# Gasoline Partially Premixed Combustion

## An Advanced Internal Combustion Engine Concept Aimed to High Efficiency, Low Emissions and Low Acoustic Noise in the Whole Load Range

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# List of Publications

## Gasoline PPC Publications
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1. **An Advanced Internal Combustion Engine Concept for Low Emissions and High Efficiency from Idle to Max Load Using Gasoline Partially Premixed Combustion**
   By Vittorio Manente, Claes-Goran Zander, Bengt Johansson, Per Tunestal and William Cannella
   SAE 10FFL-0006
2. **Effects of Ethanol and Gasoline Type of Fuels on Partially Premixed Combustion from Low to High Load**
   By Vittorio Manente, Bengt Johansson, Per Tunestal and William Cannella
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3. **Influence of Inlet Pressure, EGR, Combustion Phasing, Speed and Pilot Ratio on High Load Gasoline Partially Premixed Combustion**
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5. **Characterization of Partially Premixed Combustion with Ethanol: EGR sweeps, Low and Maximum Loads**
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6. **Partially Premixed Combustion at High Load using Gasoline and Ethanol, a Comparison with Diesel**
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7. **Half Load Partially Premixed Combustion, PPC, with High Octane Number Fuels. Gasoline and Ethanol Compared with Diesel**
   By Vittorio Manente, Bengt Johansson and Per Tunestal
   SIAT 2009 295

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8. **Gasoline Partially Premixed Combustion: High Efficiency, Low NOx and Low Soot by using an Advanced Combustion Strategy and a Compression Ignition Engine**
   By Vittorio Manente, Bengt Johansson and Per Tunestal
   International Journal of Vehicle Design, IJVD 11838

9. **Characterization of Partially Premixed Combustion With Ethanol: EGR Sweeps, Low and Maximum Loads**
   By Vittorio Manente, Bengt Johansson and Per Tunestal
   Journal of Engineering for Gas Turbines and Power (Vol.132, Iss.8), GTP 09-1183
10. **An Advanced Internal Combustion Engine Concept for High Efficiency and Low Emissions**
    By Vittorio Manente, Bengt Johansson and Per Tunestal
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I. **Validation of a Novel Self Tuning Gross Heat Release Algorithm**
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   SAE 06SETC-55

## Abstract

Environmental concerns such as global warming, the surge of crude oil prices and more stringent emission regulations are demanding the development of internal combustion engines that are highly efficient and low polluting. With classical internal combustion engine concepts such as diesel compression ignition and gasoline spark ignition engines, a trade-off exists between local and global emissions i.e. NOx, soot, CO and HC vs. $CO_2$. For instance in the middle of the 1990's when local emission regulations became tighter, the average brake efficiency, in heavy duty engine applications, dropped from 44.5 to 41.5%. The introduction of HCCI combustion, Homogenous Charge Compression Ignition, seemed to be the solution to the trade-off between local and global emissions. This concept was achieving ultra low NOx and soot, simultaneously with high efficiency. More than a decade of research has demonstrated that the viability of HCCI applications are limited to low load operations. Excessive acoustic noise, lack of direct control of the combustion and too diluted air-fuel mixture requirement make this concept not feasible to be applied to the whole load-speed engine range.

The objective of this thesis was to develop a new combustion concept capable of running over the whole load range and achieve very high efficiency, low emissions, and acceptable acoustic noise without any drawback in combustion control. The target was achieved by burning gasoline in partially premixed combustion, PPC, mode. PPC is a mix of classical diesel combustion process where fuel is injected while combustion is occurring and HCCI using a fully premixed charge. If standard diesel combustion is black and HCCI is white, PPC is some shade of gray. If it is light gray much of the fuel is mixed with air before combustion, if it is dark gray combustion is more stratified and resembling diesel diffusion combustion. Advanced running conditions were developed in order to achieve high efficiency, low emissions and low acoustic noise in the whole load range; simply injecting gasoline in a diesel engine is not enough.

The PPC concept was studied and developed using two heavy duty single cylinder engines, a Scania D12 and a Scania D13. When the concept was fully developed indicated efficiencies between 52 and 55% were reached from idle to 26 bar gross IMEP. This was accomplished keeping NOx below 0.40 g/kWh simultaneously with soot values not higher than 0.30 FSN and a maximum relative pressure rise rate below 8 bar/CAD. Gasoline PPC was also briefly studied in a light duty VOLVO D5 engine.

## Acknowledgements

Good supervision combined with a pleasant working environment and hard work usually contribute to good research results. A huge thanks goes to Bengt Johansson for his supervision during this gasoline partially premixed combustion project. In 2008 not much was known about this combustion concept and many hours of brainstorming for extended period of times were necessary for fixing the trade-off among efficiency, emissions, acoustic noise and load range. I am also really grateful for the fact he was able to take out my full potential. Though at the beginning it was a little bit tiring, I am glad I had to do the same PowerPoint four-five times or perform numerous amounts of sweeps in the lab. It was all worth and all the efforts have been paid back with the achievements. Another big thanks goes to Öivind Andersson which made me have a "research epiphany" in a chat lasted less than five minutes. He said when doing research it is important to understand everything down to a fundamental level and ask "why" until all the mechanisms that lead to a given result are fully understood. If researchers do not do that it is better if they change job. From that moment on, my way of analyzing data drastically changed. I am also thankful for the time he spent teaching me about soot and combustion chamber design. Within the gasoline PPC project Bill Cannella from Chevron deserves to be acknowledged. Without him supplying a variety of gasolines (more than 1100 liters only in 2009!) it would have taken longer before understanding the importance of the fuel molecule specs with this novel combustion concept. Within the VIMPA project (2006-08), Per Tunestål helped me a lot in cranking up with the combustion engine world and in understanding the strange phenomena that occur when combustion takes place in the milli domain at high engine speed. Even though Rolf Egnell and Martin Tunér were not my main supervisors discussing with them was really useful.

Technical staff wise, Tom Handemark deserves recognition for his work. As far as fuels were properly stored and registered, he never got upset when there was an entire engine frame to change or the coupler between the dyno and the engine broke down or when the flywheel dumper melted. He skipped many coffee breaks to fix the numerous test cell failures asap. Kjell Jonholm and Tommy Petersen were useful in helping me during the VIMPA project, the first one for building tiny components while the second for helping me with the electronic part when I was building the VIMPA test rig. Very useful were all the tips I got from Bertil Anderson when I had to fix the engine myself, I have really appreciated. Thanks also to Krister Olsson for keeping the computers always running and fixing the data logger software.

Maj-Lis Roos deserves to be acknowledged for helping me a lot with the bureaucracy, booking trips and being patient to replay to my numerous questions in almost no time.

When I started working at the combustion engine division, my former colleagues Jari Hyvönen, Leif Hildingsson, Andreas Vressner and Håkan Persson patiently helped me in understanding how to burn fuel in HCCI mode and the advantages and limitations of this concept. Even though there was not much scientific interaction it was very nice to have Sasa Trajkovic as colleague. His permanently positive attitude and funny personality helped to keep the second floor far away from an austere environment; not mentioning how far the environment was from austerity when Sasa was interacting with Magnus Lewander and Mehrzad Kaiadi. It was quite amusing to go to the conferences with these three guys. As compared to the previous three guys, there was more research cooperation with another colleague of mine: Cleas-Göran Zander. His skills in building an

outstanding control system for the Scania D13 were remarkable; in addition chatting with him was also quite interesting under a broad variety of topics. Hans Aulin is also a very nice colleague. He literally contributed to keep a warm environment; his office is always hot in winter time because of the heater. A touch of pink was added at the division when Ida Truedsson and Hadeel Solaka were hired. It was finally possible to converse about cloths, shopping and girlish/Italianish subjects in a place otherwise dominated by engines and video games endless topics. It is quite nice to hang around with them outside the working environment during the weekends together with the new arrival from Finland Teemu Anttinen. It is a pity that in the last two years I have been hyper-busy but to hang around with Kent Ekholm in my first two years in Lund was quite an experience because of his variety of activities. My colleagues from the GenDies group, Ulf Aronsson and Clement Chartier, have to be mentioned for the fruitful discussions we had regarding soot formation and how to avoid and reduce smoke. It was quite an experience try to jump from a cliff and to ski in Kirkwood at Lake Tahoe with Clement. Many thanks also go to my other colleagues for contributing to a nice atmosphere: Thomas Johansson (the king of motorbikes tuning), Patrick Borgqvist (the great LabView guru), Martin Algotsson, Uwe Horn, Peter Andersson and Noriyuki Takada.

My girlfriend, Asnate, should be acknowledged for at least a couple of reasons. The first one is for calming me down and keeping my mental balance when I am stressed at work, and also for accepting me for who I am and not who I could be thus avoiding more stress to my busy life. The second reason is because her professionalism and pickiness in doing her job motivated me to carry out my research activities as properly as possible trying to reduce the imperfections to a minimum.

Finally I would like to thank Vittorio Manente senior and Giovanni Manente for putting me on the right track since young age and inspiring all my professional and daily life.

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# 1 Introduction

## 1.1 Global Warming

Global warming is defined as the increase of the average temperature of the Earth near surface air and oceans. According to the Intergovernmental Panel on Climate Change, IPCC, most of the temperature increase since the middle of the last century was due to the increase in concentration of greenhouse gasses in the atmosphere caused by human activities e.g. deforestation and fossil fuel burning [1]. The variation of the average Earth temperature between 1961 and 1990 is presented in Figure 1.

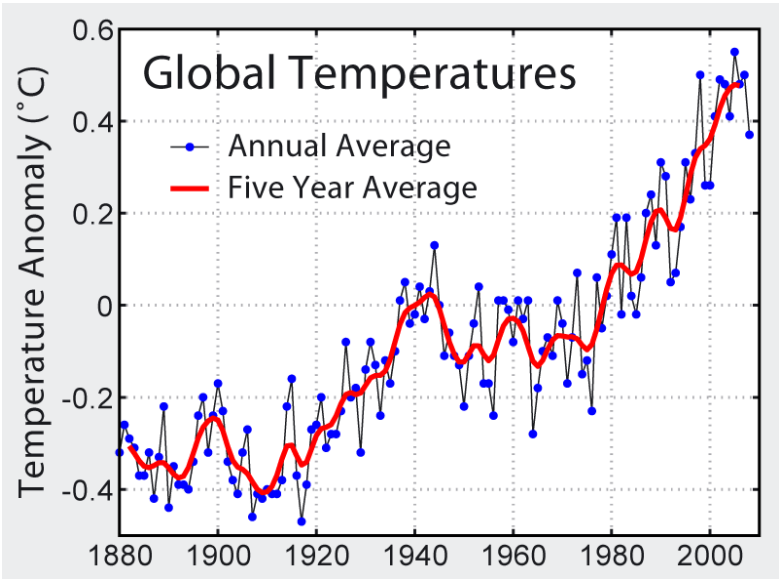


**Figure 1: Global mean surface temperature variation; the zero is the mean temperature between 1961 and 1990 [2].**

Climate models are predicting an increase of average global temperature between 1.1 and 6.4 K within the end of the current century; this spread is due both to the different sensitivity of the models to the current greenhouse gasses concentration and the use of different estimates in greenhouse gas emissions [1]. Despite these uncertainties global warming is expected to increase even after 2100 even if all the sources of emissions will instantly stop. This is because of the heat capacity of the oceans and the long life time of carbon dioxide in the atmosphere.

According to [3], the average increase of the Earth temperature will lead to a rise of the sea levels (due to the melting of the Arctic and Antarctic glaciers), an increase in the amount of precipitation and the expansions of deserts. Other possible consequences will be extreme weather events, species extinctions and changes in agricultural yields.

## 1.2 Greenhouse Gasses

The greenhouse effect is the process by which the Earth's lower atmosphere is warm up due to the absorption and emissions of infrared radiations by gasses in the atmosphere. This effect is vital for the existence of living floras and faunas on the planet Earth because it is keeps the average planet temperature at 306 K. The major greenhouse gasses in the atmosphere are: water vapor, carbon dioxide, methane, ozone and clouds[1].

Since the beginning of the industrial revolution the amount of greenhouse gasses caused by human activities is substantially increased; for instance the amount of $CO_2$ and methane is 36% and 148% higher as compared to the levels in the 18$^{th}$ century [4]. Burning fossil fuels was responsible for 75% of the increment of $CO_2$ in the atmosphere over the last 20 years; the remaining 25% was mainly due to deforestation [5]. On the other hand, anthropogenic methane production is due primarily to an increase in livestock (enteric fermentation) and decomposition of organic compounds for instance in the garbage dumps. Naturally occurring methane production is caused by clathrates in the ocean floors, leak of methane from the Earth crust and methanogenesis.

## 1.3 $CO_2$ as Major Greenhouse Gas

Even though methane is a more powerful greenhouse gas than $CO_2$, it can still be captured and burned in order to produce power. This means that the global warming might be mitigate by mostly limiting the amount of $CO_2$ released in the atmosphere either by decreasing its production or by using capture and storing techniques [6]. Figure 2 shows that most of the major greenhouse gasses emission is constituted by carbon dioxide. As described in the previous paragraph $CO_2$, caused by human activities, is mainly due by burning fossil fuels such as natural gas, petroleum and coal in order to produce power for residential, commercial, industrial and transportation purposes; see Figure 3 and Figure 4.

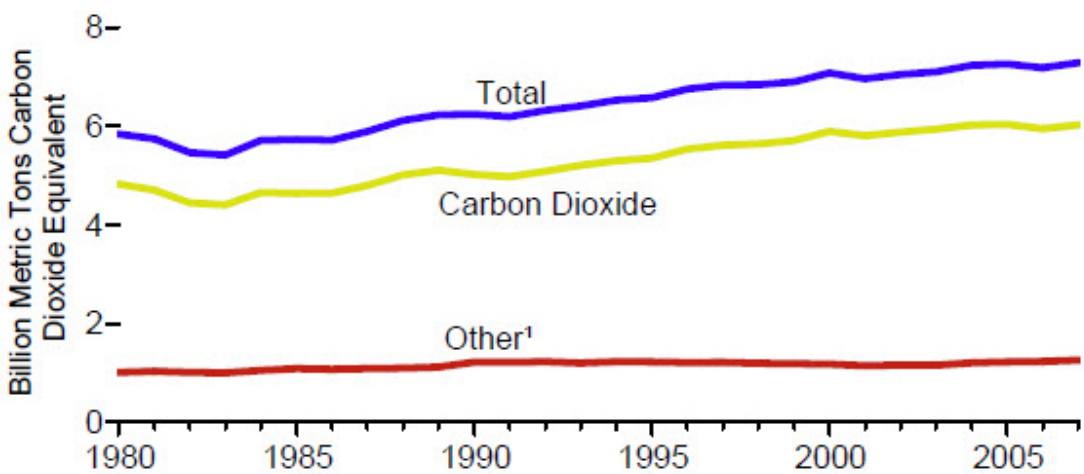


**Figure 2: Greenhouse gas emissions in the USA. Other[1] = Methane, nitrous oxide, hydrofluorocarbons (HFCs), perfluorocarbons (PFCs), and sulfur hexafluoride ($SF_6$) [7].**

[1] Clouds are not included in the water vapor category since they are made by liquid water and/or ice.

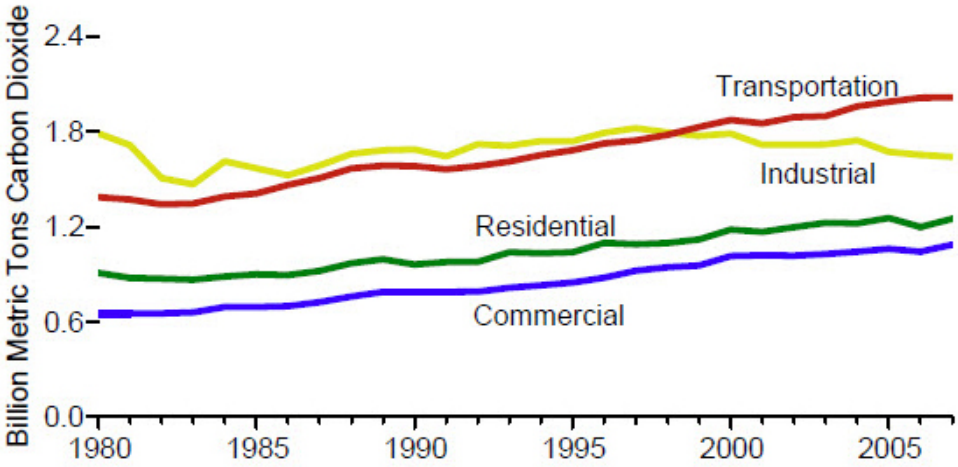


**Figure 3: $CO_2$ production in the four major economic sectors in the United States between 1980 and 2008 [7].**

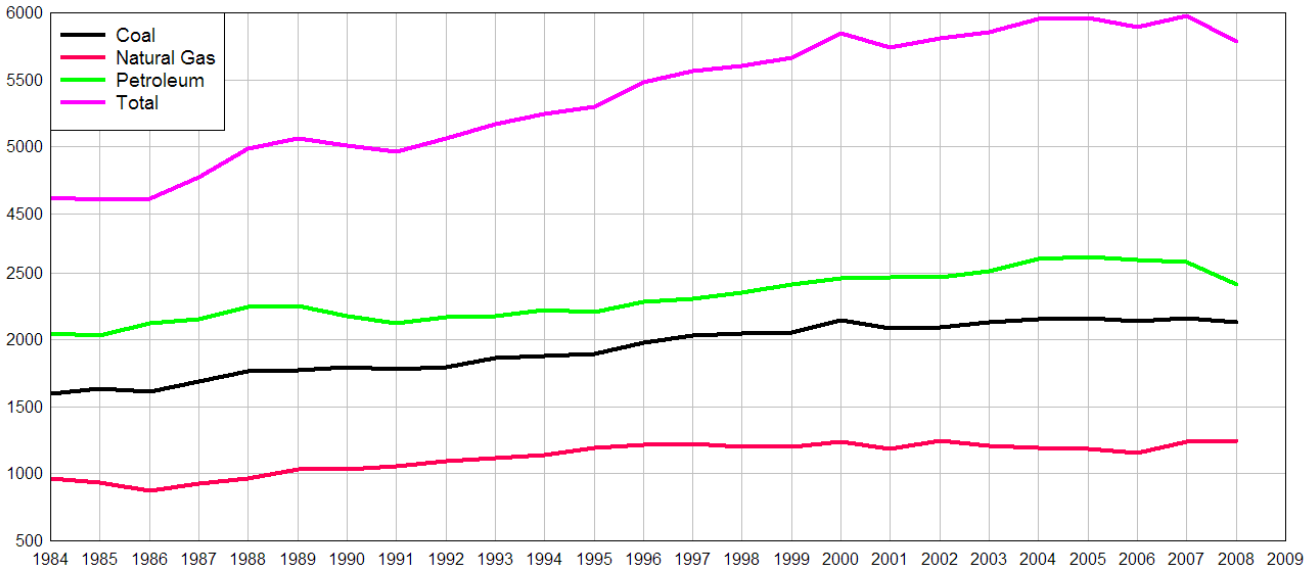


**Figure 4: Carbon dioxide emissions from consumption of different fuel sources (millions of metric tons of $CO_2$) [8].**

## 1.4 Energy Demand and Supply

Figure 5 shows the historical and projected energy demand up to 2030. According to [9] most of energy will continue to be supplied by liquid fossil fuel. The question right now is if there are enough supplies to accommodate this request.

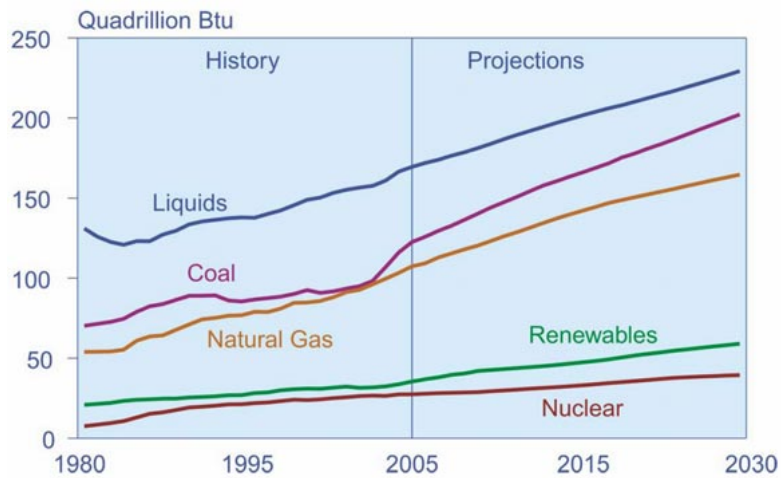


**Figure 5: Global energy demand forecast [9].**

According to the Hubbert Peak Theory in the worst case scenario, oil should be available until 2050 [10]; see Figure 6. The question now is if the oil supply beyond 2050 is really an issue. As the demand for oil increases, its price will increase as well. This means that more challenging off-shore oil reservoirs, e.g. Gulf of Mexico and costs of Brazil, will become economically viable. Also alternative hydrocarbon sources like for instance tar sands in Canada and Venezuela will become viable [11].

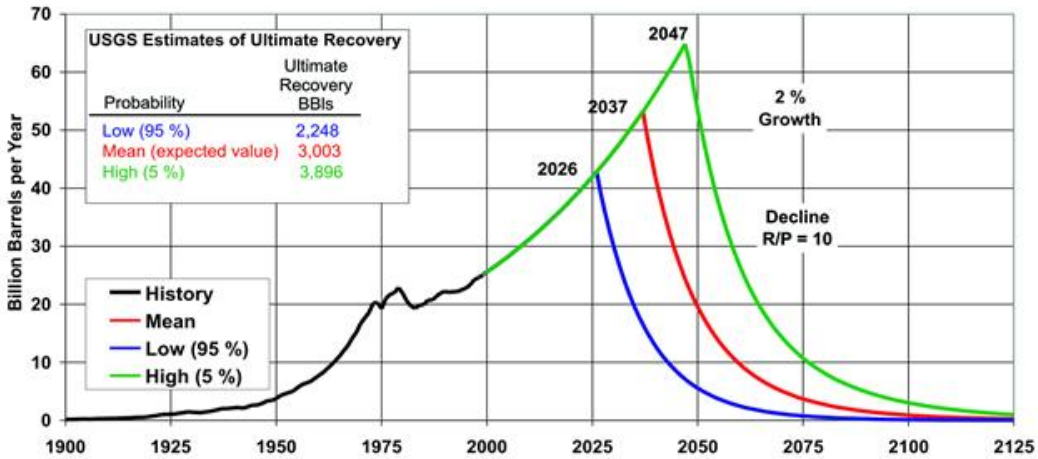


**Figure 6: Annual production scenarios with 2% growth rates and different resource levels (Decline R/P = 10).**

In the future the supply should not be an issue; the main problems will be the rise in oil cost and the carbon emissions. As previously stated the cost will rise due to the increase in demand and the more challenging locations where the oil will be extracted. On the other hand carbon emissions will be a greater concern than today because to produce oil from e.g. tar sands, shale oil, coal liquefaction is very carbon emission intensive.

## 1.5 Carbon Emissions Reduction

Because one of the main subjects of this thesis is highly efficient internal combustion engines, ICE, this section will review possible ways to reduce carbon emissions in the transportation and power generation sectors where ICEs are used to produce power.
In order to reduce $CO_2$ emissions nowadays the following alternatives are present on the market or available at research stage:

- Electric vehicles.
- Fuel cells.
- Biofuels.
- Improved vehicle aerodynamic and weight reduction.
- Highly efficient internal combustion engines.

When analyzing the above mentioned alternatives it is important to take a holistic view: well to wheel, cradle to grave analysis. For instance electric vehicles do not produce $CO_2$ emissions but if a well to wheel analysis is performed it is important to count for the $CO_2$ produced when burning fossil fuels to produce electricity.

### 1.5.1 Electric Vehicles[2]

In more than 200 years of evolution the energy density and the specific energy of the batteries[3] have improved significantly; Figure 7 shows a factor of 10 improvements in both parameters.

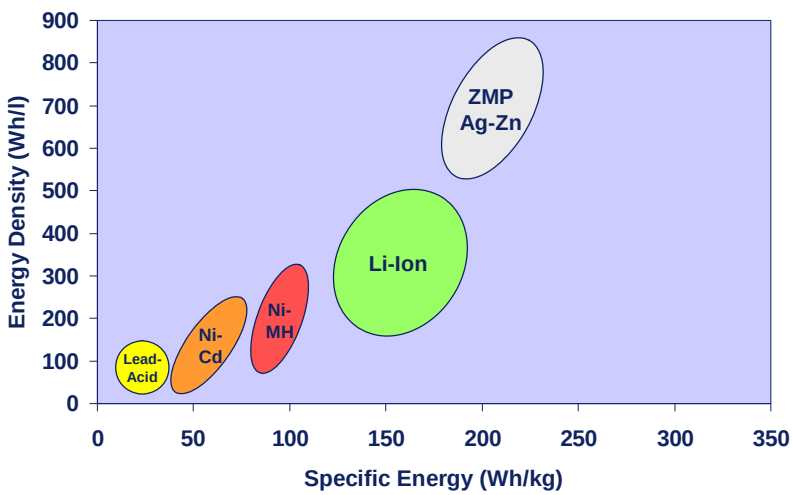


**Figure 7: Energy density and specific energy for different type of batteries [12].**

It is now important to compare the energy density and specific energy of an electric motor with the ones of a classical diesel and gasoline engines. The comparison is shown in Figure 8 and a substantial gap exists between these two energy sources. Assuming an efficiency of 20 % for an ICE[4], the resultant energy density and specific energy for an

[2] Electric vehicles consisting solely of a battery and electric motor as powertrain will be considered. Hybrid concepts, e.g. Toyota Prius, will not be discussed.
[3] They were invented by Volta in 1801.
[4] This is a very conservative number. Today a gasoline engine has a maximum efficiency of 30-35% while a diesel engine 35-40%.

ICE is at least a factor of 2 and 10 respectively higher than the best existing battery on the market. In practical terms this means that, for a given output power, an electric powertrain (motor plus battery) is much heavier than a conventional engine and for every time a classical vehicle has to stop to refill the tank an electric car has to recharge the batteries many more times.
According to [13] the maximum theoretical potential of advanced lithium-ion batteries, that have not yet been demonstrated to work, the energy density and specific energies are still only about 6 percent of crude oil.

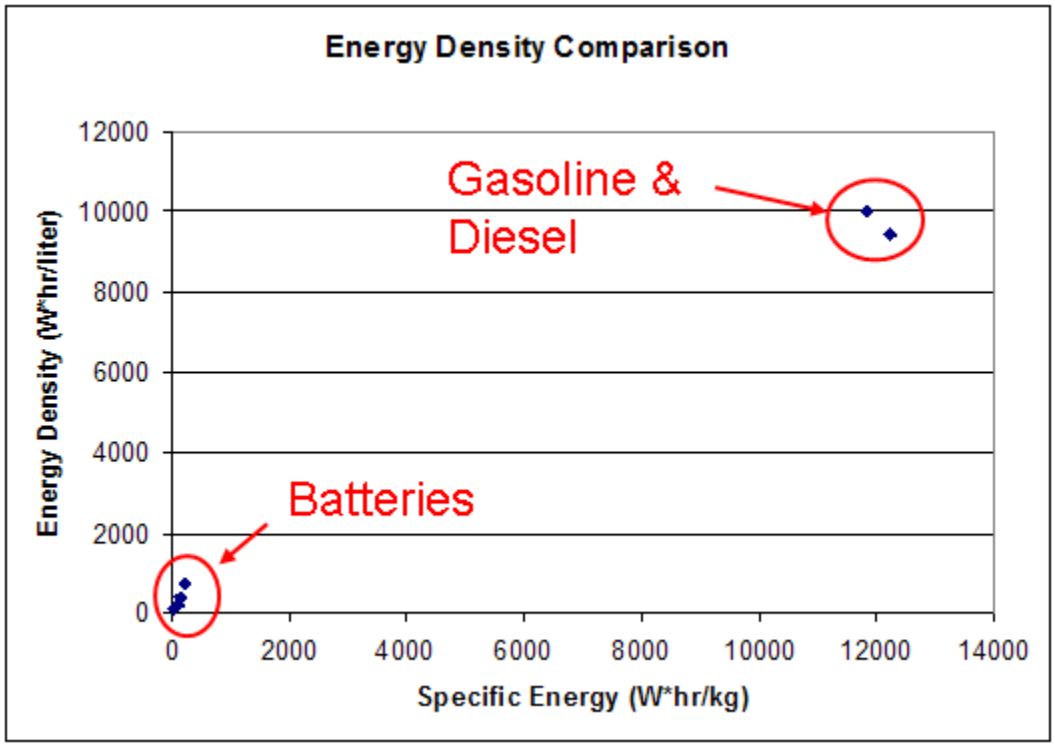


**Figure 8: Comparison between batteries and hydrocarbon fuels energy density and specific energy [12].**

It has now to be considered how the electricity to recharge the batteries of an electric vehicle is produced. Figure 9 shows that in the World 67% of the total electricity demand is produced using fossil fuels meaning that an internal combustion engine or a turbine have to be used to produce this power. The most efficient internal combustion engine in the world is the Wartsila-Sulzer RTA96-C that can reach 51.7% brake efficiency. This means that at least 48.3% of the fuel energy is lost and in addition there are energy losses on the power grid and from the battery to the wheel. In the best case this might result in an overall well to wheel efficiency between 37-41%. These numbers are comparable to a today's diesel ICE thus I personally do not see a real benefit in using fully electrical vehicles both because this thesis proves that it is possible to run a ICE at almost 50% brake efficiency and because of the limited mileage range of an electric car.

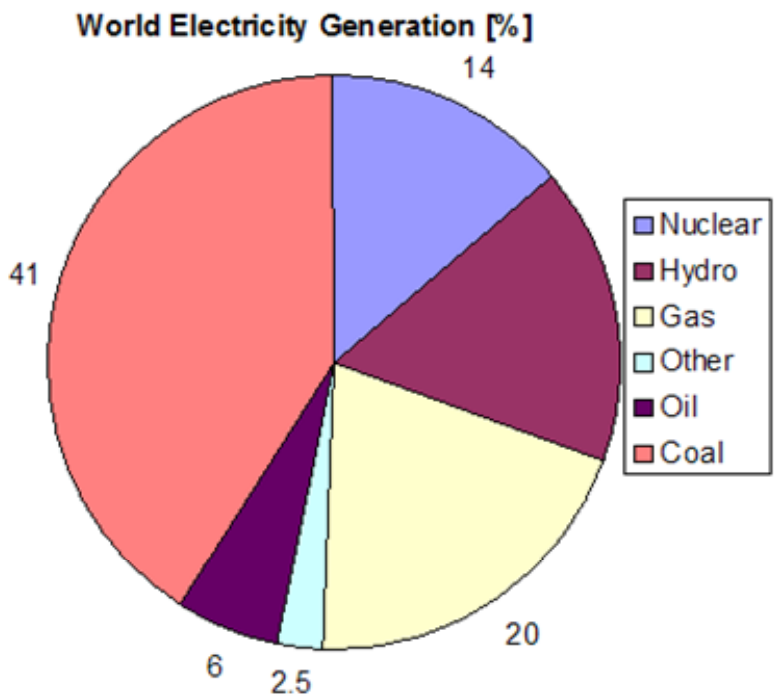


**Figure 9: Electricity generation in the world [14].**

### 1.5.2 Fuel Cells

As compared to ICE fuel cells are offering a practical efficiency of 60% [15] and there is not any emission of $CO_2$ from tank to wheels if hydrogen is used as fuel. Like in the case of the electric batteries there are many limitations concerning the fuel cells. First of all energy has to be spent to produce $H_2$ since pure hydrogen does not exist in nature [16]. Though better than electric batteries, the second issue is the energy density and specific energy that is lower than gasoline and diesel, see Figure 10, thus resulting in vehicle weight increase and frequent tank refills as explained in the previous section. The third and most important problem associated with fuel cells is that 60% efficiency is achieved at 30% of the maximum load while for an ICE the maximum efficiency is roughly at maximum load. At 100% load a fuel cell is roughly 30% efficient [18]; this limitation comes from the fact that increasing the load the current requirement increases and since the voltage on the electrodes is roughly constant this results in an increase in ohmic energy loss. As presented in [18] other issues associated with this concept are for instance:

- The energy that has to be spent to pressurize or liquefied hydrogen in order to store it.
- High energy cost. An internal combustion engine requires an investment of $30 to produce one kilowatt (kW) of power, the equivalent cost in a fuel cell is $3,000.
- If methanol is used instead of hydrogen the $CO_2$ production is still an issue and the output efficiency is lower.

