

Thesis for the Degree of Master of Science

Partially Premixed Combustion with Methanol in a Heavy Duty Engine

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Abstract

PPC is a new combustion concept that combines high efficiency with low emissions.

This thesis is based on experiments conducted with a heavy duty PPC engine that was ran on neat methanol. A fuel that might be a candidate for future engine propulsion in large scale.

By varying the inlet pressure, the corresponding inlet temperatures were found for premixed combustion, i.e. combustion in HCCI mode. Around the different operating points that were found in HCCI, the load range was investigated in PPC in the next experiment. By decreasing the temperature as the load was increased, the range of operation was extended many times compared to a constant inlet temperature. The results show a maximum load slightly above 9 bar IMEP, and the lowest load below 3 bar within the defined limitations of the load range.

The highest indicated efficiency reached, was 51 percent for 1.5 bar inlet pressure.

The emissions were also analyzed, and except for quite high levels of unburned hydrocarbons compared to emission standards, the obtained values were found to be promising.

List of abbreviations and acronyms

λ	Lambda (Relation between AF-ratio and stoichiometric AF-ratio)
A/F-ratio	Air to Fuel-ratio
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Center
CAD	Crank Angle Degree
CI	Compression Ignition
CO	Carbon monoxide
COV	Coefficient Of Variation
C_p	Specific heat Capacity
HC	Hydrocarbons
HCCI	Homogenous Charge Compression Ignition
$IMEP_g$	Gross Indicated Mean Effective Pressure
$IMEP_n$	Net Indicated Mean Effective Pressure
IVC	Inlet Valve Close
IVO	Inlet Valve Open
LHV	Lower Heat Value
NO_x	NO and NO ₂
P_{in}	Inlet Pressure
PPC	Partially Premixed Combustion
r_c	Compression ratio
SI	Spark Ignition
s_i	Specific emission for product i
TDC	Top Dead Center
T_{in}	Inlet Temperature

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1 Introduction

The work for this thesis was carried out at the Division of Combustion Engines at Lund University. It should be seen as a pre-study of how methanol works in the new combustion concept Partially Premixed Combustion (PPC), in a heavy duty engine. The goal is to generate information that can be useful for further studies regarding methanol fuel in PPC.

1.1 Outline

In order to understand the work of this thesis, it is important to understand the underlying reasons of using methanol in conjunction with PPC. To be able to catch on to all readers, the working principle for the most common engines that are used today are described. Drawbacks and advantages are also converted and compared to other combustion philosophies, in an attempt to clarify why there is still a need for new combustion engines, even though they have been around for 140 years.

In the following chapter the fuel is discussed. It has some interesting properties that is specifically interesting for a combustion engine and it is also likely to be a good candidate as a fossil free fuel for vehicle propulsion.

The experimental part that follows is the backbone of the thesis. It is basically three different kind of experiments which have the investigation of load range in common.

When future engines and propulsion systems are discussed, the majority of the people will probably not think about a combustion engine. Electric cars are getting better and more available as they become cheaper, and rapidly increase the maximum range with more powerful batteries. There is also a lot of research and development for new types of batteries that seem promising and easily outperforms the traditional lead or nickel batteries. As can be seen, heavily desired improvements of the parameters *energy density*, and the *specific energy* are both developing in the right way.

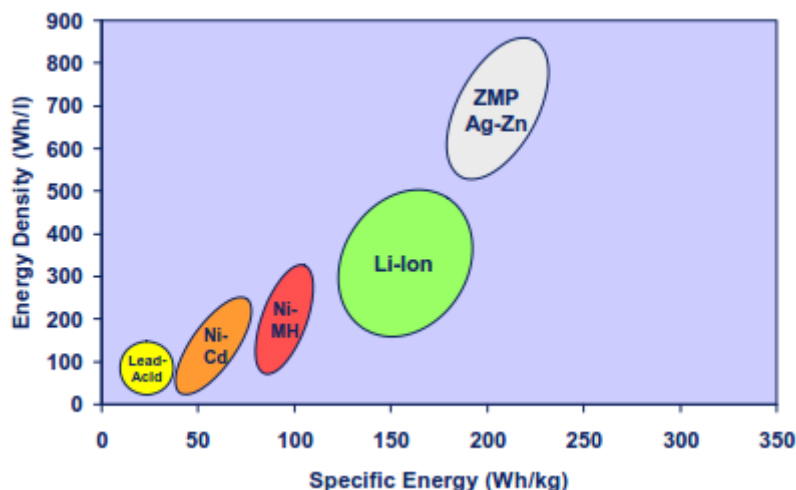


Figure 1-1. Energy density and specific energy for different types of batteries. [8]

As the batteries are getting developed, a glance at Figure 1-2 is all that is needed to understand there is a huge gap in energy capacity for a mobile electric engine versus a combustion engine.

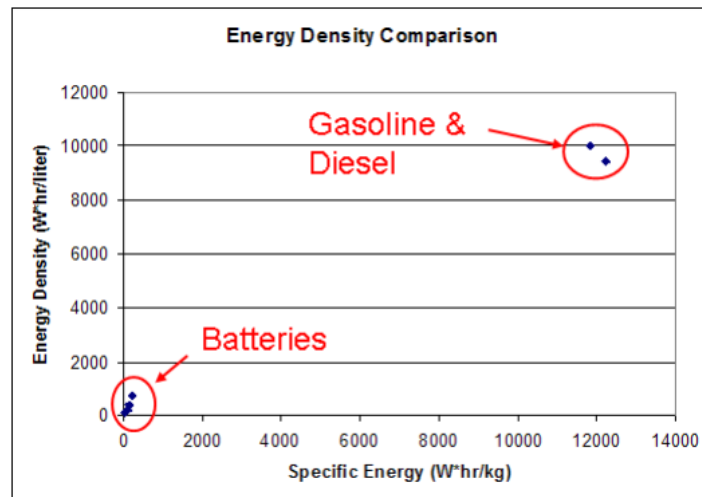


Figure 1-2. Energy density and specific energy for different types of batteries [8].

The two most common internal combustion engines that are used worldwide are the gasoline and the diesel engine. The former with spark ignition (SI) and the latter with compression ignition (CI). The SI gets decent emission values if a three way catalyst is used, and the CI shows generally good efficiency. For sure there is an interest to combine these characteristics, and one of the most promising methods for doing this is by *Partially Premixed Combustion* (PPC).

Fuel economy is something that every citizen has interest in, but even the emission levels are getting more importance. In the end of May 2015 a decision from the municipality of Oslo declared that diesel vehicles are temporary forbidden in the city if the air quality has been considered bad for two consecutive days. If this is an example that will be followed by other communities and countries, a rapid development for lower emissions is to expect.

Even though the PPC concept might lead to a production of engines which are more efficient and cleaner than today's engines, there is still a need to find new fuels for vehicles. That is due to two reasons: The increased amount of CO₂ in the air that leads to disturbances in the climate, and the reason that the oil from which today's fuels are extracted, is not infinite. Large investments have been made to try to replace parts of the gasoline consumption by ethanol. The reason for the choice of ethanol might unfortunately not be derived from criteria to find the best solution for engine operation and the fuel production from "well to wheel", but rather from economic interests from people within the line of business. A fuel that could be well suited for vehicle propulsion is another alcohol, namely methanol. It has potential for being produced in an economically sustainable way. It has only small negative effects on the environment and climate and it benefits from many properties desirable in a combustion engine [1] [2].

1.2 Aim of the Thesis

Since methanol is a promising fuel for propulsion in large scale, and PPC a combustion concept that has been found to have great potential, it is of great interest to combine these components and get some experience from it.

The goal in this thesis is to answer a few questions listed below.

- 1) Find inlet pressures and the corresponding inlet temperature to achieve premixed combustion via compression ignition.
- 2) Investigate the combustion for specific inlet temperature and pressure by varying the injection timing and amount of fuel injected in PPC mode.

- 3) Investigate the load range from the values obtained from the first experiment, but in PPC mode.
- 4) Investigate the efficiency and the emissions for the different load cases.
- 5) Announce problems regarding methanol combustion in PPC mode.
- 6) Suggest eventual improvements for further experiments.

1.3 Thesis Boundaries

An engine can basically be operated at different load and speed, but in this thesis all experiments will be conducted at a constant speed in at 1200 RPM.

All data is derived from engine operation with no EGR. This is not the common way to operate PPC, but for a pre study, it was chosen for its simplicity.

The efficiency of the gas exchange will not be considered since the boosting system is based on an external compressor system. The net power output will neither be investigated due to the setup of only one operating cylinder in a six cylinder engine.

2 Internal combustion engines

The two main classifications of the internal combustion engines are the continuous combustion engines such as gas turbines, and the intermittent engines, like piston engines. The difference in principle is that the intermittent engines have a cycle-based working process that is repeated with an intermittent combustion, unlike the continuous combustion engines which have an uninterrupted combustion. Since the experiments are based on a piston engine, these will be investigated, while the continuous combustion engine is mentioned for the last time in this thesis.

2.1 Four-stroke and two-stroke engines

The two main types of piston based internal combustion engines are the two-stroke and the four-stroke engine. A stroke is the back and forth motion in which the reciprocating piston moves in the cylinder. The two-stroke has a power stroke on each second stroke that enabling the engine to deliver power. That means one power stroke per crankshaft revolution. The four-stroke has a power stroke in one of four, i.e. one in each second crank shaft revolution. With all other parameters held equal, this means the two-stroke engine has twice the power density compared to the four-stroke engine.

Both types of engines are used as *spark ignition* (SI) and *compression ignition* (CI) engines, but the applications where they are used is widely different. The two-stroke SI engine, also known as the small two-stroke engine, is used where low weight and price is desired and high power density required - and efficiency is not a given priority. Examples of this is equipment used for gardening, such as chainsaws and hedge shears, where all weight must be held by hand. Also small motorcycles and mopeds are commonly seen with two-stroke engines.

The two-stroke diesel engine, i.e. the big two-stroke, is used in totally different applications than the small ditto. Large marine diesels and stationary engines for power generation with a stroke measured in meters, are usually of this type.

The four-stroke engine is widely used in transportation applications such as heavy duty trucks, buses, cars, motorcycles and small airplanes. It is preferable in these application due to the higher efficiency, better emission values and for its robustness compared to the small 2 stroke engine.

2.2 The working principal of a combustion engine

It is hard to give a generic explanation of the working principle of a combustion engine, and at the same time avoid confusion. Hence it was decided to describe the working principle of a four-stroke spark ignition engine, the Otto engine, and use this as a reference to describe the principle of the other common engine concepts.

To start a combustion engine it is necessary to have a certain rotational speed of the crankshaft, to which the piston is connected to via a con rod. That issue is often solved by a starter motor. The rotation of the crankshaft gives the piston a longitudinal movement which enables it to aspirate fresh air into the cylinder when the inlet valve is open. This is described as the first stroke with the first picture in Figure 2-1 on the next page.

When the piston reaches the bottom end of the cylinder, the inlet valve closes and the compression stroke starts. The compression results in high pressure and temperature in the combustion chamber as the piston reaches the top dead center. This is exhibited by the second picture in Figure 2-1.

If a gasoline engine is considered, a spark from the sparkplug initiates the combustion of the fuel and air mixture that is under pressure after the compression stroke. The heat release from the combustion causes a temperature rise and thereby a pressure increase which pushes the piston down. This is the

so called power stroke that transform the energy to work in the engine. The different principle for a diesel engine is described in paragraph 2.2.2.

After the power stroke the piston is located in the bottom end of the cylinder and will now start the gas exchange process. The exhaust valve opens in this position, and as the piston moves upwards the burned gases are blown out of the cylinder. After this stroke the next cycle starts from the intake stroke.

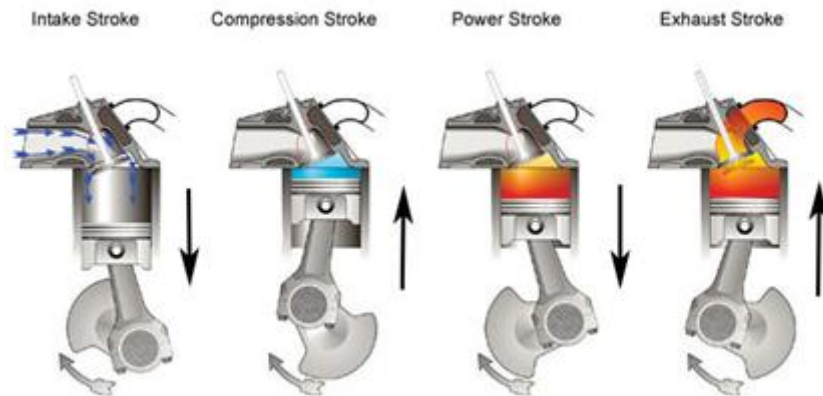


Figure 2-1 The working principle of a four-stroke SI engine.

An essential component to make the combustion engine work is the need of inertia. Since the power stroke only represents a part of the cycle, the rest of the combustion cycle is taking energy from the engine rather than supplying it from the same. Thus it is necessary to have enough energy stored in the rotating parts to open and close the valves, execute the gas exchange and in particular to compress the air with the piston. This is usually solved by a flywheel connected to the crankshaft, which also helps to smoothen out the output torque from the engine.

The diesel engine shares most of the technology with the gasoline engine. What differs is in particular the ignition process. Instead of having sparkplugs that initiate the combustion, diesel fuel is injected near the TDC and starts to combust on account of the high pressure and temperature with just a small delay from the short mixing period. This requires different properties of the fuel, since it in this case shall be easily ignited by pressure and temperature. That is the opposite from the gasoline engine, where it is important that the fuel is *not* ignited by high pressure and temperature.

The two-stroke engine has another system for gas-exchange than the four-stroke engine. But since the engine used in this thesis is a four-stroke, further information about the two-stroke can be found in other places, for example in *Combustion Engines* by Johansson et al [3].

2.2.1 SI- Engine

The four- stroke SI engine is the one that established the position of the combustion engine in our society that it has till this day, and probably for a long time ahead. It is this type of engine that Nikolaus Otto invented and patented in 1876; the reason to why it is sometimes referred to as the Otto engine.

The engine has a relatively simple construction and it can be cheaply mass produced, which makes it globally the most common engine for vehicles. [4]

2.2.2 CI-Engine

The CI engine was invented by the engineer Rudolf Diesel in 1892. This engine was, like Otto's, operated in four-strokes. The two-stroke CI engines showed up a few years later. The CI engine is in many aspects truly exceptional. It is reliable and is characterized by its long lifetime. It operates at high temperatures and pressures, which is beneficial in terms of thermodynamic efficiency. It doesn't need any high voltage ignitions system for the ignition sparks, which can interfere with navigation systems

and other applications. When the engine runs on part load, there is no need for an intake throttle that affects the efficiency due to pump losses that the SI engine has; it is only the amount of fuel injected that has to be decreased.

A CI-engine is often more expensive than a SI-engine due to the boosting and injection system. The modern injection systems are quite complex since the fuel is pressurized up to 2 000- 2500 bar, for the atomization process. The high compression ratio gives high peak pressures which in turn leads to dimensioning that is heavier and more expensive than a SI-engine of the same displacement. This is also the main reason why diesel engines have a lower maximum speed than SI engines. The piston, crankshaft and connecting rods are made heavier which limits the rotational speed due to the strength of the material. The high peak pressure also leads to higher mechanical losses. [5]

2.3 Emissions

The major drawback of a common diesel engine is the emission values. Since the fuel is ignited almost at the same time as it is injected, the mixing period is very limited, and the combustion is thereby very heterogeneous. This is shown in Figure 2-2. This means there is an area in the combustion chamber, that doesn't contain enough available oxygen for a proper combustion, and that results in unburned hydrocarbons and soot.

The emissions for heavy duty diesel engines that are regulated in law by the EU are the NO_x , CO, HC and soot [6]. What they all have in common is that they are affecting the human body and the environment in a negative way.

2.3.1 NO_x

One of the regulated emissions is the NO_x , which is a generic term for NO and NO_2 . There are different ways to form NO_x in a combustion engine, but it occurs mainly in high temperature zones with available air. The prompt NO_x is a gathering of different and complicated kinetics for how it can be formed in especially low lambda cases. Fuel NO_x is formed from the nitrogen that is a component in some fuels. High temperature is also a source for forming this particular emission, which can be referred to thermic NO_x .

NO_x contributes to acidification in nature when it reacts with water molecules in the atmosphere and forms nitric acid. This results in acidic rainfall which affects the forests and landscapes. The smog that is seen in many big cities, especially in warm climates are results from the NO_x . It causes bronchial and lung diseases like cancer bronchitis and asthma. High levels of smog also decreases the UV radiation that is necessary for the human body to form the vital vitamin D [7].

2.3.2 Unburned Hydrocarbons- HC

Hydrocarbons in the emissions means unburned or partially unburned fuel. It occurs mainly from four different sources:

1. Rich zones where the oxidation process is impeded from lack of available air.
2. Lean zones
3. Zones that are too cold leading to the combustion get quenched.
4. From the injector nozzle where the combustion cannot be kept.

Hydrocarbon emission is like NO_x also a contributor to the formation of smog [3].

2.3.3 Carbon Monoxide- CO

Carbon monoxide is formed as an intermediate component when the carbon in the fuel reacts with air to form the product CO_2 . The toxic gas CO occurs when the fuel partially reacts with oxygen but lacks enough time, temperature and oxygen to get a proper combustion. The amount of the available air has a strong dependence on the CO in the exhausts, and with a rich mixture (lots of fuel and little of air) there is almost no CO_2 formed from CO. For a common CI engine, the CO emissions are quite

moderate. But for modern CI engines, and especially in the engine concepts with low temperature and lean combustion presented later in this thesis, it becomes an issue [3].

