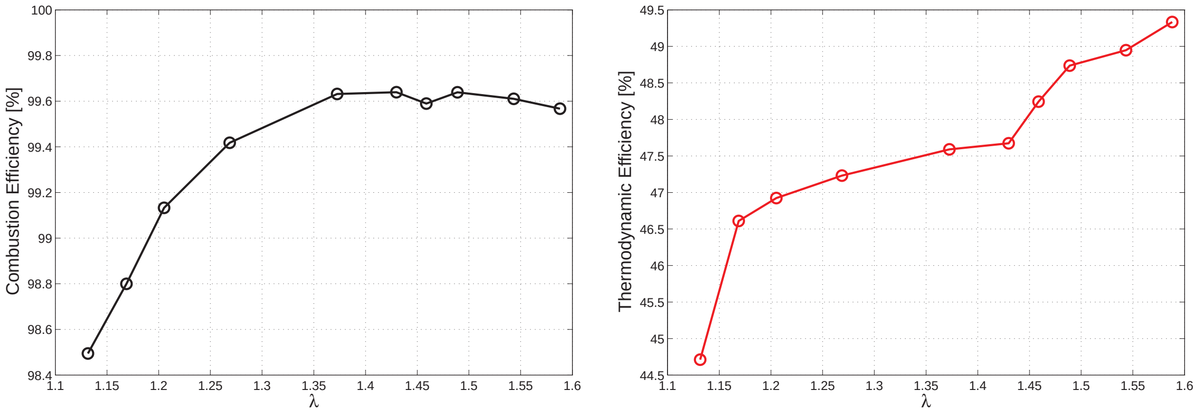



Figure 30: Efficiencies as a function of $\lambda$.(Load = 12 bar IMEP$_g$)

### 4.2.3 Rail pressure sweep

During the rail pressure sweep was lambda, load and CA50 strived to be kept constant, with increasing fuel duration, inlet temperature and phasing of second injection. It may be hard to draw clear conclusions as variations in presented data can result from change in other parameters than rail pressure. Even so the data are presented below showing interesting results, but new data may be required to isolate the effects from a rail pressure sweep.
Low rail pressure results in larger fuel droplet which have an longer mixing period, and when ignited, the combustion is quick and violent, see pressure rise rate figure 31. As CA50 is kept constant is that not visible in phasing as the inlet temperature shortens the ignition delay, figure 31 and 32. Also when increasing injection duration and decreasing rail pressure do we get a locally rich combustion which generates the noise in figure 31. Worth noting is the stability in cycle to cycle variation of the combustion throughout the experiment, see Coefficient of Variation in figure 33, which is an affect of the strive to keep lambda, load and CA50 constant as earlier mentioned. Inlet temperature is presented in figure 32, it shows a clear correlation to other results in this section. This tells us that results are more likely a combination of the effects of rail pressure and inlet temperature.