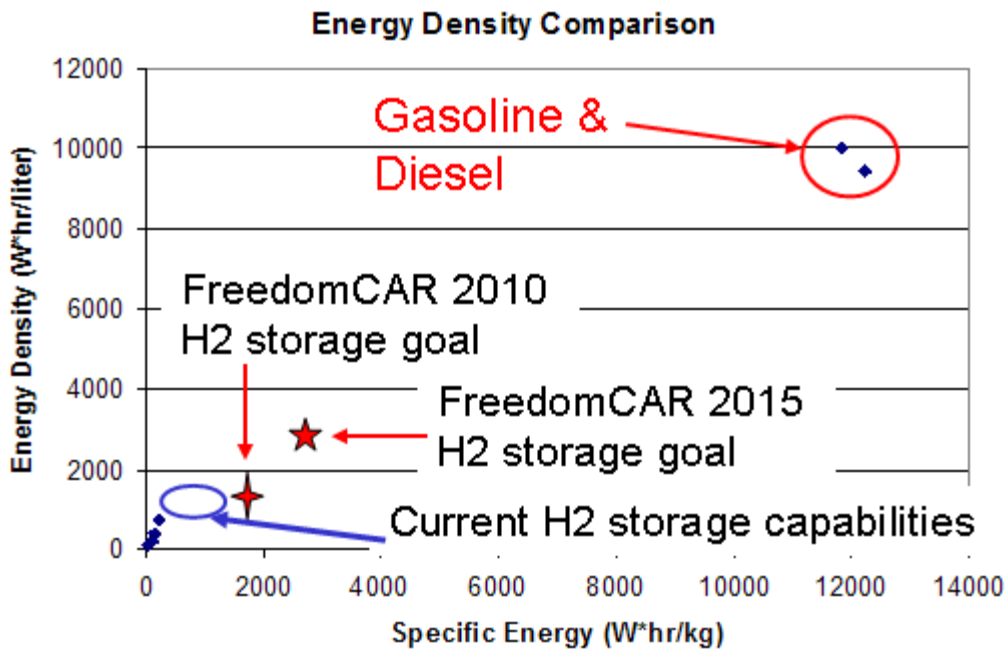


**Figure 10: Comparison among fuel cells, batteries and hydrocarbon fuels energy density and specific energy [12], [17].**

### 1.5.3 Biofuels

With very few exceptions, biofuels are an oxygenated version of hydrocarbon fuels. They do not have as high specific energy as diesel or gasoline but they are much better than fuel cells and electric batteries. The $CO_2$ emissions from biofuels is a complicated topic since there is the need of accurate accounting for the $CO_2$ that is consumed when for instance corn grows and the $CO_2$ that is produced while burning e.g. ethanol.

In [19] is presented a well to tank analysis for future non crude oil based fuel, these results are presented in Table 1. $\eta_{wtt}$ is defined as the ratio of effectively produced fuel energy and the energy supplied during the fuel production process plus the effectively produced fuel energy ( $\eta_{wtt} = \frac{m_{product} \cdot LHV}{E_{in} + m_{product} \cdot LHV}$ ), WTT-GHG is the amount of $CO_2$ produced during the well to tank process while Feedstock Specific $CO_2$ Credit is the amount of $CO_2$ that the feedstock has consumed during its growth (both these parameters are presented relative to the effectively produced fuel energy: $m_{product} \cdot LHV$ ).

Assuming that biomass based feedstock consume a considerable amount of $CO_2$ during their growth, a feedstock $CO_2$ credit can be defined. This number is independent on the fuel production chain and it is often used as an argument to strengthen the potential of biomass based energy to reduce greenhouse gas emissions from the transport industry. The account of a $CO_2$ credit for biofuels is however a topic under discussion [20]. It appears that biofuels can be a contributor to the fuel pool but are not the solution.

**Table 1: Energy efficiency and well to tank greenhouse gas emissions [19].**

| Feedstock | Product | | $\eta_{wtt}$ [%] | WTT-GHG Emissions [$g_{eq}/MJ_{fuel}$] | Feedstock Specific $CO_2$ Credit [$g_{eq}/MJ_{fuel}$] |
|---|---|---|---|---|---|
| Crude Oil | Diesel | min | 83 | 13.84 | |
| | | max | 89 | 13.84 | |
| Natural Gas | FT synthetic oil | | 63 | 16.47 | |
| | MeOH | | 68 | 11.69 | |
| Coal | FT synthetic oil | | 56 | 100.89 | |
| | MeOH | | 58 | 92.42 | |
| Biomass | FT synthetic oil | min | 43 | 5.00 | -70.80 |
| | | max | 54 | 2.40 | -70.80 |
| | MeOH | min | 45 | 5.00 | -69.10 |
| | | max | 65 | 2.30 | -69.10 |
| | Biodiesel | min | 42 | 12.30 | -75.40 |
| | | max | 53 | 71.40 | -75.40 |

### 1.5.4 Improved Aerodynamic and Weight Reduction

The power required at the wheels for a vehicle can be expressed as in Equation 1. Where $C_{rr}$ is the rolling resistance coefficient, M is the weight of the vehicle, g is the acceleration of gravity, $\rho_{air}$ is the air density, A is the front cross sectional area of the vehicle, a is the acceleration and V the speed.

**Equation 1:** $$Power = \left( Crr \cdot M \cdot g + \frac{1}{2} \cdot \rho_{air} \cdot A \cdot C_d \cdot V^2 + M \cdot a \right) \cdot V$$

Figure 11 shows that a substantial gap exists in power requirement during transient and steady state ( $M \cdot a = 0$ ) operations. More power means higher fuel consumption and consequently higher $CO_2$ emissions.

Assuming that $C_{rr}$ and A are features of a vehicle that can not be modified, an effective way to reduce the power requirement during a transient is to decrease the weight and/or the drag coefficient. The influence of vehicle weight during transient operations is shown in Figure 12. Table 2 shows the decrement in power requirement at 60 km/h if the vehicle weight decreases from 1400 kg to 1000 kg when it is accelerating at 0.496 m/s$^2$. On the other hand $\eta_{brake}$ represents the required efficiency that a vehicle whose weight is 1400 kg and its required power is 16.56 kW should have in order to emit the same amount of $CO_2$ as a vehicle of lower weight travelling at the same speed and acceleration.

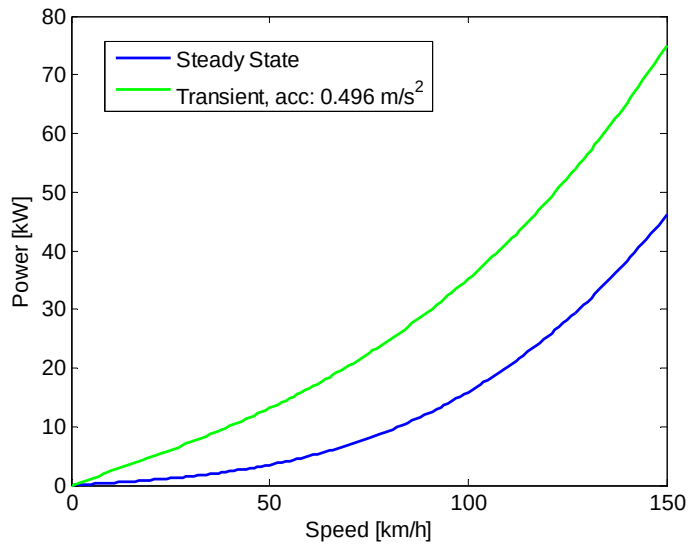


**Figure 11: Power required at the wheels as a function of the speed during steady state and transient operations[5].**

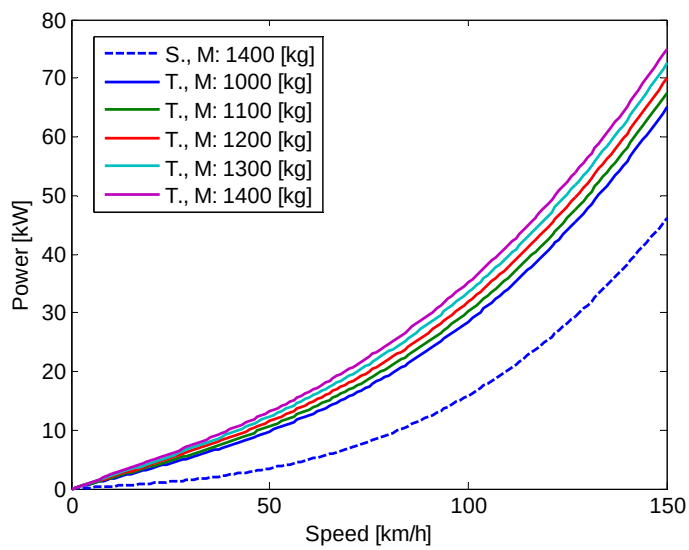


**Figure 12: Power required at the wheels as a function of the speed and vehicle weight during steady state and transient operations; acceleration: 0.496 [m/s$^2$], $C_d$: 0.35[-]. S.: steady, T.: transient.**

Figure 13 shows the power requirement decrement if the drag coefficient decreases from 0.35 to 0.15 [-]. Because the aerodynamic drag does not play a big role for speeds below 75 km/h, the power requirement demand decreases a modest 8.84% at 60 km/h if the drag coefficient is diminished from 0.35 to 0.15; see Table 3. As in the case of Table 2 $\eta_{brake}$ represents the required efficiency that a vehicle whose aerodynamic drag is 0.35 [-] and its required power is 16.56 kW should have in order to emit the same amount of $CO_2$ as a vehicle with lower drag travelling at the same speed and acceleration.

[5] It has been assumed: $C_{rr}$: 0.0106 [-], M: 1400 [kg], $\rho_{air}$: 1.18 [kg/m$^3$], A: 2.68 [m$^2$], $C_d$: 0.35 [-], a: 0.496 [m/s$^2$]. This value of acceleration was taken from the maximum acceleration in the NEDC test cycle.

**Table 2: Reduction in power requirement relative to the base case of a vehicle weighting 1400 [kg].**

| Weight [%] | 1000 | 1100 | 1200 | 1300 | 1400 |
|---|---|---|---|---|---|
| Power [kW] | 12.56 | 13.56 | 14.56 | 15.56 | 16.56 |
| Reduction [%] | 24.15 | 18.11 | 12.08 | 6.04 | 0.00 |
| $\eta_{brake}$ [%] | 43.45 | 41.34 | 39.23 | 37.11 | 35.00[6] |

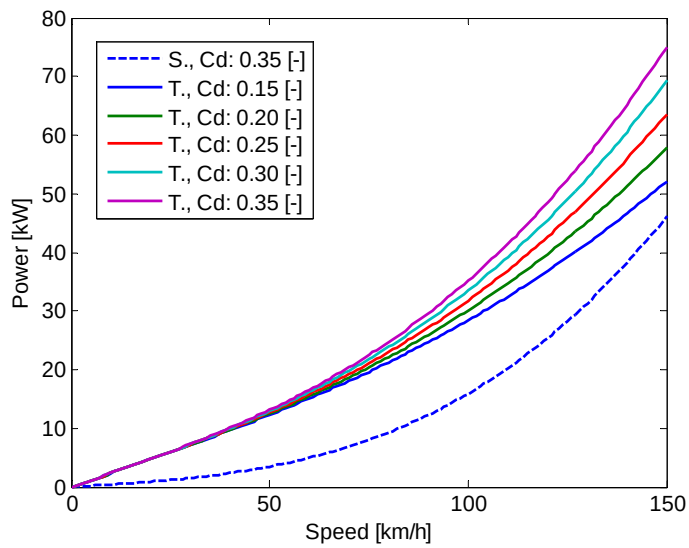


**Figure 13: Power required at the wheels as a function of the speed and drag coefficient during steady state and transient operations; acceleration: 0.496 [m/s$^2$], M: 1400 [kg]. S.: steady, T.: transient.**

**Table 3: Reduction in power requirement relative to the base case of a drag coefficient of 0.35 [-].**

| $C_d$ [-] | 0.15 | 0.20 | 0.25 | 0.30 | 0.35 |
|---|---|---|---|---|---|
| Power [kW] | 15.10 | 15.46 | 15.83 | 16.20 | 16.56 |
| Reduction [%] | 8.84 | 6.63 | 4.42 | 2.21 | 0.00 |
| $\eta_{brake}$ [%] | 38.09 | 37.32 | 36.55 | 35.77 | 35.00 |

Reducing vehicle weight and aerodynamic drag is an effective way to reduce the power requirement for a given speed and acceleration thus resulting in lower $CO_2$ emissions because of a decrease in fuel consumption. Usually a vehicle accelerates between idle and 50-70 km/h. In this speed range reducing the vehicle weight is more effective than a reduction in aerodynamic drag. A decrease in 100 kg and 0.05 in weight and drag respectively are realistic numbers, these two effects combined together would lead to 8.84% decrease in power requirement when travelling at 60 km/h and the acceleration is 0.496 m/s$^2$.

### 1.5.5 Highly Efficient Internal Combustion Engines

It has been shown that using biofuels and reducing the weight and aerodynamic drag of a vehicle are very effective ways to reduce $CO_2$ emissions in the atmosphere. If the brake efficiency of an engine can be increased from 40 to 50% it means that for a given output power $CO_2$ can be reduced up to 20%. In [72] [48] [49] the author of this thesis showed

[6] This was number was a realistic assumption made by the author.

that to reach 48-50% brake efficiency is viable and this can be accomplished by using gasoline Partially Premixed Combustion, PPC. The next sections of this manuscript will present the background towards high efficient engines, the tools to achieve simultaneously high efficiency and low emissions and finally the gasoline PPC concept that has been developed by the author between 2008 and 2009.

# 2 Towards High Efficiency and Low Emissions

## 2.1 Historical Focus

Internal combustion engines are more than hundred years old. Surprisingly Otto did not know anything about thermodynamics when he invented the four stroke cycle that is used in a large fraction of modern engines. The first person in the history who applied thermodynamics to internal combustion engines was Rudolf Diesel. His idea was to build a highly efficient engine. In order to accomplish this goal he tried to implement the Carnot cycle in which heat is added and rejected at constant temperature. Unfortunately the isothermal cycle does not generate enough work to overcome the friction of the engine thus resulting in an unviable solution. He then explored the viability of using a cycle in which heat is added either at constant volume or constant pressure. When developing his engine between 1892 and 1897 he realized that the fuel type was a key feature in order to have a highly efficient engine. At the time of the acceptance tests in 1897, Diesel's engine was able to produce 30.2% brake efficiency which was 35% higher than the Otto's engine of that time [24]. After the Otto and Diesel engine inventions, the compression ignition, CI, engine was mainly used in low speed ship and locomotive engines where fuel economy plays a big role and large size and weight can be tolerated. On the other hand the spark ignition, SI, engine was adopted in applications where the main selection criteria were low cost and high power to weight ratio, and the efficiency played a secondary role. During the 20$^{th}$ century the efficiency of the SI engine saw two big improvements. The first one was in the 1930s when leaded additives were added to gasoline in order to decrease the knocking tendency; by doing so the compression ratio of SI engines could be raised from 4:1 to 8:1. The second big step in increasing the efficiency of SI engines was in the 1970s during the oil crisis. To reduce throttle loss and increase heat capacity, Ford showed the "Programmed Combustion", PROCO, and Texaco the "Texaco Controlled Combustion System", TCCS, with lean burn and stratified charge [21] [22]. Both these concepts did not make it to production, on the other hand Honda did it with their similar system "Compound Vortex Controlled Combustion", CVCC, in 1979 [23]. In 1995 Mitsubishi introduced the gasoline direct fuel injection, GDI, with significant fuel consumption improvements at part load [26]. This engine could be operated at such lean mixtures that no throttling was needed even at part loads. During the 20$^{th}$ century CI engines saw a steady improvement in brake efficiency from 30% up to 44% [78].

In the middle of the 1990s when emission regulations became tougher there was the need to develop a combustion concept as efficient as a CI engine but with pollution levels comparable to the SI engine when a three-way catalyst is used. This was the time in which the HCCI concept, first invented by Onishi for 2-stroke engines in 1979 [27] and then in 1983 studied by Foster in a 4-stroke cycle [28], got renewed attention. With HCCI combustion a lean homogenous mixture of fuel and air is compressed until autoignition

takes place simultaneously throughout the combustion chamber near top dead center. The concept enabled to achieve efficiencies comparable to diesel engines and emissions in the range of spark ignition engines with three way catalysts; see Figure 14.

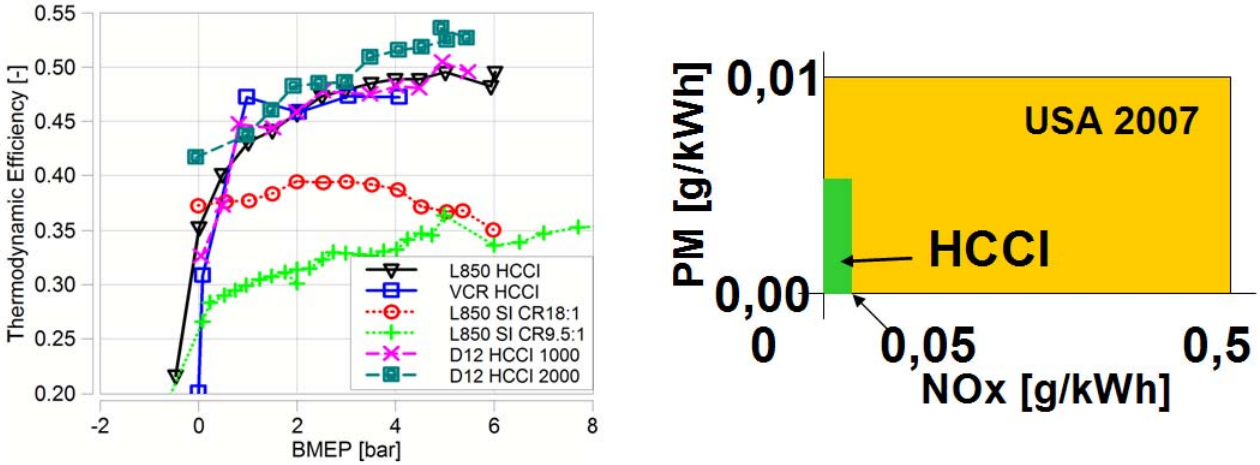


**Figure 14: Typical thermodynamic efficiency from HCCI engines (left) [25], typical NOx-particulate emissions from HCCI combustion (right) [29].**

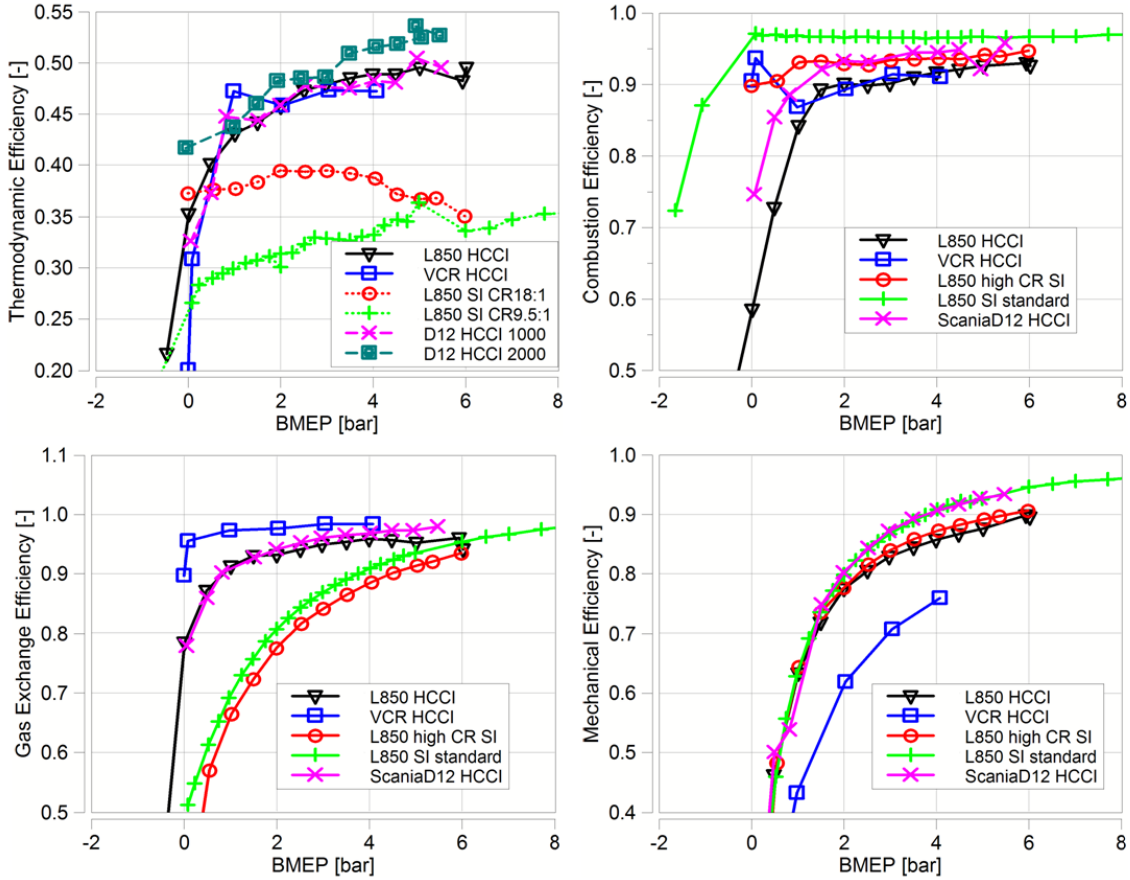


**Figure 15: Combustion, thermodynamic, gas exchange and mechanical efficiencies of three HCCI engines in comparison to SI operation [25].**

In 1998 the first results with supercharged HCCI at 14 bar IMEP were showed [30]. This was the first ever demonstration of high load HCCI outside the traditional low load range. In 2001, in [31], the viability of running a HCCI engine at 20 bar IMEP in a multi cylinder truck engine using closed loop combustion control was demonstrated. In recent time the viability of HCCI combustion at 16 bar BMEP and 16 bar IMEPg was demonstrated in [97] and [108] respectively. Much work was devoted to better understand the HCCI combustion process. Laser diagnostics as well as modeling were applied. It was concluded that combustion is far from homogeneous (even if the charge is perfectly mixed) and the fluid flow in the cylinder plays a significant role [31] [32] [33] [34][35] [36] [37] [109]. The brake efficiency of an engine is the product of combustion, thermodynamic, gas exchange and mechanical efficiencies. In [25] it was shown that HCCI can reach a thermodynamic efficiency of 50% or even slightly higher but there still are some issues with combustion, gas exchange and mechanical losses. In most operational points the combustion efficiency is around 90% and due to the relatively low load applicability range both gas exchange and mechanical efficiencies are rather low; see Figure 15. This results in an overall brake efficiency in the range of 32-42%; see Figure 16.

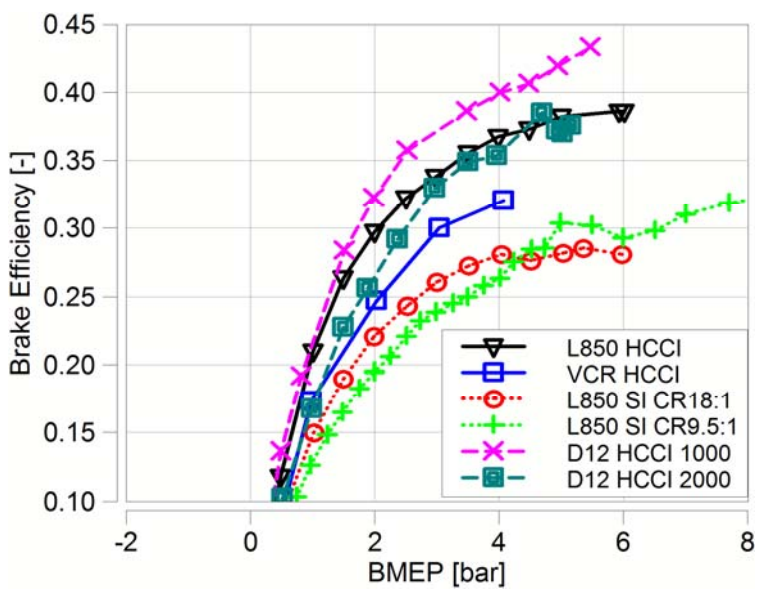


**Figure 16: Brake efficiency of three HCCI engines in comparison to SI operation [25].**

New combustion solutions outside the HCCI field were also studied in the first decade of the 21st century: Spark Assisted Compression Ignition, SACI and Partially Premixed Combustion, PPC. Spark Assisted Compression Ignition combustion can be considered as an intermediate process between regular flame propagation as in a Spark Ignition engine and auto ignition as in HCCI. Extensive investigation of SACI has been conducted, e.g. [38] [39] [40] [41] [42], but the applicability is limited since it is very hard to master; it has excessive cycle to cycle variations, high pumping loss and relatively high NOx. Partially Premixed Combustion, PPC, can also be considered as an intermediate process. PPC is a mix of the classical diesel combustion process where fuel is injected while combustion is occurring and HCCI using a fully premixed charge. With PPC, fuel is injected during the compression stroke and then mixed with air to some extent before

combustion starts. Ideally all the fuel should be injected before combustion starts but at the same time fuel injection should not be too early to generate violent combustion once reactions start. Very low NOx and soot can be achieved even at 15 bar IMEP net [43]. The drawback of diesel PPC is the fact that in order to have simultaneously low NOx and soot at high load, it is necessary to use 80% of EGR which lowers the combustion efficiency down to 85% and also the use of a low compression ratio piston (in this case 12:1) limits the expansion ratio. Using a thermodynamically better compression ratio like the standard diesel engine 18:1 will increase both pressure and temperature in the cylinder and thus substantially reduce the ignition delay. This will strongly limit the maximum diesel PPC load.

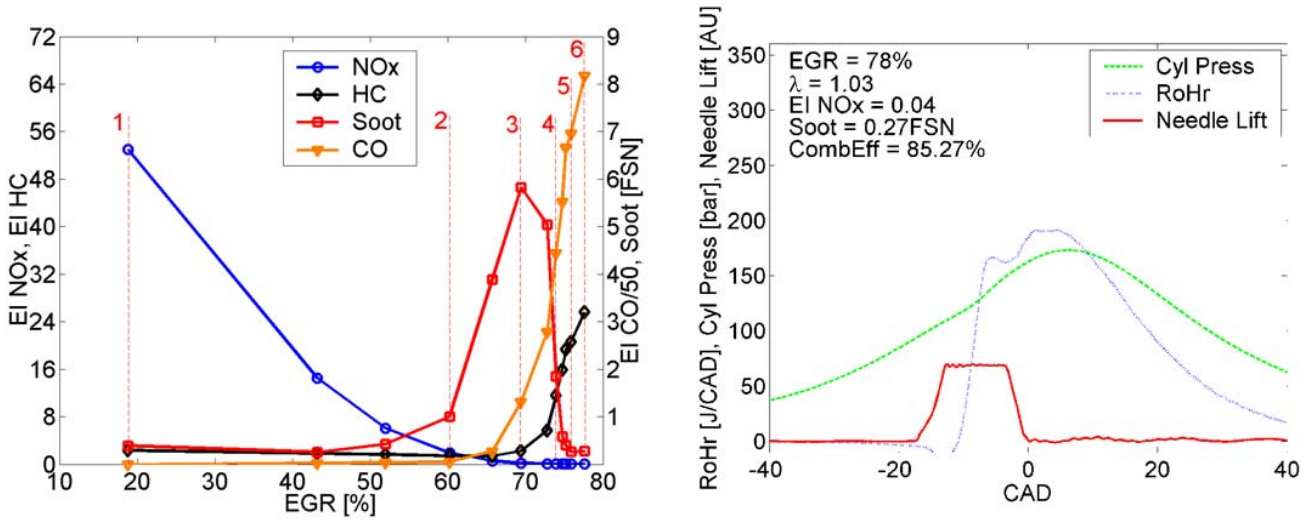


**Figure 17: Emissions trend as a function of EGR at 15 bar IMEP net (left). Rate of heat release, needle lift and cylinder pressure (right) at the maximum EGR rate [43].**

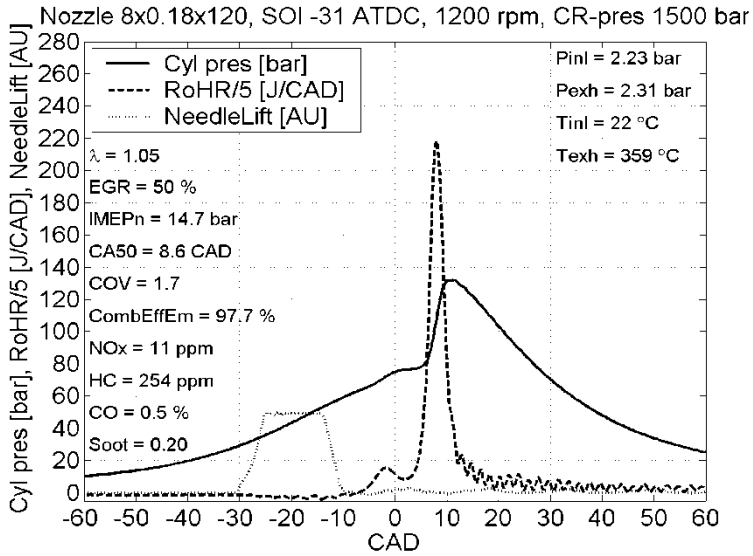


**Figure 18: Partially Premixed Combustion with low cetane diesel fuel [110].**

In 2005 the potential of using low cetane number (21) fuels in PPC was shown for the first time in [110]. Figure 18 shows the combustion behavior using a low cetane number diesel fuel in a heavy duty Scania D12. Fuel was injected at 30 CAD BTDC but with the long ignition delay combustion started rather late giving 50% heat released at an ideal 8.6 CAD ATDC. The long ignition delay meant that fuel had time to mix and thus NOx was only 11 ppm even with a moderate EGR amount of 50% at a load of 15 bar net IMEP. Also soot was low with only 0.2 FSN. The combustion efficiency approaches 98% even when running very close to stoichiometric with 50% of EGR.
Starting from 2008 it was then decided to study low cetane fuel PPC. All the steps followed to develop a robust and reliable gasoline Partially Premixed Combustion concept capable to achieve very high efficiency, low acoustic noise and low emissions were produced by the author and the results are presented in the next sections.