2.3.4 Particulate matter - PM

Particulate matter is not a specific defined substance, but rather is a mixture of different non-gaseous emissions. Most of the particulate matter in the exhausts consists of soot, which is composed of carbon. It is formed in high temperature zones where there is a shortfall of oxygen. The available oxygen in these areas may only be enough to oxidize the hydrogen atoms, and leaves the carbon unoxidized. In a diesel engine, this is most likely to occur in the middle of the spray, inside of the diffusion flame. This and the other emission formations are seen in the Figure 2-2 below. A part of the PM can also be derived from lubrication additives and wear of the engine.

It is connected to health effects like lung cancer, respiratory and heart and blood vessels diseases [3] [4].

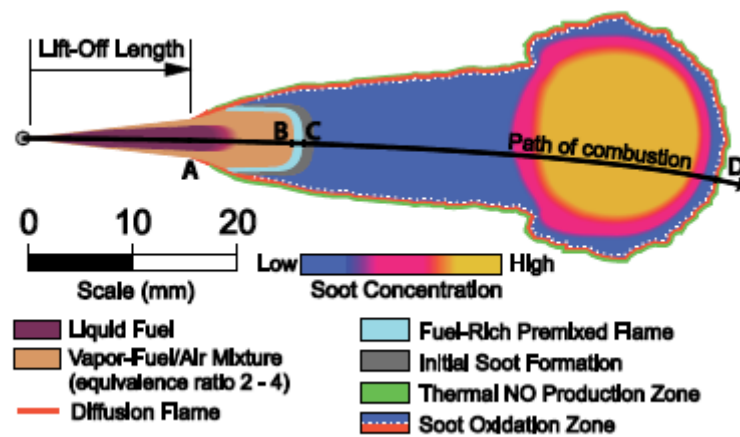


Figure 2-2 The formation of emissions from a diesel spray [6].

2.4 Advanced combustion concepts

The main advantage of the SI engine compared to the CI engine is the emission values. If a three way catalyst is used, the amount of NO_x , HC and CO emissions can be held at a minimum. This is due to the premixed fuel and air that is combusted homogeneously in the cylinder at $\lambda=1$.

Since the SI-engine is limited to a compression ratio of 10-12 because of the risk of knock, a compression ignition system is still desired since it has the potential to give a much better fuel economy. The correlation of thermal efficiency and the compression ratio is described in equation 1 below. Figure 2-3 is the equation (1) plotted and shows quite clearly how important a high compression ratio is. [4]

$$\eta_{Th} = 1 - \frac{1}{r_c^{\gamma-1}} \quad (1)$$

η_{Th} is the thermal efficiency, which is the efficiency of the transformation of heat in the cylinder, to work, on the piston. This is without doubt the most critical source of energy loss in a combustion engine. r_c is the compression ratio: the maximum volume in the cylinder divided by the smallest

volume when the piston moves up and down. The higher the compression ratio is, the longer expansion on the piston gets. γ is the specific heat capacity of the fluid, 1.4 for air and a bit lower for exhaust gases. The plot does show the thermal efficiency based on $\gamma = 1.4$ though.

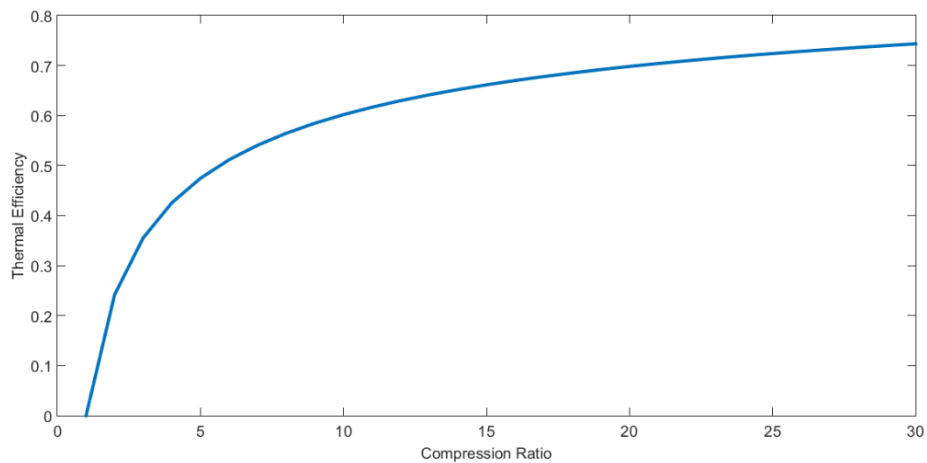


Figure 2-3 Thermal efficiency as a function of compression ratio.

So the conclusion is, in order to get good efficiency, the compression ratio must be high, but the combustion shall not create too high emissions. Solution for these demands requires new combustion engine concepts.

2.4.1 HCCI- Homogenous Charge Compression Ignition

In 1979 Onishi invented the strategy that today is referred to as HCCI, *homogenous charge compression ignition*, and his work is based on a two-stroke cycle. Onishi himself didn't use this appellation, instead he called the strategy *active thermo atmosphere combustion*, ATAC. In 1983 Foster studied this concept with a four-stroke cycle, and it was him who gave it the name that is used nowadays, HCCI. [8] This is a combustion strategy that is achieved by homogenous compression ignition. That means the fuel and air is well mixed before the ignition. HCCI has no flame front unlike the SI-engine; the oxidation, i.e. combustion starts everywhere almost at the same time. But just like in a CI engine, the engine lacks spark plugs and the combustion is initiated by just compression. For a stoichiometric combustion, the combustion would be hard to distinguish from just a knocking SI engine, and the engine would be destroyed in seconds. So the combustion must be slowed down, and this is usually done by high amounts of exhaust gas recirculation, EGR and a lean air/fuel composition. With the right amounts of excess air and EGR, the oxidation duration can last as long as for a SI engine. In Figure 2-4 below, the heat release in a HCCI engine is shown schematically and compared to the combustion in a SI engine.

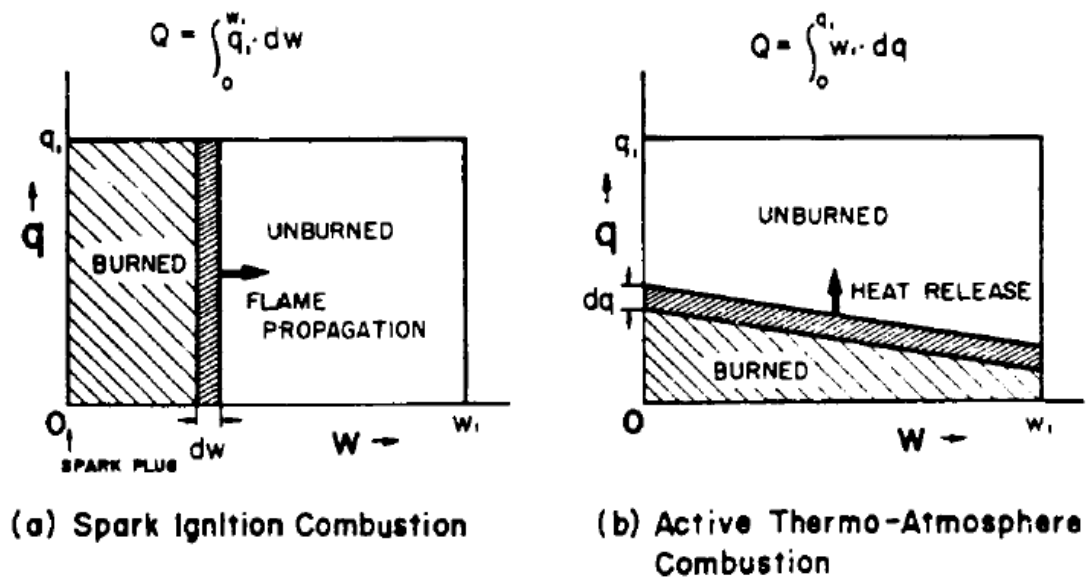


Figure 2-4 Combustion process a) in a SI engine and b) in a HCCI engine [8].

The description of the heat release in the SI engine (left picture) is rather straight forward. It takes place where the flame front is located and the arrow pointing right shows the flame propagation in the combustion chamber. The w indicates the distance the flame travels and the Q the total amount of the heat release. The right picture on the other hand shows that the combustion takes place everywhere at the same time (almost everywhere, since the line of heat release is not completely horizontal), but that the oxidation process is not infinitely short. The reason to why the heat release line is not totally horizontal is due to the fact that even in homogenous charge compression ignition, the mix is not totally homogenous, and that some parts of the combustion chamber is warmer than others. The combustion is initiated somewhere - and explains the leaning heat release line.

Something quite odd about the HCCI engines is that they are supposed to use a very lean mixture of fuel and air. It runs at a λ between 3 and 9, depending on the load. That means 3 to 9 times more air than necessary for a complete combustion. A SI engine with normal flame propagation would be impossible to run under these conditions. This flexibility in the λ value is a big advantage for the HCCI. Compared to the SI engine that uses a throttle on part load to limit the power, the HCCI can just lean out the fuel-air-mixture even further, and thereby avoid the pump losses that the SI engine experiences. And since the cylinder is filled with so much air, the temperature rise is quite low, and so are the heat losses.

To be able to run on HCCI mode, the temperature must reach levels high enough for the heat releasing reactions. The temperature at which the combustion starts is dependent on the fuel properties, the air/fuel composition, EGR-levels, and the cylinder pressure. To make the HCCI engine run properly, it is necessary to start the combustion very close to the TDC in the cylinder. The main parameters that can affect the temperature at the TDC are the compression ratio and the inlet temperature. An increase of any of them results in a higher temperature in the combustion chamber. This implies that the HCCI engine needs some kind of controlling system. To make sure that the combustion takes place at the right time, the system must have some kind of feedback from the combustion, and a capability to change the situation in order to reach the right temperature. According to Johansson the difference between too early ignition and misfire can be temperature differences less than 5°C. But even if the

controlling works well, the range of operation is still small, and this is probably the biggest drawback of the HCCI [4].

HCCI is the only combustion principle that lacks significant NO_x emissions. This is due to two reasons:

1. The fuel is highly diluted with excess air and EGR, which means that a given amount of fuel has to heat up a lot of mass. This makes the highest temperature relative to other combustion strategies very low.
2. The combustion is close to homogenous. That means there are no zones with exceedingly high temperatures that create a lot of NO_x . In a SI engine, the temperature close to the spark is about 400 degrees warmer than the coolest part of the combustion chamber, which makes the temperature distribution quite inhomogeneous. And for the diesel combustion, the temperature distribution is just like the rest of the combustion, very heterogeneous.

Regarding the emissions, the NO_x and PM are superb for being a combustion engine, but the CO and HC emissions can reach high values. This is due to the low combustion temperature and insufficient time to complete the oxidation of the fuel to CO, and further to the final state of CO_2 . The efficiency of combustion is only about 90-95 % for a typical HCCI engine [4].

2.4.2 PPC- Partially Premixed Combustion

Partially Premixed Combustion, PPC is a strategy of combustion that can be applied in a common CI-engine. The combustion process is something between HCCI and the combustion taking place in a common CI engine. The greatest advantages compared to the standard diesel engine are the better emission values and the higher thermal efficiency. The advantage compared to HCCI is the controllability, but also capability to reach even higher efficiency. This is due to the partly stratified combustion that makes the heat release a bit slower than the HCCI combustion. Theoretically, the best combustion should be infinitely short, giving all the force from the combustion to the piston in the best position of the cycle to get the most efficient work output. But a combustion that is too fast leads to high heat transfers and mechanical stresses, hence the high efficiency benefit can be lost. The combustion timing can be controlled by injection timing and when it comes to emissions, unburned hydrocarbons and carbon monoxide are much lower compared to HCCI. [9]

The history of this concept is not very old. The first known experiments was back in 2006, made by Kalghatgi [8] where the used fuel was diesel oil. Unfortunately the load range is limited in a PPC with a constant compression ratio. Working with diesel fuel and a compression ratio of 16, the load range is limited to about 5 bar IMEP gross. And in the upper limit of the load range, the amount of EGR starts to be unacceptably high according to Manente [8]. What was later found, was that gasoline like fuels are more suitable than the diesel fuels for the PPC, thanks to the higher resistance to auto ignition. [10]

In the process the fuel is injected near the TDC and is allowed to mix with the air before it ignites by the high pressure and temperature in the cylinder. In this concept it is possible to keep separation between start of combustion (SOC) and end of injection (EOI), but the definition is the region between truly homogenous combustion (HCCI) and diffusion controlled combustion (diesel). [11]

To visualize the different combustion strategies that are mentioned they are shown in Figure 2-5. As a function of SOI, the emissions of NO_x and HC are shown, and that the PPC strategy is suitable for a tradeoff between NO_x and HC emissions.

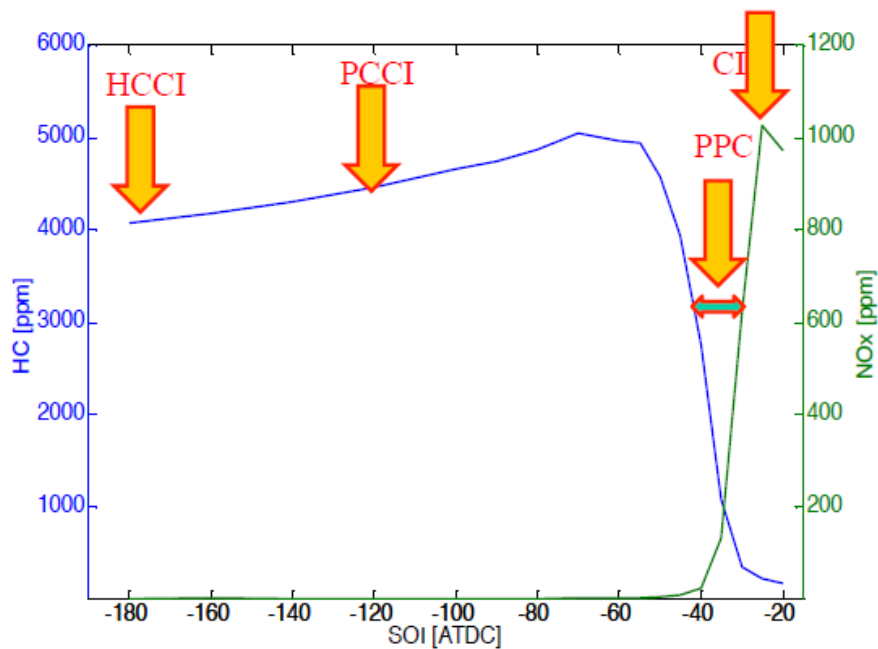


Figure 2-5. HC and NO_x as a function of SOI for different combustion strategies. [11]

To be able to achieve a positive mixing period, there are a few tricks to modify the conventional diesel cycle to PPC. The common method is to have high percentage of cooled EGR. That will lower the temperature in the combustion chamber and delay the combustion process and will also slow down the kinetics of the process. The EGR is commonly defined as the concentration of CO₂ in the intake divided by the CO₂ concentration for the exhaust gases. At EGR fraction of 60% the formation of soot is prevented. Unfortunately, this high rate of EGR also prevents the oxidation of soot, CO and HC, since the temperature in the combustion chamber is not enough, to fully oxidize the fuel. [12]

The later injection timing of PPC is preferred compared to the HCCI or PCCI (premixed charge compression ignition) since it gives an extra degree of freedom for the combustion strategy. In other words, the injection timing could be adjusted with different loads and speeds in order to keep the combustion partially premixed without exceeding any critical values, like the maximum cylinder pressure or pressure rise rate. The injection can also be adjusted in order to find the best tradeoff between HC and NO_x emissions, which is illustrated in Figure 2-5.

Another method to delay the combustion is to use fuels that have lower cetane number (CN), making them more resistant to auto ignition than diesel fuels. Since it delays the ignition, it will also improve the mixing. The level of EGR must be adjusted when another fuel is used, and the rule for CN and level of EGR is: Lower CN requires a lower level of EGR [13].

2.4.2.1.1 The PPC combustion

Since the work output from a combustion engine is derived from heat release, the work is correlated to thermal efficiency. The part of heat that is not transformed to work either ends up as heat flux through the cylinder, or escapes with the exhaust gases. One attempt to decrease one of the two heat losses could be to delay the combustion and thereby decrease the time for the high temperature gas

to dissipate through the cylinder walls. This could be a method that would decrease the heat flux and thereby increase the efficiency. Unfortunately, the less amount of heat that is transferred through the cylinder walls is very likely to escape through the exhaust system instead, and no extra work output gain would be achieved. The same thing is true for the opposite strategy, an earlier combustion for lowering the exhaust heat losses, would increase the heat flux. This implies that this is just a tradeoff. Regarding the heat losses, the only thing that is actually decided in terms of combustion timing, is where the losses should take place [4].

To decrease the heat losses in the cylinder there is a point to look at the temperature distribution in the combustion chamber. Fridriksson et al made useful CFD- calculations on the distribution of fuel and temperature in a diesel and a PPC-cycle which are seen in Figure 2-6 and Figure 2-7. These show the distribution of temperature with a non- isolated cylinder. The two combustion chambers are not entirely comparable though, since the simulated hardware differs a little. For instance, the compression ratio of the common CI engine is 16.8 and for the PPC it is only 15.5. The load in the CI is also higher, 26.2 compared to the PPC which is only 20.8.