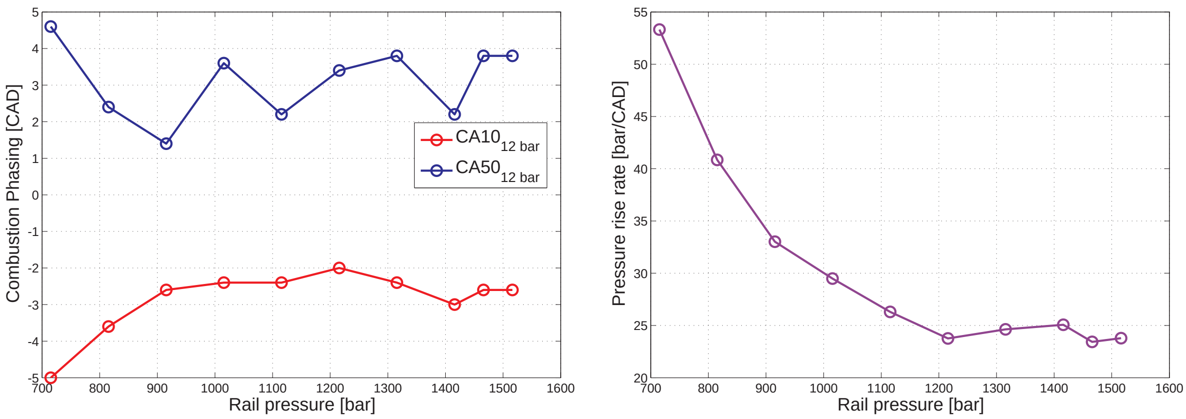



Figure 31: Phasing and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of rail pressure. (Load = 12 bar IMEP$_g$)

Emissions in figure 33 also shows a trend that are more connected with inlet temperature rather than rail pressure. The low emissions at low rail pressure (and high inlet temperature) are the result of a combustion with good efficiency but to a cost of a pressure rise rate that is way over the expectable level. Soot are not presented as it is zero.

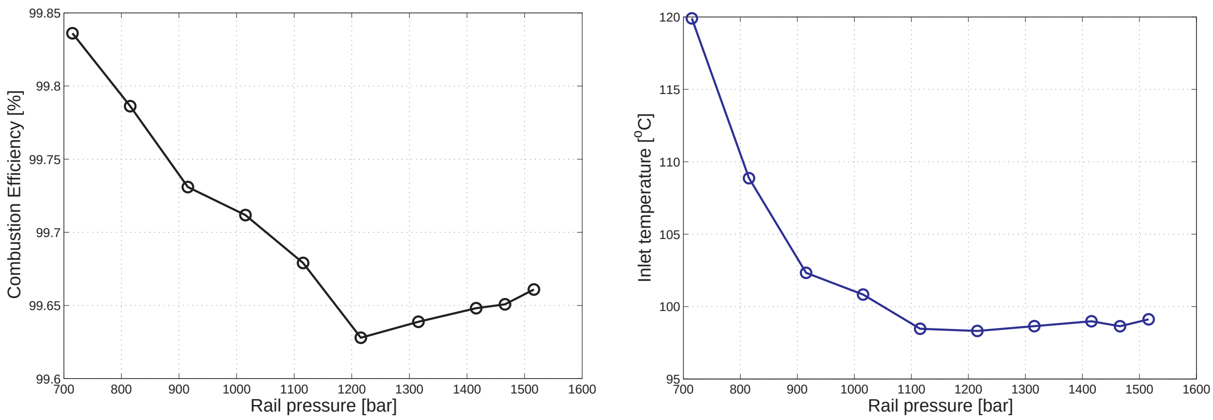



Figure 32: Combustion efficiency and inlet temperature as a function of for rail pressure. (Load = 12 bar IMEP$_g$)