## 2.2 Review of High Efficiency and Low Emissions PPC Combustion Concepts

Since 2006 there has been increased interest in developing combustion concepts that can offer indicated efficiency in the order of 50% or above, achieve very low NOx and smoke, and run the whole load range (up to 25-30 bar IMEPg) without drawbacks in combustion control and acoustic noise. So far two main concepts have been shown: gasoline partially premixed combustion and gasoline-diesel PPC. The first one was introduced by Kalghatgi in 2006 (though something similar was shown in 2005 in [110]) and later extensively studied at Lund University followed by some work performed at the University of Wisconsin while the second concept was proposed and in depth studied by Kokjohn, Splitter and Hanson once again at the University of Wisconsin. Very similar to the Wisconsin fumigation concept is a variant of the HEDGE project proposed by Southwest Research Institute in 2004, in which a highly diluted stoichiometric gasoline-air mixture is ignited by means of a micro diesel injection.

### 2.2.1 Gasoline Partially Premixed Combustion

Low NOx and soot can be simultaneously achieved if combustion takes place under homogenous or very low stratified conditions e.g. HCCI or PPC combustions. When working with a compression ignition engine running with diesel fuel and having a compression ratio in the range of 15-16, the load region in which the engine can run in PPC mode is limited to 5-6 bar gross IMEP. If this upper load PPC boundary has to be increased a piston with much lower compression ratio has to be used and in addition an intolerable amount of EGR has to be introduced in order to keep separated the start of the combustion and the end of injection thus resulting in low stratification when ignition occurs [43].
In order to keep the end of injection and the start of the combustion sufficiently separated at high load without using a low compression ratio piston and/or lot of EGR, it might be convenient to use a fuel that is more resistant to ignition as compared to diesel. In 2006 Kalghatgi proposed to inject gasoline fuel in a compression ignition engine. In his concept fuel-air mixing is promoted by the high resistance of gasoline to ignite. Kalghatgi studied this gasoline PPC concept both in heavy and light duty engines. In [67] he ran a single cylinder Scania D12 up to 14.86 bar gross IMEP achieving 46% indicated efficiency, soot 0.36 FSN and 1.21 g/kWh of NOx. In these experiments the engine had a

compression ratio of 14, it was running at 1200 rpm using regular European pump gasoline and all the tests were performed using single injection and 25% of EGR. In this paper he emphasized the importance of the fuel-air stratification to achieve ignition in gasoline PPC. For the same fuel amount, if injection occurs very early in the cycle the combustion will result very unstable or misfire might occur. On the other hand if the injection event is near TDC, the more stratified mixture will result in a more stable combustion. In [68], using the above mentioned HD engine, Kalghatgi studied the influence of an early pilot injection as a tool to reduce the maximum pressure rise rate and the IMEP fluctuations. Using 23% of fuel in the pilot at -150 TDC and 25% of EGR, he ran the engine up to 15.95 bar gross IMEP achieving 0.07 FSN of smoke, 0.58 g/kWh of NOx and a 179 g/kWh as gross indicated specific fuel consumption (ISFC). A remarkable 47% gross indicated efficiency (ISFC: 174 g/kWh) was reached at 12.86 bar gross IMEP while keeping 0.39 g/kWh of NOx and 0.19 FSN of soot. Starting from 2009 the focus shifted from heavy duty to light duty engines. Using a half liter single cylinder engine with a compression ratio of 15.8, in [81], it was shown that at 12 bar gross IMEP and 2000 rpm, NOx can be controlled by the amount of EGR both for diesel and gasoline while the amount of soot produced is directly proportional to the ignitibility properties of the fuel (i.e. different ignition delay); a gasoline with a ON of 95 produces almost 7 times less soot than a gasoline with 84 as ON. In this paper it was shown that a gasoline with an ON of 84 can run at 3000 rpm in PPC mode while a higher octane number fuel can not because the ignition delay is too long and misfire occurs. At 12 bar gross IMEP the ISFC was in the range of 203g/kWh and this number seemed to be fuel properties independent. At the end of 2009 the fuel effects were studied at 4 and 10 bar gross IMEP in an EGR sweep using diesel and four gasoline fuels with different ON (91, 84, 78 and 72) [82]. At lower load all the fuels showed the same NOx trend, soot was almost zero for the gasolines while for diesel exponentially grew up to 2 FSN using 50% of EGR. As in the case of 4 bar gross IMEP, at 10 bar IMEP all the fuels showed the same trend for NOx emissions. Using a gasoline with 91 as ON, there was not any increase in soot while gasolines with a lower octane number showed an increment in particulate emissions when EGR increases but not as much as for diesel fuel. In the current work Kalghatgi stated that the best fuel for gasoline partially premixed combustion should have an octane number between 75 and 85 if the engine layout is kept constant. In this paper the efficiency of the engine was not discussed. Fuel effects were also studied in [83]. The question that was now to be answered was if the volatility of the fuel plays a role in NOx and soot formation or if low NOx and low soot can be achieved as far as there is enough separation between end of the injection and start of the combustion. An American diesel fuel was blended with 63 and 75% of an aromatic compound, Shellsol A100, thus resulting in a mixture with a ON in the range of 70 but a boiling point similar to diesel (T50 ~473 K ). These two fuels were compared to an European and an American diesel, and to gasolines with 91, 84 and 73 as octane number. The comparisons were carried out with two EGR sweeps at 4 and 10 bar gross IMEP at different engine speeds. Two main conclusions were presented in this paper. In the Kalghatgi gasoline PPC concept as long as combustion starts after the end of the injection, if CA50 and the ignition delay match the emissions (NOx & soot) will be the same regardless fuel volatility and composition thus his PPC concept is solely a function of the fuel ignitibility property rather than its volatility. This is the main difference with the concept proposed in this thesis. With the

Lund concept, combustion might start during the injection but as long as there is the correct λ and EGR combination, and the fuel has a boiling point range of gasoline, NOx and soot are simultaneously avoided. The second main Kalghatgi's conclusion was that increasing the aromatic percentage in the fuel will help to increase the gasoline PPC upper load limit. Aromatic fuels become relatively more resistant to autoignition compared to paraffinic fuels as the pressure increases while the temperature is held constant; this was in line with the results presented in his Octane Index concept [84].
In 2009 gasoline partially premixed combustion was studied by Hanson et al. at the University of Wisconsin [85] using a heavy duty engine with a compression ratio reduced from 16.1 to 9.1. Two load points were studied: 11.5 and 6.5 bar net IMEP at 1300 rpm. At the higher load the influence of pilot and main injection timings, EGR, inlet pressure and pilot-main ratio were studied. The lower load point was studied using single injection and two EGR rates, 0 and 30%. Using 40% of EGR and single injection at 11.5 bar IMEP an extraordinary 49.63% net indicated efficiency was reached while at 6.5 bar IMEP the efficiency was 50.20%. Such high efficiency was achieved keeping NOx and soot well below EPA high way heavy duty 2010 standards.

### 2.2.2 Dual Fuel Partially Premixed Combustion

According to Besonette [97] the HCCI load range can be extended by varying the fuel composition. For instance at 16 bar BMEP the fuel should have a cetane number of 27 while at 3 bar BMEP the cetane number should be at least 45. HCCI modeling at the University of Wisconsin showed that, for a given load and engine layout, the efficiency is not maximized neither with pure gasoline or pure diesel [86], for instance at 11 bar net IMEP the lowest indicated specific fuel consumption is reached using PRF80 and 50% of EGR. The basic idea at Wisconsin University is that different load-speed points require different fuel reactivity thus the best combustion concept should consist in port injecting gasoline and direct injection of diesel thus the reactivity of the mixture is varied by adjusting the diesel-gasoline ratio. Similar to the Kalghagi's concept, the combustion should start when the injection event is over.
In [86] simulations of dual fuel partially premixed combustion were carried out at 6 and 11 bar IMEP net; gasoline was simulated using iso-octane while diesel with n-heptane. At 1300 rpm, for a given load, the net indicated specific fuel consumption was plotted as a function of fuel reactivity (different PRF) and EGR. As previously mentioned the ISFC is minimized for a given combination of PRF and EGR, and this combination is load/speed dependent. It was stated that with dual fuel PPC, the combustion phasing is controlled by varying the fuel reactivity, i.e. ON, while the maximum pressure rise rate by adjusting the level of fuel stratification i.e. start of diesel injection. The second part of the paper was spent in validating the model results. The injection parameters were chosen using KIVA GA optimization tool [89]. According to this optimizer, optimum conditions are achieved by splitting the direct injection in two events. A first pulse at -60 TDC while the second at -33 TDC, 60% of the total diesel amount has to be injected in the first pulse; this numbers were found roughly load independent. The experimental results showed that both at 6 and 11 bar net IMEP, NOx and soot were far below US10 legislation. At lower load the net indicated efficiency was 46.55% while at higher load 49.02%. In both cases it was mentioned that the maximum pressure rise rate was within reasonable levels. Modeling work helped to understand the ignition process. According to the simulations,

at both loads, there are two separate combustion stages. Ignition occurs at the highest concentration of diesel. For diesel fuel, low temperature reactions are followed by a high temperature heat release. Gasoline does not ignite until diesel transitions to thermal ignition (high temperature reactions). In [87] sweeps in position and percentage of the first and second diesel direct injection, gasoline-diesel ratio and effective compression ratio, were carried out at 9 bar net IMEP, using the dual fuel PPC concept proposed in [86]. The experimental results were simulated in order to understand how combustion starts and evolves. By using a compression ratio of 16.1:1, 89% of the total fuel amount port injected (gasoline fuel), 62% of the total diesel amount in the first injection at -58 TDC while the remaining at -37 TDC, it was possible to achieve a remarkable 57% gross indicated efficiency (53% net indicated), NOx and soot well below US10 legislation and a maximum pressure rise rate of 9 bar/CAD. In [88] an optical investigation was carried out at 4.5 bar IMEP, 1300 rpm together with some simulation work, in order to understand the ignition process of dual fuel PPC. The injection parameters were the same as in [86] [87] i.e. 62% of diesel in the first direct injection at -58 TDC while the remaining at -37 TDC; 68% of the total fuel amount was port injected using as fuel gasoline. EGR was not used. By using two points location optical access (bowl and squish), the natural emitted light was analyzed trough Fourier Transform Infrared Spectroscopy (FTIR) in the 2-4.5 µm spectral region. According to this work combustion evolves as follow. The most reactive fuel globally initiates combustion at a common crank angle. Reactions are progressing from the most to the less reactive fuel. The reactions of the more reactive fuel steadily progresses from LTR (Low Temperature Reactions) to HTR (High Temperature Reactions). For the less reactive fuel LTR starts then they stop for 4-5 CAD and then the HTR starts. It was found that the speed of the high temperature reactions for gasoline and diesel is the same. In the conclusions it was stated that the fuel reactivity gradient in the combustion volume can be used to increase the delay from the most to the less reactive fuel regions thus extending the duration of the heat release rate; by doing so the maximum pressure rise decreases and higher load operations are possible as shown in [86] [87].

Dual fuel experiments were also performed at Southwest Research Institute within the HEDGE (High Efficiency Diluted Gasoline Engine) project. The HEDGE concept consist of using high boost (engine downsizing and higher efficiency), high EGR rate (knock mitigation, less throttle loss, less NOx, less heat transfer) and stoichiometric operation (the 3-way catalyst can be used) [90]. The concept was studied both in heavy and light duty engines [92] [93] [94] [95] [96]. The original concept was developed to use as ignition source a spark plug. Using a heavy duty engine in [91] the use of the spark limited the dilution limit to 30% of EGR and the load to 9 bar BMEP. When the ignition source was switched to a micro diesel pilot injection the dilution limit could be increased up to 40% and the load up to 10 bar BMEP at 1200 rpm because of the lower tendency to knock due to the higher dilution. The measured brake efficiency of this single cylinder engine was 35% while estimating this parameter from the friction data coming from a multi cylinder engine would have lead to 39% brake efficiency.

# 3 Tools to Achieve High Efficiency, Low Emissions and Low Acoustic Noise[7]

## 3.1 High Efficiency

### 3.1.1 High boost

By applying the first law of the thermodynamics to the power stroke leads to:

**Equation 2:** $$Q_{fuel} + H_{in} = W_{i\_g} + H_{ex} + Q_{ht} + Q_{chem} + H_{BB}$$

Where:

- $Q_{fuel}$ is the fuel energy delivered per cycle; $m_{f\_cycle} \cdot H_{LV}$.
- $H_{in}$ is the internal energy of the system; $c_p \cdot (m_a + m_f + m_{EGR}) \cdot (T_{cylinder} - T_{reference})$.
- $W_{i\_g}$ is the gross indicated work; $\oint P \cdot dV$
- $H_{ex}$ is the exhaust enthalpy; $c_p \cdot (m_a + m_f + m_{EGR}) \cdot (T_{exhaust} - T_{reference})$.
- $Q_{ht}$ is the heat transfer to the combustion chamber walls; $h_c \cdot A_{wall} \cdot (T_{cylinder} - T_{wall})$.
- $Q_{chem}$ is the combustion inefficiency (CO, HC and $H_2$ in the exhaust).
- $H_{BB}$ is the blow by loss.

Equation 2 can be rearranged as:

**Equation 3:** $$W_{i\_g} = Q_{fuel} + H_{in} - H_{ex} - Q_{ht} - Q_{chem} - H_{BB}$$

For a given fuel energy per cycle, $Q_{fuel} = const$, the output work, $W_{i\_g}$, might be increased by increasing the internal energy of the system i.e. higher inlet pressure. Assuming $H_{BB}$ and $Q_{chem}$ negligible, the increase of $W_{i\_g}$ as a function of inlet pressure depends solely on the internal energy, heat transfer and exhaust loss.
A simple thermodynamic model was build in order to quantify how $W_{i\_g}$ is affected by the inlet pressure and fuel energy per cycle[8]. The following assumptions were made:

1. CA50: 8 [ATDC].
2. CA90-10: 15 [CAD].
3. Engine geometry: standard Scania D13.
4. Inlet temperature: 303 [K].
5. Reference temperature: 298 [K].
6. Engine speed: 1250 [rpm].
7. Differential pressure exhaust minus inlet: 0.25 [bar].
8. Cylinder wall temperature: 450 [K].

[7] Throughout the text mean effective quantities (e.g. Fuel MEP) and different type of efficiencies will be mentioned. The definition of these parameters can be found in 7.3.
[8] The results obtained with the model presented in this section have the only objective to describe trends without giving precise quantitative information.

9. Heat transfer modeled with the Woschni equation [44]. The C1 and C2 constants were tuned to match the experimental results in [49].
10. The rate of heat release has been approximated with a Wiebe function.
11. EGR is added in order to have 1.35 as λ. If the inlet pressure was not enough to have λ without EGR higher than 1.35, EGR was set to zero.
12. The combustion efficiency was assumed to be 100%.
13. Lower heating value 43.8 MJ/kg, stoichiometric air fuel ratio 14.68.

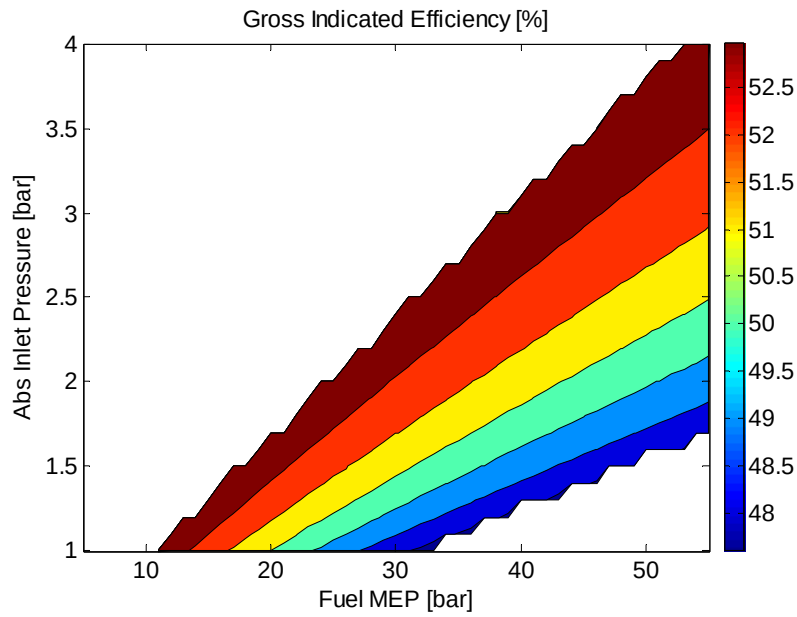


**Figure 19: Gross indicated efficiency as a function of Fuel MEP and boost level.**

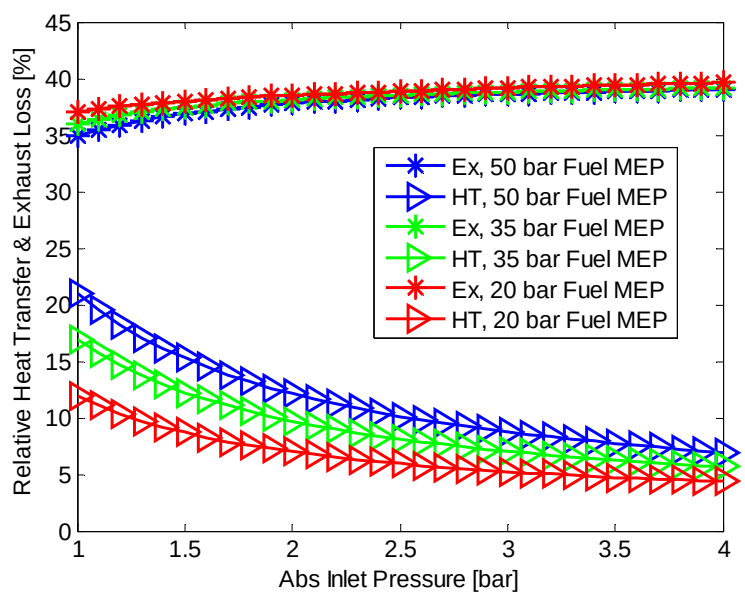


**Figure 20: Relative[9] heat transfer, HT, and relative exhaust loss, Ex, as a function of boost at three different fuel rates.**

[9] These two quantities are relative to the input energy per cycle.

Theoretically the inlet pressure can be set to an arbitrary value, in practice it is limited by the efficiency of the turbo system. Maximum practical values of turbo efficiencies are in the range of 55%. In the following diagrams the upper white area is due to inlet pressure values that would require turbo efficiency higher than 55% while the lower white area is due to running conditions in which λ is below 1.1.
The gross indicated efficiency as a function of inlet pressure and fuel mean effective pressure is shown in Figure 19. As expected by increasing the boost the efficiency substantially increases for a given Fuel MEP. Figure 20 shows that the increase in efficiency is due to the relative heat transfer that decreases faster than the increase in exhaust loss.

### 3.1.2 Combustion Phasing and Combustion Duration[8]

The previous paragraph showed that increasing the boost is an effective way to improve the efficiency. For a given boost level the efficiency can be maximized if the combustion is properly phased and it has the correct duration. Too early combustion phasing or too short combustion duration will result in high combustion temperature thus more heat transfer. On the other hand late combustion phasing or long combustion duration will lead to a decrease in effective compression ratio thus more exhaust loss. Using the model described in the previous section a sweep in CA50 and CA9010 was carried out at 50 bar Fuel MEP, 50% of EGR and 3.7 bar absolute inlet pressure. The exhaust and heat transfer trends previously described can be seen in Figure 21 and Figure 22. The minimizations of heat transfer and exhaust loss result in a maximization of the gross indicated efficiency when the combustion phasing is roughly at 6 ATDC and the combustion duration is at most 25 CAD long (see Figure 23); this result is in line with what the author found in [77]. As described in [45] the CA50 and CA9010 in which the gross indicated efficiency is maximized is a function of the load.

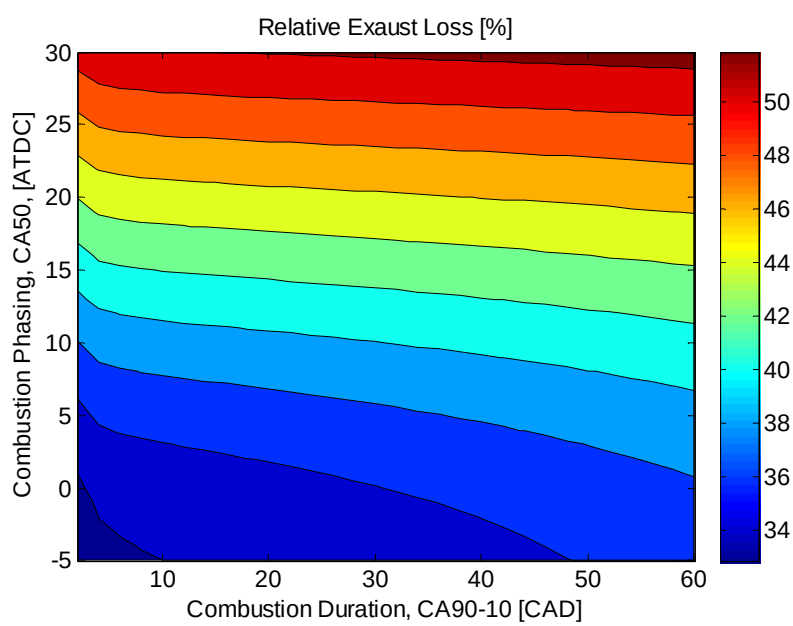


**Figure 21: Relative exhaust loss as a function of combustion duration and phasing at 50 bar Fuel MEP.**

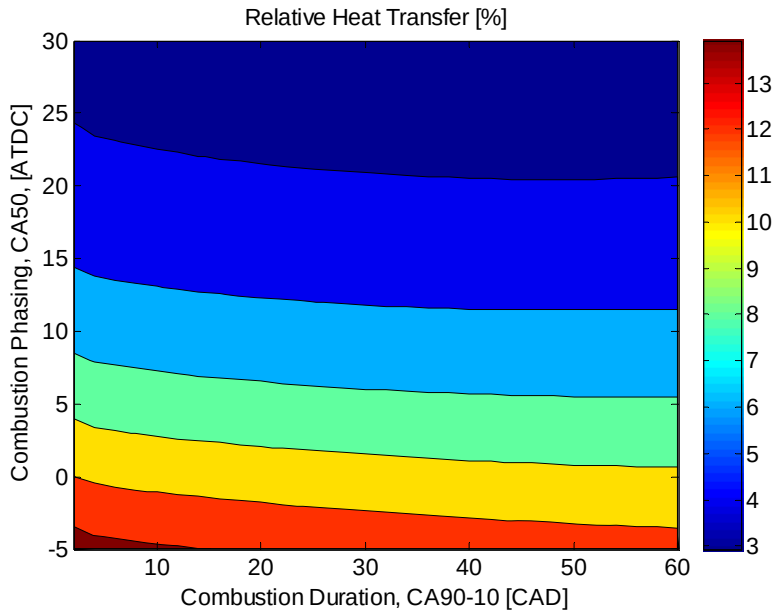


**Figure 22: Relative heat transfer as a function of combustion duration and phasing at 50 bar Fuel MEP.**

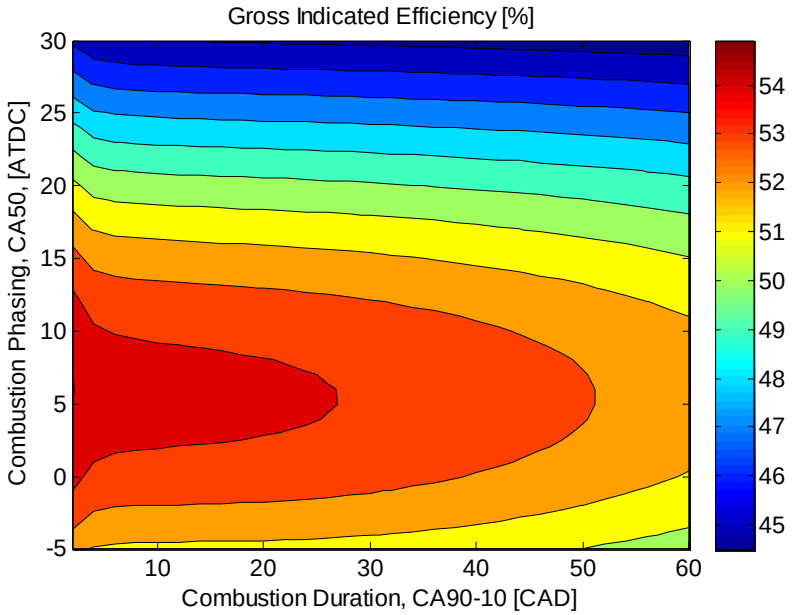


**Figure 23: Gross indicated efficiency as a function of combustion duration and phasing at 50 bar Fuel MEP.**

### 3.1.3 Squish as Thermal Insulator

Heat transfer can be reduced by minimizing the temperature difference between the combustion chamber walls and cylinder gasses or by decreasing the thermal conductivity of the combustion chamber walls. With stratified mixture e.g. classical diesel combustion or PPC most of the reactions are taking place in the bowl region; the combustion volume is thus divided in two main zones: a hot core (the bowl volume) and a much cooler area (the squish volume). Despite the different running conditions the purpose of Figure 24 is to give an idea of the different temperature field in homogenous and stratified combustions. If most of the reactions are kept in the bowl volume, the squish area acts as a thermal insulator and the efficiency of the engine can be increased because of a reduction in heat transfer; this way of thinking leads to the conclusion that a narrower umbrella angle is more beneficial than the one used in standard diesel engines.
In [66] a sweep was made in nozzle umbrella angle using partially premixed combustion. The heat transfer relative to the input energy per cycle steadily decreases from the baseline case, 155° UA, to 80° UA; see Figure 25. Little variation can be observed in exhaust loss while, as a consequence of the deterioration of the combustion efficiency, there is a drastic increase in combustion loss when the umbrella angle decreases from 100° to 80°. In [66] it was argued that for the combustion chamber under examination the most suitable umbrella angle was the 120° one. With 120° there is a decrease of 6% in heat transfer relative to the baseline case, 155°.

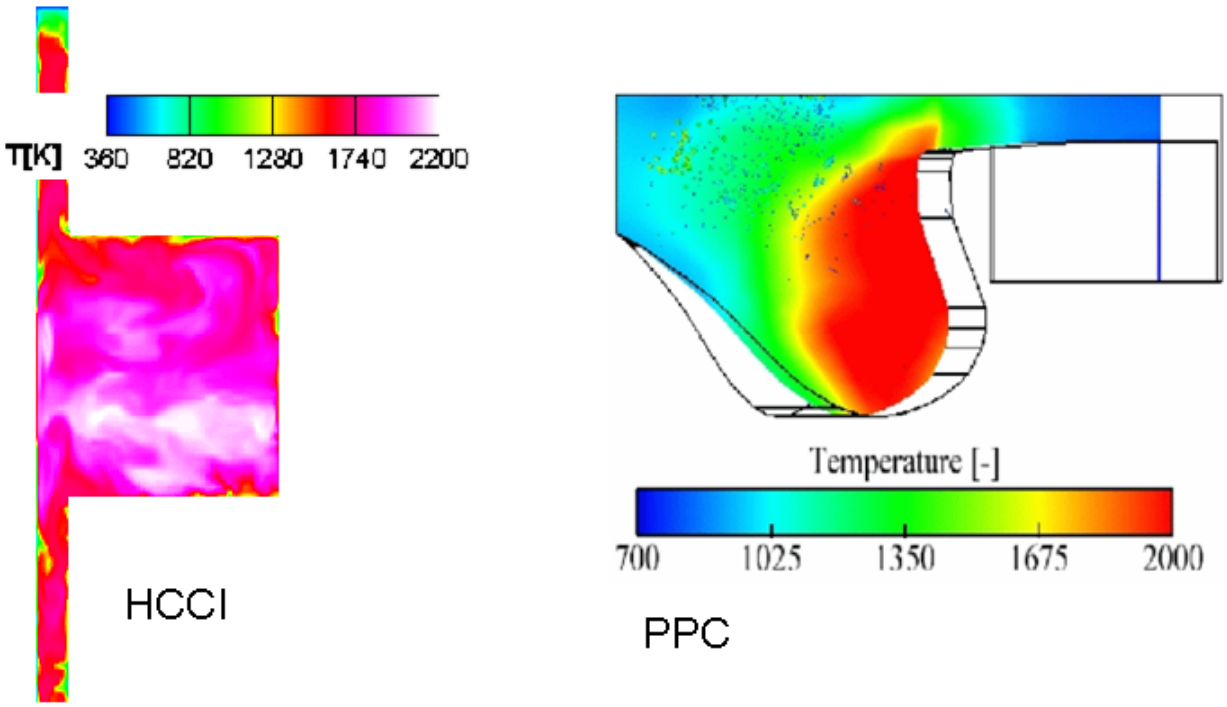


**Figure 24: Temperature distribution in HCCI and PPC combustions [65] [66].**

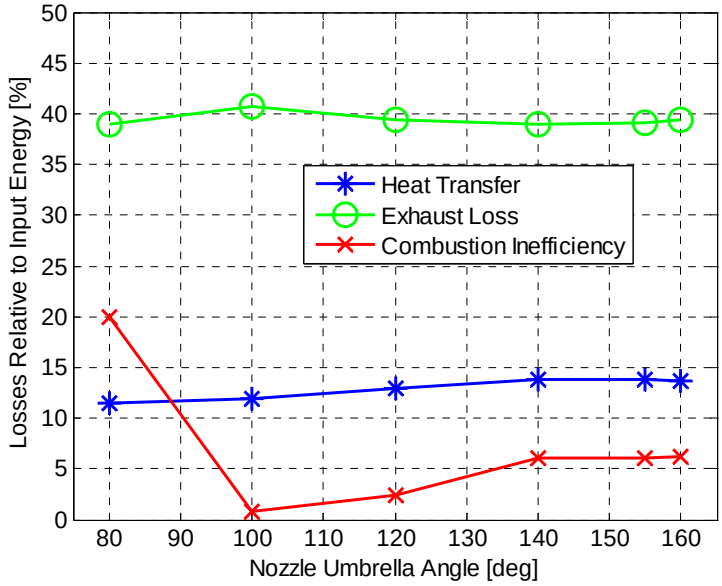


**Figure 25: Combustion, heat transfer and exhaust losses relative to the input energy per cycle as a function of the nozzle umbrella angle [65].**

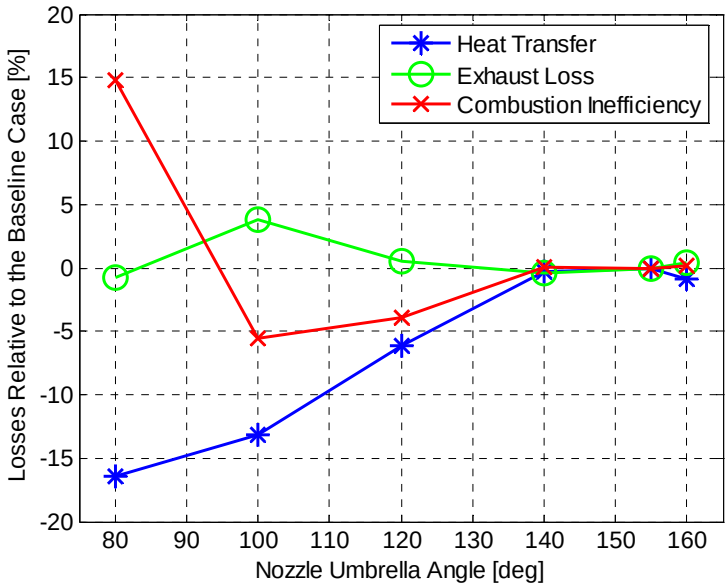


**Figure 26: Variation of heat transfer, exhaust and combustion losses relative to the baseline case (155° UA) [65].**

## 3.2 Low Emissions

### 3.2.1 Proper λ and EGR Combination: Reduction of CO, UHC and NOx.

In order to have simultaneously low CO, UHC and NOx, the maximum temperature inside the combustion chamber during the cycle should be between 1500 and 2000 K; see Figure 27 and Figure 28. If the maximum temperature is higher than 1500 K CO and UHC are oxidized while if the temperature is below 2000 K NOx formation is avoided. Figure 29 shows the maximum cycle temperature as a function of EGR and lambda for a homogenous mixture which burns at constant volume[10]. According to the diagram if λ is between 1.3 and 1.6 and EGR is in the range 45 – 55%, the maximum cycle temperature is always within the above mentioned temperature range thus it should be theoretically possible to achieve simultaneously low CO, UHC and NOx, furthermore this observation seems to be load and fuel independent.