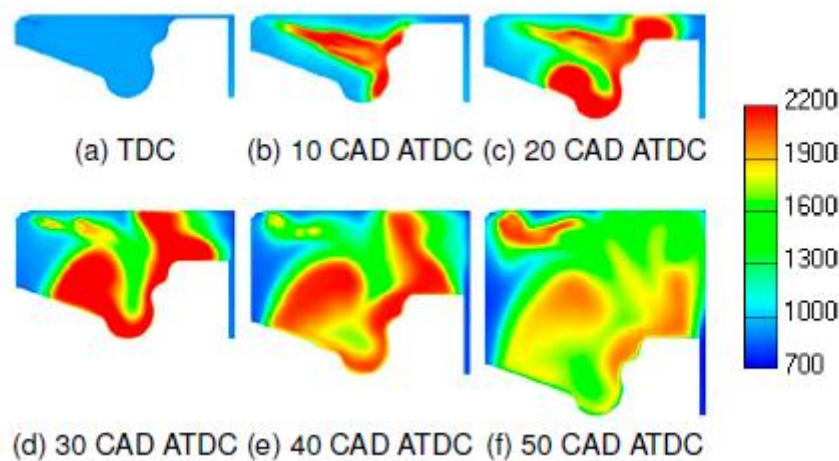


Figure 2-6. Temperature distribution [K], in the combustion chamber in a CI-engine at different crank angles near TDC. [14]

The blue part in picture (a) in the figure represents a part of the combustion chamber, and the white part is the piston, with the typical bowl of a diesel engine. In picture (b) the fuel is injected, and the immediate combustion of it shows an increased temperature.

What is seen is that the temperature distribution is very inhomogeneous and the temperature ranges from below 1000 K to well above 2000 K. Except the fact that this will result in a lot of different kinds of emissions, the highest temperatures are distributed close to the cylinder walls. This leads to high heat fluxes that are major source of energy losses in a combustion engine.

In Figure 2-7, PPC combustion is shown and it is obvious that the temperature distribution is not homogenous, but much closer to be so, than for the common diesel combustion. The maximum temperature is below 2000 K, and the warmest regions are not located near the boundaries of the combustion chamber, i.e. the liner, cylinder head and the piston.

Since the combustion duration is shorter in PPC than in a standard CI, the heat transfer decreases, which contributes to a potential of higher thermal efficiency. Figure 2-7 also shows that the temperature is lower during combustion, especially near the boundaries of the combustion chamber which leads to lower heat flux. These are the main reason for the possibility to get very good efficiency values for PPC.

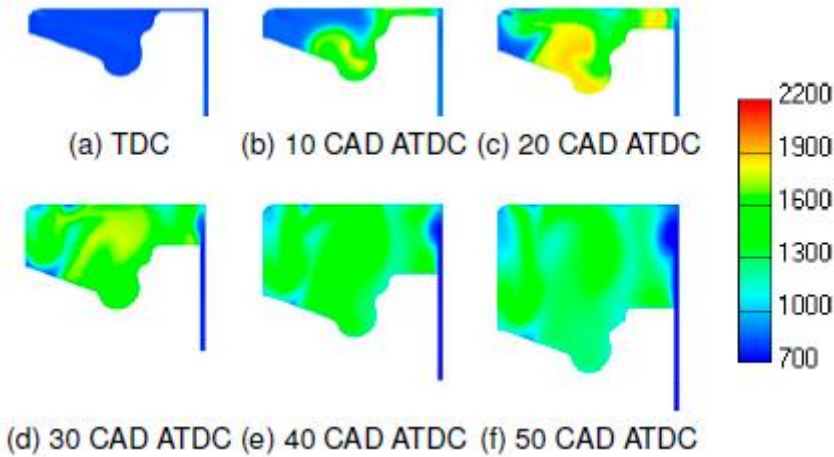


Figure 2-7 Temperature distribution in the combustion chamber at PPC combustion. [13]

3 Methanol

3.1 Why methanol?

The emissions from the SI engine are well mitigated since the use of the three way catalyst (TWC) has become standard. This requires stoichiometric combustion, which has been pursued in CI engines for a while. A big obstacle for achieving stoichiometric combustion and get good emission results is the formation of soot when conventional fuels are used, like the Swedish diesel fuel MK1. [15]

Methanol is the simplest alcohol with the chemical formula CH_3OH that tells there is just a single carbon atom in each molecule, and four hydrogen atoms. Every molecule also contains one atom of oxygen, which implies efficient combustion. That means low numbers of unburned hydrocarbons, CO and practically no soot. Even the amount of NO_x is generally low, which is a result of the low flame temperature [16].

Methanol is biodegradable and is quickly absorbed in water. It has a huge benefit compared to other fuels regarding its production flexibility. That means it can be obtained from different sources in different ways. One way is to extract it from wood, and that is why it is known as wood alcohol. Today most of the methanol is produced from natural gas, but it can also be obtained from biomass, coal, waste, but also from the CO_2 pollutions from power plants [17] [18].

Compared to the usual fuels like gasoline and diesel oil, methanol has a high heat of vaporization. Since the heat value for fuels are defined with calorific methods, meaning how much a specific amount of water is heated with a specific amount of the fuel, nothing is said about the combustion process [19]. A fuel like methanol that has a very high heat of vaporization compared to the lower heat value (LHV) can be more useful regarding energy than the number the energy content tells. In other words, the work output for every engine cycle gets higher with methanol, if it is compensated for the fuels heat value, compared to many other fuels. That also implies higher efficiency since a greater amount of work can be obtained for the same input of total fuel energy. This is explained in Figure 3-1.

If Q is the mass of fuel times the LHV, and W is the work done, the efficiency η is defined by equation (2).

$$\eta = \frac{W}{Q} \tag{2}$$

Table 3-1 shows the comparison of LHV and heat of vaporization for gasoline, diesel fuel and methanol. The ratio between these are shown in the bottom tuple, and is in this context the most interesting parameter.

Table 3-1 Heat values for three different fuels.

	Diesel Fuel	Gasoline	Methanol
LHV [kJ/kg]	42 450	43 960	19 930
Heat of vaporization [kJ/kg]	267	330	1098
Latent Heat/ LHV [%]	0.63	0.75	5.5

In Figure 3-1 below the lighter gray area is the compression work, and the expansion work represent the same plus the darker gray area. The area of the banana shaped figure represents the difference

between compression work and expansion work, in other words- the energy output from one cycle. The methanol is injected in the end of the cycle to represent PPC injection strategy. The cooling effect from the vaporization leads to a relative temperature drop during the compression phasing, and since the pressure follows temperature, the compression work decreases.

The proportions of this extra work output area is exaggerated, but should do the work as an illustration.

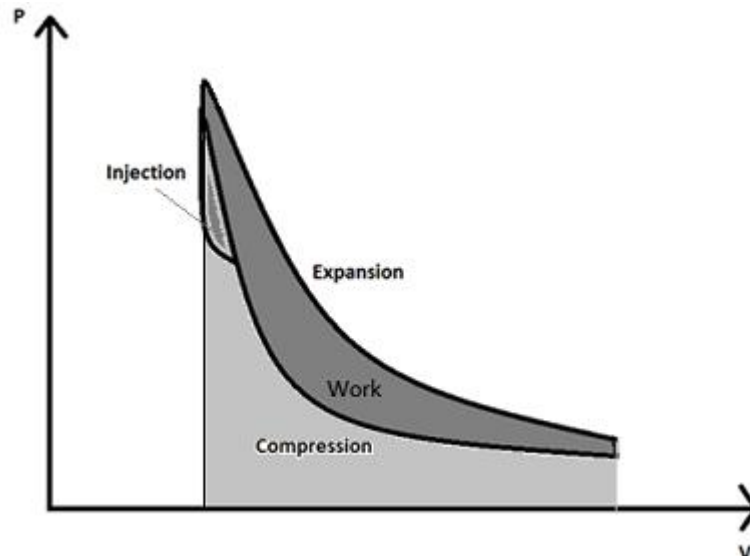
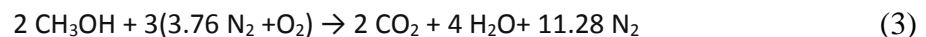


Figure 3-1. Pressure and volume diagram of combustion cycle of methanol.

Having this high heat of vaporization is not always an advantage in engine operations. It is also the main reason why methanol and alcohols in general are resistant to cold start. [1]

The methanol combustion in air follows the reaction formula (3).



The air-to-fuel mass ratio for stoichiometric combustion when methanol is combusted in air is 6.45. This is a remarkable low value compared to other fuels. Gasoline and diesel fuel exhibits an air-to-fuel ratio between 14.5 and 15, and pure hydrogen a ratio of 34. This means a lot more fuel can be injected in the cylinder for a given amount of air. The main difference is the oxygen content in the methanol molecules leading to this property [20] [21].

The fuel has a great potential for becoming a major contributor to the fuels for vehicles, but it is also associated with some drawbacks. The flame burns with an almost invisible color, which in some contexts can cause great danger, since no actions may be taken to an ongoing fire. It is also toxic for the human body. If it is consumed, one of the products from the digestion is formic acid, which has such a low pH-value that it causes blindness, and eventually death. The methanol is also corrosive and can be a big problem for many plastics and metals, when for example gasoline is changed to methanol and affects different devices.

4 Experiments

4.1 The engine setup

The experiments were made with a modified Scania D13 in-line six cylinder heavy duty engine. This is seen in Figure 4-1 below. As can be seen from the picture, only the cylinder head for cylinder 1 and 6 are mounted on the engine. For the PPC experiments the engine operations took place in the left cylinder, closest to the electric machine, and it is marked with the number 3.

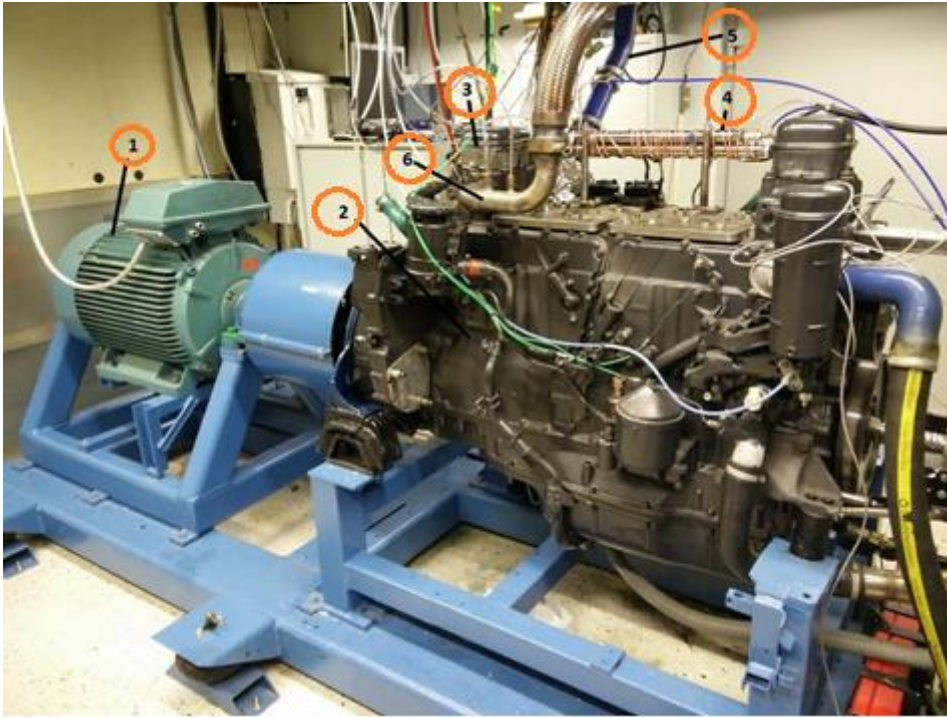


Figure 4-1 The engine rig in the motor lab. 1: The dyno. 2: The Scania D13 engine. 3: The cylinder head for the working cylinder. 4: The common-rail bar with a copper tube wound around for cooling of the fuel after the high pressure pump. 5: Silicone tube for the inlet air. 6: Exhaust pipe.

Since the engine is modified to only operate on one cylinder and the other five pistons in the non-combustion cylinders are just motored, the expectation of friction losses are higher than the specific friction losses of an engine that would have been operated on all 6 cylinders. For that reason, the indicated load is more important in the following experiments than the torque output, i.e. the brake mean pressure. So for the following experiments a torque meter is therefore not used. Engine properties are shown in Table 4-1.

Table 4-1 Engine specifications.

Displaced Volume [cm³]	2124
Bore [mm]	130
Stroke [mm]	160
Connecting rod [mm]	255
Compression Ratio	15
Swirl Ratio [-]	2.1
IVC [CAD BTDC]	141
EVO [CAD ATDC]	137

4.1.1 Pressure control and EGR

In the lab there is a central system for compressed air of about 10 bar. The inlet pressure to engine is controlled by a valve that is mounted before the blue silicone tube that is seen in Figure 4-1 above. The backpressure is also controlled by a valve, and in order to introduce EGR, and also to simulate a turbo charger, the value of this must be kept higher than the inlet pressure.

4.1.2 Measuring systems

The engine is connected to an electric machine that has two main functions. It works as an electric starter motor and also as a dyno. When the dyno is given a set value for the speed, it keeps this speed constant independent on the load of the combustion engine. This means that the load can be increased without any acceleration of the engine speed.

The different emission products: HC, CO, NO_x and CO₂ emissions were measured with a Horiba measurement system MEXA-9100EGR. The λ values used in the results are based on the emission data.

With different sensors it was possible to measure a lot of parameters which are of interest for engine operation. Following parameters were all used:

- In-cylinder pressure
- Air inlet temperature
- Inlet pressure
- Exhaust pressure
- Injection pressure
- injection duration
- Fuel temperature
- Oil, water and cylinder head temperature
- Emissions

4.1.3 The fuel flow system

To be able to calculate the engine efficiency, it is necessary to know how much fuel that is injected. To get this information there are a number of solutions, to which a fuel flow meter might be the most intuitive one. In this test cell a fuel scale was used instead. The advantage compared to a flow meter is the reliable and accurate results - at a low cost. The drawbacks are that the fuel cannot be supplied continuously from the fuel storage to the "scale tank". It must be filled repeatedly, and cannot provide useful data during that time. It also demands the operator to keep an eye on the scale over time since the tank cannot, in any moment totally run out of fuel, since it would cause the high pressure pump to run without fuel. That affects the efficiency of the experimental work, or in other words, takes more time, and focus from not essential items.

Figure 4-2 describes this experimental setup for the fuel system. The red tube that is marked (2) in the figure is the fuel supply line to the system. This is connected to a pressurized 200 liter barrel in the fuel storage, and is manually handled with a tap (1) to fill up the 10 liter plastic can with 99.85 % pure methanol fuel (3). From this can the fuel is pumped via the tube marked (4) to the metallic “scale tank” (5). The weight of this is measured by the scale from which it is hanging (6). This scale provides real time measurement of the weight of the fuel. (7) is the circulation pump that provides the engine with fuel from the scale tank.

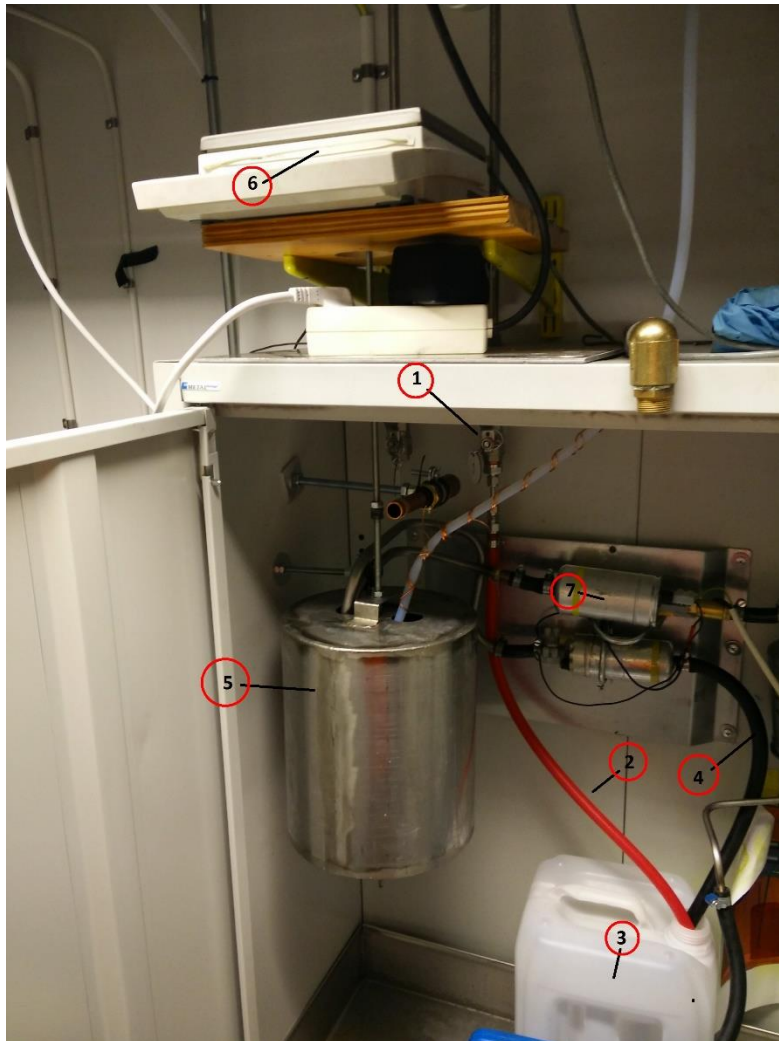


Figure 4-2. The fuel supply system. 1: Tap to open the fuel flow from the pressurized fuel supply line. 2: The connecting tube from the fuel line to the plastic intermediate storage. 3: Plastic container that is used as an intermediate storage of fuel. 4: Tube in which the fuel goes through when the pump is manually operated to fill up the scale tank. 5: Fuel tank connected to a scale from which the engine is supplied. 6: Scale that is used to get real time measurement of the fuel consumption. 7: Circulation pump that provides the engine with fuel from the scale tank.

4.1.4 The piston

The contours of the used piston that resulted in a compression ratio of 15:1 is shown in Figure 4-3.