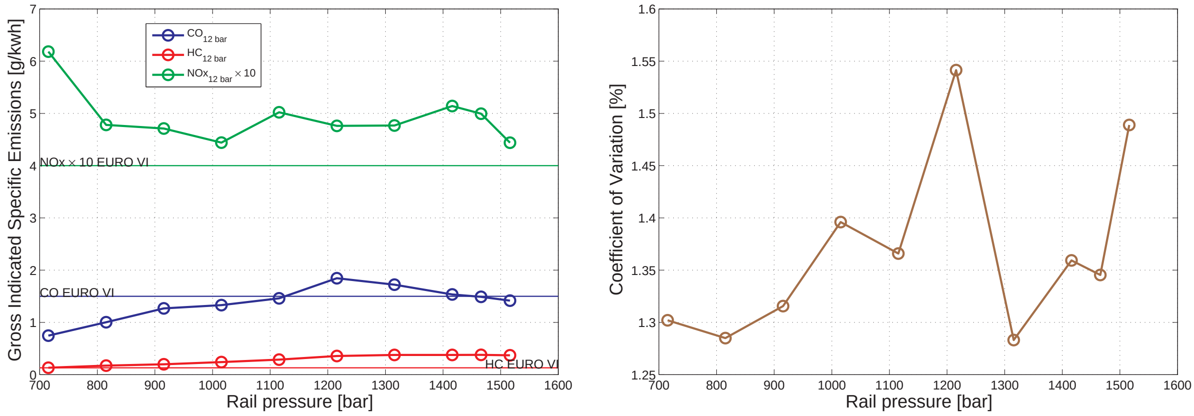



Figure 33: Emissions and Coefficient of Variation as functions of rail pressure. NOx readings show unreasonable high reading compared to other results, and are probably measurement errors. Based on other sweeps, one can assume that the NOx variations are correct, but are more likely to be around a value of 1.6 g/kWh. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on $NO_x$)(Load = 12 bar $IMEP_g$)

#### 4.2.4 EGR sweep

EGR works as a inert gas to lower combustion temperature, as well as its make the fuel mixture leaner. As seen in figure 34 is the ignition delay getting shorter as EGR is lowered. This allows the engine to be filled with more air, keeping the lean mixture, but with higher lambda that increases combustion temperature and noise, see figure 34. This also makes the combustion inconsistent as seen in figure 35 and as phasing of CA10 appears before second SOI can it more be resembled with HCCI. Worth noting is that second SOI does still have control over CA50 as EGR is decreasing. The control over combustion is lost between 55-60 % EGR and with the increasing level does the ignition delay extend as there are less oxygen. The pressure rise rate (figure 34) in this region increases probably due to CA10 is close to TDC.

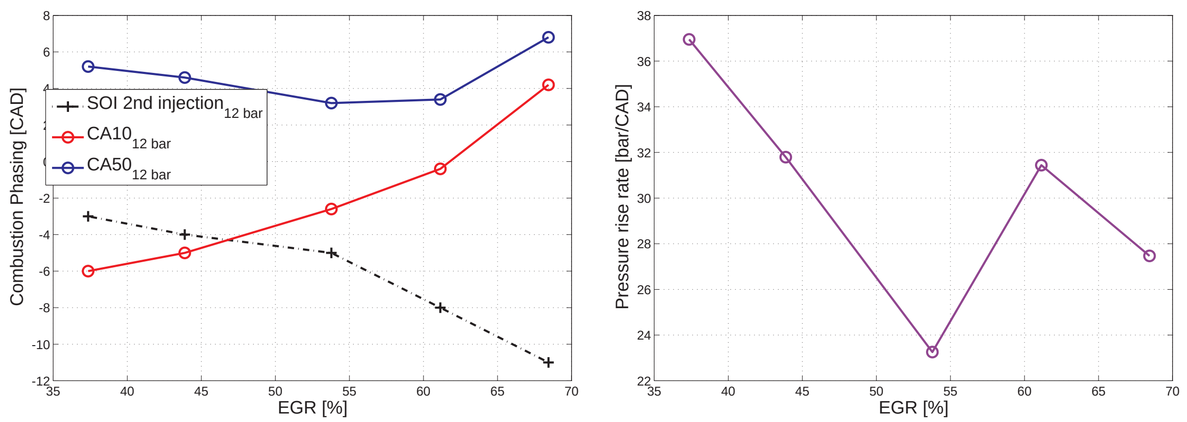



Figure 34: Phasing with 2nd SOI and pressure rise rate ($\delta P/\delta CAD$) as a function of ratio of EGR. (Load = 12 bar $IMEP_g$)

With the high EGR level does the efficiency drop, see figure 35, also a consequence by the lack of oxygen. This follow in the emissions in figure 36 as we see increasing levels of CO, due lacks air to oxidise and HC, due to poor combustion. NOx follows the trend as earlier, with a high combustion temperature do we get increasing NOx readings, and vice versa. The EGR sweep has been repeated since there was irregularities in soot measurement, both repetitions gave soot readings but as there was no consistent trend between readings nor earlier sweeps are these results ignored and assumed to be zero.