This concept was successfully applied both to a single cylinder Scania D12 and a Scania D13 using a variety of different fuels at different load points; the results are shown from Figure 30 to Figure 33. It is interesting to note that this strategy works not only with gasoline type of fuels but with diesel as well. So far in the literature the CO-NOx trade-off in diesel PPC has been considered as un-avoidable obstacle if low temperature combustion has to be used in the whole operating load range e.g. [43].

It has to be remembered that in real life applications the maximum amount of EGR is limited by the available inlet pressure which is a function of the available energy at the turbine. By increasing the amount of EGR the exhaust temperature decreases thus less energy is available to drive the compressor. Assuming that the maximum turbo system efficiency is 55% and the relative excess of air is 1.35 Figure 34 shows the maximum EGR rate as a function of Fuel MEP and compression ratio. Higher compression ratio means more expansion resulting in less available energy at the turbine thus less EGR can be used. Using a compression ratio of roughly 14 allows having high enough temperature in the exhaust to drive the turbine to boost the engine enough to use 45-50% of EGR. It has to be kept in mind that it is not convenient to decrease the compression ratio below 14 in order to allow the use of more than 50% of EGR in order to avoid penalties in gross indicated efficiency; see Figure 35.

[10] If combustion takes place around TDC (CA50 ~5-10 [ATDC]) and its fast enough (CA90-10 ~5-15 [CAD]), it might be considered occurring at roughly constant volume since the piston does not move significantly. $T_{MAX} = T_{SOC} + \dfrac{H_{LV}}{c_v \cdot AF_{st} \cdot \left[ \lambda \cdot \left( 1 + \dfrac{EGR}{1 - EGR} \right) + 1 \right]}$.

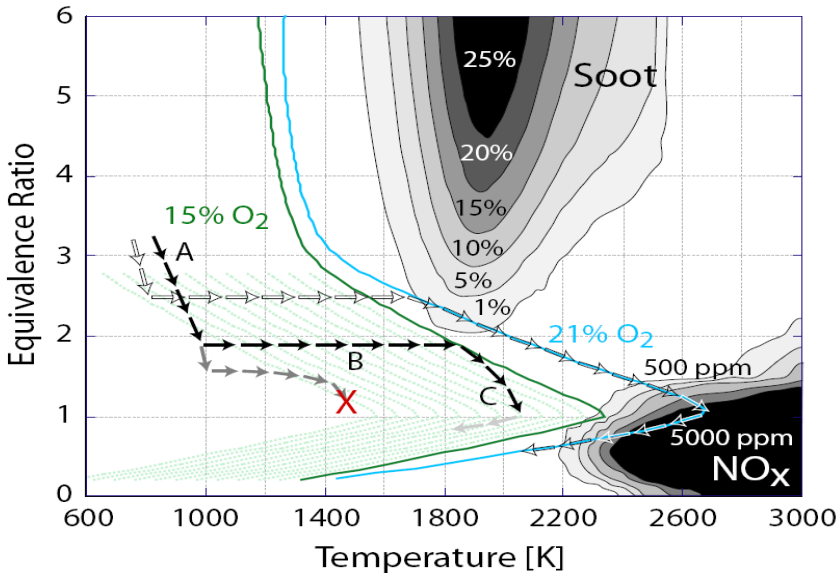


**Figure 27: Soot and NOx production zones as a function of temperature and equivalence ratio. The diagram is obtained from a homogenous reactor simulation of air / n-heptane mixture [46].**

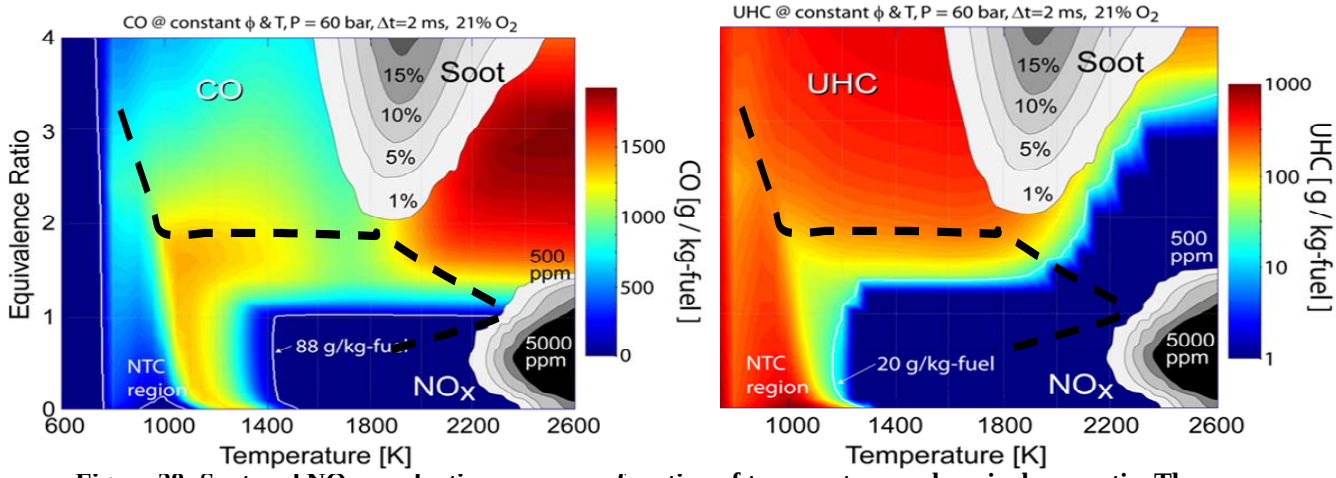


**Figure 28: Soot and NOx production areas as a function of temperature and equivalence ratio. The left plot contains the specific CO while the right one UHC formation zones [47].**

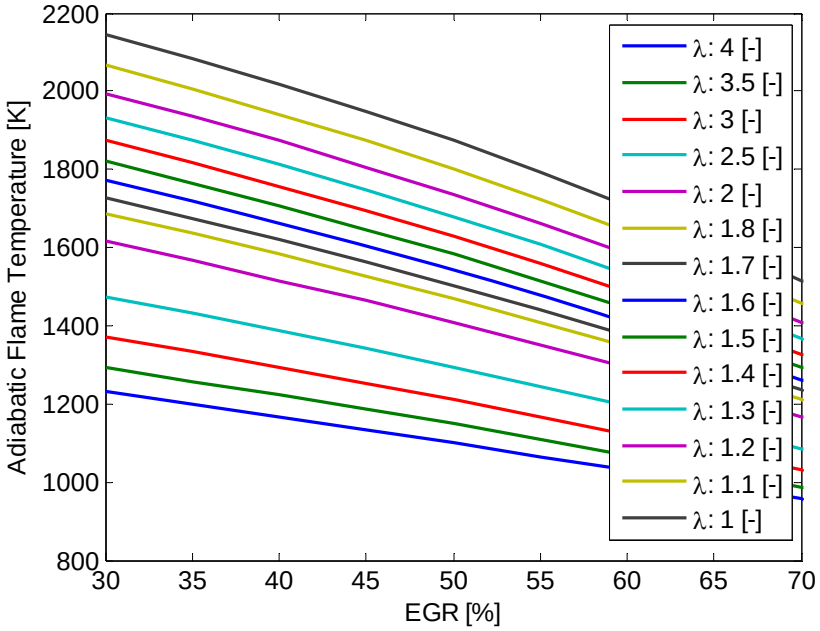


**Figure 29: Adiabatic flame temperature as a function of EGR and λ calculated assuming constant volume combustion.**

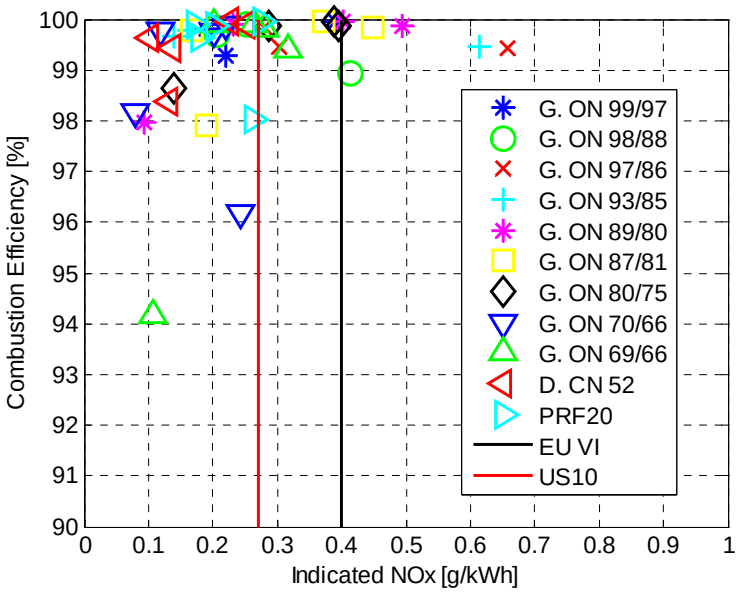


**Figure 30: Combustion efficiency as a function of specific NOx and fuel type in the Scania D13 between 5 and 26 bar gross IMEP [49].**

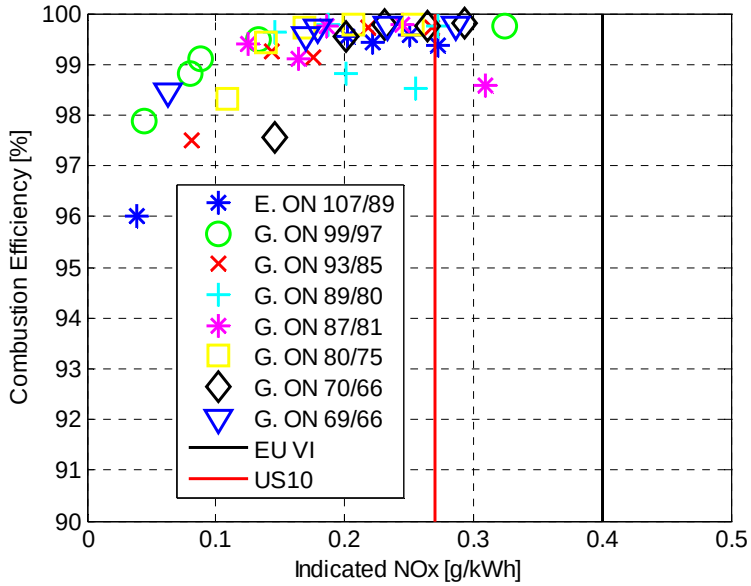


**Figure 31: Combustion efficiency as a function of specific NOx and fuel type in the Scania D12 between 5 and 18 bar gross IMEP [48].**

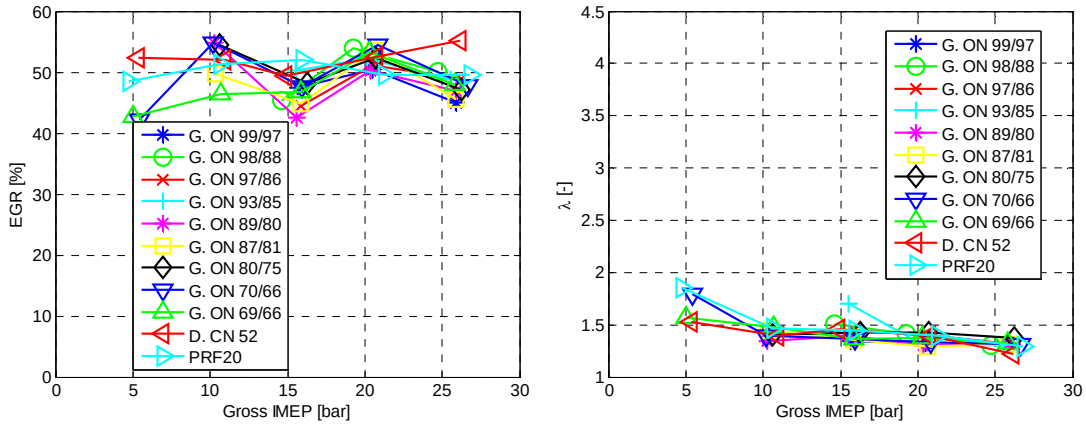


**Figure 32: λ and EGR as a function of load and fuel type in the Scania D13 [49].**

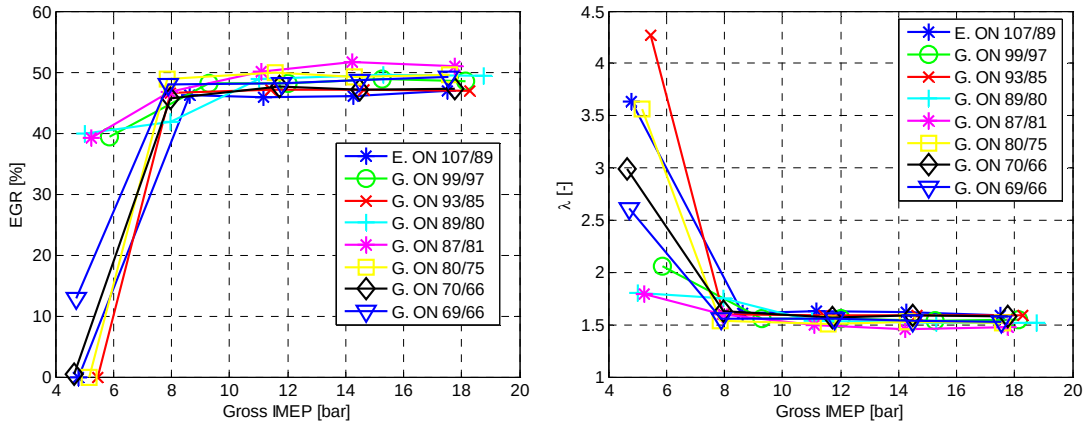


**Figure 33: λ and EGR as a function of load and fuel type in the Scania D12 [48].**

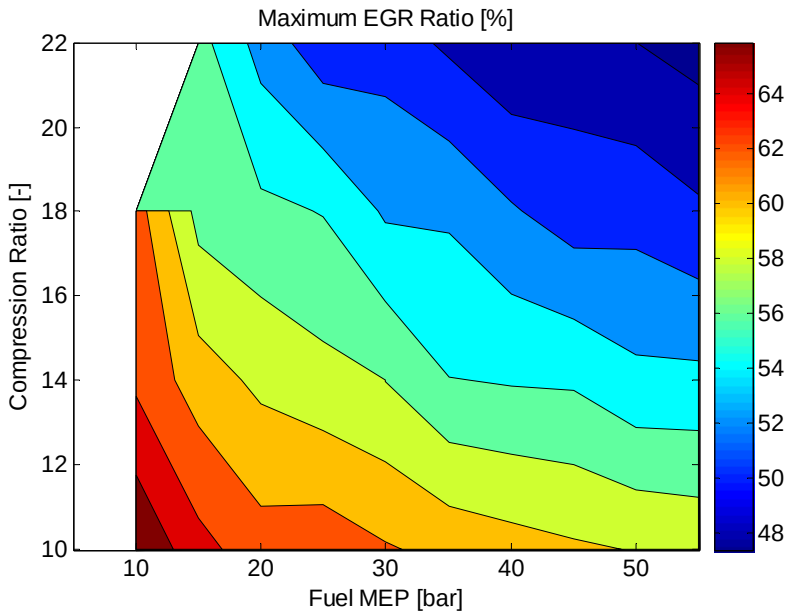


**Figure 34: Maximum EGR rate as a function of Fuel MEP and compression ratio. The model used is the one described in 3.1.1; the assumptions missing in 3.2.1 can be found in 3.1.1.**

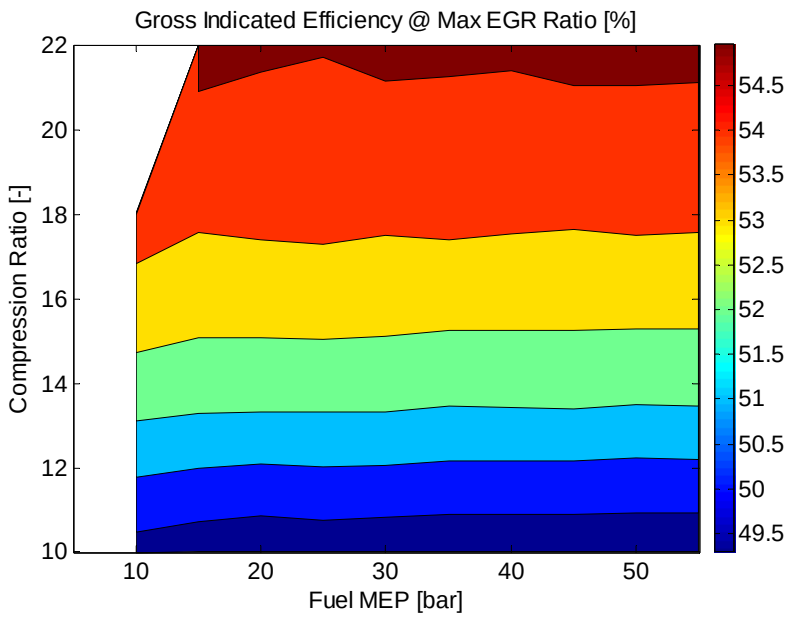


**Figure 35: Gross indicated efficiency at maximum EGR rate as a function of Fuel MEP and compression ratio.**

### 3.2.2 Injection System: Low Soot

It has been shown in the previous paragraph that to decrease NOx low enough, EGR has to be used up to 50%. For a given stratification level, increasing EGR leads to higher soot production because the temperature is not high enough to fully oxidize particulate matter (Figure 27). According to [58] the use of EGR increases the liquid jet penetration thus it is more likely that the fuel will impinge on the piston or combustion chamber walls

resulting in higher soot production because of pyrolysis. It is commonly accepted that smaller nozzle holes and high injection pressure are effective tools to limit particulate matter production in low temperature combustion. With smaller holes the liquid spray length is shorter thus it is less probable that the fuel jet will impinge on the combustion chamber surface [59] [60] and according to [59] [60] [61] the local equivalence ratio decreases for a given spray length. The mean fuel drop diameter decreases with smaller orifices but this has an almost negligible effect on soot production in normal engine running conditions [60]. Higher injection pressure relatively influences the spray length; on the other hand it increases the mixing in the head vortex which is the place where most of the soot is formed in LTC [58] and it also reduces the residence time of the fuel in the jet [61]. By using small orifices and high injection pressure smokeless combustion can be achieved at low load in a heavy duty diesel engine [61] and in [60], for the same NOx, a reduction of 94% of soot was achieved at 50% load (high EGR case) in a diesel engine. In [60] the particulate matter reductions were greatest at lower load conditions with minimal improvements at 100% load where the baseline engine already had high injection pressure.

### 3.2.3 Fuel Type: Low Soot

In 3.2.2 the effectiveness of using smaller nozzle holes and high injection pressure in reducing soot was shown. In [49] it was discovered that the fuel molecule plays a fundamental role in soot production. Using 50% of EGR and 1.35 as relative excess of air, very low soot can be achieved in the whole load range (i.e. from idle to 25 bar gross IMEP) if a fuel has an average carbon length like gasoline; the cetane number, i.e. ignition delay, does not play a major role.

In [49] it was noticed that for the same inlet conditions diesel was producing much more soot as compared to the nine gasolines under examination; see Figure 36. It was thought that this might have been due to:

- Ignition delay, more mixing.
- Presence of $O_2$ in the fuel molecule.
- Air entrainment in the spray.
- Liquid spray length.
- Fuel composition.

Each of the five hypothesis was analyzed to understand which one was the main factor influencing the soot formation process.

If fuel and air are not sufficiently mixed prior combustion, soot can be formed in the rich combustion zones. The mixing period, CA10-EOI, is an effective parameter to measure how premixed is the fuel-air mixture when combustion takes place; see Figure 37. If this number is bigger than zero it means that combustion starts when the injection event is completed. Fuels G. ON 69/66 and G. ON 70/66 are the only two gasolines with a mixing period similar to diesel but slightly longer. It might be argued that this small difference is sufficient to avoid soot production. In order to rule out this hypothesis PRF20 was tested since it has roughly the same CN as diesel [50] but a T50 similar to gasoline. The mixing period and the rate of heat release (RoHR) of PRF20 and diesel are matching quite well; see Figure 37. Despite this agreement the soot levels are very different; this implies that mixing is not the primary cause of this broad gap. Oxygen in the fuel molecule can be also excluded since G. ON 69/66 and G. ON 70/66 do not contain any $O_2$.

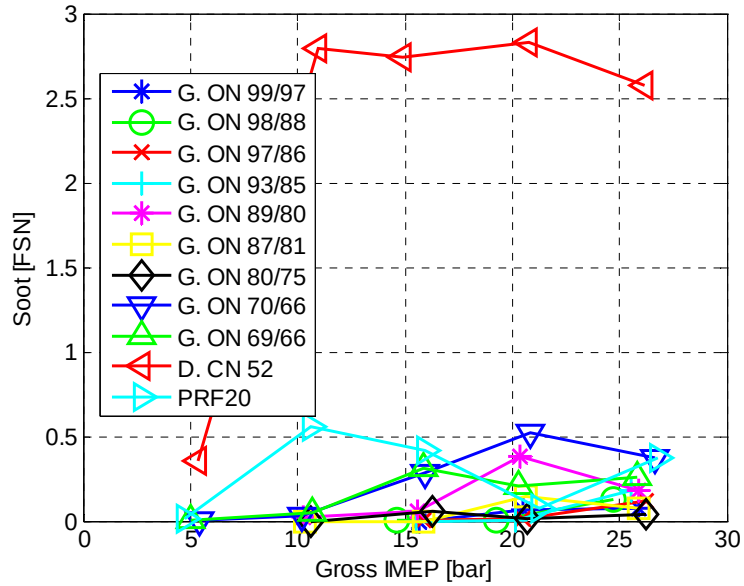


**Figure 36: Soot as a function of load and fuel type [49].**

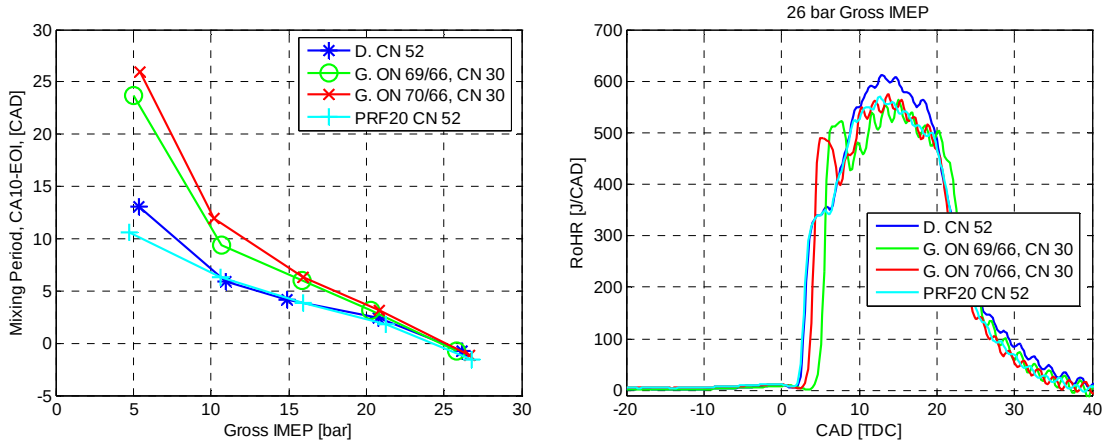


**Figure 37: Mixing period as a function of load (left) and Rate of Heat Release (right) for diesel MK1, PRF20, G. ON 69/66 and G. ON 70/66 [49].**

Air entrainment, liquid spray length and fuel composition seem to be the most likely cause of this gap. Unfortunately neither optical diagnostic and/or simulations were performed, thus the previous theories had to be backed up using the available literature.
According to [51] the use of PRF0, which is roughly as volatile as gasoline, in a CI engine can reduce the liquid spray length up to 60% at 10 bar gross IMEP as compared to diesel MK1. Figure 38 shows the liquid spray length and angle using PRF0 and MK1 for the same inlet condition; the liquid spray length for the more volatile fuel is significantly shorter as compared to diesel. Less liquid spray length means less probability to impinge the spray on the wall and/or piston, especially in LTC regimes, thus soot formation due to pyrolysis can be reduced or avoided.

The spray cone angle plays an important role in soot formation, the larger this parameter is the more air is mixed up with fuel prior combustion, thus less soot is produced. For the same running conditions, according to [51] and [52] the volatility of the fuel has almost no influence on the spray cone angle.
It is believed that the difference in soot levels between the gasolines and diesel MK1 was due neither to liquid impingement nor air entrainment. If diesel would have impinged on the wall or piston there would have been substantially higher CO and HC emissions, and a pronounced tail in the RoHR. On the other hand if the spray cone angle would have been different, the first part of the RoHR would not have matched for PRF20 and MK1. This discussion leads to the conclusion that the difference in soot levels between gasoline and diesel is primarily caused by molecular differences.
Laser extinction and planar laser-induced incandescence were used in [53] to measure the soot volume fraction in a steady state spray for PRF0 and diesel under the same ambient conditions but slightly different injection pressure to match the ignition delay. Figure 39 shows that diesel produces 10 times more soot than PRF0 and it has a much broader space distribution than the high volatile fuel.

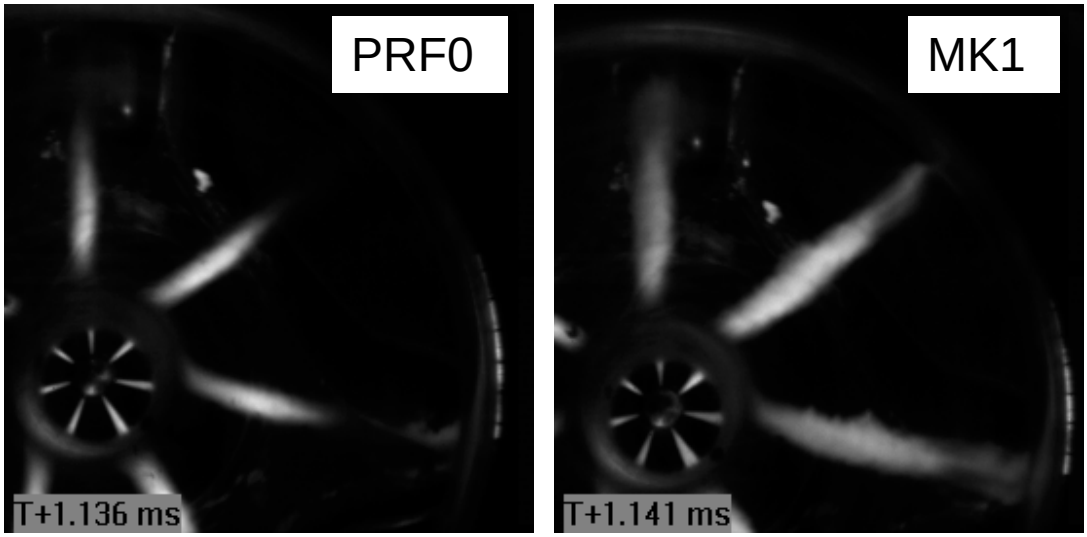


**Figure 38: Liquid spray shape using PRF0 and MK1 for the same running conditions using 100% EGR [51].**

It might be argued that American diesel has much more aromatics than MK1, ~30% vs. ~3%, thus resulting in higher soot production as shown in Figure 39. According to [54] aromatics and polyaromatics are not the exclusive precursors of soot. Naphthenes, which are a hydrogenated version of aromatics and polyaromatics, are also prone to produce lot of soot; MK1 contains lot of cycloparaffins [55]. This founding is also supported by [56] which states that, for typical engine running conditions, the soot formation process starts with a fuel molecule containing 12 to 22 carbon atoms and H/C ratio of about 2; a typical gasoline has an average carbon length of 7 while diesel 16. A research performed at

Cummins, [57], found that soot formation is not affected at all by the cetane number of the fuel but rather by its T50[11]; see Figure 40.

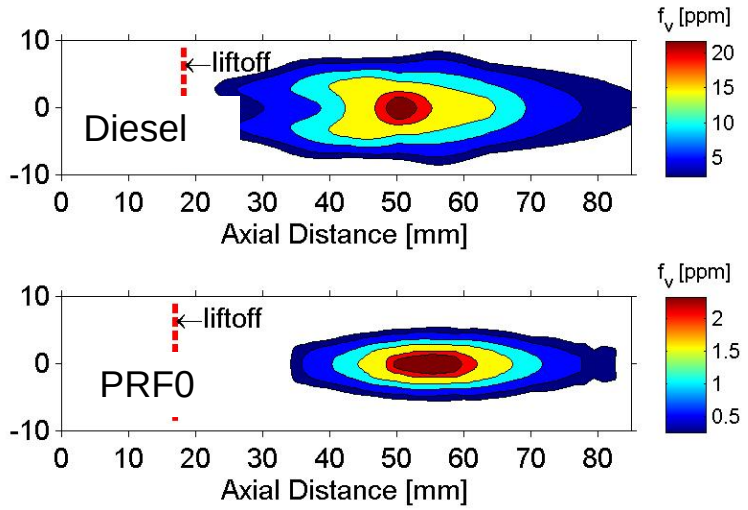


**Figure 39: Soot volume fraction for diesel and PRF0 under the same running conditions measured in a spray bomb [53]. Note the different scale in soot volume fraction between the upper and lower figure.**

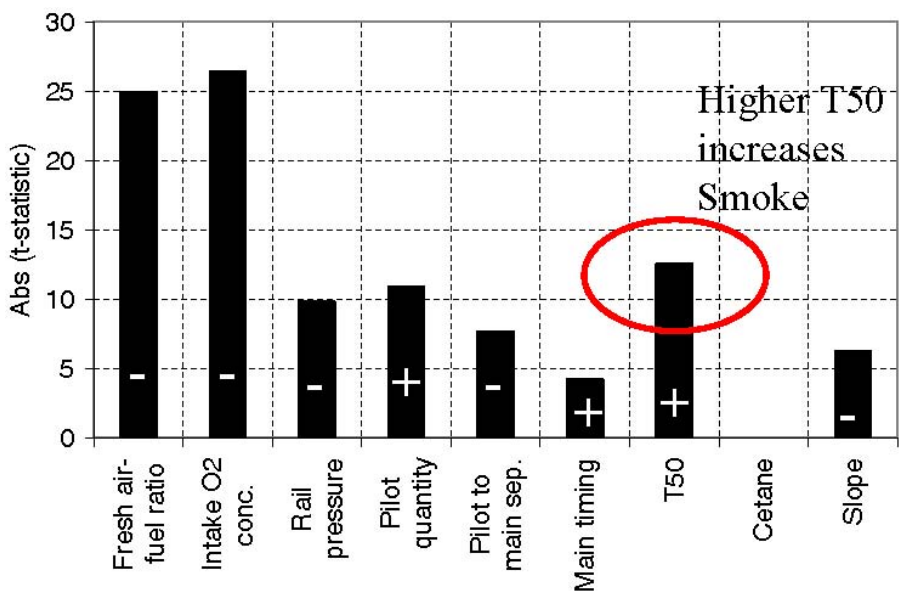


**Figure 40: Influence of the running conditions and fuel parameters on soot out emissions [57][12].**

[11] Typical T50 for gasoline is 368 K while for diesel 513 K. T50 is proportional to the average carbon length in the fuel molecule.
[12] Slope is defined as T90-T10.