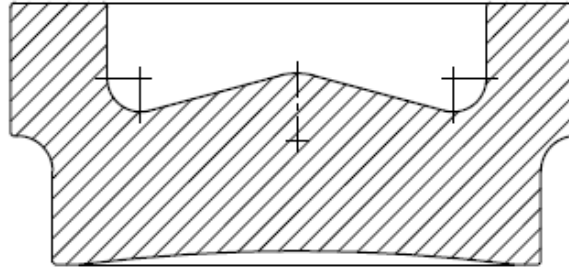


Figure 4-3. The contours of the piston used for the experiments. It gives a compression ratio of 15:1 in the used cylinder head.

4.1.5 The injector

The injector that was used is an 8 holes 207 PPH (pounds per hour) that was suited for alcohols. The first sweeps were made with an injector with the same specifications, but not suited for methanol. However the fuel flow properties were still the same.

4.1.6 Execution

Before every experiment, the emission analyzer was calibrated with a calibration gas before the measurements were conducted, to secure as high reliability as possible for the obtained emission values. These are not only interesting in terms of the amount of pollutants that the engine produce, they are also used to calculate the combustion efficiency.

To secure as reliable values as possible, every point was saved in a steady state condition, meaning no variation over time in engine temperature, inlet temperature, inlet pressure and exhaust pressure. This required a certain time for the engine to warm up, and as the engine was running, time to stabilize after every change of any parameter, before a point was saved.

The operation points were saved with 300 cycles, to which the heat release were based on, from the in-cylinder pressure. The heat release equation (4) follows the description in Heywood [22]

$$\frac{\partial Q}{\partial \theta} = \frac{\gamma}{\gamma - 1} P \frac{\partial V}{\partial \theta} + \frac{1}{\gamma - 1} V \frac{\partial P}{\partial \theta} + \frac{\delta Q_{losses}}{\delta \theta} \quad (4)$$

4.2 Theory

The auto ignition temperature is different for each value of the in cylinder pressure. This can be described by Figure 4-4 which shows at what temperature and pressure hydrogen reacts with oxygen at stoichiometric conditions. To the right side of the explosion limit, the reaction is violent.

For hydrocarbons like methanol the shape is slightly different, but the principle is the same [23].

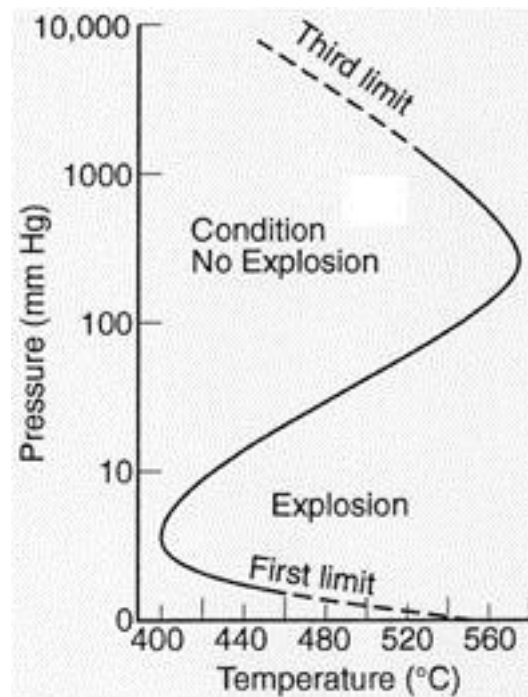


Figure 4-4 Explosion diagram for hydrogen and air under stoichiometric conditions. The relation between mm HG and Bar is 0.00133, meaning 10 000 mm HG =13.3 bar. [35]

To match the conditions in the experiments, apart from the different fuel, the third limit of the explosion line must be extrapolated, since the pressure in a cylinder after compression is almost an order of magnitude higher than the highest pressure shown in the diagram in Figure 4-4. At off-stoichiometric conditions the explosion line is also affected.

4.3 Test plan

In this thesis the load range was defined by two limits. The upper limit was set by the maximum *pressure* or the maximum *pressure rise per CAD* in the cylinder, depending on which occurred first. The lower limit was set to the point where the combustion was still stable, which was defined by a low value of the coefficient of variation (COV) of IMEP, and where the CA50 occurred in a satisfactory early stage. This was mainly due to three reasons: to get a fair comparison between different operating points when the lowest load was sought. The efficiency is negatively affected with a too late combustion, and that the combustion could not be kept stable with later heat release, especially for the lower pressure cases. The specific values of the limits are shown in Table 4-2.

The COV is a mathematical measurement of the variation, using statistic data. In this thesis it is used to describe how big the variation from cycle to cycle is regarding IMEP gross. A low number means that the engine runs stable and with low cycle to cycle variations in indicated load.

CA50 is a measure of where the heat release takes place. This is a common parameter used in engine diagnostics and is defined as the piston angle ATDC for which 50 percent of the heat has been released in the combustion.

The maximum cylinder pressure of 220 bar was set to save the engine from damage. The highest value for $dP/dCAD$ was set to save the pressure sensors, which are sensitive to violent pressure increase. It also prevents vigorous vibrations and engine noise to have a reasonable low pressure rise limit.

In Table 4-2, the limits of the investigated load range are shown.

Table 4-2. General limitations for the experiments conducted.

	P_{max} [bar]	$dP/dCAD$ [bar/CAD]	COV_{max} [-]	CA50 [CAD ATDC]
Upper load limit	220 bar	40	-	-
Stable Combustion				
Lower load limit	-	-	5	10

To achieve combustion it is necessary to have high pressure and temperature due to the methanol's resistance to auto ignition, especially for low load. Therefore an electric heater was used to heat up the inlet air, to increase the temperature in the combustion chamber.

4.3.1 First Experiment – Find the inlet temperature for HCCI

In the first experiment different points regarding inlet pressure and inlet temperature were investigated in order to achieve combustion. The operation was in premixed mode (HCCI), meaning early injection timing, SOI at 140 CAD BTDC. The idea was to use the obtained values to define the PPC operation range in the following experiments.

Since a homogenous charge is the hardest condition for auto ignition for a given amount of fuel regarding the temperature, the obtained values from this experiment were meant to be the temperature for the lowest load in PPC mode, where the highest inlet temperature is used.

The amount of fuel injected was decided by the predetermined λ value. Since no EGR was used in the experiments, the combustion process is reluctant to being slowed down. Simple pre studies of the engine behavior with methanol lead to a choice of lambda 5. This rather high value was chosen since richer blending lead to quite aggressive heat releases when the inlet pressure was increased. Priority was given to investigate the whole range from 1 to 3 bar in inlet pressure, and as a result this high value was chosen. The used values are compiled in table 4-3.

Table 4-3. The conditions for the premixed combustion experiment.

P_{in} [Bar]	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
$P_{injection}$ [Bar]	1000	1000	1000	1000	1000	1000	1000	1000	1000	1000	1000
λ [-]	5	5	5	5	5	5	5	5	5	5	5
SOI [BTDC]	140	140	140	140	140	140	140	140	140	140	140
T_{in} [°C]	Determined through experimental evaluation										

4.3.2 Second experiment- Varying load during constant inlet temperature and pressure

A specific inlet temperature and pressure was chosen from the results of the first experiment for further investigation of the ignition temperature when the load was varied. This experiment was conducted in PPC mode.

The only parameters that were changed was the injection duration, common rail pressure, and SOI. The SOI timing was the only parameter changed in order to keep the pressure rise rate and the maximum cylinder pressure down.

SOI at 23 CAD BTDC was chosen because this turned out to be at a point where the lowest load could be found, i.e. a point where the combustion was first started for a given amount of injected fuel. That is due to a good compromise between the time needed to vaporize the fuel before combustion, and the level of stratification desired, to keep a low local lambda – which is beneficial for starting the combustion.

4.3.3 Third experiment – PPC load range for different inlet pressures with a varying inlet temperature

The experiment was started in the upper temperature end, meaning starting with lowest possible load at the temperature obtained in HCCI mode from experiment 1. As the load was increased, the temperature was decreased as the combustion was continuously ongoing till any of the load range limits were crossed. The load range limitations are set to the ones expressed in Table 4-2 in the first paragraph of this chapter.

The load was primarily adjusted by the injection duration, and when it reached a value of about two milliseconds, the common rail pressure was increased, and the injection timing was instead held rather constant. This was done in order to keep the separation of injection and combustion. An earlier start of injection caused too high pressure rise rates, so the only solution was to keep the injection time shorter by higher injection pressure.

Since the lowest load occurs at a late combustion timing (with a constant temperature decided from experiment 1), the lowest “approved” load occurs at 10 CAD ATDC. An increase of injected fuel advances the injection timing and rises the rate of heat release per CAD. When the

The SOI was adjusted to make sure that CA50 occurs rather at the same CAD as the load, expressed in bar. For example, 5 bar IMEP should result in CA50 at 5 CAD ATDC. This is due to the requirement for the maximum pressure in the cylinder, which makes it unsuitable to have a high heat release too near the TDC. At the lowest load point it is set to around 20 CAD BTDC, depending on where the combustion first starts, at a given inlet temperature, pressure and amount of fuel injection.

Table 4-4 Operation plan for the third experiment.

P_{in}[Bar]	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00
P_{injection}[Bar]	1000	1000	1000	1000	1000	1000	1000	1000	1000
SOI [BTDC]	20	20	22	18	20	20	23	23	23
T_{in} [°C]	Starts from the temperature obtained in the first experiment								

4.4 Expected results

4.4.1 First experiment

If the process of compressing air and fuel in the cylinder is considered adiabatic, meaning no heat exchange with the surroundings before combustion, the temperature is increased in accordance with the equation of adiabatic compression (9) below.

$$T_2 = T_1 \frac{V_1}{V_2}^{\gamma-1} \quad (4)$$

T₂ is the temperature at the start of combustion, T₁ the inlet temperature, V₁ the cylinder volume before compression and V₂ the cylinder volume at start of combustion. γ is the specific heat for air and it has a value of 1.4 kJ/(kg*K).

This equation tells that the temperature in the cylinder after compression but before start of combustion is only a function of the inlet temperature, not the inlet pressure. So for a given compression ratio, the temperature after compression is only dependent on the inlet temperature.

Regarding the inlet temperature and pressure correlation, a decreasing temperature should correspond to a higher inlet pressure, according to Figure 4-4. The range of inlet pressures are most probably within the range where a higher inlet pressure requires a lower inlet temperature, but the correlation is unknown.

Since lambda is kept constant, and the inlet pressure is increased, meaning more air in the cylinder, the result must be a higher amount of fuel injection. Consequently, the load is expected to increase with an increase of the inlet pressure.

4.4.2 Second experiment

The expectations from the experiment regarding constant inlet pressure and inlet temperature is that the upper load limit will be decided by the pressure rise rate, or the maximum in cylinder pressure. The lower will be decided as the limit where COV exceeds the value 5, or CA50 exceeds 10 CAD.

4.4.3 Third experiment

The lowest temperature where the combustion starts in PPC mode, is expected to be lower than in the first experiment in HCCI mode. This is due to the PPC has richer zones where the combustion is initiated. The pressure rise rate is expected to decrease with a lower inlet temperature, making it possible to extend the load range by increased load, without exceeding the pressure rise rate limit. A lower inlet pressure causes the cylinder temperature to get lower after compression which in turn slows down the reaction when the fuel starts to combust.

4.4.4 Comparison to other results

The four figures in Appendix A show the results from a sensitivity analysis made with the same engine, but with gasoline fuel. These figures are used as a tool to predict and analyze the results of the methanol experiments conducted in the project for this thesis.

The sensitivity analysis is based on a varying inlet temperature and injection timing. The x-axis shows the SOI for all the four figures, and the y-axis show indicated efficiency, specific NO_x, specific HC and specific CO emission.

5 Results

5.1 Required inlet temperature for different inlet pressures

The required inlet temperature is in this case defined as the inlet temperature for which the premixed fuel starts to combust after compression, for a given inlet pressure. This correlation is shown in Figure 5-4.

To obtain a result from experiments, as few parameters as possible should be changed to be able to see a good correlation between the changed parameter, in this case the inlet pressure, and the function, the inlet temperature. The choice of keeping the CA50, λ and the COV constant was due to the similarity in the combustion. A constant lambda means there is same proportions of fuel and air, as the amount of air intake increases with the inlet pressure. That means more fuel injection and thereby an increase in load. The tendency with an increase of pressure and constant lambda was an advance in ignition timing. This was avoided by the decreasing temperature, so the CA50 could be held constant at around 5 CAD ATDC. COV of IMEP was held rather constant, even though it appears to be a result of the timing of CA50.

The load was increased as the inlet pressure is increased. This was done in order to get stable and reasonable early heat release, but it is also a result of the lambda that was held constant.

Table 5-1. The experiment plan containing values of the in-parameters and the resulting inlet temperature in HCCI mode.

P_{in}[Bar]	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
P_{injection}[Bar]	1000	1000	1000	1000	1000	1000	1000	1000	1000	1000	1000
λ [-]	5	5	5	5	5	5	5	5	5	5	5
CA50[ATDC]	4	4	4	4	4	4	4	4	4	4	4
SOI [BTDC]	140	140	140	140	140	140	140	140	140	140	140
T_{in} [°C]	155	152	149	144	139	134	131	127	123	119	115

Since it is experimental results presented in this chapter, the values of λ , CA50 and COV are not perfectly precise. The imperfections from the experiments are shown in Figure 5-2 to Figure 5-3. This should not vastly affect the desired result from this experiment, the inlet temperature required to get a stable combustion.

The heat releases with the corresponding cylinder pressures of all different inlet pressures are shown in Figure 5-1.

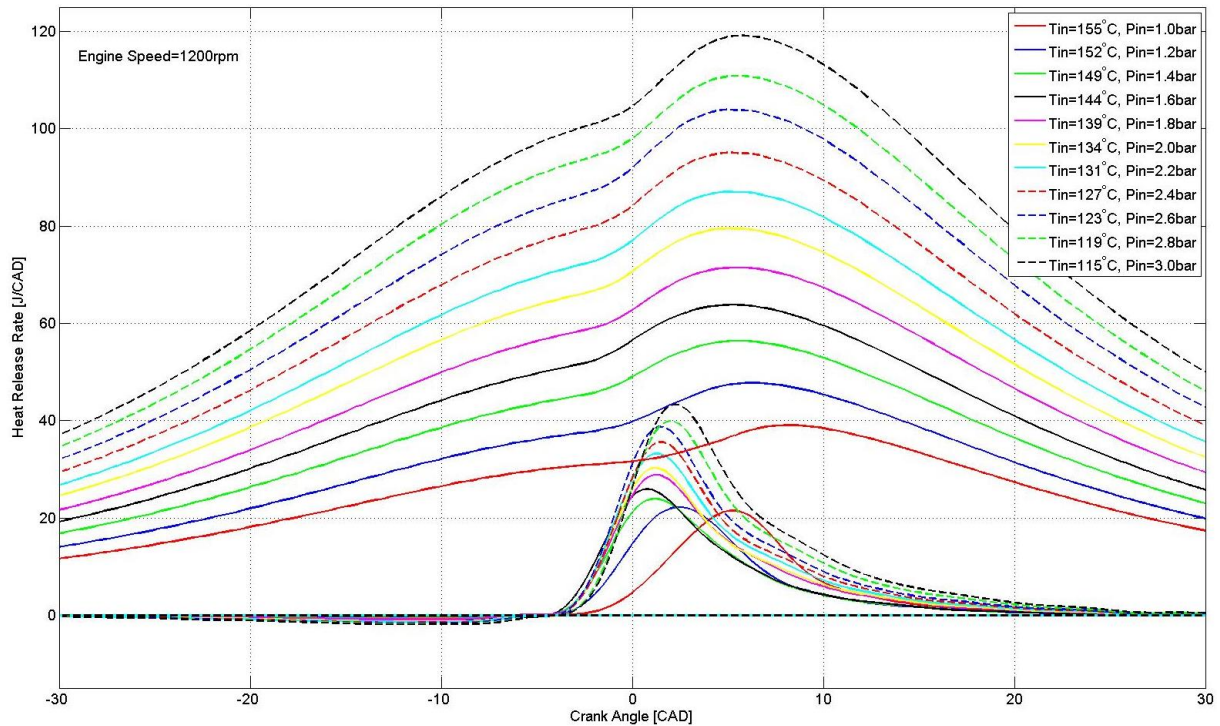


Figure 5-1. Heat release rate and cylinder pressure as a function of crank angle for different inlet pressures.

Priority was given to keep the λ value and CA50 as constant as possible. But the system was quite sensitive and the post process showed that the first point at 1 bar inlet pressure exhibits too low value of λ and too late CA50. The rest of the values are rather stable and differs only slightly from the set values. A constant λ also means that the amount of fuel, and the load should be increased in the same manner, which is clearly seen in Figure 5-2.

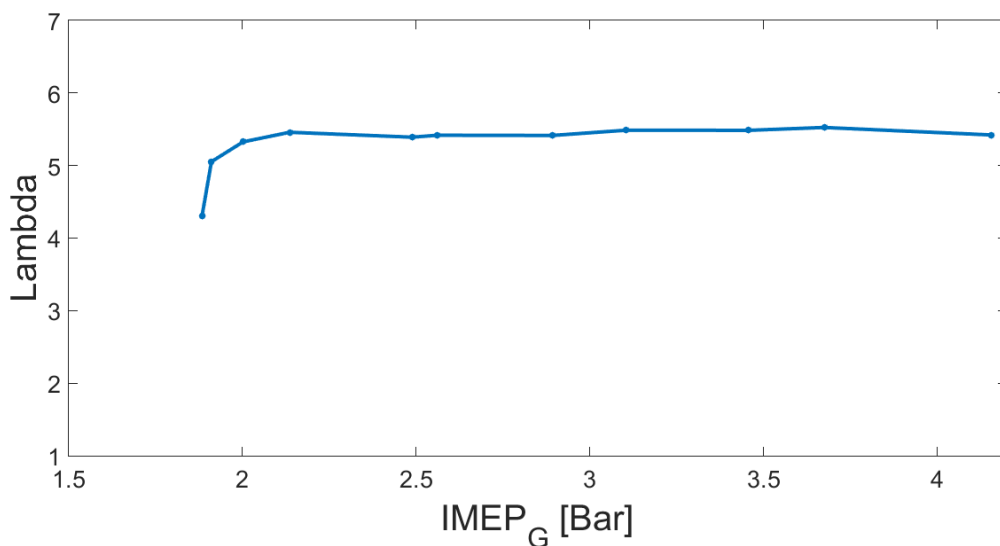


Figure 5-2. Lambda as a function of IMEP.