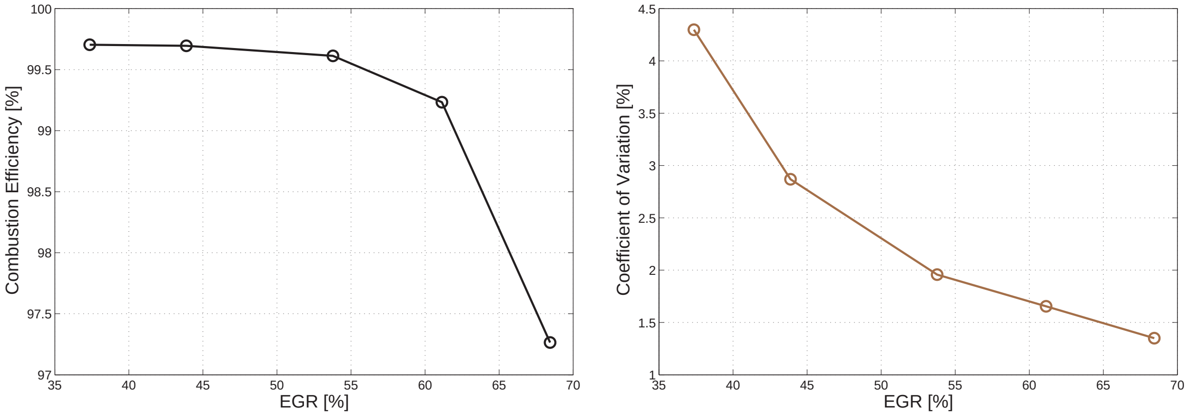



Figure 35: Combustion efficiency and Coefficient of Variation as a function of ratio of EGR. (Load = 12 bar IMEP$_g$)

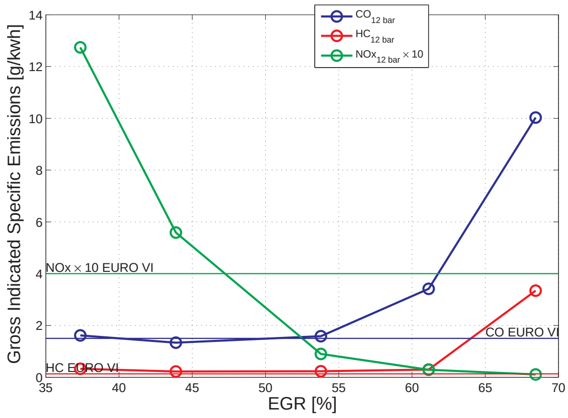
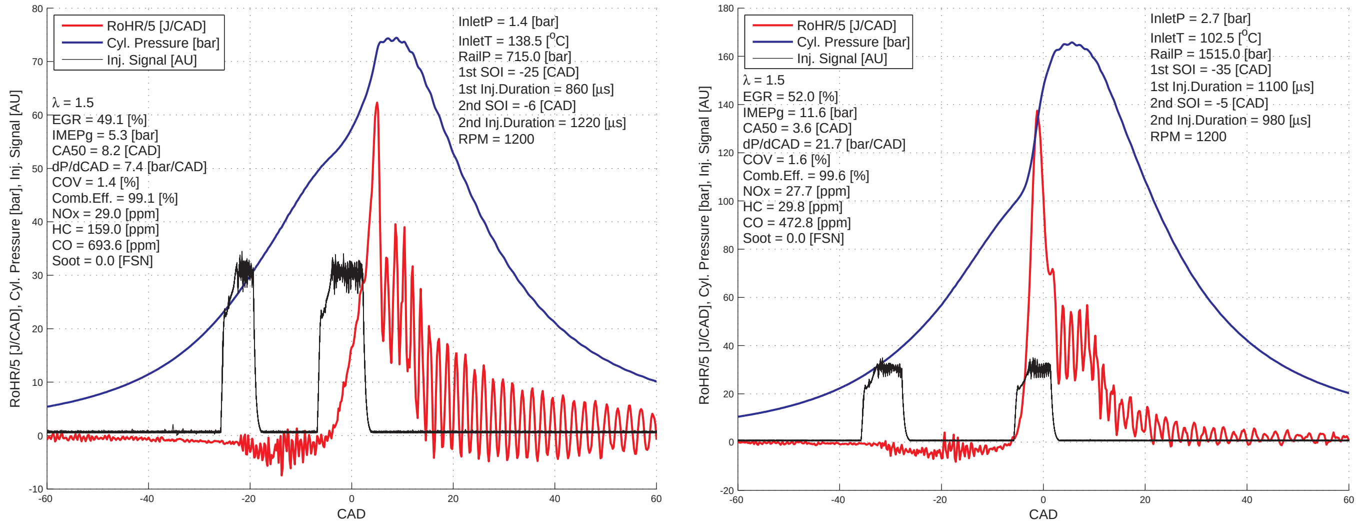