## 3.3 Low Acoustic Noise

### 3.3.1 Promote Low Temperature Reactions and Control the Combustion Rate

It is believed that high efficiency and low emissions have to be achieved simultaneously with acceptable acoustic noise which is conventionally measured as maximum pressure rise rate, MPRR. Depending on the combustion mode, there are many ways to keep the engine acoustic noise within acceptable levels e.g. use of pilot injection, retarded combustion phasing, thermal stratification, fuel stratification, or use of a fuel with two stage ignition [62] [63] [64].

The previous paragraphs showed that high efficiency and low emissions are simultaneously achieved by using lots of EGR and boost. Increasing the inlet pressure is an effective way to promote low temperature reactions, LTR. If single injection is employed, increasing the magnitude of LTR is an effective way to reduce the acoustic noise because LTR have the same effect as using a pilot injection. Using a variety of fuels, inlet pressures and speeds, HCCI combustion was characterized by the author in a Scania D12. Keeping the energy delivered per cycle constant, Figure 41 shows that the maximum rate of heat release can decrease by increasing the amount of LTR trough an increment of inlet pressure from 1.02 to 1.92 [bar]. Independently of the octane number the MPRR decreases when the boost is increased at constant fuelling rate. On the other hand the rate at which MPRR decreases as a function of boost depends mainly on the fuel composition; see Figure 42. For instance the decrease in MPRR when increasing boost is faster for fuels containing lot of n-paraffins as compared to fuels containing lot of e.g. aromatics or iso-paraffins; this is because n-paraffins are more prone to increase the LTR activities when the inlet pressure increases as compared to for instance aromatics or iso-paraffins.

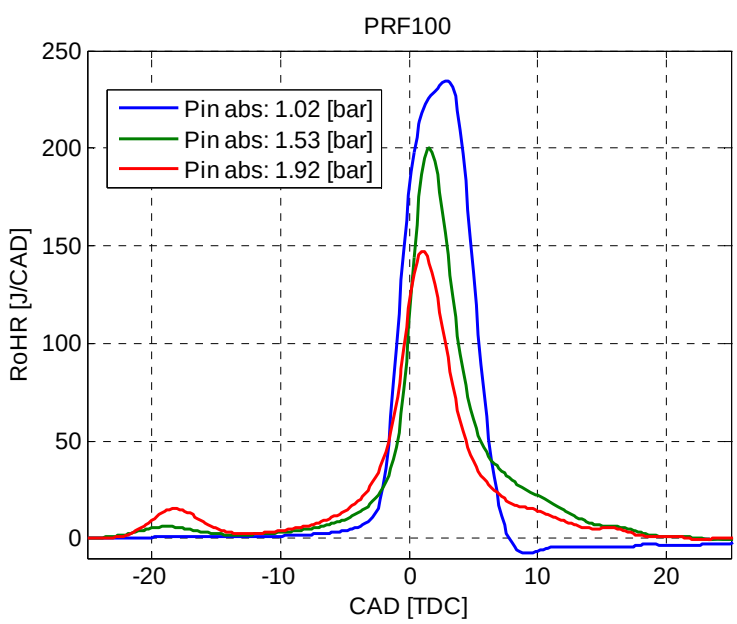


**Figure 41: Rate of heat release at 830 [rpm] as a function of boost using PRF100.**

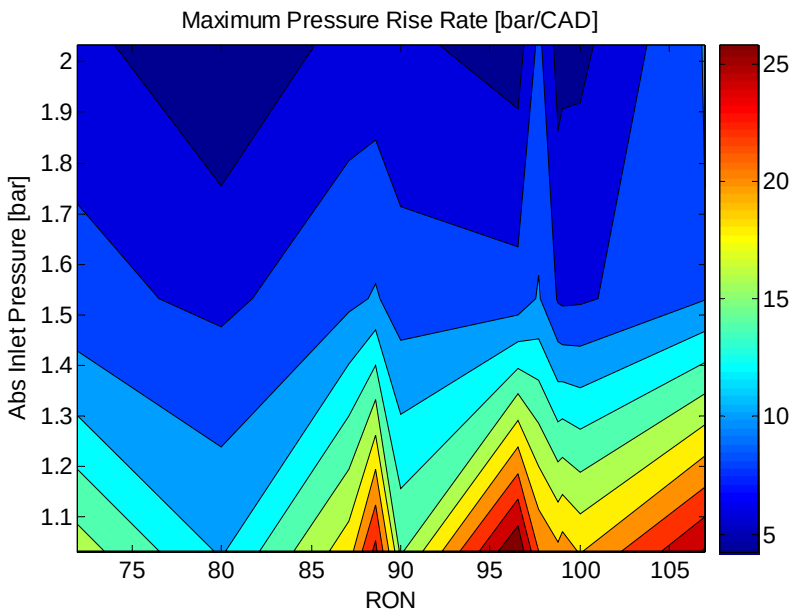


**Figure 42: Maximum pressure rise rate as a function of boost and RON at 830 [rpm].**

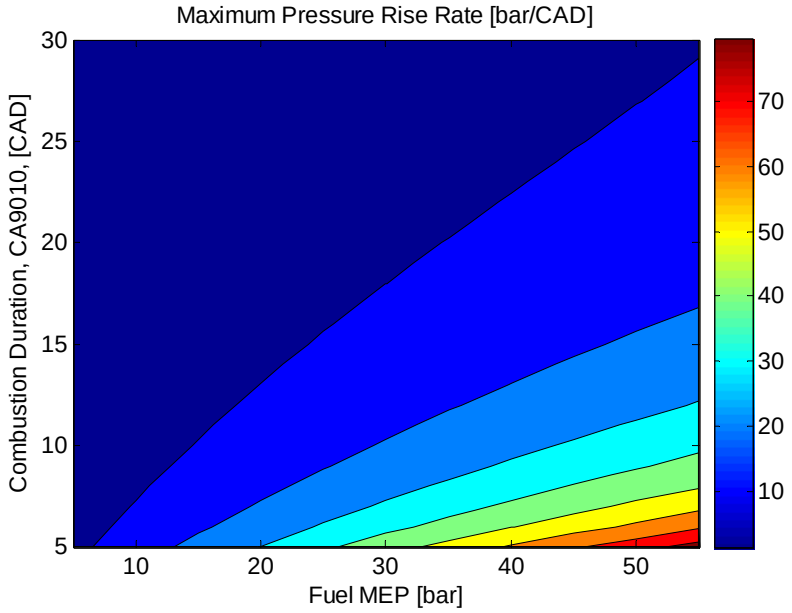


**Figure 43: Maximum pressure rise rate as a function of the combustion duration and fuel MEP.**

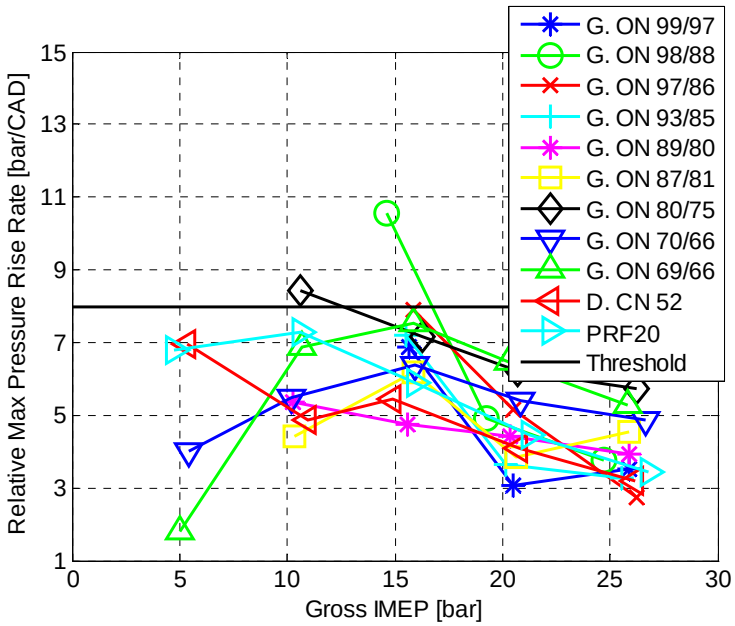


**Figure 44: Relative maximum pressure rise rate measured in the load sweeps performed using the Scania D13 [49].**

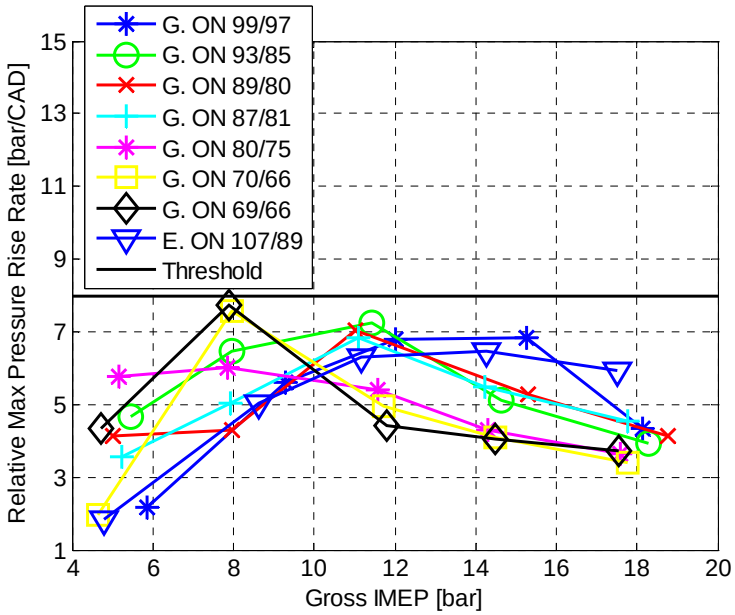


**Figure 45: Relative maximum pressure rise rate measured in the load sweeps performed using the Scania D12 [48].**

For practical applications the boost level can not always be increased to the desired value since the exhaust enthalpy decreases with increased boost, assuming a reasonable pressure difference between intake and exhaust. In mixing controlled or partially mixing controlled combustion, controlling the rate at which fuel burns is a very effective way of controlling the maximum pressure rise rate. The combustion rate can be easily controlled by adjusting the common rail pressure and injection duration. When decreasing the combustion rate, the combustion duration increases thus resulting in less acoustic noise.

Assuming the use of 50% EGR and CA50 at 8 [ATDC], the model described in 3.1.1 was used to compute the MPRR as a function of load and combustion duration (the inlet pressure was a linear function of the load). If the combustion duration is between 15 and 20 CAD, relatively low maximum pressure rise rate can be achieved even at maximum load.

The amount of boost used by the author in [48] and [49] combined with the appropriate injection pressure and duration allowed maintaining the normalized maximum pressure rise rate[13] within heavy duty engine acceptable levels (8 bar/CAD) even when running the engine at or close to MBT; see Figure 44 and Figure 45.

## 3.4 Results on Gasoline Partially Premixed Combustion

### 3.4.1 Experimental Set-Up

**Engine-** The experimental results reported in this thesis were carried out in two different engines, a single cylinder Scania D12 and a six cylinder Scania D13 modified to run only with one cylinder active; see Figure 46. Both engines had a flat cylinder head and they were equipped with bowl pistons; see Figure 47. Detailed geometrical properties of the engines can be found in Table 4. In both set-ups the boost was simulated, an external compressor was compressing air to the desired pressure and a waste gate valve was used for keeping the pressure to exactly the desired level. A backpressure valve was mounted in the exhaust manifold in order to raise the pressure high enough to allow the flow of EGR into the intake. A heater placed prior the inlet manifold was used to keep the intake temperature to the desired value.

**Injection system-** The D13 and D12 were equipped with two very different common rail injection systems; the first engine had a modern XPI common rail while a first generation Bosch prototype common rail injection system was fitted on the second engine; see Table 5. In both cases the fuel flow was measured with a gravity scale with two digits precision from Sartorius and each operative point was sampled for at least two minutes.

**EGR-** The EGR used in this paper is defined as the ratio of carbon dioxide in the intake and exhaust $EGR = \frac{CO_{2\_\text{intake}}}{CO_{2\_\text{exhaust}}}$. The exhaust gasses were cooled down before being introduced in the intake. EGR was mixed with fresh air prior being heated up with the above mentioned heater.

**Emission Measurements Systems-** In the D13 and D12 engines, NOx, soot and HC were measured respectively with a chemiluminescent analyzer from ECO Physics CLD 700 EL unit, an AVL 415 opacimeter and a heated flame ionization detector from JUM. A Cussons system was used in the D12 test cell to measure CO and $CO_2$ by non-dispersive infrared analyzer and $O_2$ with a paramagnetic analyzer while in the D13 rig the same emissions were measured with a device from Maihak. Each analyzer was calibrated with an appropriate calibration gas before every set of measurements.

[13] See Appendix 7.2 for the definition of *normalized maximum pressure rise rate*. When showing the results sometimes it has been referred to the absolute maximum pressure rise rate rather than its relative value. This is because this concept was developed in late 2009 in order to demonstrate that when running an engine at 26 bar IMEP gross using 3.60 bar absolute inlet pressure, 20 bar/CAD as maximum pressure rise rate is not a high value.

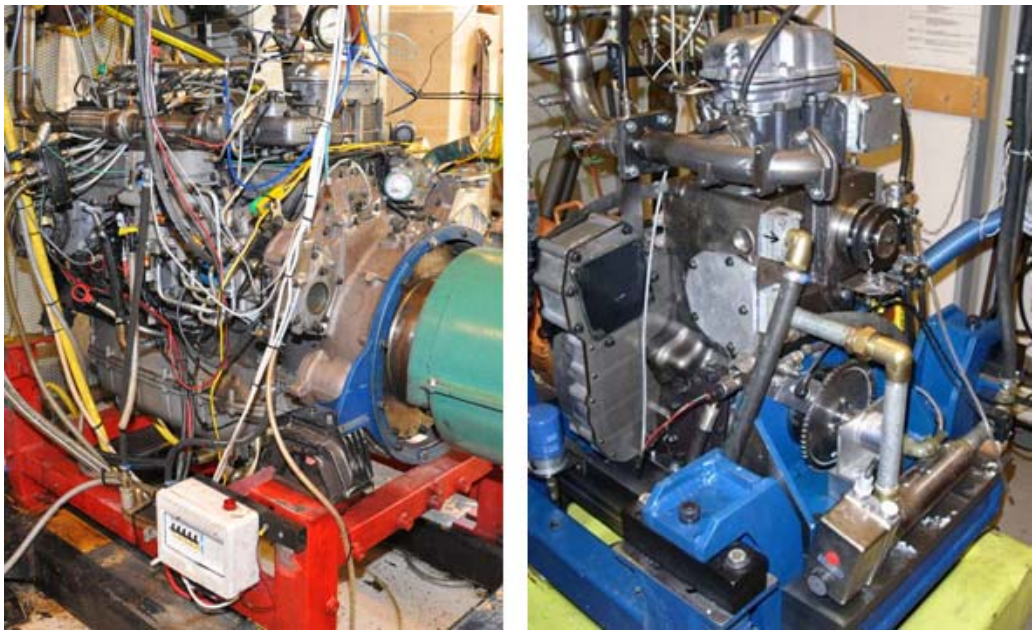

**Figure 46: Scania six cylinders 13 liters "D13" (left) and Scania single cylinder 2 liters "D12" (right).**

**Figure 47: From the left to the right, high compression ratio piston D12 (17.1:1), low compression ratio piston D12 (14.3:1), high compression ratio piston D13 (17.3:1).**

**Table 4: Geometrical properties of the Scania D13 (left) and D12 (right).**

| | | |
|---|---|---|
| Vd | 2124 | [$cm^3$] |
| Stroke | 160 | [mm] |
| Bore | 130 | [mm] |
| Con Rod | 255 | [mm] |
| rc, high | 17.3 | [-] |
| rc, low | n.a. | [-] |
| Swirl Ratio, low | 2.095 | [-] |
| Swirl Ratio, high | 3.57 (est.) | [-] |
| IVC | 40 | [ABDC] |
| EVO | 50 | [BBDC] |

| | | |
|---|---|---|
| Vd | 1966 | [$cm^3$] |
| Stroke | 154 | [mm] |
| Bore | 127.5 | [mm] |
| Con Rod | 255 | [mm] |
| rc, high | 17.1 | [-] |
| rc, low | 14.3 | [-] |
| Swirl Ratio, low | 1.7 | [-] |
| Swirl Ratio, high | 2.9 | [-] |
| IVC | 29 | [ABDC] |
| EVO | 34 | [BBDC] |

**Table 5: Injection system specifications; XPI (left) and Bosch (right).**

| | | |
|---|---|---|
| Spray Angle | 148° | [deg] |
| Orifices | 8 | [-] |
| Orifice Diameter | 0.19 | [mm] |
| Rail Pressure | 0-2400 | [bar] |

| | | |
|---|---|---|
| Spray Angle | 120° | [deg] |
| Orifices | 8 | [-] |
| Orifice Diameter | 0.18 | [mm] |
| Rail Pressure | 1500-1600 | [bar] |

**Table 6: Main properties of the fuels used during the experiments.**

| | RON | MON | C | H/C | O/C | LHV [MJ/kg] | A/F stoich |
|---|---|---|---|---|---|---|---|
| FR47335CVX | 99 | 97 | 7.04 | 2.28 | 0 | 44.3 | 15.1 |
| FR47332CVX | 98 | 88 | 6.61 | 2.06 | 0.07 | 39.7 | 13.44 |
| FR47337CVX | 97 | 86 | 7.53 | 1.53 | 0 | 42.1 | 14.03 |
| FR47338CVX | 89 | 80 | 7.21 | 1.88 | 0 | 43.5 | 14.53 |
| FR47330CVX | 87 | 81 | 7.2 | 1.92 | 0 | 43.5 | 14.6 |
| FR47331CVX | 93 | 85 | 6.9 | 1.99 | 0.03 | 41.6 | 14.02 |
| FR47336CVX | 70 | 66 | 7.1 | 2.08 | 0 | 43.8 | 14.83 |
| FR47334CVX | 69 | 66 | 7.11 | 1.98 | 0 | 43.8 | 14.68 |
| FR47333CVX | 80 | 75 | 7.16 | 1.97 | 0 | 43.7 | 14.65 |
| Ethanol, 99.5% | 107 | 89 | 2 | 3 | 0.5 | 29.7 | 9 |
| PRF20 | 20 | 20 | 7.2 | 2.28 | 0 | 44.51 | 15.07 |
| Diesel MK1 | n.a. | 20 | n.a. | 1.87 | 0 | 43.15 | 14.9 |
| Generic Gasoline | 95 | n.a. | n.a. | n.a. | 0 | 44.8 | 14.5 |

**Fuels-** Thirteen different fuels were used in the experiments and they are shown in Table 6. Two of them were Swedish commercial diesel and gasoline, the FR4733#CVX were nine fuels in the boiling point range of gasoline supplied by Chevron, and then ethanol and PRF20 were also tested for comparative purposes. In order to facilitate the communicability of the results sometimes the gasoline fuels and ethanol are labeled as "Z. ON X/Y" where Z. stands for either G. (gasoline) or E. (ethanol), ON is the abbreviation for octane number, X is the RON while Y is the MON. Swedish MK1 was labeled as: "D. CN 52" while PRF20 remained unchanged.

### 3.4.2 Introduction to the Results

To develop the highly efficient, low polluting, low noise and easy to control combustion concept proposed in this manuscript it was not a straight forward process since only three works were presented on gasoline partially premixed combustion when this research started in 2008 [67] [68] [110].

The idea of injecting a high octane number fuel in a CI engine was conceived to extend the upper load limit of HCCI combustion without the need of using too much EGR and/or too low compression ratio. The first thing that was done at Lund was to understand the upper load limit of gasoline PPC. This was carried out by running a standard Scania D12 (see Figure 46) using conventional pump gasoline (RON ca 95) [69]. It was discovered that the maximum load achievable was roughly 11.5 bar IMEP. This was the maximum load at which it was possible to separate the end of the injection with the start of the

combustion. Very low soot and NOx were achieved as compared to diesel but the efficiency was too low, the maximum pressure rise rate was unacceptably high and the combustion was tricky to control. To overcome these problems an advanced injection strategy was developed [70]. Using the standard Scania D12 and regular pump gasoline, the upper load limit was increased up to 16 bar IMEP which was the maximum load that the dyno could handle. Very low NOx, acceptable pressure rise rate and efficiency comparable to standard diesel engines were achieved at the above mentioned load. Unfortunately drawbacks were found in the high amount of CO and UHC because a significant amount of fuel was injected in the first injection (ca 50% at -60 [TDC] at 16 bar IMEP), and soot was unacceptably high. Ethanol was also tested using the same running conditions and load [71]. This fuel performed much better. Despite the overlap between injection and combustion events, good efficiency, low NOx and low maximum pressure rise rate were achieved as in the previous case but this time soot, CO and HC were simultaneously low. To decrease CO, UHC and soot in gasoline PPC it was decided to substantially decrease the fuel amount in the first injection and to use more boost to have high enough oxygen in the intake to reduce soot [72]. Using nine fuels in the boiling point range of gasoline provided by Chevron a load sweep was carried out between 1 and 12 bar gross IMEP using the standard Scania D12 and ca 50% of EGR. Low NOx, acceptable pressure rise rate and much less soot as compared to the fuels tested in [70] were simultaneously achieved throughout the sweep. Surprisingly it was possible to achieve gross indicated efficiencies between 52 and 57% in the load range under examination! High efficiency, low emissions and acceptable pressure rise rate were achieved almost independently of the gasoline ignitibility properties. At this point it was understood that high efficiency, low emissions and low acoustic noise can be simultaneously achieved in the whole load range by choosing an appropriate combination of $\lambda$ and EGR (inlet pressure information is redundant once these two parameters have been fixed) rather than the gasoline type. In order to increase the maximum load the compression ratio of the Scania D12 was decreased from the standard 17.1 down to 14.3. The $\lambda$ and EGR combination was improved and it was possible to run the engine from low load up to 18 bar IMEP [48]. The better selection of $\lambda$ and EGR allowed a further reduction of soot as compared to [72] while NOx and efficiency were comparable to [72]. The research carried out in [48] showed that as long as there is the right combination of $\lambda$ and EGR it does not really matter if there is an overlap between combustion and injection event because high efficiency, low emissions and low acoustic noise can always be simultaneously achieved independently on the gasoline type. Because of limitations of the D12 test cell, it was decided to apply the gasoline PPC concept developed so far to a single cylinder Scania D13; see Figure 46. This new test cell allowed achieving a maximum load of 25 bar IMEP and in addition the engine was equipped with a brand new and up to date injection system. The previously mentioned gasolines were tested from idle to 25 bar IMEP using the standard Scania D13 and comparisons were carried out with diesel [49]. This latest research showed that, for the exactly the same running conditions, the main difference between gasoline and diesel is in the amount of soot produced and a slightly better efficiency for the more volatile fuel. With an appropriate selection of $\lambda$ and EGR both gasoline and diesel have low NOx, low acoustic noise and very high efficiency throughout the whole load range.

In summary these two years of research showed that very high efficiency, low emissions, low acoustic noise and good combustion control can be simultaneously achieved in the whole load range if:

- The appropriate λ and EGR combination is used.
- The fuel has an octane number of roughly 70 and it has the same boiling point range of gasoline.
- The right piston, nozzle umbrella angle and swirl are used.
- The engine is equipped with a modern injection system.

Kinetically controlled combustion did not seem to give any benefit as compared to mixing controlled combustion; it resulted in more drawbacks than advantages! Also the use of a pilot injection seemed to play a secondary role.
More detailed explanations on how the concept presented in [48] and [49] has been developed can be found from 3.4.3 to 3.4.5.

### 3.4.3 Classical Partially Premixed Combustion

The target of classical diesel partially premixed combustion is to achieve a full separation between the end of the injection and the start of the combustion. By doing so it is possible to avoid rich and stoichiometric air-fuel pockets and in this way low NOx and low soot can be simultaneously achieved. To promote this separation a significant amount of cold EGR and/or very low compression ratio must be used, although this has severe consequences on engine efficiency, CO and HC. By using a fuel which is more resistant to ignition than diesel fuel it should be possible to achieve a separation between combustion and injection event without using much EGR or lowering the compression ratio.
In [69] the aim was to understand the maximum load achievable in the un-modified Scania D12 using classical partially premixed combustion fuelled with pump gasoline (95 RON). When trying to maximize the load, boost has to be increased and this has a negative effect on the ignition delay because as seen in Figure 48 the compression energy increases (area below the curve). To decrease the reactivity of the mixture more EGR has to be used but the higher the amount of EGR the more the inlet pressure has to be increased.
It was found that, for the compression ratio under examination, using 33% of EGR and λ 1.05, the maximum load achievable in classical PPC using the unmodified Scania D12 was 11.5 bar at 1100 [rpm]. Figure 49 shows the cylinder pressure, rate of heat release and injection signal in the above mentioned point; a substantial gap exists between end of the injection and the combustion event. In this research gasoline was compared to ethanol and diesel. Because ethanol has a lower heating value than gasoline and diesel, 30% of the fuel was injected at -60 TDC in the pilot. The three fuels were compared using exactly the same EGR, λ, IMEP, CA50[14] but different inlet temperatures. To keep the combustion alive ethanol needed 368 K as inlet temperature while diesel and gasoline 313 K. Figure 50 shows that with all the three fuels it is possible to achieve very low NOx, comparable to HCCI levels, both because of the EGR amount and the fact that the engine is running very close to stoichiometric (there is not much $O_2$ to form NOx). In Figure 51 CO, HC and soot are shown. Gasoline and ethanol have very low values of soot

[14] It was adjusted by modifying the start of the injection.

as compared to diesel, ca 0.5 vs. 9.5 FSN, but for different reasons. In the case of gasoline it was because of the homogeneity of the mixture when combustion took place while for ethanol it was because its molecular structure it is not prone to produce lot of particulate [56] [75]. Figure 52 shows that with ethanol there is not a complete separation between combustion and injection events.

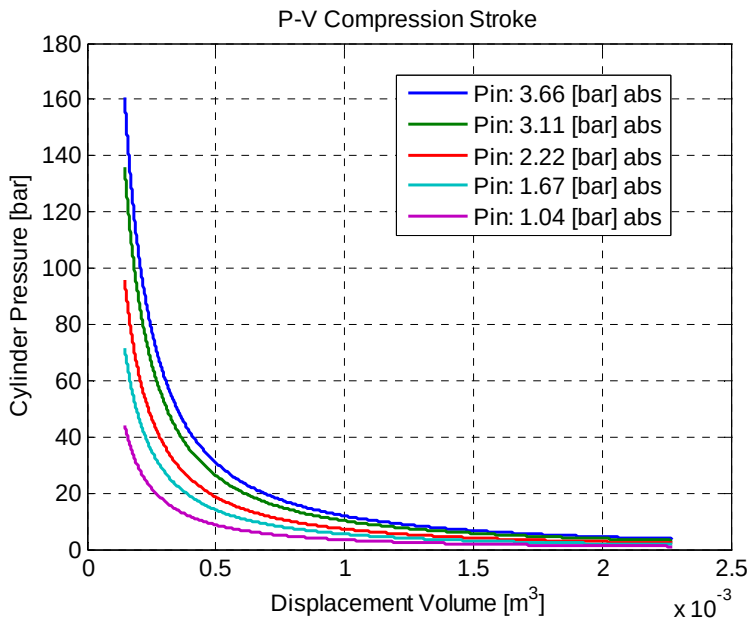


**Figure 48: Cylinder pressure vs. volume during compression stroke for different boost levels.**

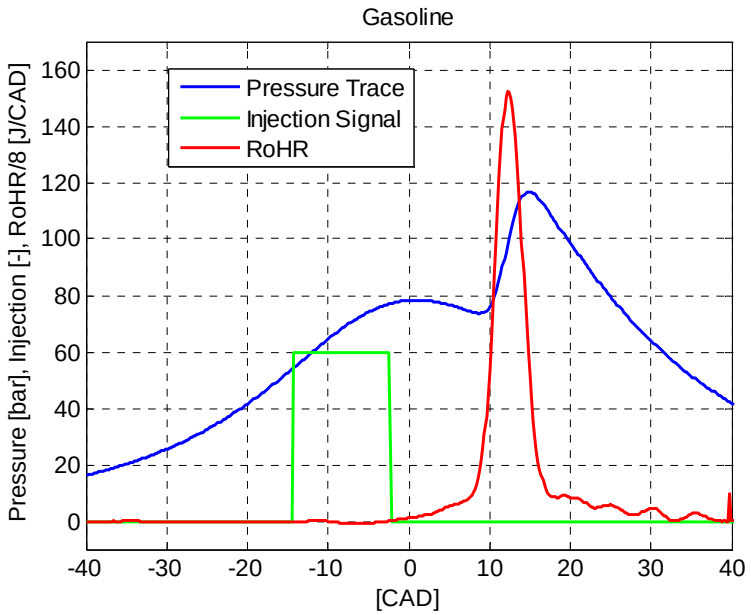


**Figure 49: Pressure trace, rate of heat release and injection signal for gasoline, IMEP: 11.5 bar, EGR: 33%, λ: 1.05 [-].**