CA50 was the result of a continuously decreased temperature. When it reached the desired value of 4 CAD, the operating point was saved. The cycle to cycle variations implies that it is hard to get perfect results from the experiments. Even though the CA50 was meant to be constant independent of the inlet pressure, there was a trend of increasing CA50 as the inlet pressure is risen, apart from the first

few points. The situation changes as the 300 cycles are being saved, so the result does not apply perfectly to the situation when the saving process was started.

What is seen in Figure 5-3 is that the combustion in the lowest pressure point occurs a little late compared to the other points. In the meantime Figure 5-2 shows a little too low value of the

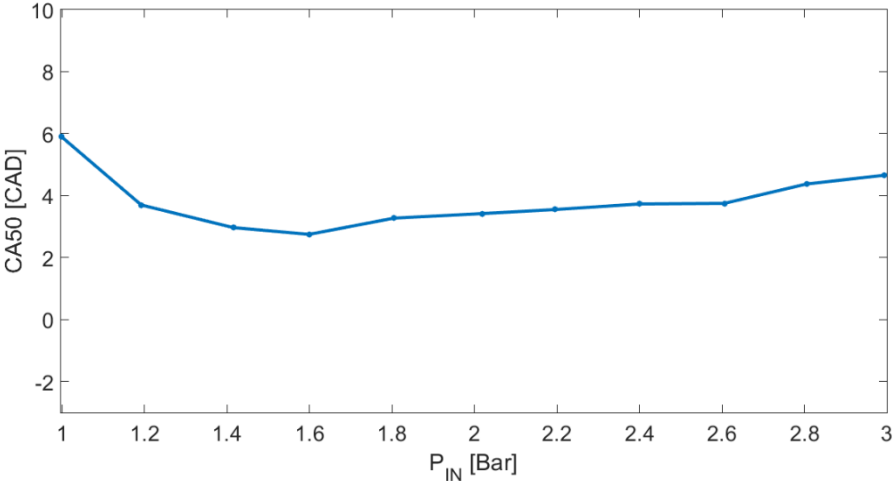


Figure 5-3. CA50 as a function of inlet pressure.

corresponding λ value in the same point. This is a result of imperfections in the execution of the experiment. That being said, it does not affect the following experiment too much if the temperature is adjusted slightly upwards for the lowest inlet pressure point compared to the result from this experiment.

From the equation tool in Matlab the linear equation in the range between 1 and 3 bar was defined by

$$y = 21x + 180 \tag{5}$$

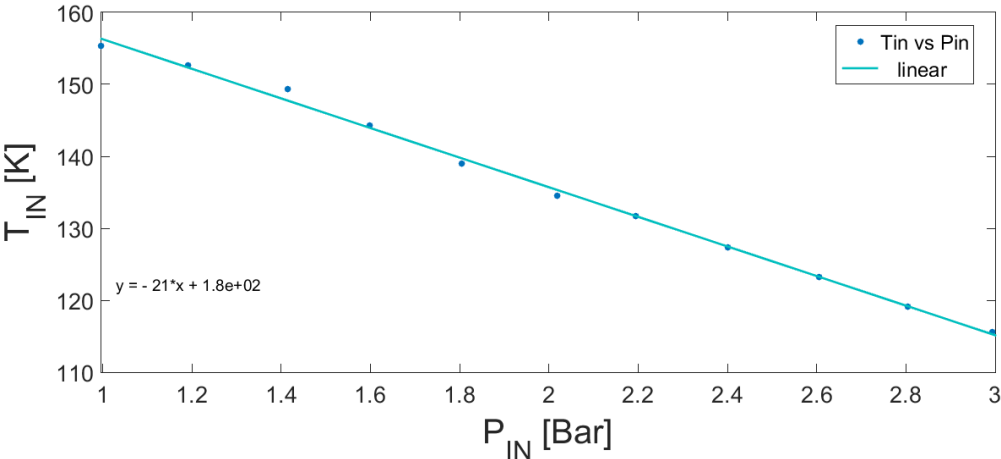


Figure 5-4. Inlet temperature as a function of inlet pressure

meaning the required inlet temperature decreased by 21 °C/bar, in inlet pressure. Even though the input values are not perfect, Figure 5-4 shows a result with good linearity. The value of the residuals are just above 2, when a linear line was inserted. This shows that the range of operation is within an area where the gradient of the explosion line is proportional to the cylinder temperature, seen in Figure 4-4.

Constant λ during increase of inlet pressure resulted in an increase of load with good linearity, seen in Figure 5-5.

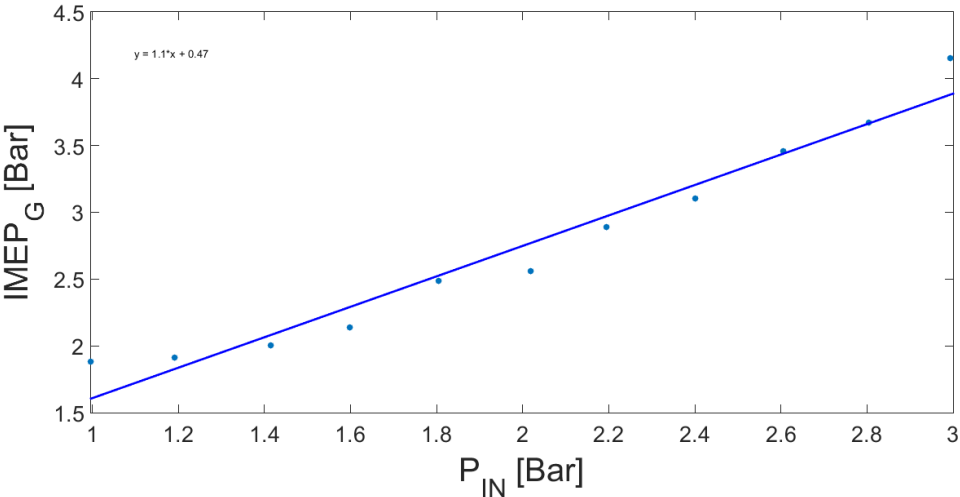


Figure 5-5. IMEPg as a function of inlet pressure.

The equation of the trend line reads

$$y = 1.1x + 0.47 \tag{6}$$

meaning that the load increases by 1.1 bar for every 1.0 bar increase of inlet pressure. To confirm it is the fuel that starts to combust at a lower temperature as the pressure goes up, Figure 5-6 and Figure 5-7 should be studied.

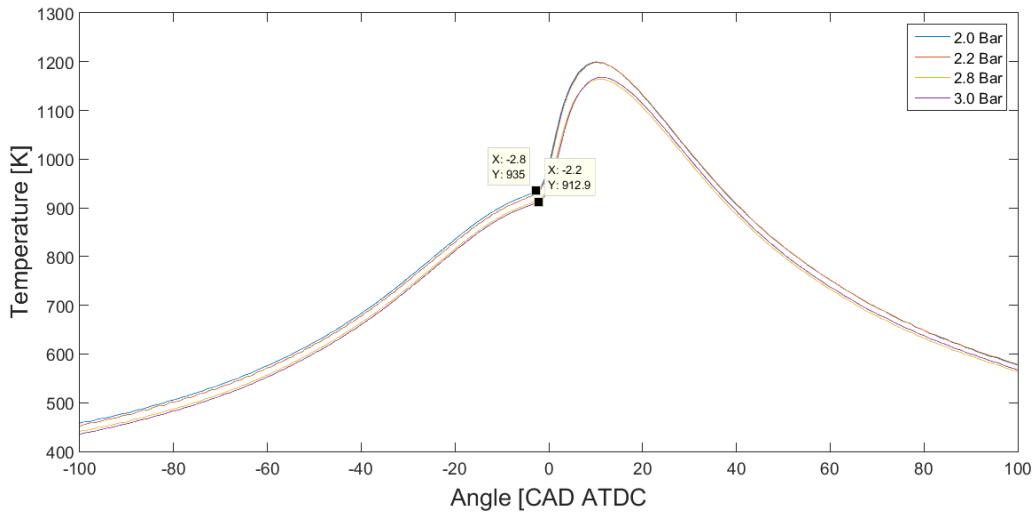


Figure 5-6. Temperature as a function of crank angle.

The start of combustion seems to start at a temperature of 913 K with 3 bar inlet pressure, and at 935 K with 2 bar inlet pressure. This figure is not a result of measurements, but a model of the in-cylinder temperature from the heat release equation, and should work as a tool to see how the temperature of the different operation points behave. It shows that the combustion actually starts at a lower temperature when the pressure is higher. This result together with the result seen in Figure 5-4 confirms that the operation is within the corresponding range with a gradient that is constant, seen in Figure 4-4 (referred to as the “third limit”), but for methanol and at lambda 5.

The scale makes it hard to actually see the different lines, so Figure 5-7 (which is a zoomed in version of Figure 5-6) is therefore inserted, in order to more clearly show the individual temperature curves.

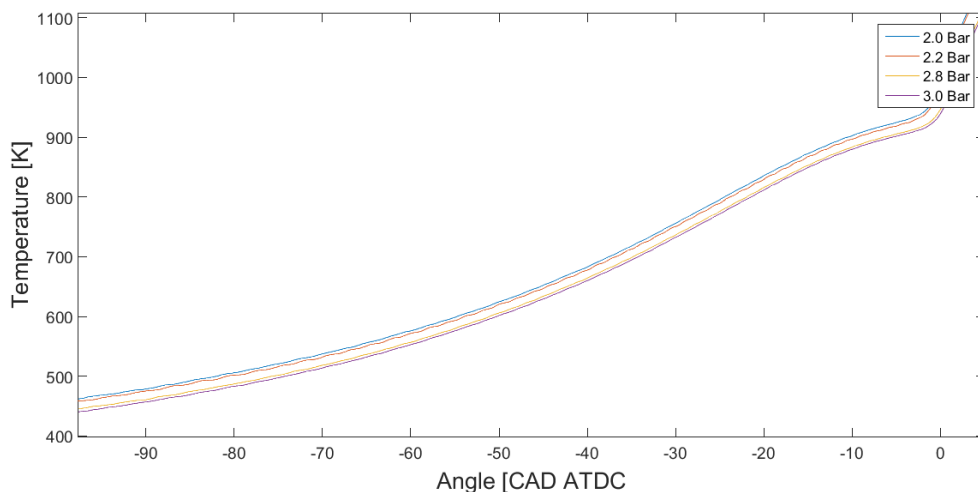


Figure 5-7. The picture shows a zoomed in version of Figure 5-6 during the compression stroke, this shows that the temperature lines are not crossed during the compression.

5.2 Constant inlet pressure and temperature

This second part of the experiments is an investigation of varying amount of fuel injected with constant inlet pressure and inlet temperature.

The limitations are the same as in the former experiment part; the lower limit is set by CA50 before 10 CAD ATDC, and a COV of IMEP below 5. The upper limit is set by the pressure rise rate and the maximum in cylinder pressure.

Table 5-2. Requirements and results from the load range experiment of constant inlet pressure and temperature.

Inlet Pressure [Bar]	1.5
Inlet Temperature [°C]	150
CA50 [ATDC]	< 10
COV (IMEP) [%]	< 5
SOI [BTDC]	23-12
Injection Pressure [Bar]	1000-1500

Figure 5-8 shows the heat release as a function of crank angle degree for four different load points. What is noticeable is that the lowest load, which corresponds to the earliest injection timing, shows a low temperature combustion before the greater heat release from the main combustion.

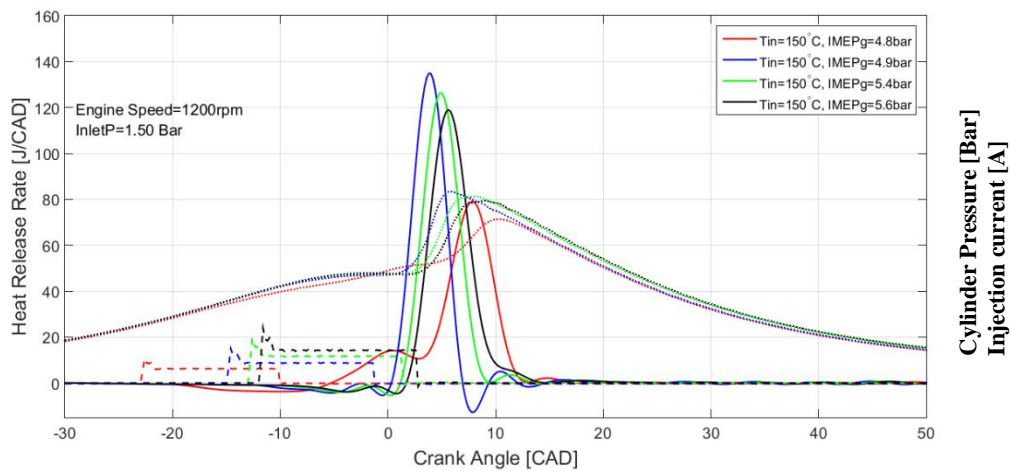


Figure 5-8. Heat release rate, injection timing and cylinder pressure as a function of crank angle for 1.5 Bar inlet pressure.

The corresponding temperature plot in Figure 5-9 shows a quite interesting result, even at the highest temperature for the highest load point, the maximum temperature only reaches 1765 K in PPC mode. As earlier discussed, this implies low numbers of soot and NO_x and high efficiency.

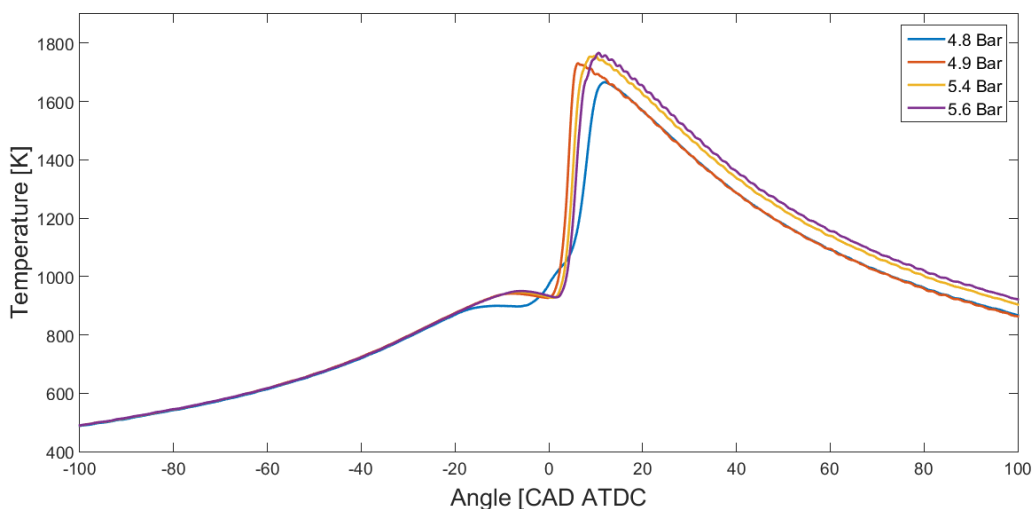


Figure 5-9. In-cylinder temperature as a function of crank angle degree for four different loads at 1.5 bar inlet pressure and 150 °C inlet temperature.

Figure 5-10 is a zoomed in version of Figure 5-9 where the focus is on start of combustion. The endothermic reaction when methanol is vaporized is actually lowering the temperature before the piston reaches the TDC when methanol is injected. In the lowest load case the temperature is held rather stable from about 17 to 3 CAD BTDC. This is called isothermal compression and decreases the compression work, which is discussed in 3.1.

Figure 5-10 also shows that an earlier injection makes the start of combustion occur at lower temperature, since the temperature rise during compression is interrupted by the cooling effect from

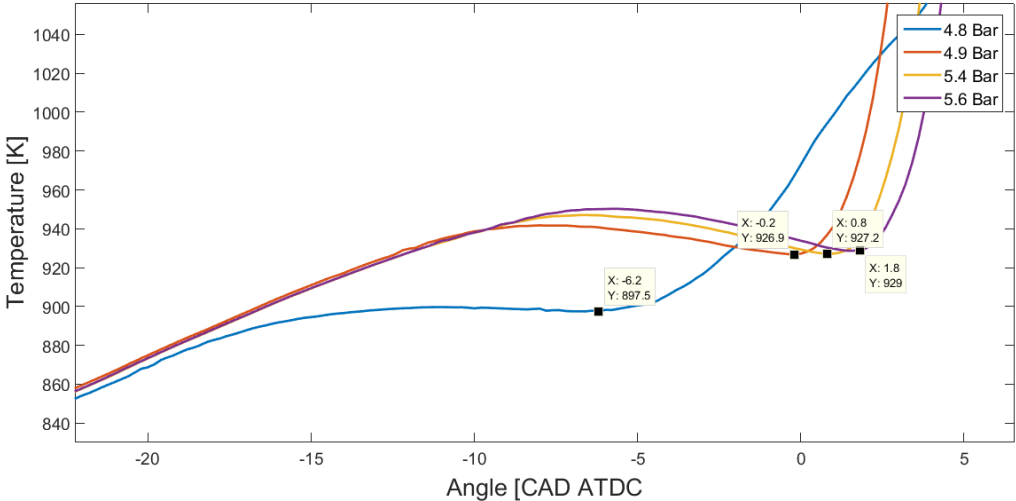


Figure 5-10. In cylinder temperature as a function of crank angle degree for four different loads at 1.5 Bar inlet pressure and 150 °C inlet temperature Zoomed in.

the injection. According to the plotted results the temperature at the start of combustion occurs at 897 K when the fuel was injected at 23 CAD BTDC and 927 – 929 K when the fuel was injected 15 – 12 CAD BTDC. This implies that the lowest ignition temperature is not only pressure correlated, it is also time dependent for methanol PPC.