Figure 36: Emissions as a function of EGR. **Euro VI emissions are given in Brake Specific Emissions [g/kWh].**(Note multipliers on NO$_x$)(Load = 12 bar IMEP$_g$)

# 5 Conclusions

This thesis presents a combustion analysis of Ethanol as fuel in the PPC operation mode, EPPC. Experiments have been carried out in form of parameter sweeps to determine suitable injection control strategies and sensitivity analyses on all operating parameters. The latter to find operating points with good efficiency, low emissions, and to learn how sensitive combustion is to parameter variations.




(a) Low load=5 bar IMEP$_g$

(b) Half load=12 bar IMEP$_g$

Figure 37: Example of EPPC combustion from SOI sweep in section 4.1.1, for 5 and 12 bar load.

With the results in hand, one can clearly tell that the key feature to facilitate a satisfactory EPPC combustion is to have the correct temperature at the correct time. EPPC need a high enough inlet temperature for the fuel mixture to ignite, a low combustion temperature to prevent NOx formation, as well as a lean mixture to prevent soot, and still have control.

As experiment was carried out at half load with double injection, see figure 37, it was early found that more or less an evenly distributed injection strategy gave; control over the combustion, high efficiency, low noise while keeping NOx and soot low. The double SOI are set to have the first injection around -35 CAD and second injection at -5 CAD. In terms of control the second injection is in control over the combustion duration. Results have shown that SOC or CA10 mainly is dependent of the combination of inlet temperature, first SOI, and lambda. The SOC starts as soon as a proper air and fuel mixture for a certain temperature is reached, e.g if the lambda is constant, can SOC be advanced or retarded by increasing or decreasing inlet temperature. With the pilot injection you can control the ID/mixing period with a more or less constant SOC. To gain control over the combustion at 5 bar IMEP$_g$ was the injection duration shifted so that the duration of the second injection is larger than the first injection. The trend shows that with increasing load is fuel ratio shifting from major injection on second injection, to first injection, but in common for each load is that the first injection is the combustion trigger and second injection is the controller. One can say that the combustion can be separated in two parts, the first more violent kinetically ignited combustion, and the second injection which gets partially kinetically ignited mixture which continues in a partially stratified injection creating a diffusion combustion. This can be seen in figure 37 as a slight hump in the rate of heat release just after the maximum peak. Single injection was early ruled out to since it has a very small operating window, due to the fast kinetically ignited combustion.

Results from the sensitivity sweeps show that a combination of inlet temperature, inlet pressure and EGR contributes with a sufficient ID and SOC before TDC, and we get a broader operating range, figure 37. Inlet temperature has shown to have big influence over the whole combustion sequence and is a crucial parameter for EPPC control. Emissions shows overall low values in NOx and undetectable soot, and combustion efficiencies shows high values when combustion is under control.

## 5.1 Future work

High load sweeps since these have not been done due to limitations in the experimental set up. To determine combustion limitations and characteristics.

RPM sweep to find characteristics.

A more thorough investigation of SOC to determine a solid control strategy.

Repeat proven operation points in optical studies to get more knowledge about EPPC combustion.

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