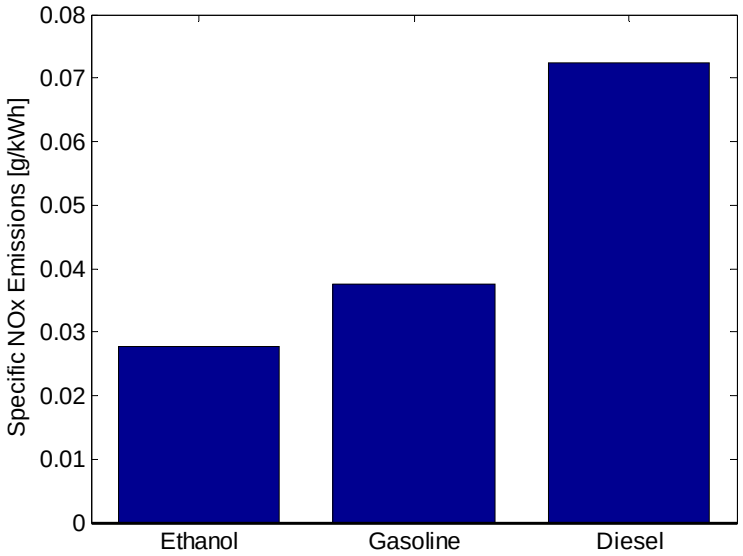


**Figure 50: Specific NOx emissions at 11. 5 bar IMEP, EGR: 33%, λ: 1.05 [-].**

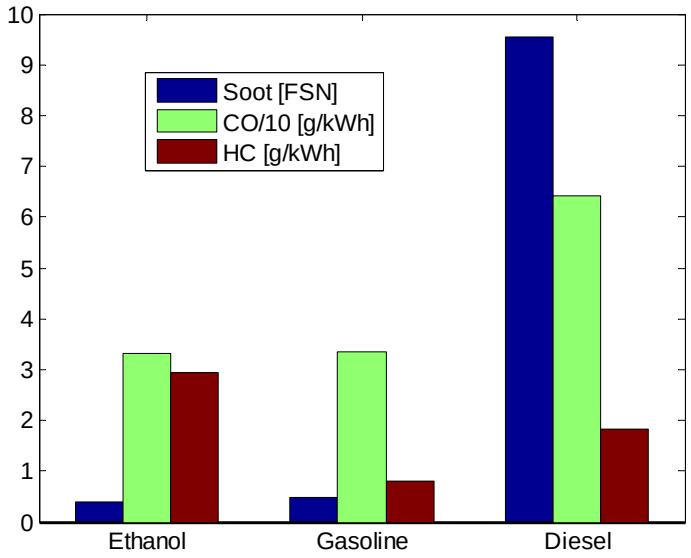


**Figure 51: Specific CO and HC emissions together with soot, IMEP: 11. 5 bar, EGR: 33%, λ: 1.05 [-].**

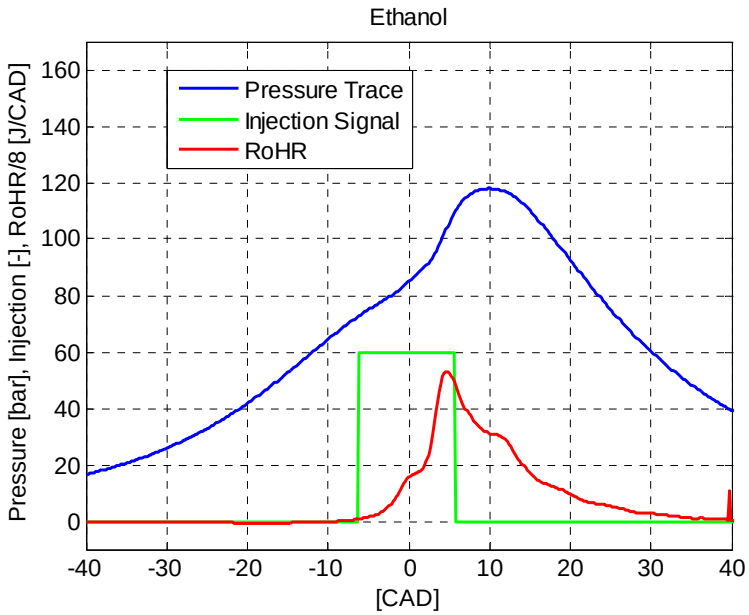


**Figure 52: Pressure trace, rate of heat release and needle lift for ethanol, IMEP: 11. 5 bar, EGR: 33%, λ: 1.05 [-].**

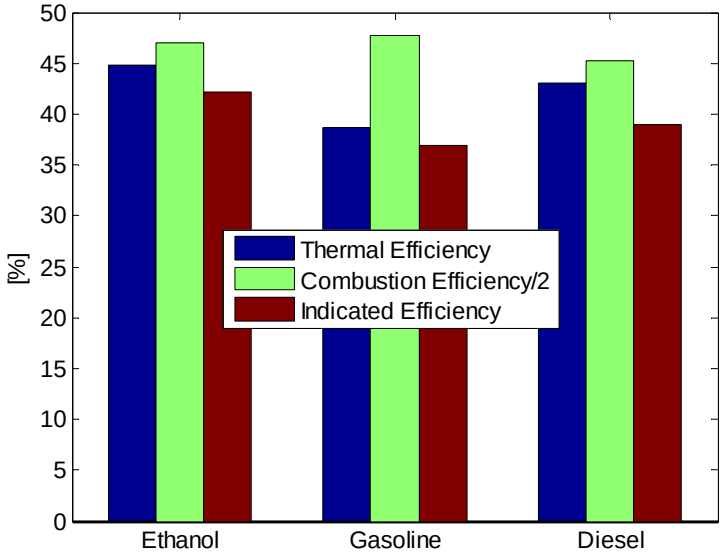


**Figure 53: Combustion, thermal and net indicated efficiencies for ethanol, gasoline and diesel; IMEP: 11. 5 bar, EGR: 33%, λ: 1.05 [-].**

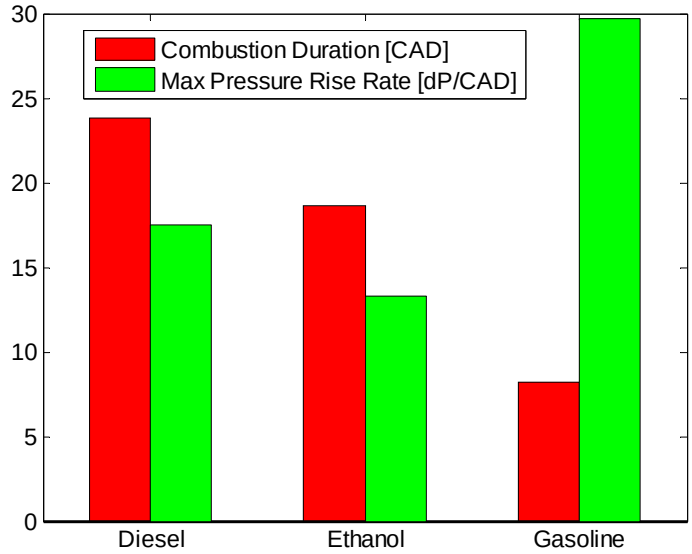


**Figure 54: Maximum pressure rise rate and combustion duration for ethanol, gasoline and diesel; IMEP: 11. 5 bar, EGR: 33%, λ: 1.05 [-].**

In that work it was shown that gasoline PPC is capable of simultaneously achieving low NOx and low soot and an upper load limit of 11.5 bar IMEP which is roughly 4-5 bar higher than a conventional HCCI engine. The drawback of this concept was found when the efficiency and maximum pressure rise rate were examined. Figure 53 showed that the net indicated efficiency of gasoline is roughly 40% while diesel and ethanol are approaching 44%. Also the maximum pressure rise rate in
Figure **54** is very close to 30 bar/CAD for gasoline while for ethanol and diesel is roughly 12.5 and 17.5 bar/CAD respectively. In [69] it was argued that the lower efficiency in the gasoline case was due to:

- The high pressure oscillations after combustion which can break the thermal boundary layer and thus increase the heat transfer [73], [74].
- The short combustion duration which is responsible for higher combustion temperature thus more heat lost to the walls.

### 3.4.4 Advanced Injection Strategy

Despite the very low emissions of NOx and soot using classical gasoline PPC combustion, the results achieved in [69] were not satisfying enough because the efficiency was too low, the maximum pressure rise rate too high and the maximum load obtained was only 11.5 bar IMEP. To decrease the maximum pressure rise rate, improve the fuel consumption and increase the maximum load it was decided to use a pilot injection. One key issue was to determine the optimal timing of the pilot injection. Early during compression stroke is beneficial because there is enough time for fuel and air to mix and increase the degree of homogeneity. This should produce less soot but unfortunately more NOx because of the higher temperature during the premixed part of the combustion; on the other hand placing the pilot injection late during compression stroke leads to the opposite result in terms of NOx and soot. To determine the optimal pilot injection timing in [70] a start of pilot injection sweep was carried out between -80 and -20 [TDC] at 14.8

bar gross IMEP, 0% of EGR, 20% of the fuel in the first injection and Swedish pump gasoline as fuel. Despite the soot values, it was decided that the best location of the pilot injection was -20 TDC because NOx, MPRR and the gross ISFC are simultaneously low; see Figure 55.

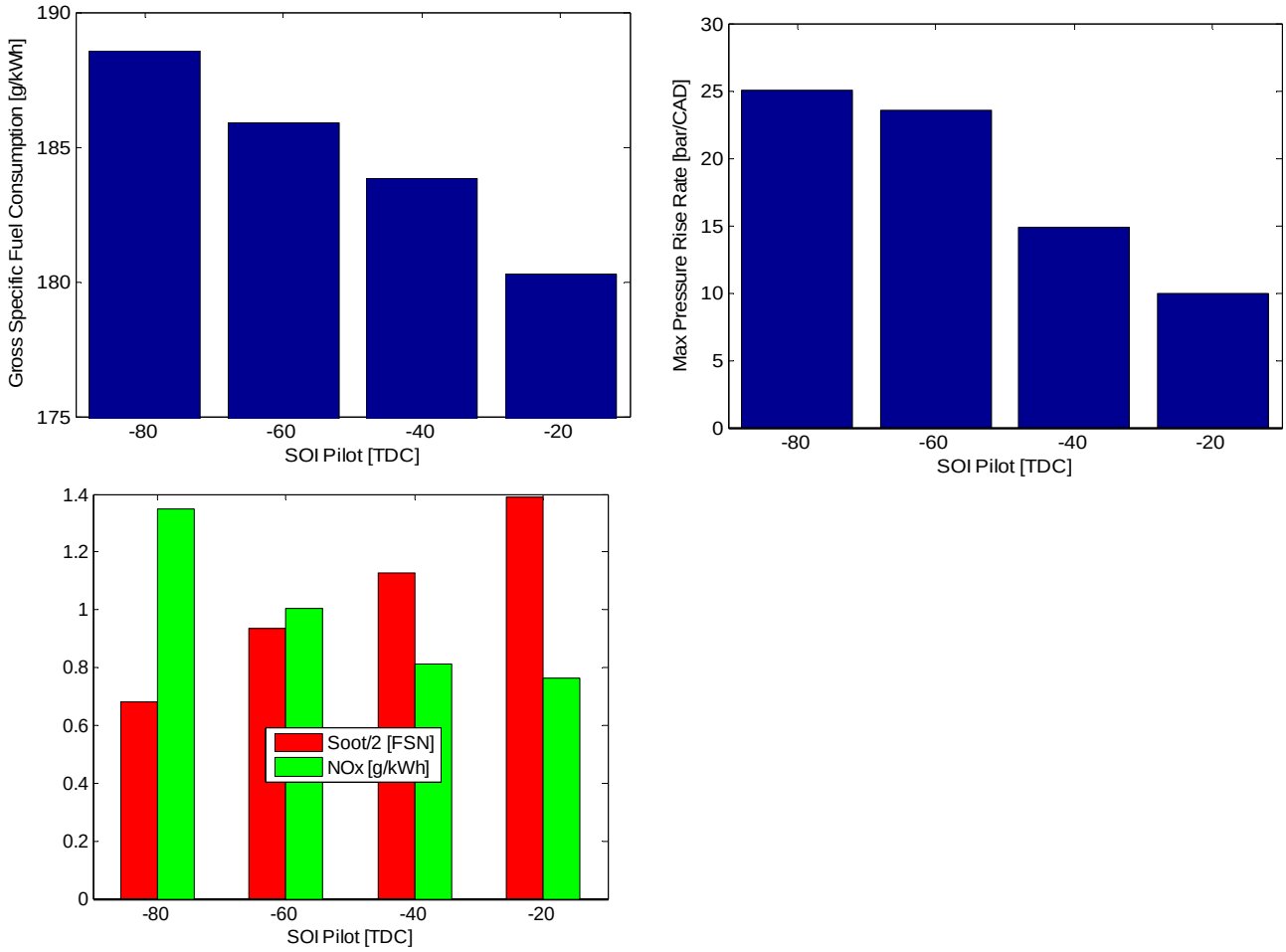


**Figure 55: Gross specific fuel consumption, maximum pressure rise rate, soot and NOx as a function of $SOI_p$; IMEP: 14.8 bar, EGR: 0%.**

Based on these results it was decided to compare ethanol, diesel and gasoline at 14.80 bar IMEP using 25% of EGR and λ 1.56. All the three fuels had a pilot placed at -20 TDC. Diesel and gasoline were running with 20% of fuel in the first injection while ethanol, because it has a lower heating value, with 36%. The duration of the main injection was kept constant for the three fuels.

Despite the emission values, it was interesting that the diesel equivalent gross ISFC of ethanol was 165 g/kWh (~50% $\eta_{gross}$); see Figure 56. In order to understand why this is happening, the rate of heat release in Figure 57 was analyzed for the three fuels. The diagram shows that gasoline and diesel are burning in diffusion controlled mode while with ethanol a significant amount of kinetically controlled combustion is followed by a diffusion combustion tail. As shown in many HCCI papers kinetically controlled combustion has the main advantage of fairly high thermal efficiency which leads to high

indicated efficiency (low fuel consumption) as compared to classical mixing controlled or flame propagation combustion processes. The question is why with gasoline it was not possible to do the same e.g. in [69] or when the pilot was placed at -80 TDC (see Figure 55). Because optical diagnostic was not available it was necessary to rely on plausible guesses based on the available data. When the pilot is placed at -80, the rate of heat release trace in Figure 58 suggests that in the combustion chamber there are many zones with a near to stoichiometric air/fuel ratio which resulted in a violent release of heat as it can be seen from the initial slope of the RoHR. As shown by Tsurushima, [73] [74], pressure oscillations after combustion are able to break the thermal boundary layer thus enhancing the heat flux which results in higher fuel consumption. For instance the poor efficiency of gasoline in [69] is thought to be the result of heavy knocking in a fairly HCCI combustion mode. In the case of ethanol 35 % of the fuel was injected in the pilot injection; considering its long ignition delay it is possible to assume that at the moment of the ignition there were not many fuel rich zones and this resulted in a smooth and kinetically controlled heat release rate as it can be seen in Figure 57. LIF measurements and/or chemiluminescence images are needed for validating this hypothesis.

Based on the results found for ethanol an advanced injection strategy was developed in order to run high octane number fuels in a compression ignition engine. Similar to UNIBUS [76] the advanced strategy consisted of two injections, the first is very early in the compression stroke to create a homogeneous mixture while the second one is around TDC to create a certain level of stratification for achieving autoignition. The concept might be conceived as a hybrid between UNIBUS and fumigation combustions, the difference consists in the fact that at high load (~ 16 bar gross IMEP) up to 54.54 % of the fuel is injected in the pilot then a certain amount of EGR is used to prevent autoignition of the first injection and finally around TDC the remaining fuel is injected and autoignition is achieved through charge stratification. The concept was tested using gasoline and it was compared with diesel. In order to have a fair comparison the two fuels were tested with two different injection strategies. Diesel had a pilot injection of 10% of the total fuel at -15 TDC and the remaining amount of fuel at -10 TDC. For gasoline, 54.54 % of the fuel is injected in the pilot injection at -60 TDC while the rest was injected at -5 TDC. For both fuels the amount of EGR was set to 37 % while $\lambda$ was 1.23. The running conditions were the followings: engine speed 1100 rpm, combustion phasing (CA50) 8.51 TDC and constant fuel flow for both fuels: 35.22 bar Fuel MEP. Figure 59 shows the pressure trace, rate of heat release and injection signal for gasoline and diesel under the running conditions previously described. With diesel, classical diffusion combustion is achieved and no sign of any premixed spike is visible thanks to the use of the pilot injection. In the case of gasoline the combustion is mainly kinetically controlled with a small tail of diffusion combustion most probably due to the overlap between the last moments of the injection and combustion. In the case of gasoline fuel, due to a small overlap between combustion and main injection (which resulted in a certain level of stratification) it was possible to have a low maximum pressure rise rate of 14.67 bar/CAD.

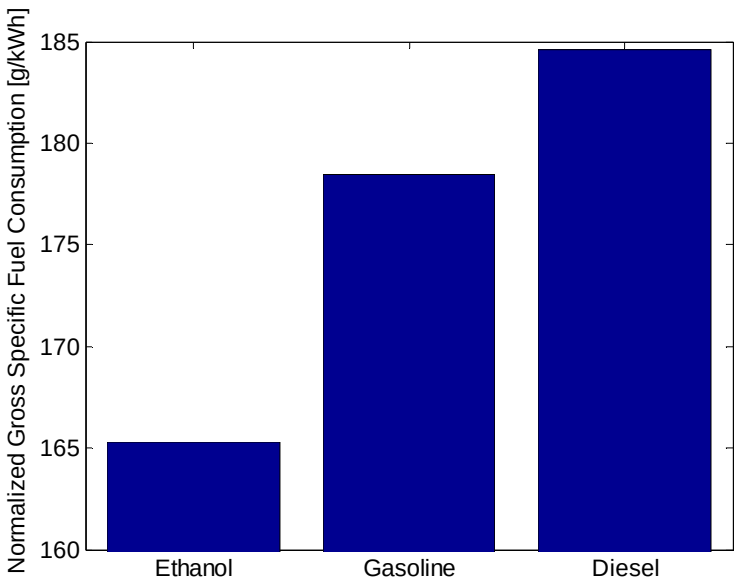


**Figure 56: Diesel equivalent gross indicated specific fuel consumption for ethanol, gasoline and diesel at 14.80 bar IMEP.**

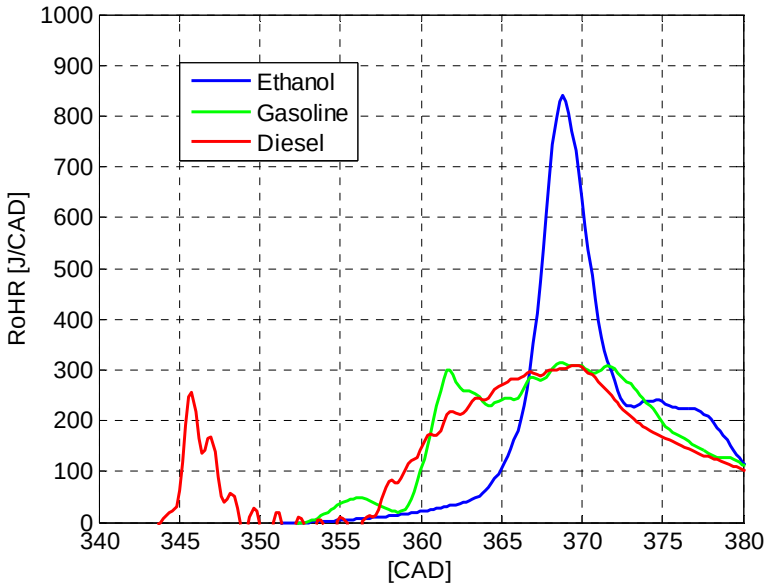


**Figure 57: Rate of heat release traces for ethanol, gasoline and diesel at 14.80 bar IMEP.**

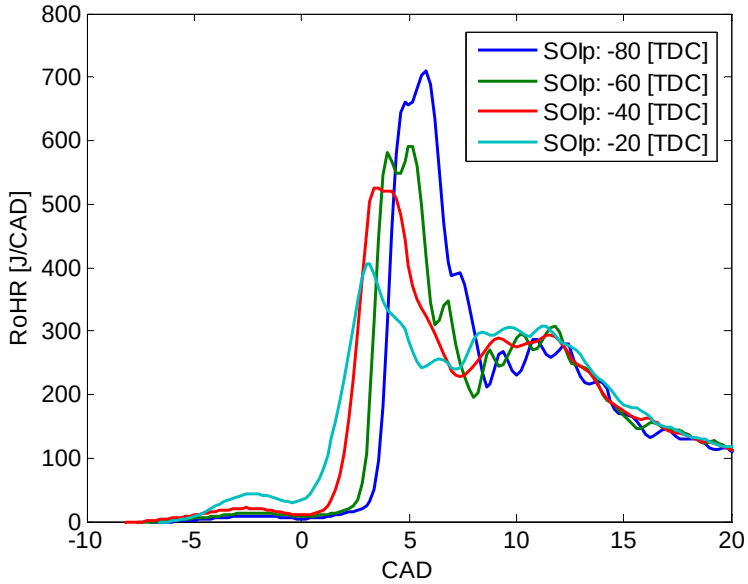


**Figure 58: Variation of the shape of the rate of heat release with different start of pilot injection timings using gasoline.**

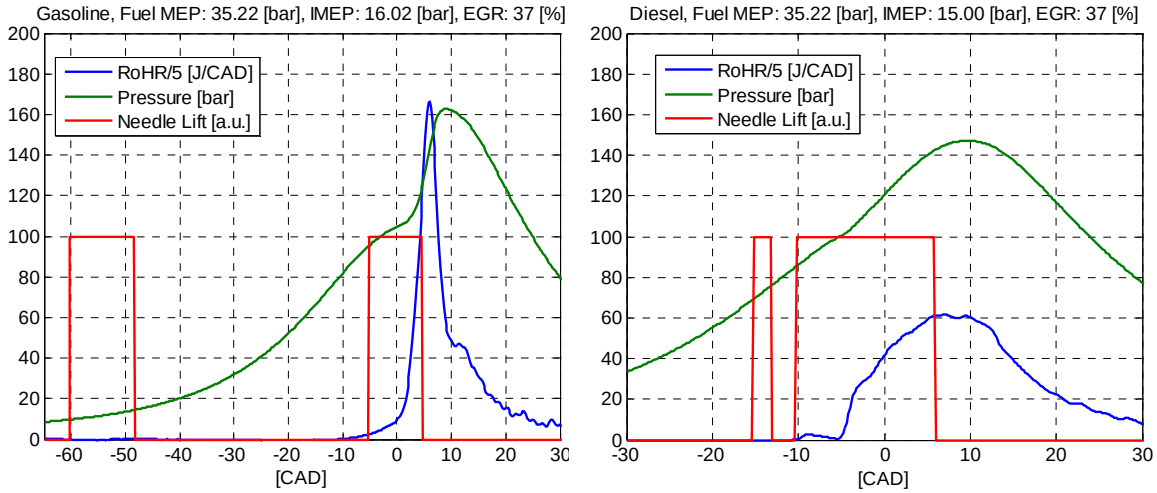


**Figure 59: Ensemble averaged cylinder pressure, rate of heat release and injection signal for gasoline (left) and diesel (right).**

In Figure 60 the emission levels are presented for gasoline and diesel. In terms of HC and CO gasoline is showing higher levels as compared to diesel mainly because of the high amount of fuel in the first injection. In terms of NOx gasoline is able to achieve 0.12 g/kWh while diesel 0.17; the reduction of almost 30% has to be attributed to the higher level of homogeneity when combustion took place. In terms of soot the FSN levels are high for both fuels: 5.85 and 7.40 for gasoline and diesel respectively. Once again because of the lower heterogeneity during combustion with gasoline, soot are 21 % lower as compared to diesel.

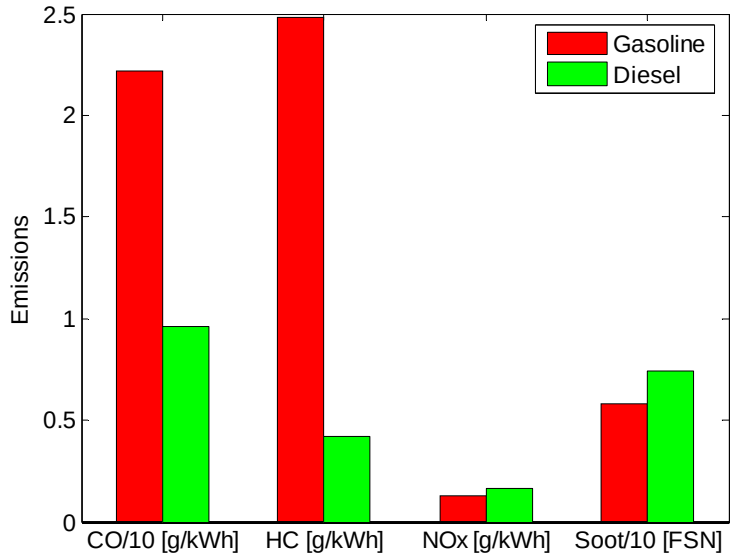


**Figure 60: CO, HC, NOx and soot for gasoline and diesel.**

Figure 61 displays the gross indicated specific fuel consumption both for gasoline and diesel, the difference is significant with values at 174.03 and 185.69 g/kWh respectively. It is thought this was possible because of higher expansion ratio in the case of gasoline; its combustion duration, CA90-10, was 16.84 CAD while for diesel 23.34 CAD. Also the radiative heat transfer is reduced for gasoline because of less soot is produced as compared to diesel as well as the convective heat transfer is decreased for gasoline because with a shorter combustion duration it is possible to avoid high combustion temperature during the expansion stroke.

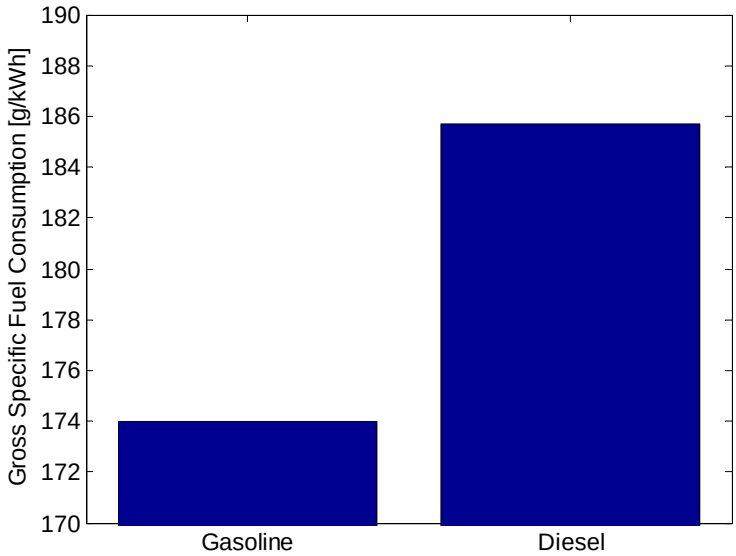


**Figure 61: Gross specific fuel consumption for gasoline and diesel.**

Due to the advantages of this injection strategy, it was decided to apply this concept to ethanol [71]. Because ethanol and gasoline have different ignitibility properties, a pilot ratio and a pilot position sweeps were carried out at the maximum load allowed by the dyno. These two sweeps were performed in order to understand if the settings found for gasoline are valid for ethanol as well.

In [71] the engine was run as follow: 1100 rpm, 323 K and 2.38 bar as inlet temperature and absolute pressure respectively, 42% of EGR (λ: 1.27 [-]), CA50 13 [TDC] (kept constant by adjusting the start of the second injection) and Fuel MEP 35.67 bar. As shown in Figure 62 in the operating window under examination NOx was almost as low as in the case of gasoline. The main difference between these two fuels was found in the soot level. Very low soot was produced with ethanol mainly because of its molecular structure [56] [75].

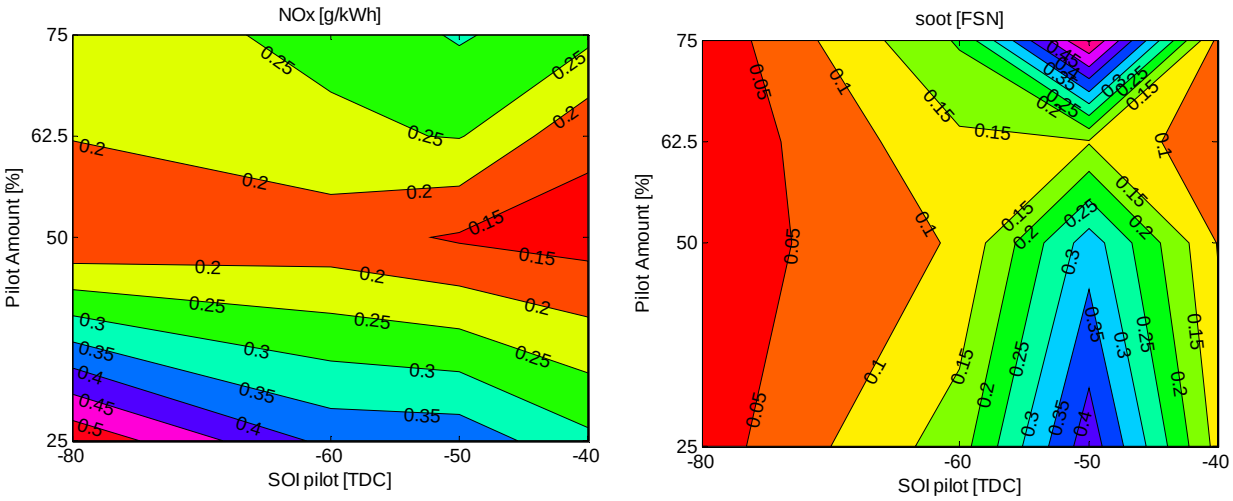


**Figure 62: gross indicated specific NOx (left) and soot (right) as a function of SOIp and pilot-main ratio.**

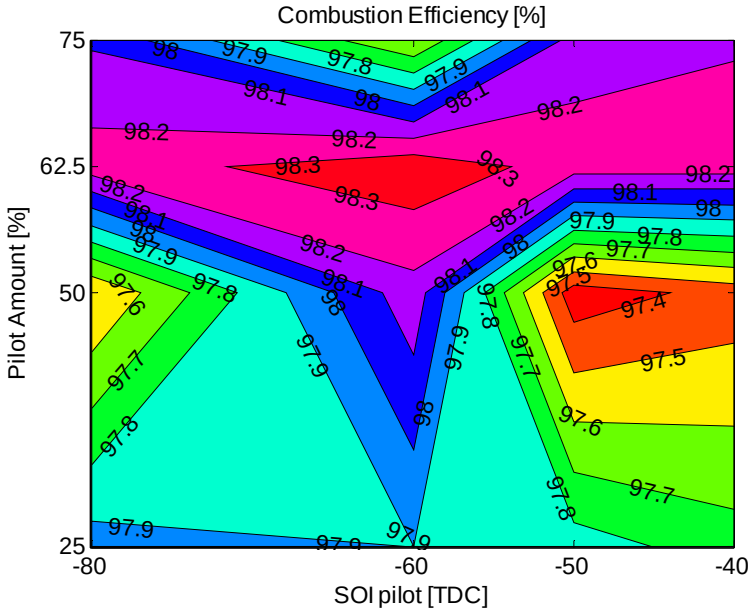


**Figure 63: Combustion efficiency a function of SOIp and pilot-main ratio.**

In the operating range under examination very low CO and HC were obtained. In the worst case the combustion efficiency was 97.4% (see Figure 63). As shown in Figure 64, fairly independent of the pilot position, with a pilot-main ratio smaller than 50% the gross indicated efficiency was higher than 47% as in the case of gasoline in Figure 61.

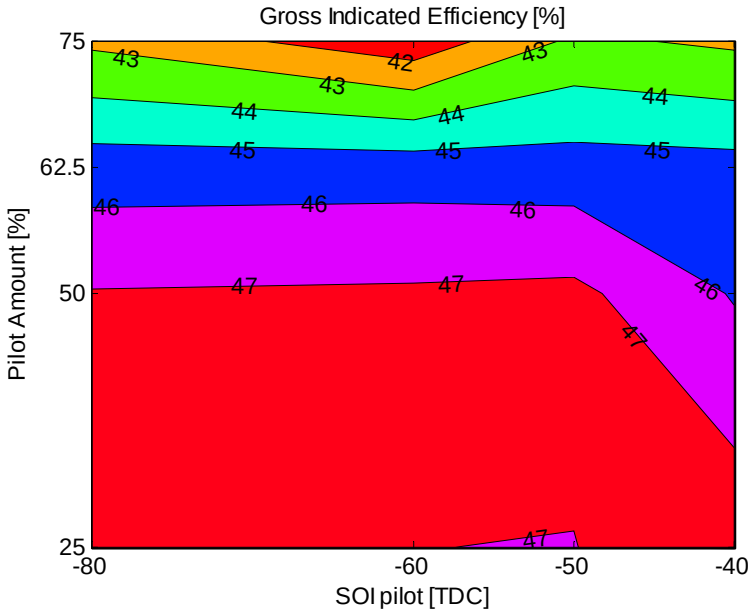


**Figure 64: Gross indicated efficiency a function of SOIp and pilot-main ratio.**

To find the best injection settings, the following constraints were used: NOx smaller than 0.20 g/kWh, gross indicated efficiency higher than 45%, maximum pressure rise rate below 15 bar/CAD and soot as low as possible. This optimization showed that the best injection settings at this load are very close to the ones of gasoline in [70]; see Table 7.