5.3 Decreased temperature in order to extend the load range

In this part the results are presented that were obtained by decreasing the temperature as the load was increased, to keep the pressure rise rate down when the load range was investigated.

5.3.1 Heat release rate

An interesting result to study is how the injection of methanol in Figure 5-11 clearly gives a negative contribution to the heat release rate, which is a result that is discussed in 3.1. What is also seen in the figures is the separation between the injection timing and the heat release, which is typical for PPC.

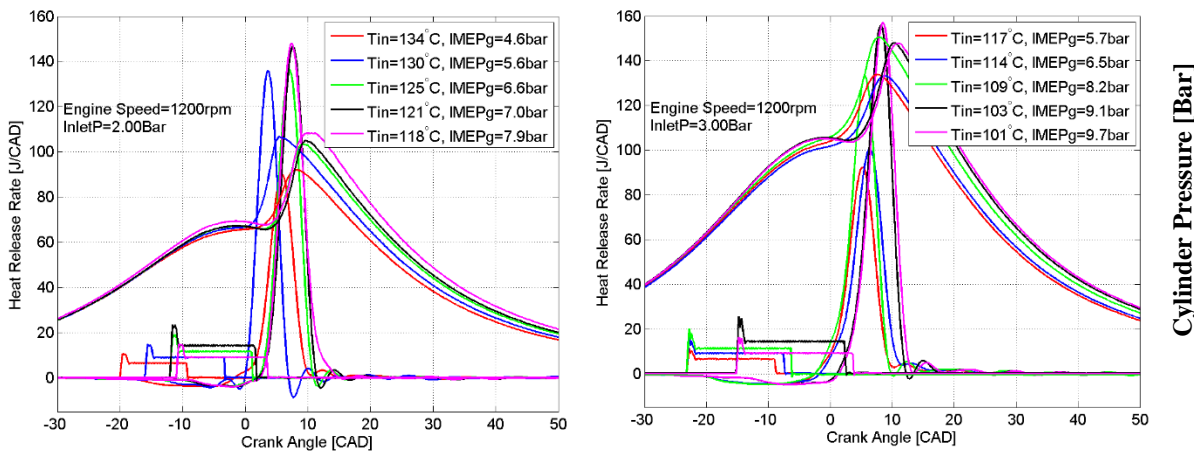
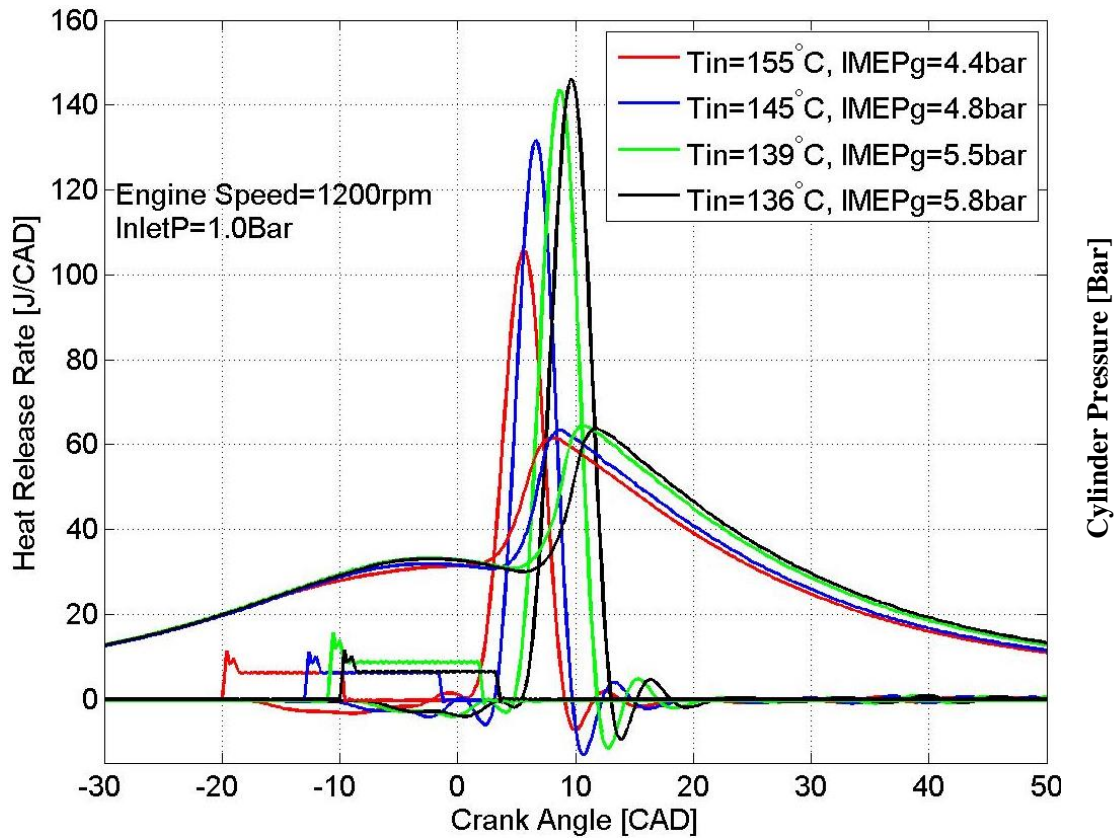


Figure 5-11. Heat release rate and cylinder pressure as a function of crank angle. Injection timing is seen as the square figures before heat release. At the top: Inlet pressure of 1.0 bar. Left: 2.0 Bar inlet pressure. Right: 3 Bar inlet pressure.

5.3.2 Load range

Since an extra degree of freedom was achieved when the inlet temperature was decreased in order to extend the load range, it is possible to optimize a combustion strategy for all considered parameters. If Figure 5-12 is considered, the traces of the load as a function of the inlet temperature is shown.

For any load, between slightly below 3 bar IMEP and up to 9 bar, there is a point within the colored area that represent possible operation points, represented in Figure 5-13.

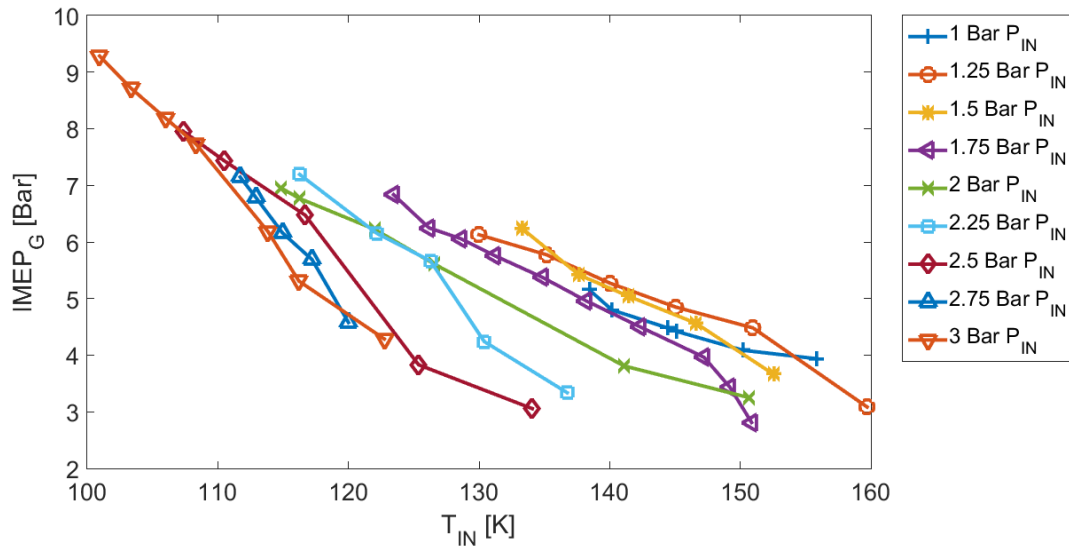


Figure 5-12. Indicated load as a function of inlet temperature.

Different aspects of emission levels and efficiency are discussed in 5.4 to decide which operating points are preferred within the colored area seen in Figure 5-13.

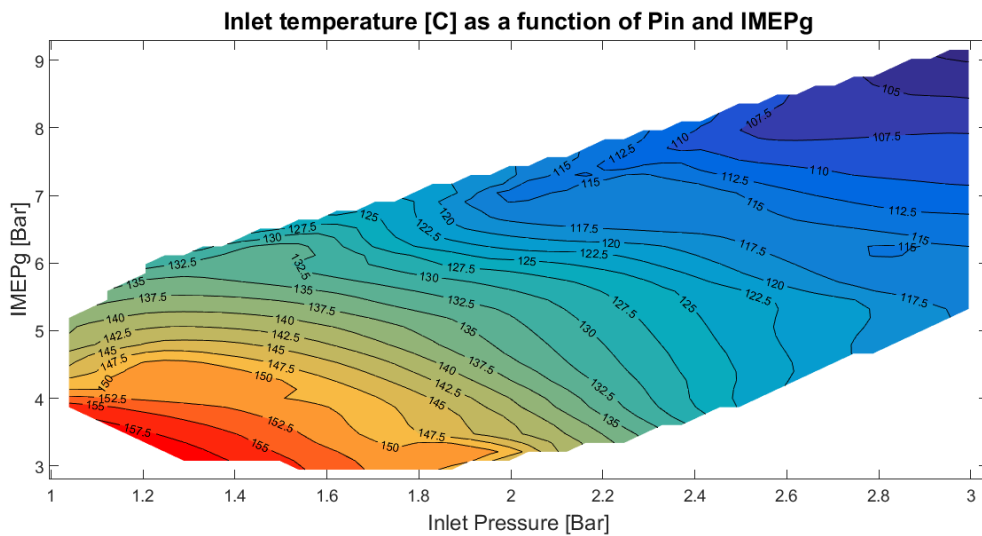


Figure 5-13. Load range for different inlet pressures, and the corresponding inlet temperature.

According to Figure 5-12 the trend seems to be a wider load range for higher inlet pressures, and that the temperature does not have to be greatly decreased as the load is increased. For example, to change the load from 4 to 9 bar with 3 bar inlet pressure, the temperature has to be decreased from 123 to 102 °C. That results in an increase of the load of 5 bar, for a temperature decrease of 21°C. That corresponds to 0.24 bar/degree.

The same calculation for 1 bar inlet pressure with a load range between 3 and 6.4 bar and a corresponding inlet temperature range from 160 °C to 130 °C, results in 0.11 bar/°C in decreased temperature. This example shows that the load range for 3 bar inlet pressure is almost twice the load range for 1 bar inlet pressure, and the temperature condition has to be changed less than half compared to the natural aspirated case, when the load is increased.

5.3.3 Emissions

The EU standards for heavy duty vehicles are shown in Table 3-1 where Euro VI is the prevailing requirement. From Euro I to Euro VI the levels of allowed NO_x has been most drastically reduced, which is of importance when research is made with new engine concepts.

Table 5-3. Emission regulations in EU. From 2013 the strict rules defined in Euro VI must be met. [6]

EU Emission Standards for Heavy-Duty Diesel Engines: Steady-State Testing

Stage	Date	Test	CO	HC	NO _x	PM	PN	Smoke
			g/kWh			1/kWh	1/m	
Euro I	1992, ≤ 85 kW	ECE R-49	4.5	1.1	8.0	0.612		
	1992, > 85 kW		4.5	1.1	8.0	0.36		
Euro II	1996.10		4.0	1.1	7.0	0.25		
	1998.10		4.0	1.1	7.0	0.15		
Euro III	1999.10 EEV only	ESC & ELR	1.5	0.25	2.0	0.02		0.15
	2000.10		2.1	0.66	5.0	0.10 ^a		0.8
Euro IV	2005.10		1.5	0.46	3.5	0.02		0.5
Euro V	2008.10		1.5	0.46	2.0	0.02		0.5
Euro VI	2013.01	WHSC	1.5	0.13	0.40	0.01	8.0×10 ¹¹	

a - PM = 0.13 g/kWh for engines < 0.75 dm³ swept volume per cylinder and a rated power speed > 3000 min⁻¹

Different emissions are heavily affected by the inlet temperature and the load which is seen in the following emission diagrams. All values presented in this thesis are shown without any after treatment.

Emissions could be presented in different ways, depending on the purpose and what kind of emission that is considered. According to human health, the concentration measure of PPM is an interesting way to present it. But for a heavy duty engine that shall execute a work, it is more fair to put the emission in relation to the work done. For that reason the NO_x, CO and HC are presented in the unit g/kWh. The particulate matter measurement was not available for most experiments, but a few results are found in Figure 8-1 in Appendix B.

5.3.3.1 NO_x

Figure 5-14 shows a strong correlation between load and the levels of NO_x. The highest levels occurs at low inlet pressure and the highest load for the corresponding inlet pressure. Since low inlet pressure means high inlet temperature in this experiment - and high load means high temperature after combustion, seen in Figure 5-8. This also becomes rather logical according to the formation of thermal NO_x discussed in 2.3.1.

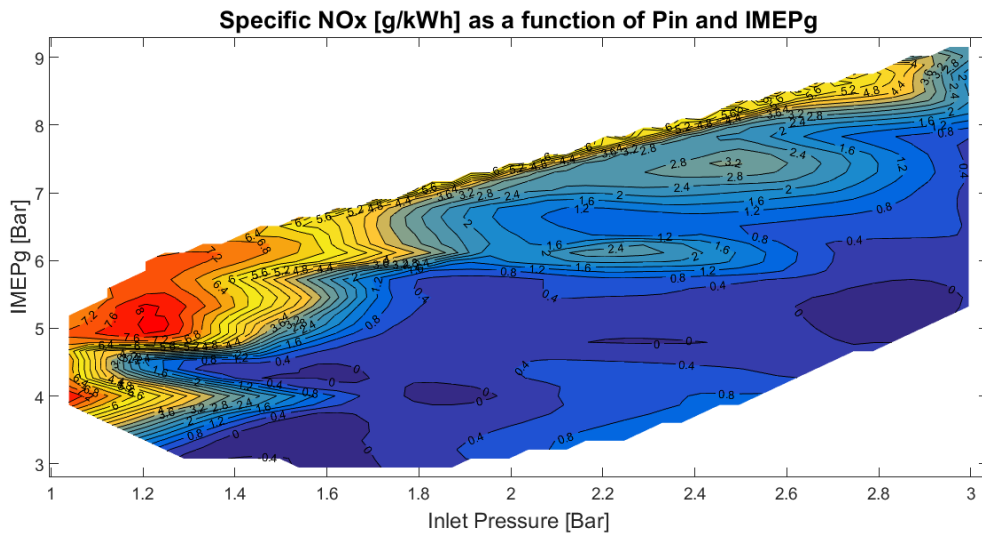


Figure 5-14. Specific NO_x as function of inlet temperature.

If Figure 5-15 is also taken into account, an obvious connection to λ is seen. To minimize the exhaust NO_x the inlet temperature should be low, and the λ kept high. Figure 5-15 also shows that high inlet pressures lead to lower λ values, and thereby lower NO_x levels.

Since most of the NO_x is formed where the combustion takes place in common diesel combustion, where λ is close to 1, the correlation between the global lambda measured in the exhausts and the formation of NO_x does not have to be clear. But in PPC combustion where the combustion occurs at lambda higher than one, there is a distinct correlation between the lambda and the amount of formed NO_x, according to Figure 5-15. A higher lambda means lower combustion temperature, and the specific NO_x formed is significantly lower in the higher lambda end of the diagram.

The Euro VI limits of NO_x emission is 0.4 g/kWh. This is not fulfilled in most operating points but for λ values higher than 4, nearly all operating points are close to or below this limit.

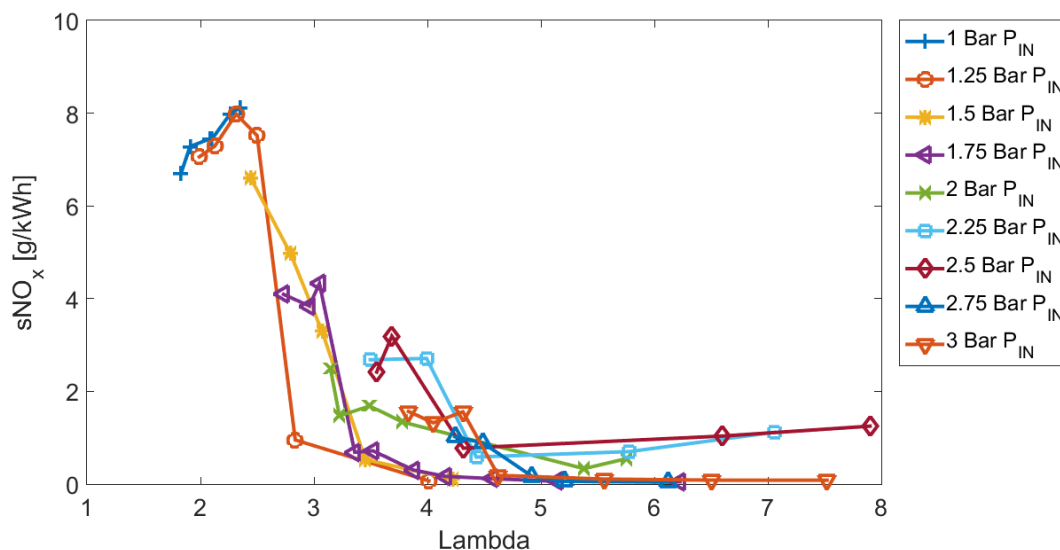


Figure 5-15 Specific NO_x as a function of Lambda.

5.3.3.2 CO

The results from carbon monoxide measurements in Figure 5-16 show an opposite trend compared to the NO_x emissions. Lowest specific emissions are obtained when the inlet pressure is low, and the load is highest possible (seen as lowest inlet temperature for each colored line). The decrease of the specific emission with an increasing load is expected since the combustion temperature increases, which facilitates the oxidation of CO. The great influence of the inlet pressure - that sets the λ hinders the oxidation of carbon monoxides when the combustion gets too lean.

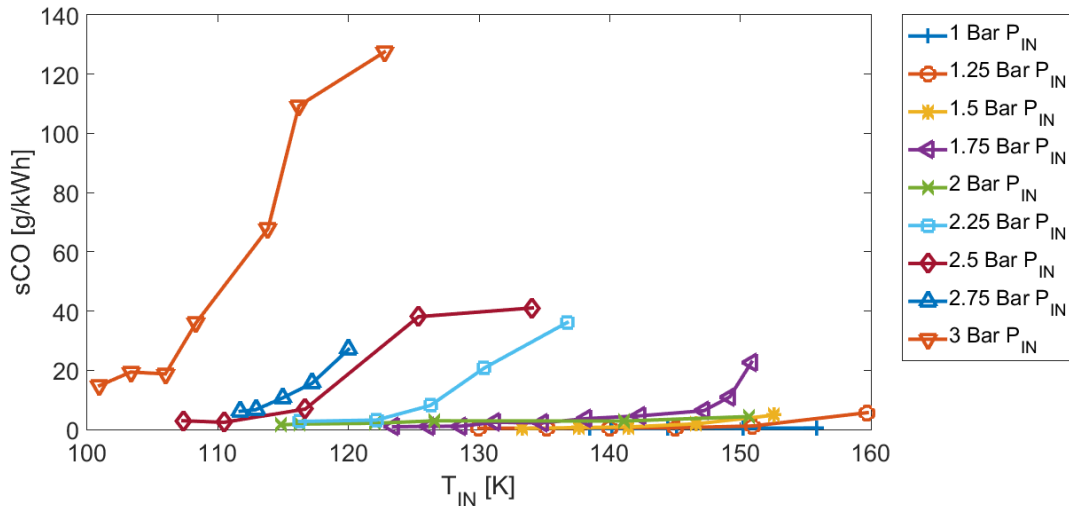


Figure 5-16 Specific carbon monoxide emissions as a function of inlet temperature.

In order to better see the trend of the curves that are within or close to the permitted emission levels, Figure 5-17 is inserted. Without after treatment, the Euro VI limit of 1.5 g/kWh is only reached at the lower inlet pressures at their corresponding high load, seen as clear blue color in Figure 5-17.

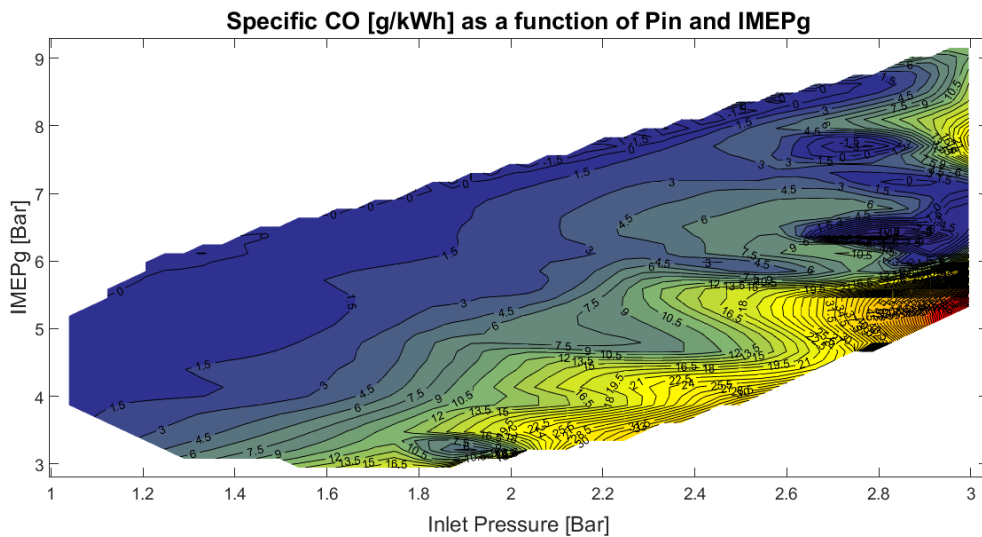


Figure 5-17. Specific CO within the load range.

5.3.3.3 HC

The trend of unburned hydrocarbons is that lower inlet temperature, which means higher load for a specific inlet pressure, results in lower specific HC according to Figure 5-18. Other trends are hard to interpret from this particular figure.

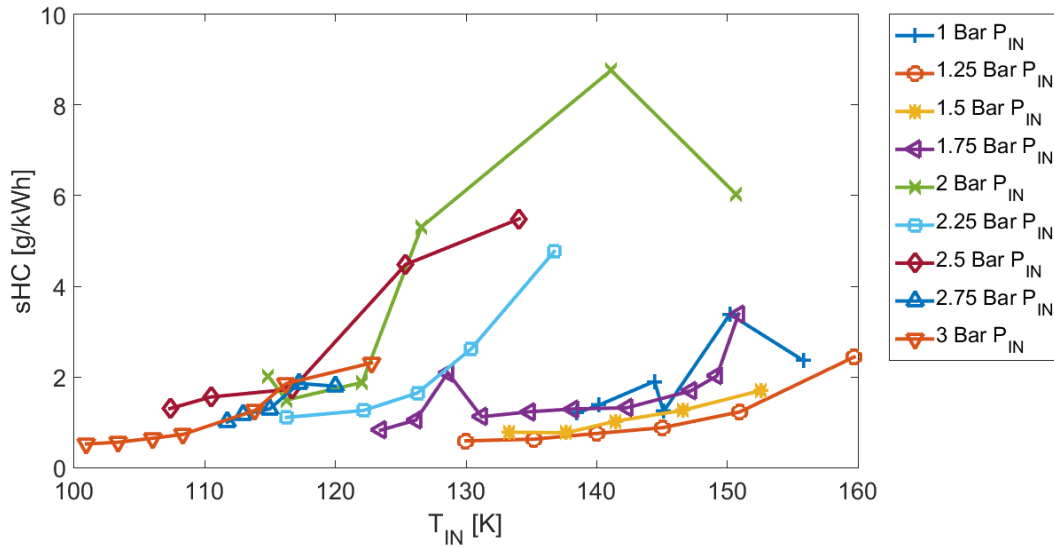


Figure 5-18. Specific emission levels of unburned hydrocarbons as a function of inlet temperature.

Figure 5-19 confirms that the level of unburned hydrocarbons is dependent on the load, but also on the inlet pressure. The emission level of hydrocarbons seems lower for a given load at lower inlet pressures. It makes sense since lower inlet pressures correspond to higher inlet temperature, leading to higher in cylinder temperatures. Since the combustion takes place at lean conditions, most of HC is formed if the temperature gets too low [3].

Compared to the Euro VI restriction of maximum unburned hydrocarbon levels of 0.13 g/kWh, this operation strategy was not successful. Single points at maximum load shows HC values close to this level, but most of the operating points are far from being acceptable without after treatment.

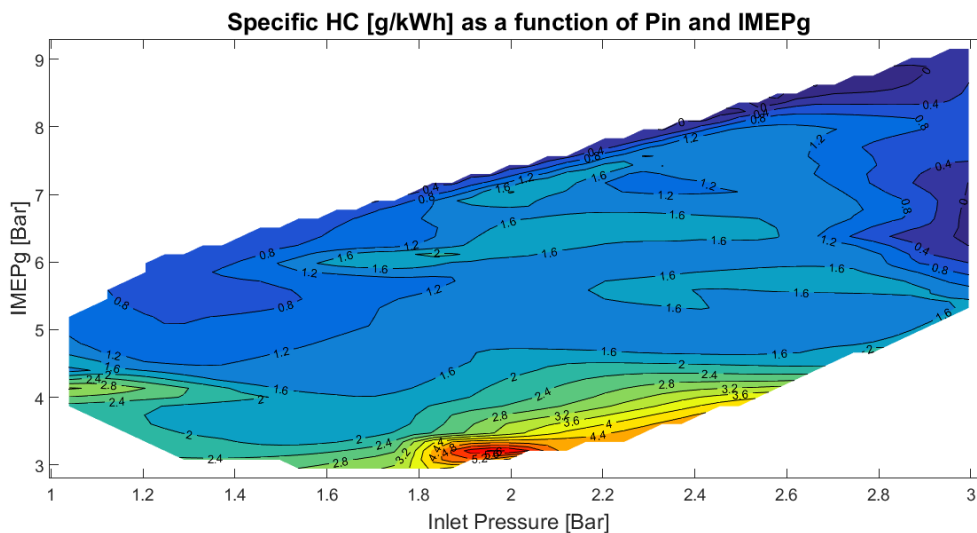


Figure 5-19. Specific HC within the load range.

HC is thereby the biggest issues in this particular setup, since most of the operating points exhibits HC emissions in an order of magnitude over the permitted levels in the experiments.

5.3.4 Efficiency

An important parameter for premixed compression ignition is the combustion efficiency. A very clear relation between λ and the combustion efficiency is seen in Figure 5-20. Lower lambda means higher combustion efficiency, and what is also seen is that the vast majority of the operating points corresponds to a combustion efficiency well above 98 %.

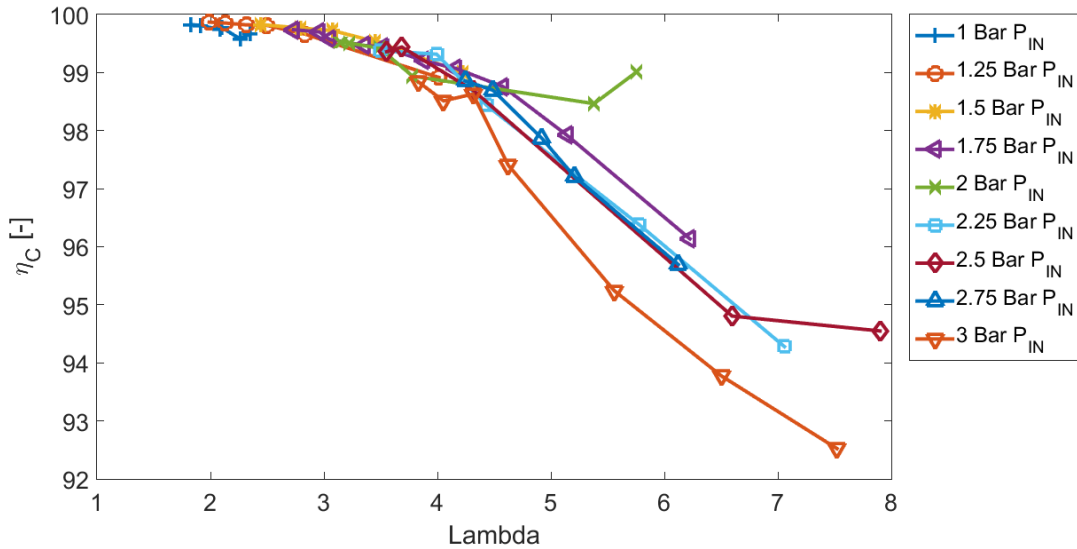


Figure 5-20. Combustion efficiency as a function of lambda.

Since no torque meter was used, there is no possibility to see the brake mean efficiency, instead the indicated efficiency is presented. The indicated net mean efficiency cannot be presented either, since the gas exchange was helped by an external compressor. Hence the presented efficiency is the indicated gross mean efficiency, which is seen in Figure 5-21.

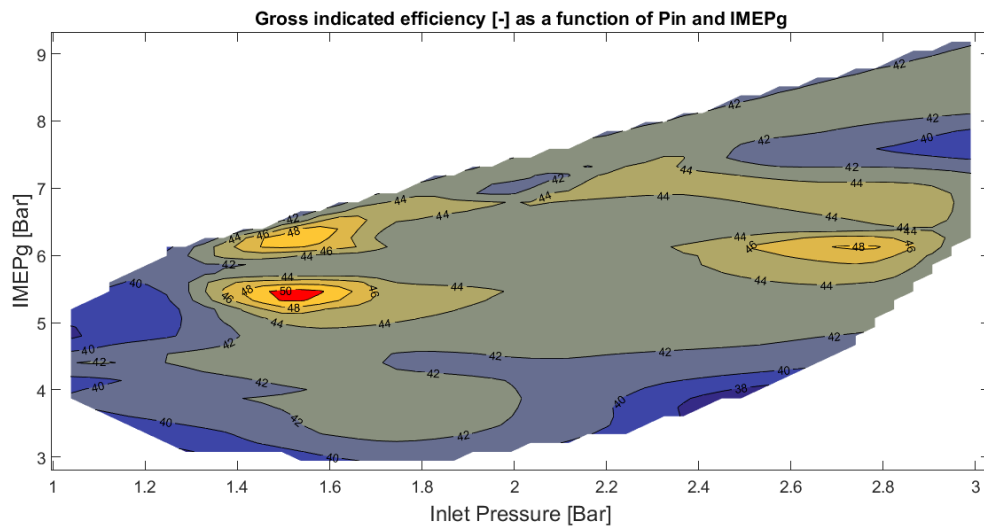


Figure 5-21 Indicated efficiency as a function of load. Gross indicated efficiency within the load range. The lowest load point at 3 Bar inlet pressure is not included due to fuel flow problems.

The highest value of the indicated efficiency was 51 % at 5.4 bar load with an inlet pressure of 1.5 bar. The better part of the operating points are above 42%, but as discussed in paragraph 2.4 the efficiency could be higher with a higher compression ratio.

The reason the lowest load point for 3 bar inlet pressure was not included in Figure 5-21 was due to problems with the experimental equipment, the measurement of the fuel flow in this case. The fuel flow measurement affects the efficiency and therefore this particular point is not included.

5.4 The best operating point

The best operating point is a point that combines many different desired values. In this case the highest possible load, at the highest efficiency with best tradeoff between the emissions is sought.

It is not convenient to have too high inlet pressure for a production engine since it requires a more expensive and bigger turbo machine. Therefore the best point is searched at the lowest possible inlet pressure.

At 1.5 bar inlet pressure, there are two areas that exhibit high efficiencies according to Figure 5-21. At maximum load, 6.5 bar IMEPg, the efficiency is above 48 percent. CO as well as HC shows relatively low values in this particular point, but the NO_x level is about 6 g/kWh, which must be considered high.

Another alternative is 2 bar inlet pressure and 7 bar gross IMEP. The efficiency is about 44 percent, which is acceptable, but the tradeoff between the emissions is good. The CO level is below 1.5, NO_x below 2.4 and HC below 1.6 g/kWh.

5.5 Sources of error

Figure 5-21 shows the efficiency as a function of the entire load range, and represents important results for the thesis. Care should though be taken when it is read too literally. The “islands” of higher efficiencies seen in a red tone might be the result of fortunate tradeoff of low heat transfer and exhaust energy losses. But the fluctuations of the fuel flow and the reading of the same might be a parameter that affects the result in the same magnitude. The local values that differs much from nearby points, are probably primarily an unwanted effect from the imperfections of the fuel flow measurements.

The valves which were adjusted to get the right inlet pressure and back pressure to the engine, fluctuated a few percent and has probably contributed to a less accurate result, according to efficiency as well as the different emission levels. It is hard to approximate the magnitude of effect that it makes on the result.

The emission system was regularly calibrated and examined to make sure it exhibited as accurate values as possible. Variations of the same operating point could still be seen though.

To estimate the impact from these different sources of error, a sensitivity analysis for each source could be conducted. Obtained results would make possible to quantitatively estimate the errors from the experiment, and enable to do better conclusions about the errors and what is actually differences in result. Lack of time limited this kind of investigation.

6 Conclusions and discussion

The most important conclusion from these experiments is that it is possible to run PPC on neat methanol, and this thesis is thereby the first report this can be announced.

High inlet temperatures were required to achieve combustion with the used compression ratio, but this required temperature is lowered when either the load or the boost is increased, and especially if they both are.

As expected, the soot formation was very low and the NO_x emissions that use to be a problem in CI engines is not far from the Euro 6 upper limit in many load points without after treatment.

What was obtained from the first experiment with HCCI combustion was how much the inlet temperature could be decreased with an increase of the inlet pressure. The function was very well suited to a straight line with the curve gradient of -21 °C/Bar. That is interpreted as the methanol exhibits a linear path on the explosion diagram for the specific pressures, temperature and lambda that the methanol fuel and air mixture was exposed to.

From the second part of the experiments, where the inlet pressure as well the inlet temperature was held constant in PPC mode, it was a rather small space for operating without exceeding any defined limit. The experiment was kept with a separation between the injection and the start of combustion in typical PPC manner. When neat methanol is combusted, there is no actual need for keeping this injection-combustion separation. The main purpose of the PPC is to keep the soot levels at a minimum, which is of great importance when diesel fuel is combusted. But when methanol is used, there is no risk for large soot production due to the molecules oxygen content and single carbon structure. This was known during the experiment time, and consequently the lack of a working soot meter in all load points was not an issue.

To extend the load range for PPC the inlet temperature was decreased as the load was increased to avoid reaching the maximum pressure rise rate in the third experiment. This is not the common way to handle the combustion delay that is desired in PPC. Since it is not very convenient to have an inlet temperature of about 150 °C, another piston would greatly improve the experiments. The required temperature in the combustion chamber for auto ignition would be reached without this high inlet temperature if the compression ratio was increased. Since methanol has a very high resistance to auto ignition, a higher compression ratio would probably be suitable. It would be interesting to see how it would affect the lowest inlet temperature of air to get auto ignition. The indicated efficiency could also be expected to increase since the thermal efficiency is improved with higher compression ratio.