**Table 7: Best setting for ethanol PPC at 16.81 bar gross IMEP.**

| | | |
|---|---|---|
| Pilot | 50 | [%] |
| Main | 50 | [%] |
| SOIp | -60 | [TDC] |
| | | |
| $\eta_{ind\ gross}$ | 47.13 | [%] |
| | | |
| NOx | 0.17 | [g/kWh] |
| soot | 0.0080 | [g/kWh] |
| CO | 5.65 | [g/kWh] |
| HC | 2.34 | [g/kWh] |
| | | |
| dP | 9.95 | [bar/CAD] |

### 3.4.5 High Efficiency and Low Emissions Scania D12

The injection strategy developed in 3.4.4 enabled achievement of low NOx, acceptable maximum pressure rise rate, relatively good efficiency and a pretty high maximum load. In the case of gasoline the main drawback was the high levels of soot while with ethanol the particulate level was within acceptable levels. One practical issue of the concept proposed in 3.4.4 is that if for instance in real life application the EGR system is not fast enough and/or the ambient temperature is too high, the first injection might react during the compression stroke thus resulting in a thermodynamic disaster. In addition, injection of 50% of the total fuel amount at -60 TDC at high load will result in lot of CO and UHC because of partial quenching in the squish area. At this point of the research, the main issues were to significantly reduce soot, CO and HC, to develop an injection strategy that would not have led to problems in real life applications and to further increase the engine efficiency. NOx was not considered to be a problem because it could be easily controlled with EGR.

As first step it was decided to use much less fuel in the first injection without eliminating it in order to avoid the problems described in 3.4.3. By doing so, less fuel is confined in the squish region thus resulting in less CO, less UHC and slightly higher efficiency is obtained because of the reasons described in 3.1.3. It was also decided to increase the boost in order to use roughly 50% of EGR and $\lambda$ in the range of 1.4-1.5. As described in Chapter 3 these running conditions should enable low NOx-CO-HC, high efficiency and low maximum pressure raise rate to be obtained simultaneously. To reduce soot, the swirl ratio was increased from 1.7 to 2.9 by closing the tumble inlet port; the common rail and the injector nozzle could not be modified because no other hardware was available for the Scania D12. Finally it was decided to reduce the mixing period in order to avoid the maximum pressure rise rate issue described in 3.4.3. Even though the mixing period was reduced it was decided to keep a few CAD separation between the start of the combustion and the end of the injection because at the time when [72], [48] and [77] were written it was believed that, at every load, kinetically controlled combustion is a better way to burn fuel as compared to the mixing controlled combustion.

In [72] using the standard Scania D12 piston, rc 17.1, 12 bar IMEP was the maximum load at which the combustion was mostly kinetically controlled. In Figure 65 the gross indicated efficiency is shown. For loads higher than 4 bar gross IMEP the gross indicated efficiency is always higher than 50% for all the fuels tested (except in one point in which it is believed there might have been a fuel flow measurement error). The efficiency has a peak of 57% at roughly 8 bar IMEP with two fuels (FR47333CVX and FR47338CVX) having RON in the range of 80-89. After this load point all the fuels show a slightly deterioration in efficiency. Despite the use of 50% of EGR, the combustion efficiency is always above 98% for loads higher than 7 bar IMEP (see Figure 66) thus proving all the theories described in 3.2.1. Because of the appropriate $\lambda$-EGR combination the specific NOx presented in Figure 67 is below 0.32 g/kWh in the whole load range except for two fuels at low load (FR47330 and FR47335CVX). In terms of soot, for loads higher than 8 bar IMEP, the amount of particulates is mainly a function of the fuel properties. At 12 bar IMEP three clusters of points can be spotted. A first group (FR47335CVX and FR47332CVX) has soot below 0.5 FSN, the second one FR47331CVX and FR47337CVX having RON=93-97) between 1.5 and 2.5 and the third one (the remaining fuels having RON=69-87) between 3.5 and 5. The chemical reasons behind the different

behavior of these fuels were not investigated. In [72] it was argued, and later proved in [49], that high soot values (more than 1.5 FSN) were mainly due to the obsolete injection system rather than fuel-air mixing. With all the nine fuels, at 12 bar IMEP, the mixing period was always between -2 and 10 CAD; see Figure 69.

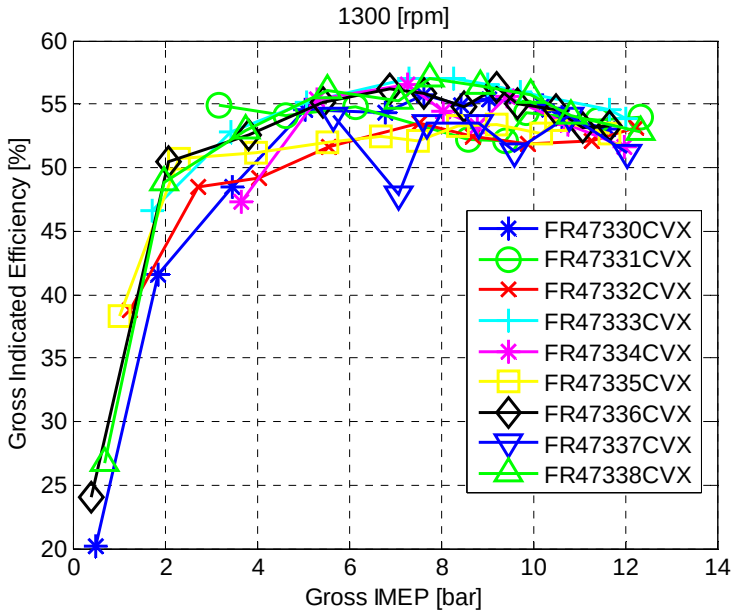


**Figure 65: Gross indicated efficiency as a function of load and fuel type. Scania D12, high compression ratio (rc=17.1:1).**

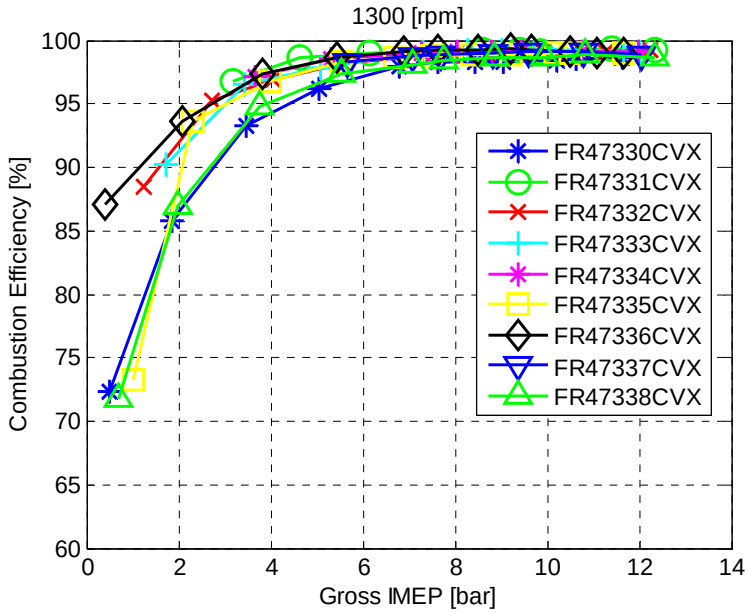


**Figure 66: Combustion efficiency as a function of load and fuel type. Scania D12, high compression ratio (rc=17.1:1).**

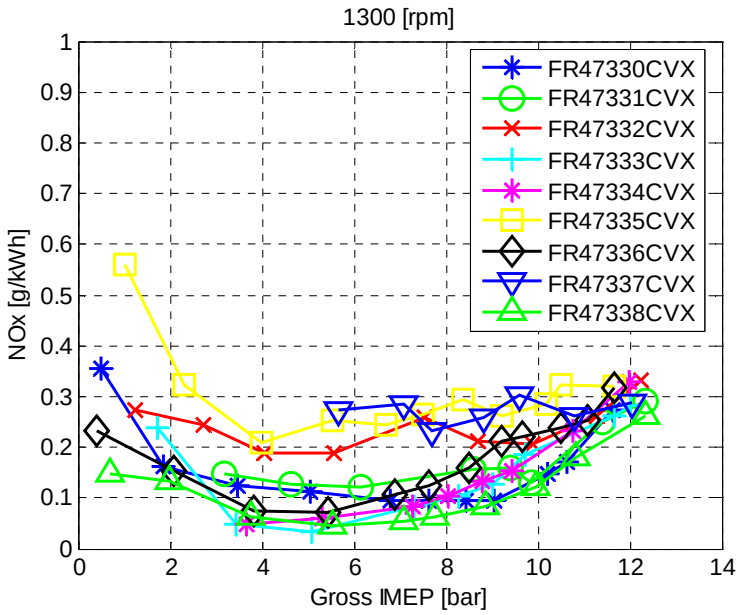


**Figure 67: Specific NOx as a function of load and fuel type. Scania D12, high compression ratio (rc=17.1:1).**

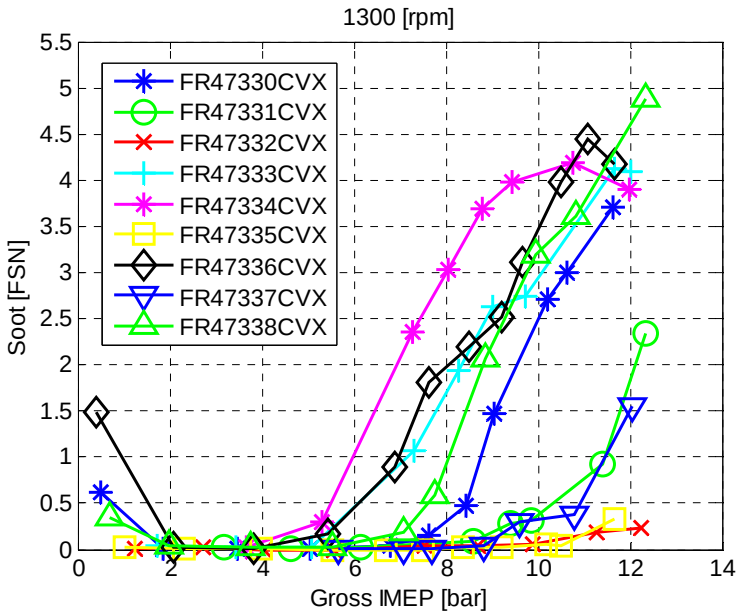


**Figure 68: Soot as a function of load and fuel type. Scania D12, high compression ratio (rc=17.1:1).**

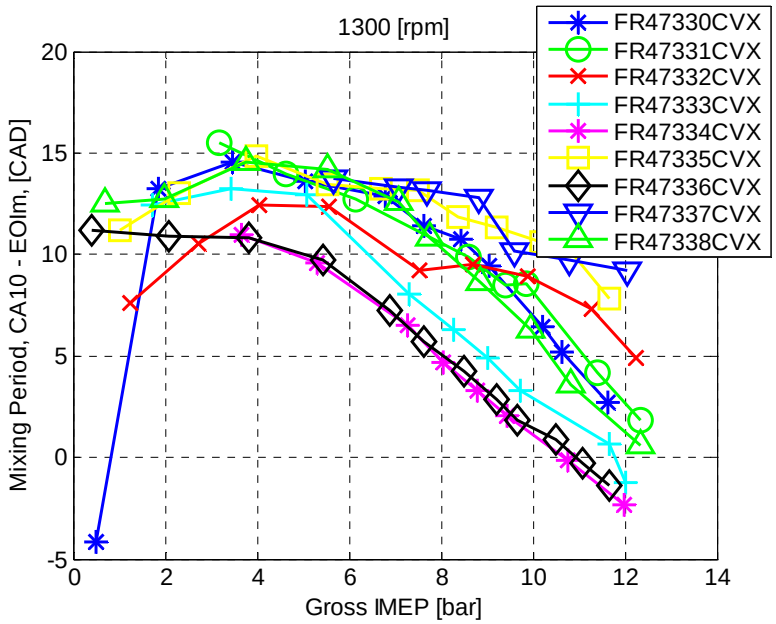


**Figure 69: Mixing period as a function of load and fuel type. Scania D12, high compression ratio (rc=17.1:1).**

To increase the maximum engine load up to the limit of the dyno (ca 15 bar BMEP), in [48] the compression ratio of the Scania D12 was decreased from 17.1 to 14.3. This was done to keep a positive mixing period at maximum load thus achieving kinetically controlled combustion in the whole load range. By decreasing the compression ratio, the load could be increased from 12 to 18 bar gross IMEP. Because of the good results achieved in [72], the EGR-λ combination described in 3.2.1 was used in [48] as well. In [48] λ was better tuned. It was noticed that soot production is strongly influenced by small variation in relative excess of air rather than minor variation in EGR rate. With the experimental set-up under examination, increasing λ from 1.48 to 1.52 resulted in very different soot emissions. Despite the similar mixing period at high load (see Figure 71 and Figure 69) and the same engine speed (1300 rpm), in [48] because of a better tuning of λ, in the worst case soot is 2 FSN (see Figure 70) which is roughly 60% less compared to Figure 68. Despite a reduction of at least 60%, 1-2 FSN of soot are still considered relatively high, as it will be shown in 3.4.6 this problem can be easily eliminated with an up to date injection system. It is interesting to note that the use of ethanol enables almost zero soot (below 0.1 FSN) at 18 bar gross IMEP. The gross indicated efficiency is once again very high; see Figure 72. For loads higher than 12 bar IMEP it reaches a steady value between 54 and 56%. The slightly decrease in peak efficiency as compared to the study performed in [72] was attributed to a decrease in expansion ratio. As explained in 3.2.1 because it was always possible to burn the fuel-air mixture within the temperature range of 1500 K and 2000 K, low NOx and high combustion efficiency were achieved between 4 and 18 bar gross IMEP for the fuel tested; see Figure 73 and Figure 74.

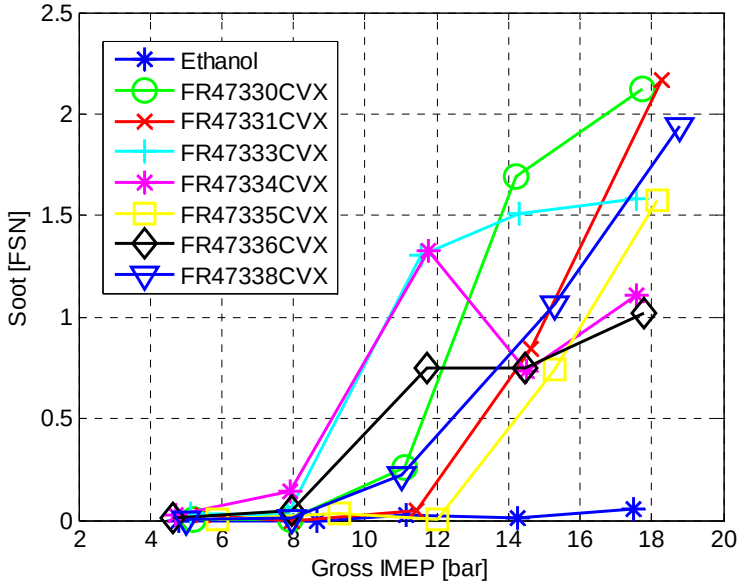


**Figure 70: Soot as a function of load and fuel type. Scania D12, low compression ratio (rc=14.3).**

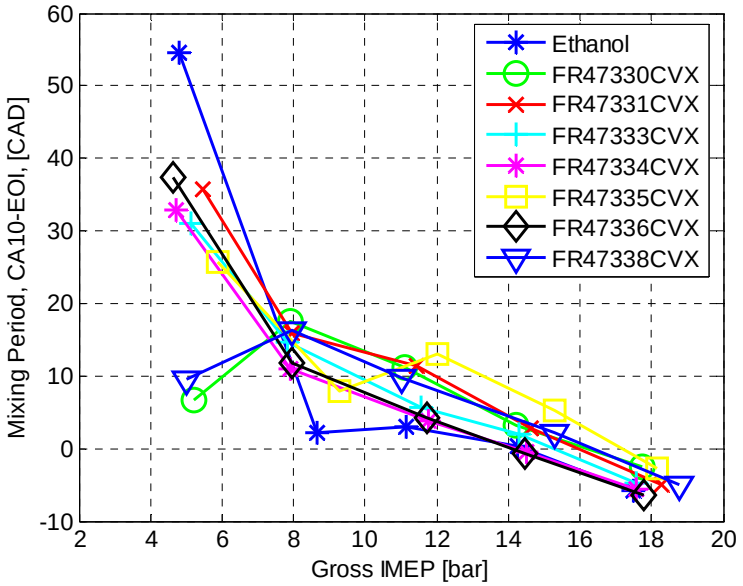


**Figure 71: Mixing period as a function of load and fuel type. Scania D12, low compression ratio (rc=14.3).**

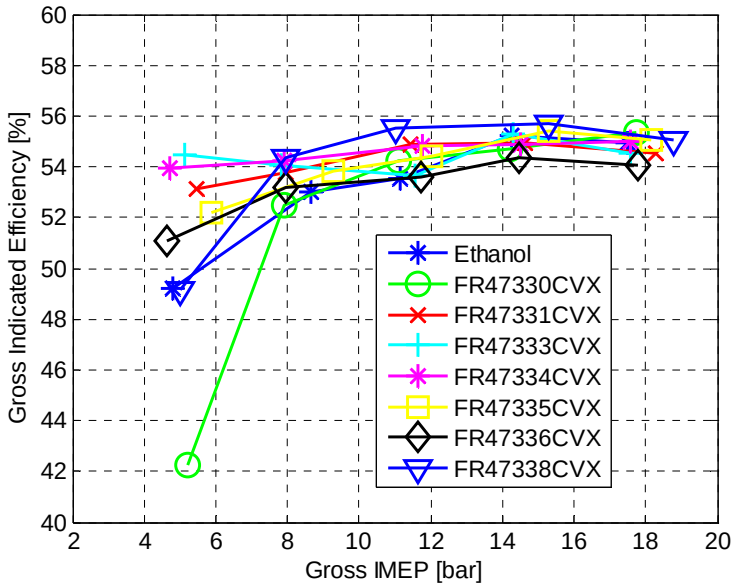


**Figure 72: Gross indicated efficiency as a function of load and fuel type. Scania D12, low compression ratio (rc=14.3).**

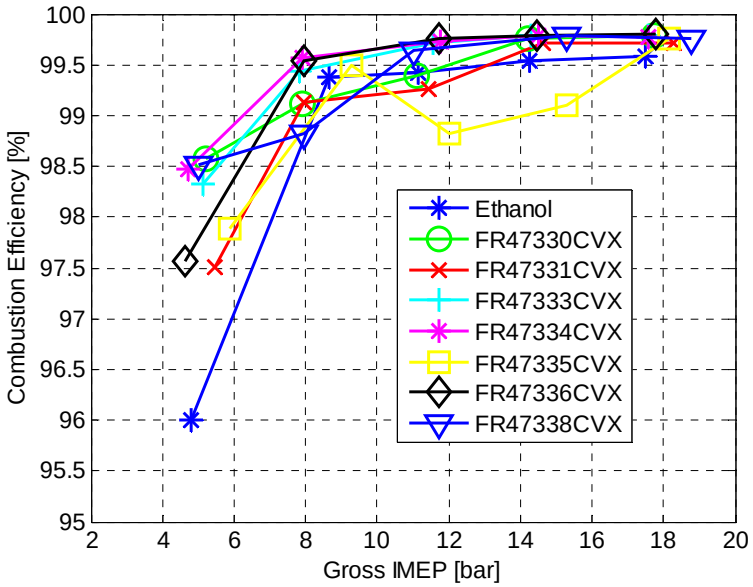


**Figure 73: Combustion efficiency as a function of load and fuel type. Scania D12, low compression ratio (rc=14.3).**

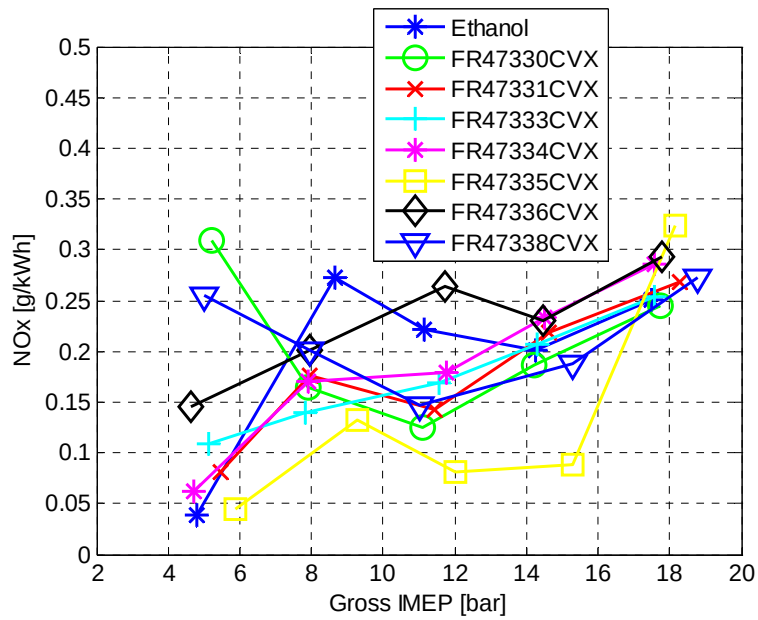


**Figure 74: Specific NOx as a function of load and fuel type. Scania D12, low compression ratio (rc=14.3).**

In [77] further studies were conducted to determine if there are better running conditions (CA50, λ, EGR and pilot ratio) in order to further increase the efficiency and reduce soot, NOx and maximum pressure rise rate. At the maximum load allowed by the dyno, fuels FR47335CVX, FR47338CVX and FR47334CVX (RON 99, 89 and 69 respectively) were tested in the following sweeps:

- Inlet pressure.
- EGR.
- Pilot ratio.
- Combustion phasing.

A least squares optimization algorithm was applied to all the acquired data. The gross indicated efficiency, maximum pressure rise rate, NOx and soot were empirically modeled as a function of $P_{ratio}$, CA50, EGR and λ. A second order model was used to build up this system of four equations (see Equation 4); each fuel had a different system of equations.

**Equation 4:**

$$Y = a_{ij} \cdot \sum_i X_i \sum_j X_j + const; \quad X = [CA50 \quad EGR \quad \lambda \quad P_{ratio}];$$
$$Y = [\eta_{gross} \quad dP_{\max} \quad NOx \quad soot]$$

The inlet pressure was not included in the X variables since if EGR and λ are known the use of this parameter is redundant. This second order model is able to predict fairly well the Y parameters (see Figure 75) ; for all the three fuels the goodness of the fit, $R^2$, for all the four output parameters is higher than 0.99.

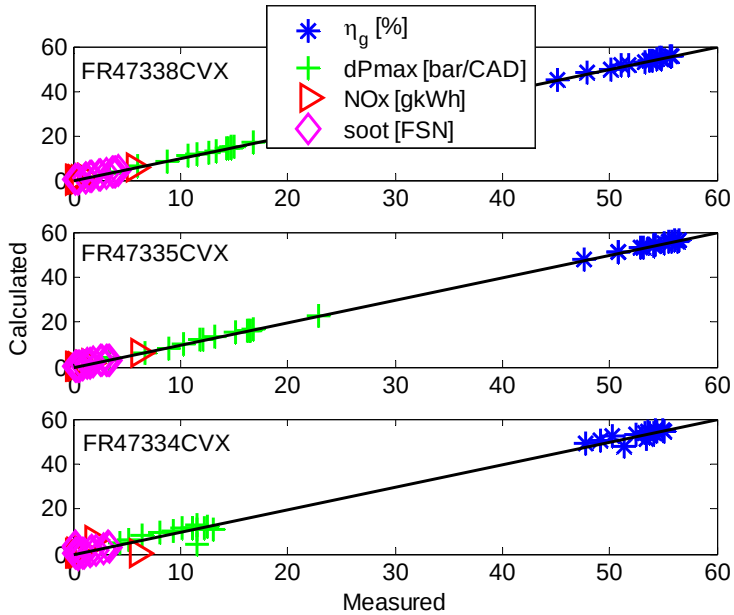


**Figure 75: Predicted versus measured data using three different gasoline fuels.**

By assuming the following constraints: gross indicated efficiency higher than 54%, maximum pressure rise rate below 15 bar/CAD, NOx below 0.30 g/kWh and soot below 1.50 FSN, the model calculated that for the load under examination the best possible settings are the ones shown in Table 8. According to the model, for all the three fuels, the optimal settings are a pilot ratio of 0 %, CA50 between 2 and 4 aTDC, EGR between 44 and 52 % and λ between 1.54 and 1.58. This confirms that the settings used in [48] and [72] are the best possible. It is important to emphasize that these settings are valid for the experimental set-up used in this research and that a different engine might behave differently.

**Table 8: Calculated inputs and outputs from the proposed empirical model.**

| FR47338CVX | | | FR47335CVX | | | FR47334CVX | | |
|---|---|---|---|---|---|---|---|---|
| Pratio | 0 | [%] | Pratio | 0 | [%] | Pratio | 0 | [%] |
| CA50 | 3.19 | [aTDC] | CA50 | 2.04 | [aTDC] | CA50 | 3.98 | [aTDC] |
| EGR | 45.62 | [%] | EGR | 51.6 | [%] | EGR | 51.34 | [%] |
| λ | 1.54 | [-] | λ | 1.58 | [-] | λ | 1.57 | [-] |
| | | | | | | | | |
| $\eta_{g\_ind}$ | 54.65 | [%] | $\eta_{g\_ind}$ | 56.18 | [%] | $\eta_{g\_ind}$ | 54.32 | [%] |
| dPmax | 13.9 | [bar/CAD] | dPmax | 14.41 | [bar/CAD] | dPmax | 11.49 | [bar/CAD] |
| NOx | 0.18 | [g/kWh] | NOx | 0.23 | [g/kWh] | NOx | 0.19 | [g/kWh] |
| soot | 1.26 | [FSN] | soot | 1.21 | [FSN] | soot | 1.43 | [FSN] |

### 3.4.6 High Efficiency and Low Emissions Scania D13

The analysis of the data obtained in [77], [48] and [72] indicated that high efficiency is achieved mainly because of the λ, EGR, CA50 and CA90-10 combinations rather than burning fuel in kinetically controlled combustion mode. For instance the rate of heat release presented in Figure 76 shows that at 18 bar gross IMEP fuel FR47334CVX is burning mainly in mixing controlled mode and despite that the gross indicated efficiency is always between 53 and 54%. For the same data set, the analysis of the combustion duration and phasing, see Figure 77, suggests that the efficiency is mainly influenced by the phasing of the combustion rather than its duration (if CA90-10 is within reasonable values); this is also supported by the modeling work presented in Figure 23. This finding has remarkable implications because this means that by using the correct:

1. Rail pressure (that is proportional to the combustion duration, CA90-10)
2. Start of injection (that is directly correlated to the combustion phasing, CA50)
3. λ and EGR combination

it is always possible to achieve very high efficiency, directly control the combustion event and keep acceptable maximum pressure rise rate at every load and speed, and in addition because of the well chosen λ and EGR values the emissions are not an issue. Also if the injection system can reach high enough pressure there is no need to use the double injection strategy presented in 3.4.4.

The concept proposed in 3.4.5 was tested in a standard Scania D13 at 1250 rpm between 5 and 25 bar gross IMEP. The engine was un-modified: standard swirl, piston and nozzle. The engine was ran with single injection and the previously mentioned λ and EGR combination.

As expected the combustion efficiency is always higher than 98% throughout the load sweep for all the fuels tested. At 5 bar IMEP two fuels have efficiency below 98%. High combustion efficiency was achieved keeping NOx, in the worst case, below 0.65 g/kWh. As for the Scania D12 also in the D13, the NOx-combustion efficiency trade-off has been strongly mitigated; see Figure 78. In most of the load points with some fuels it was possible to keep NOx both below US10 and EUVI emissions regulations. For all the fuels tested, the gross indicated efficiency, presented in Figure 79, was between 52 and 55%. It is believed that this parameter is slightly lower as compared to Figure 65 and Figure 72 mainly because the wider umbrella angle (as shown in 3.1.3) contributes to increase the heat transfer. Figure 80 presents the rate of heat release for the fuels under examination at maximum load. As can be observed all the fuels are burning in mixing controlled combustion mode and despite that an average 53% gross indicated efficiency was reached at the 25 bar IMEP load point. In Figure 81 the mixing period is presented. Using roughly 50% of EGR, at maximum load, this parameter is in the range of 0 CAD meaning that combustion starts immediately after the injection ends. This has positive consequences for both soot and CO. In the first case, during combustion, rich fuel pockets are avoided thus less particulate is produced according to Figure 27, on the other hand because of the increase in mixing, CO production due to undermixing is avoided. Soot as a function of the relative excess of air at 20 and 25 bar gross IMEP is presented in Figure 82. It can be observed that gasoline fuels are capable of running very close to stoichiometric, λ ~ 1.35, and still keep soot below 0.5 FSN while diesel shows values in the range of 2.5-2.75 FSN. This finding suggest that one of the many advantages of running gasoline PPC is the

improved power density of a heavy duty compression ignition engine since usually at maximum load the relative excess of air is roughly 1.5-1.55. As shown in Figure 83 using gasoline fuels in partially premixed combustion enables to drastically reduce the NOx-soot trade-off in the whole load range; in some load points for some fuels the soot bump can be noted. Diesel produces roughly 10 times more soot than the gasolines under examination between 5 and 25 bar IMEP; the reasons behind this behavior were explained in 3.2.3.