In order to slow down the combustion, EGR could be used instead of excessive air, which lambda 5 in the experiments implies. The specific heat of EGR is much higher than the specific heat of air, which is useful when the maximum load is sought, without exceeding the limits of the engine. This would probably make it possible to reach the maximum defined pressure in the cylinder at high load, instead of just reaching the maximum pressure rise rate that is limiting the load. Such experiments were actually conducted, but the valves for backpressure and the EGR line were not stable enough to make it possible to increase the EGR levels carefully enough with methanol combustion - every increase above just a few percent EGR killed the combustion, so no results from that campaign is presented in this thesis.

It would also be interesting to take a look at the brake mean efficiency by mounting the dyno to a torque meter. Since the experiment was ran on a single cylinder engine, the friction losses would probably be in such a high order that the absolute value of the torque meter would be pointless to consider, but the tendency of increased load with increased fuel injection could still be observed. Maybe this kind of combustion with a rather steep increase in heat release and pressure would cause massive friction losses, which cannot be observed by only investigating the IMEP.

Methanol has interesting properties that is desired for PPC combustion apart from the high octane number. So it could also be investigated with an ignition improver as an additive. The heat release would probably appear smoother and the inlet temperature could be lowered, with most of the methanol's properties kept constant, like the high oxygen and hydrogen content that prevents soot to form as well as the high heat of vaporization.

The load range could also be extended by just keep injecting fuel even though the there is no separation between the injection timing and the heat release. The methanol fuel should not cause severe soot problems even with a pure dissipation combustion. Larger injector nozzles would also be interesting to investigate, to be able to do the opposite – a fast and massive fuel injection without too high injection pressure, to keep the stratification when aiming for higher loads.

It would also be of interest to do the experiments at different engine speeds.

7 Appendix A

Gasoline PPC

Efficiency

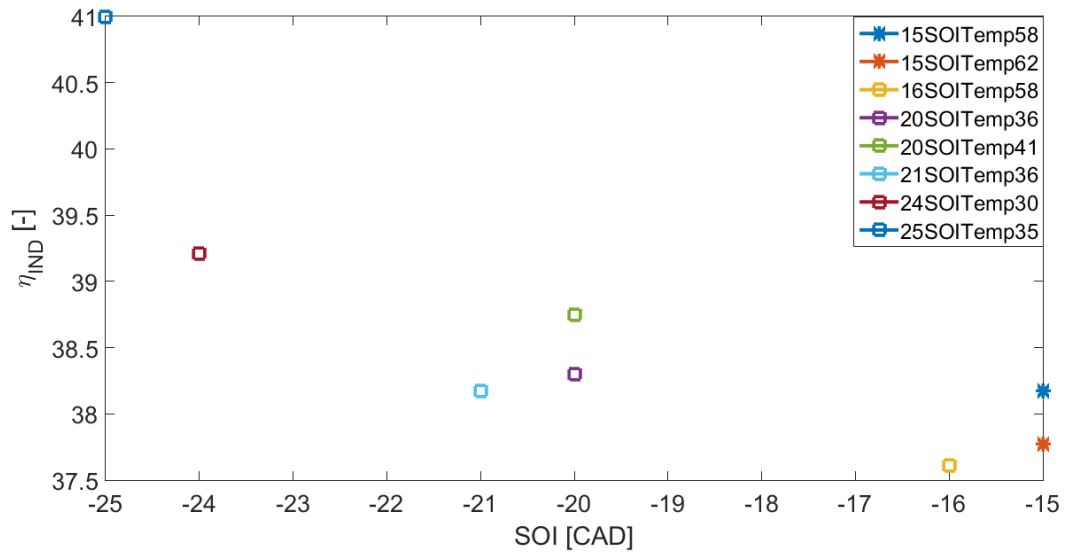


Figure 7-1. Indicated efficiency as a function of injection timing

Carbon monoxide, CO

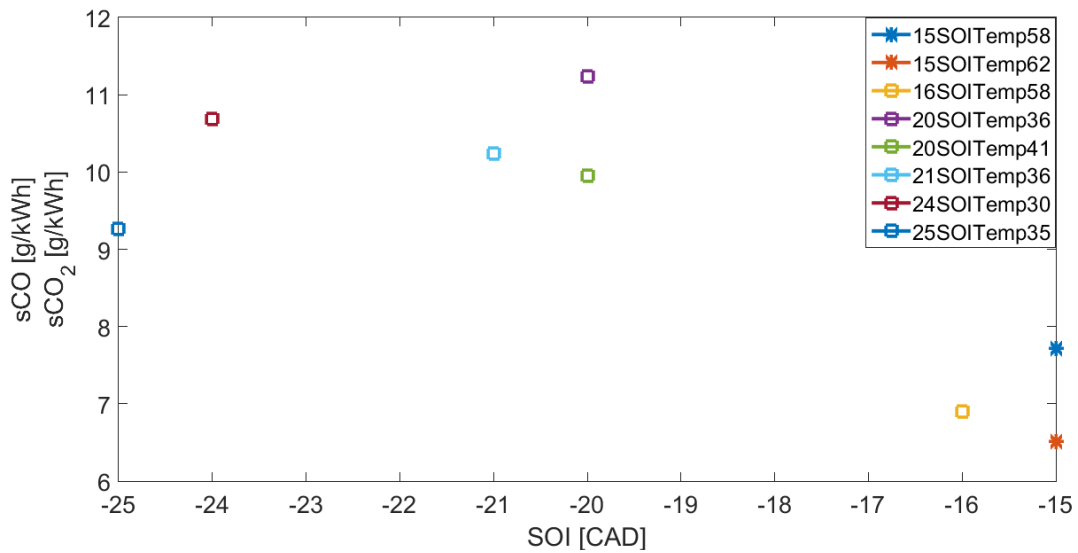


Figure 7-2 Specific CO as a function of injection timing.

Nitro oxides, NOx

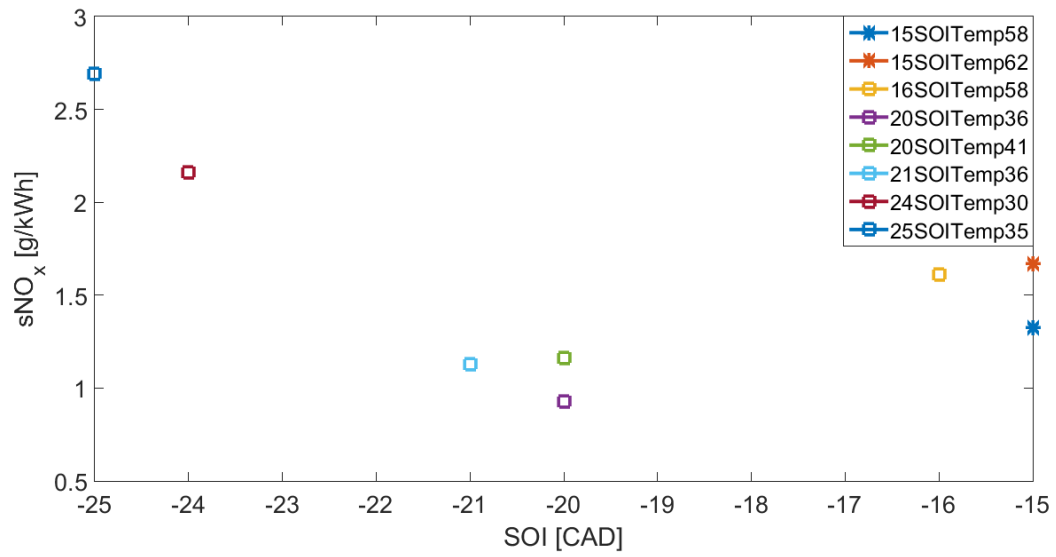


Figure 7-3. Specific NO_x as a function of injection timing.

Hydrocarbons, HC

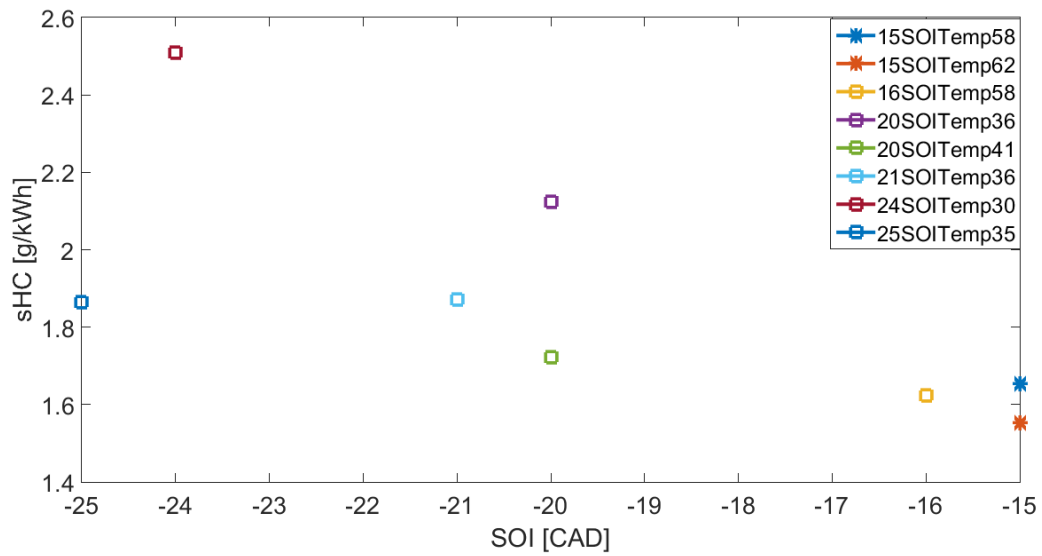


Figure 7-4. Specific HC as a function of injection timing.

8 Appendix B

Methanol PPC

Soot

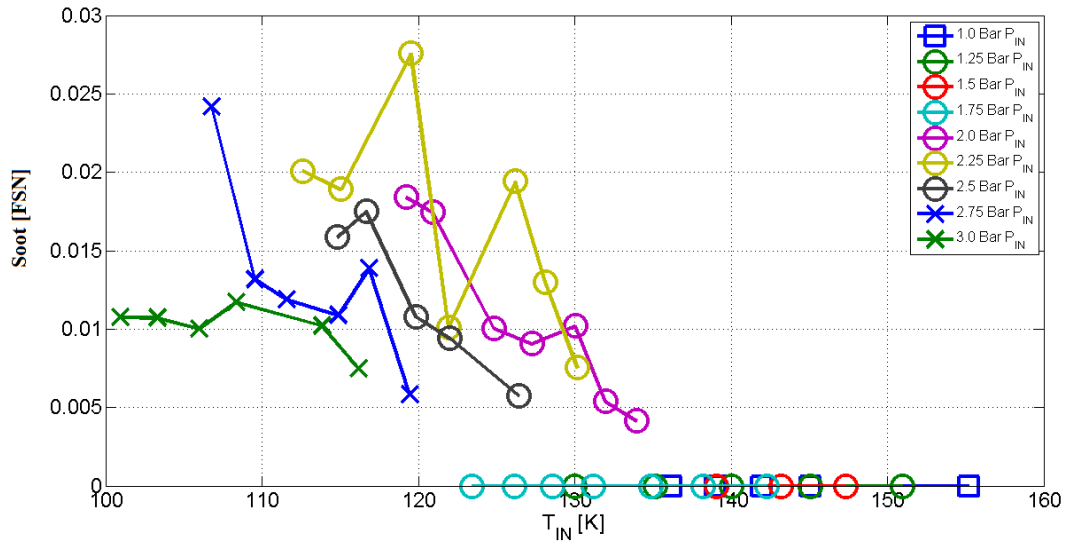


Figure 8-1. Soot emissions as a function of inlet temperature and inlet pressure.

Load range

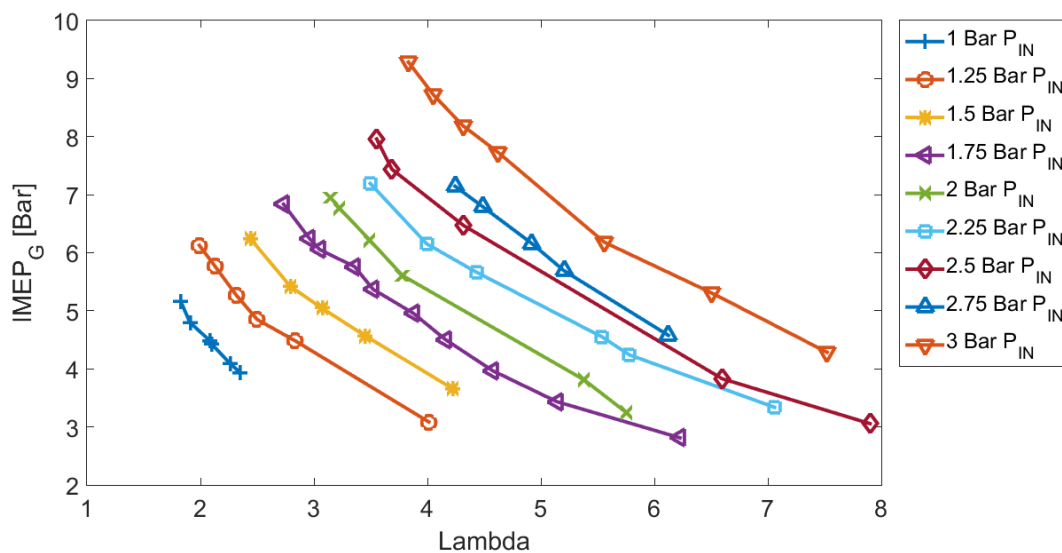


Figure 8-2. Load as a function of Lambda for different inlet pressures.

Efficiency and Specific NO_x

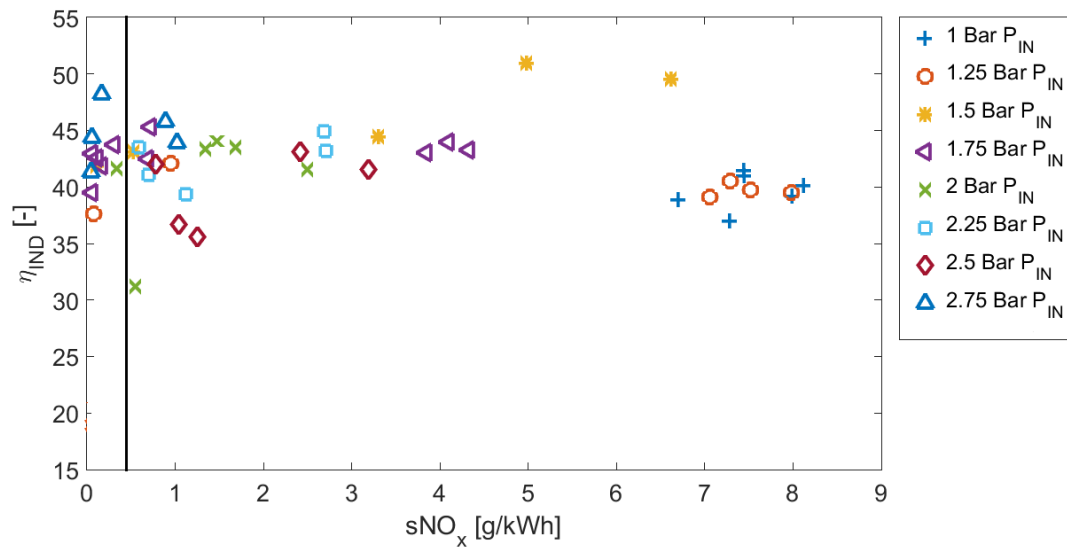


Figure 8-3. Gross indicated efficiency as a function of NO_x for different inlet pressures. Euro VI limit of NO_x highlighted with a vertical line.

Pressure rise rate and lambda

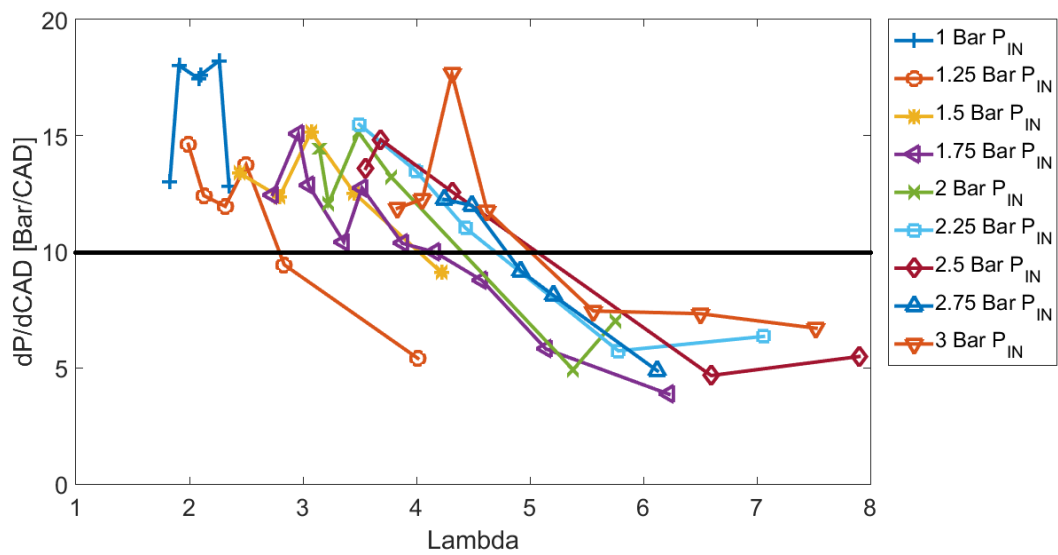


Figure 8-4. Pressure rise rate as a function of lambda. dP/dCAD=10 is highlighted to show the upper limit for production diesel engines.

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