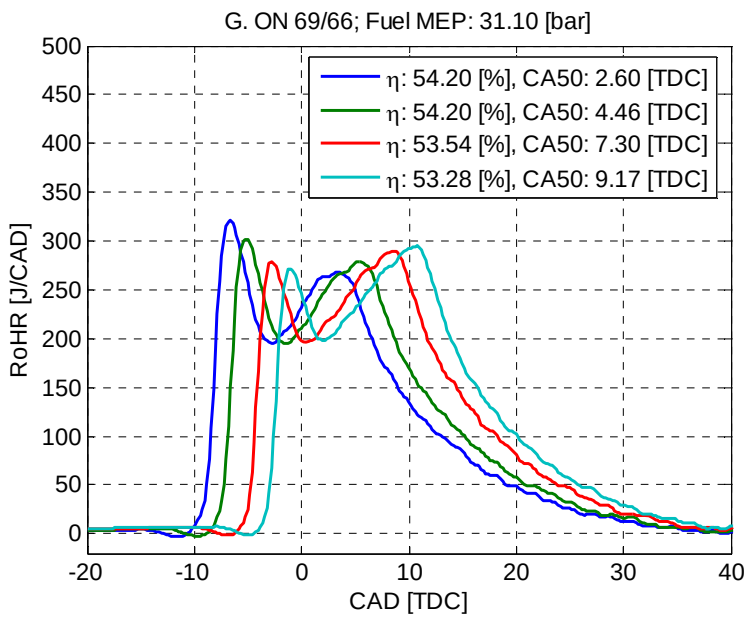


**Figure 76: Rate of heat release traces as a function of CA50. Scania D12, low compression ratio (rc=14.3:1) [77].**

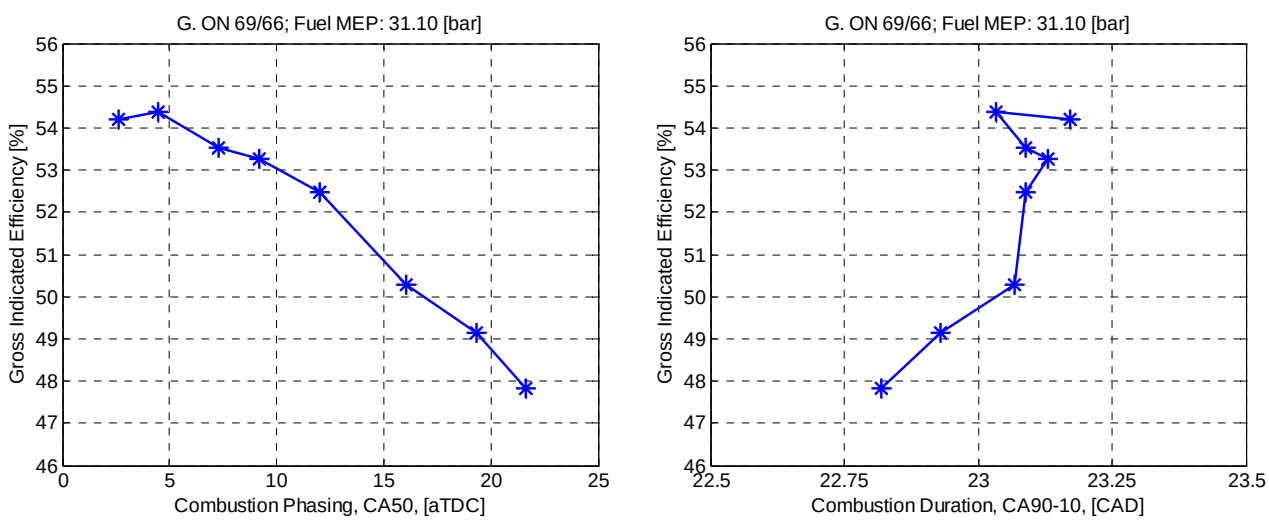


**Figure 77: Gross indicated efficiency as a function of CA50 (left) and combustion duration (right). Scania D12, low compression ratio (rc=14.3:1) [77].**

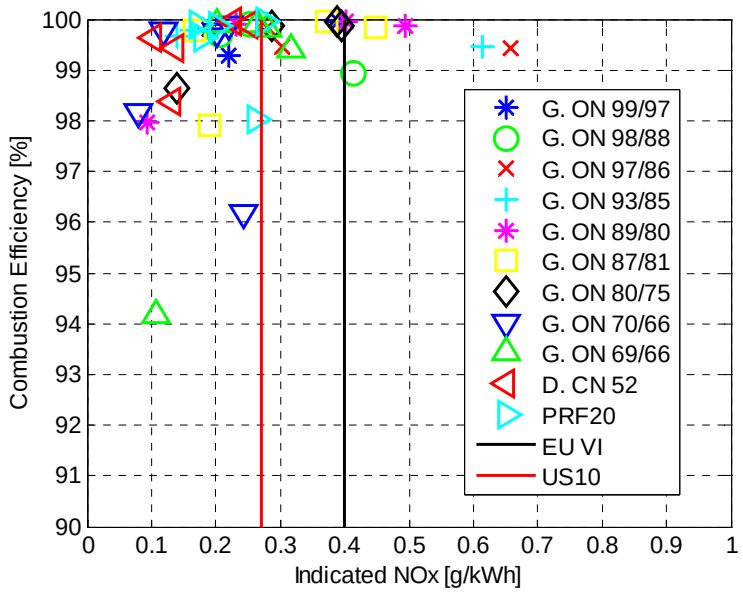


**Figure 78: Combustion efficiency vs. NOx for different type of fuels from 5 to 25 bar gross IMEP.**

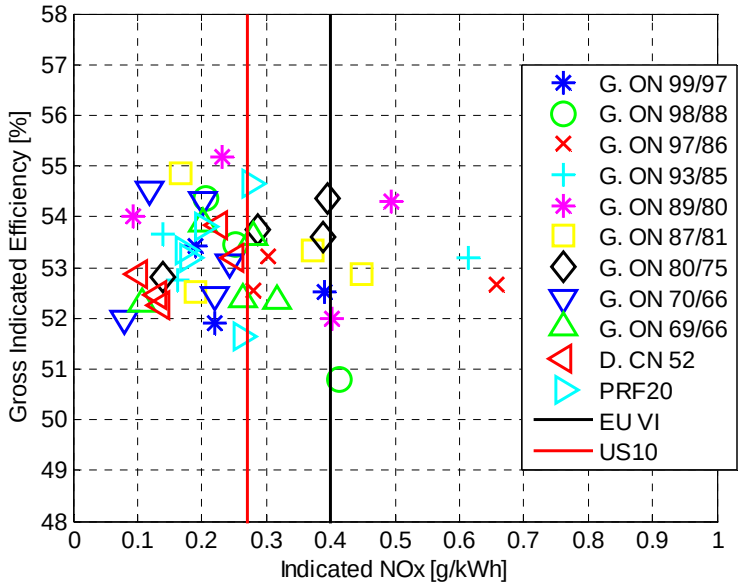


**Figure 79: Gross indicated efficiency vs. NOx for different type of fuels from 5 to 25 bar gross IMEP.**

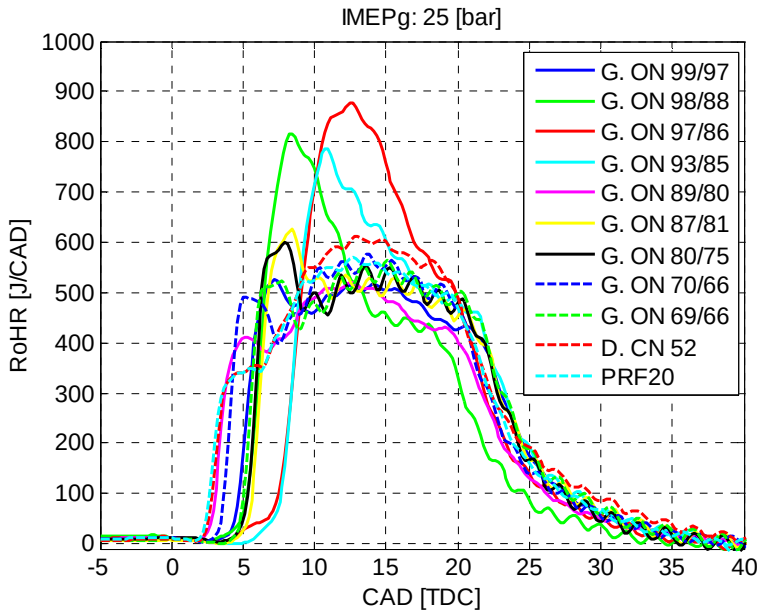


**Figure 80: Rate of heat release at 25 bar gross IMEP for different fuels.**

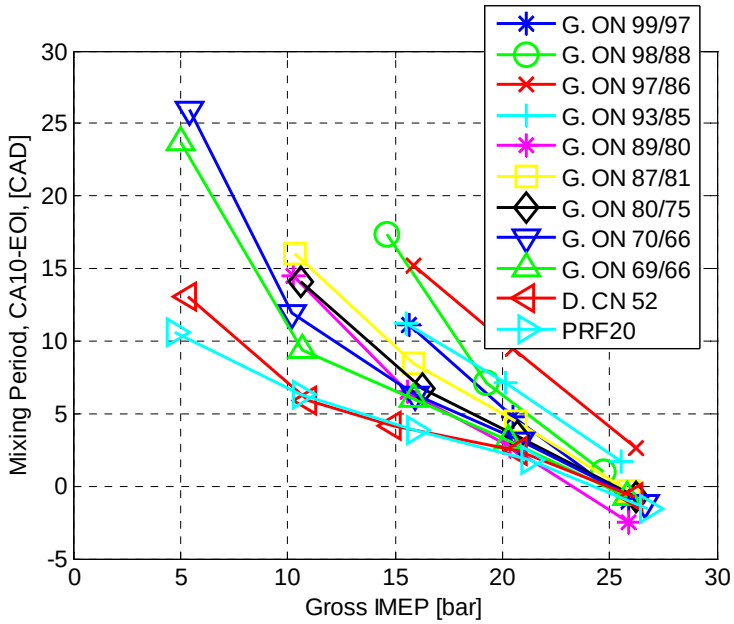


**Figure 81: Mixing period as a function of load and fuel type.**

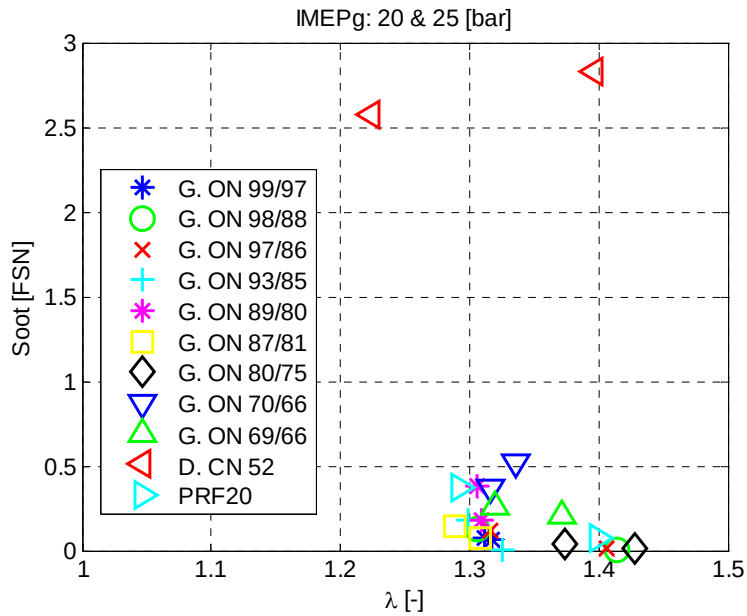


**Figure 82: Soot as a function of λ for different fuels at 20 and 25 bar gross IMEP.**

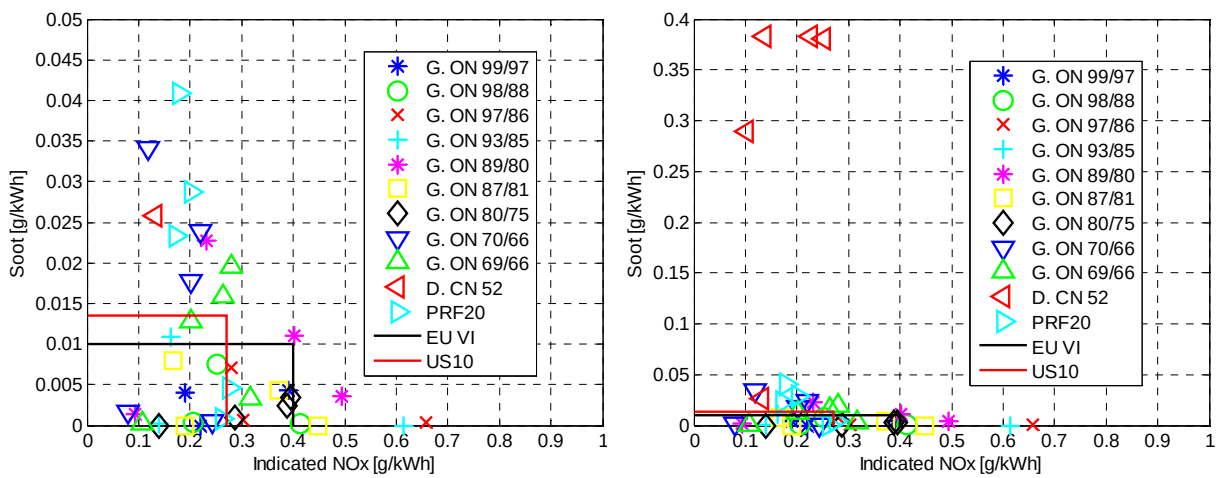


**Figure 83: Soot vs. NOx for different fuels from 5 to 25 bar gross IMEP. The left figure is a zoom of the right one.**

To demonstrate the viability of the concept over the entire load range, idle was also tested (2.5 bar gross IMEP and 600 rpm) using fuel FR47334CVX (G. ON 69/66) in a combustion phasing sweep. As shown in Figure 94, combustion efficiency in the range of 93% was achieved using only 303 K as inlet temperature while the gross indicated efficiency was roughly 40% when CA50 was between 3.5 and 7 [ATDC]; see Figure 84. In Figure 85 the legislated emissions are presented. The indicated NOx and soot are very low both below US 10 and EU VI legislations. On the other hand CO and HC are in the ranges of 32.5 and 17.5 g/kWh respectively as a result of relatively poor combustion efficiency.

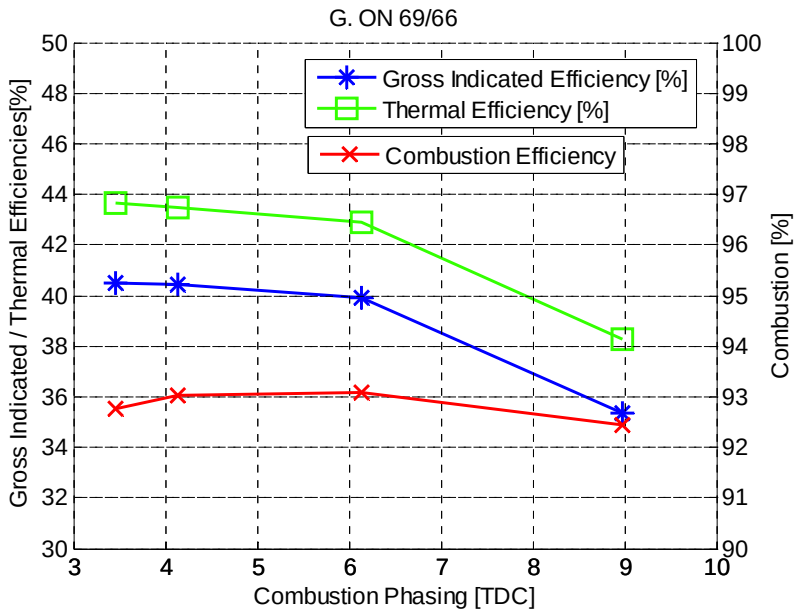


**Figure 84: Gross indicated, thermal and combustion efficiencies versus CA50 at idle conditions.**

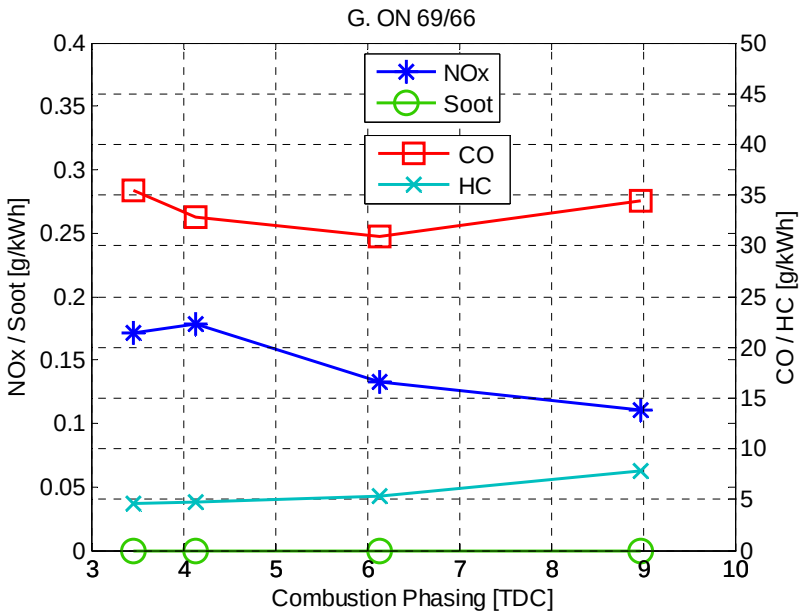


**Figure 85: Indicated NOx, soot, CO and HC versus CA50 at idle conditions.**

### 3.4.7 Optimum Fuel for Gasoline Partially Premixed Combustion

Figure 86 shows the stable operating load region as a function of the octane number using the standard Scania D13 and the modified Scania D12. The low load limit was identified as the point in which ambient inlet temperature (303 K for the D13 and 308 K for the D12) was not enough to achieve stable combustion.

For the partially premixed combustion concept proposed in this manuscript, the best fuels (among the ones that have been tested) are gasolines with an octane number in the range of 70 (FR47334CVX and FR47336CVX). The selection was mainly based on the fact that both fuels can run the whole load range using ~50% of EGR and λ ~1.5:

- Without any need of inlet temperature higher than ambient.
- Without producing excessive acoustic noise; see Figure 44 and Figure 45.
- The gross indicated efficiency is fairly high and soot is pretty low simultaneously with low NOx; see Figure 87.

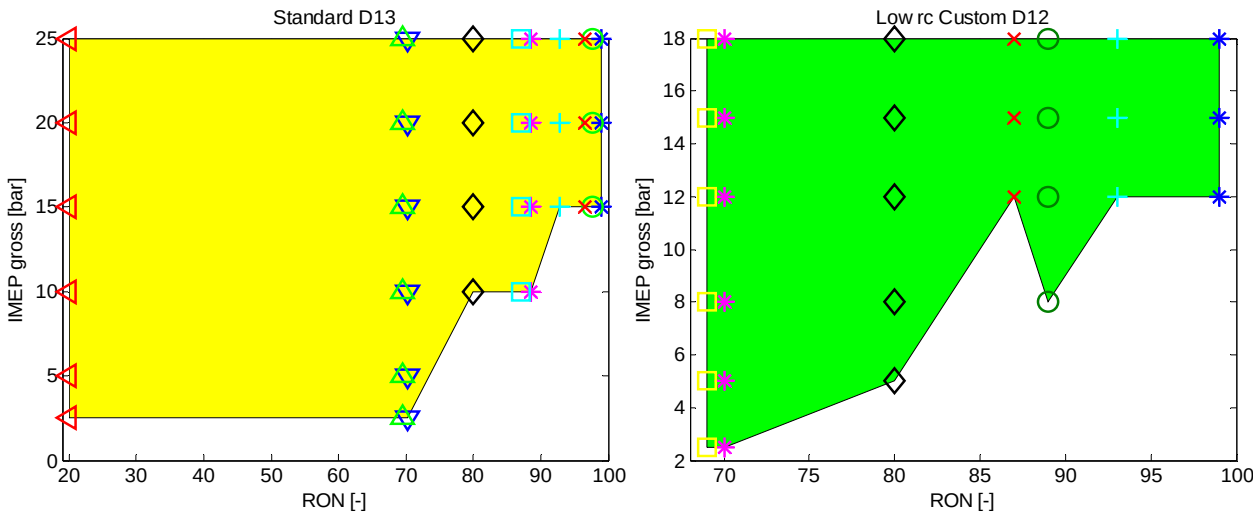


**Figure 86: Stable operating load range for different types of gasolines using the standard Scania D13 (left) and the modified Scania D12 (right).**

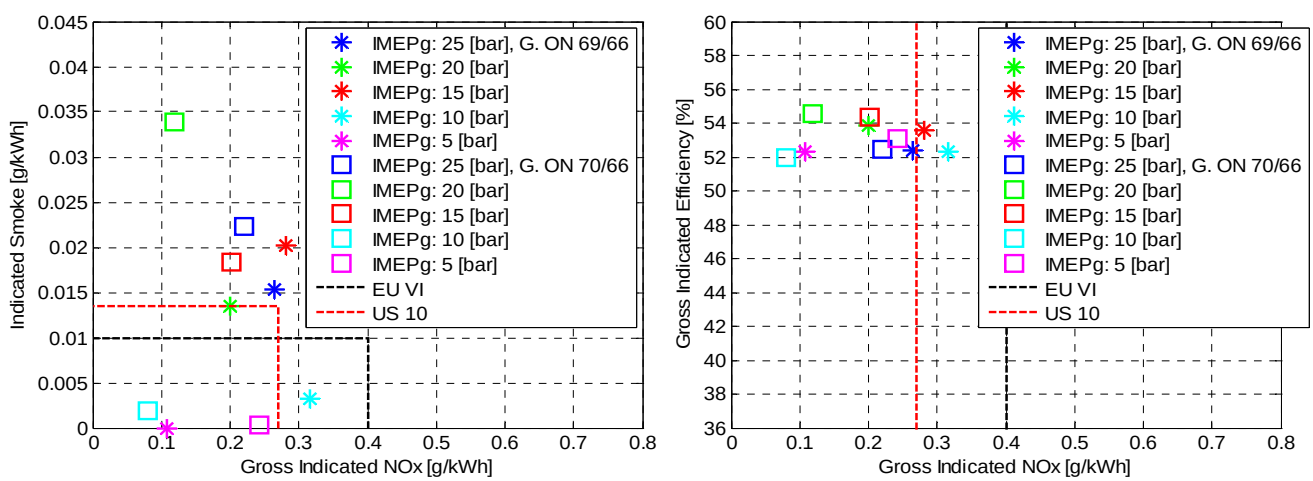


**Figure 87: NOx-soot trade-off (left) and NOx-gross indicated efficiency trade-off (right) at different loads for the two best fuels using the Scania D13.**

### 3.4.8 Why Not Optimize the Engine for a Given Fuel?

It might be argued that a market gasoline with an octane number in the range of 70 is not available. When developing, almost from scratch, an advanced internal combustion engine concept for the long term future (10-20 years span) there are two main variables: engine layout and fuel type. At this point either the fuel type is kept constant and the engine layout is adjusted in order to accommodate emissions and efficiency requirements, or the fuel type can be modified and the engine design is kept unvaried. In both cases I believe that in the future the engines will run with high levels of EGR to decrease NOx and heat loss to the walls, and a fuel in the boiling point range of gasoline because petrol has much less tendency to soot than diesel when lot of EGR is used. During the second year of this gasoline PPC research, the main objective was to keep the engine layout as unchanged as possible and to find out the most suitable fuel for this concept. On the other hand if the fuel octane number is constrained to 85-95, an alternative concept is the one proposed in Figure 88. The hybrid mode proposed in Figure 88 consists in running the engine from 100% load to the minimum ignitibility limit using the PPC concept proposed in the thesis. At lower load there are two alternatives:

1. To use a variable compression ratio mechanism to increase the compression ratio in order to supply to the fuel-air mixture enough energy to achieve ignition. In this way the combustion mode can still be the PPC concept proposed in the thesis.
2. To use a glow plug in order to promote ignition of a mildly stratified combustible mixture. The glow plug is preferred to the spark plug because of ignitability issues when the fuel-air mixture is highly diluted.

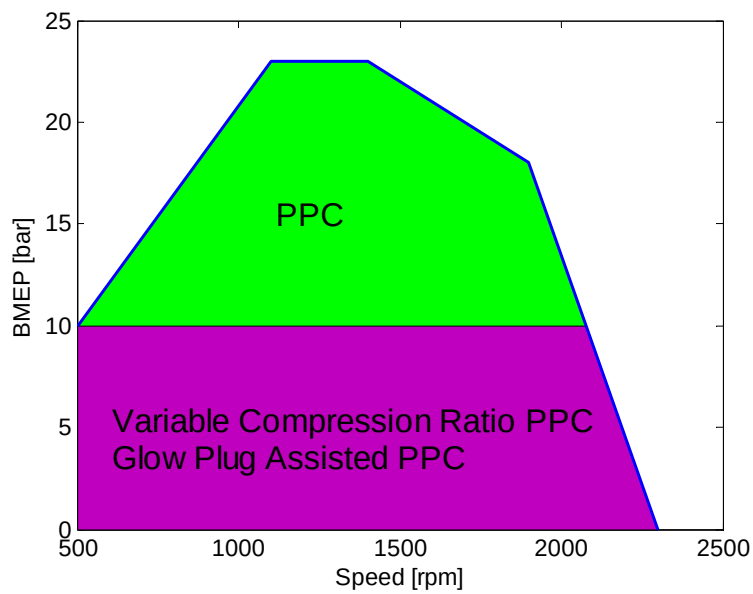


**Figure 88: Hybrid PPC combustion concept using medium or high ON gasoline fuels.**

### 3.4.9 Is 50% Brake Efficiency Viable in a Heavy Duty Truck Engine?

In Figure 89 the brake thermal efficiency for heavy duty engine applications is presented. The diagram shows a steady increase in efficiency up to roughly 44.5% in 2000 then because emissions regulations were getting tighter the efficiency steeply dropped down to 41.5%.

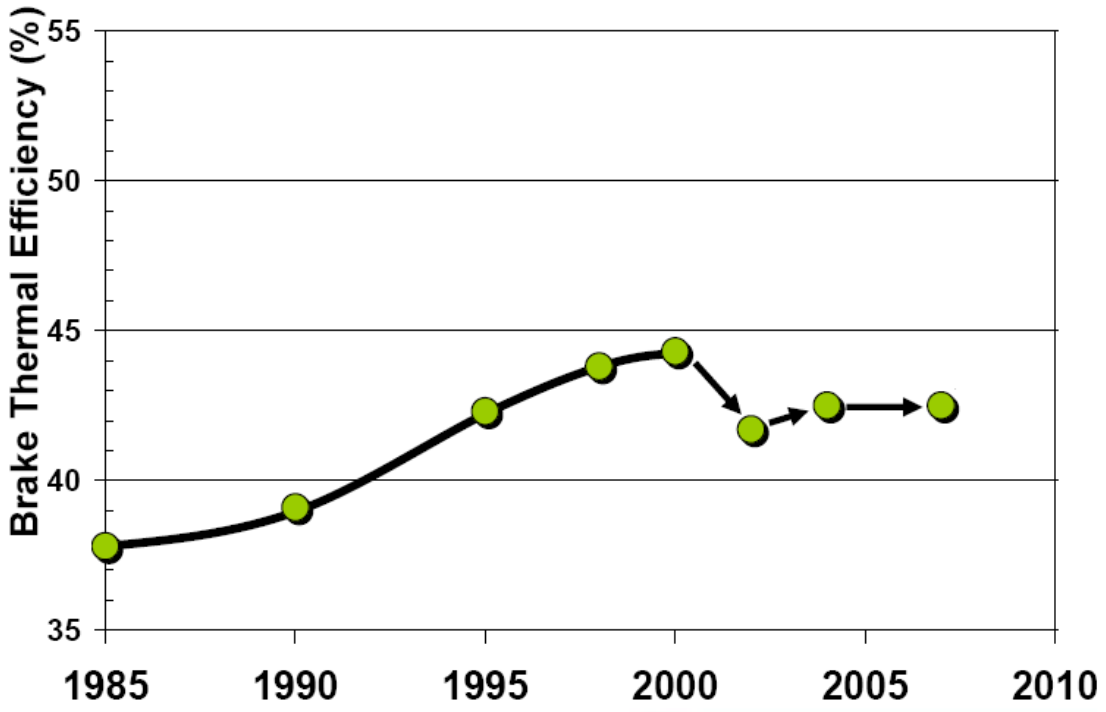


**Figure 89: Historical perspective of heavy duty brake efficiency [78].**

The gasoline PPC concept proposed so far enabled very low emissions together with very high gross indicated efficiencies [48], [49] and [72] to be attained. It is of interest now to compute the brake efficiency in order to understand if it is possible to achieve values comparable to the year 2000, see Figure 89, or even higher. The brake efficiency was estimated using the pumping and friction losses from a standard six cylinder Scania D13. It was decided not to use the measured pumping and friction losses because in [48], [49] and [72] the experiments were carried out on a single cylinder engine and it is known that friction and pumping work are quite different if measured in a single cylinder compared to a standard engine; see Figure 90. Even for the same differential pressure, Pex-Pin, the gas exchange efficiency is lower with a single cylinder engine because of the tuning and the different intake and exhaust manifold volumes; this parameter is shown in Figure 91.

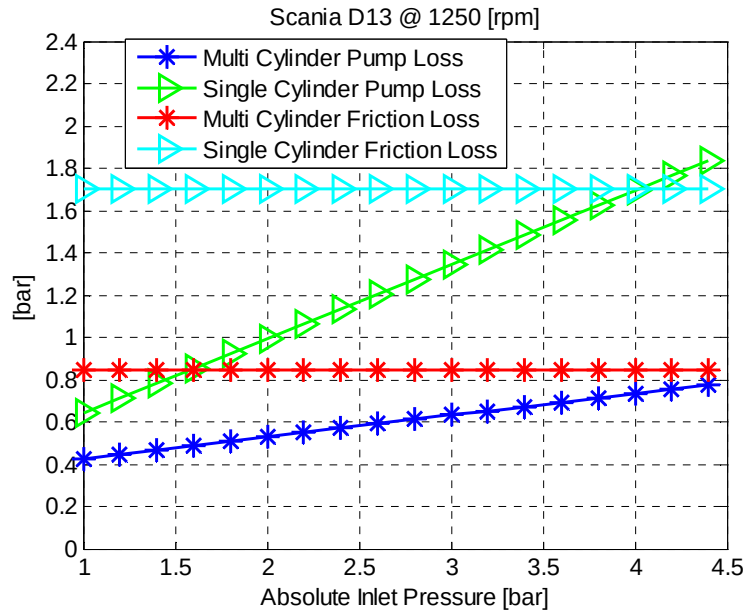


**Figure 90: Friction and pumping losses as a function of the absolute inlet pressure using both a single and a multi cylinder engine.**

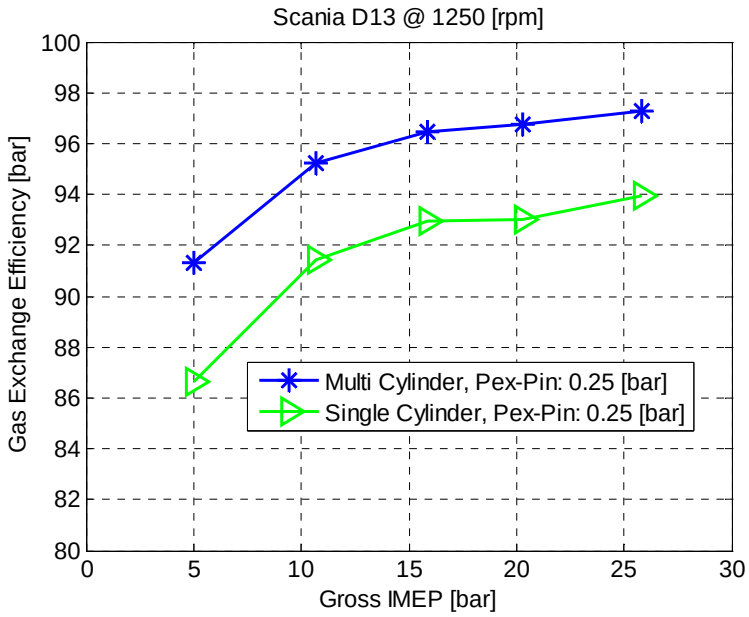


**Figure 91: Gas exchange efficiency as a function of load in a single and a multi cylinder engine.**

The brake efficiency was estimated for fuels FR47333CVX and FR47334CVX. The first one was the most suitable in the Scania D12 running with the high compression ratio piston and in addition this fuel was the one that reached 57% gross indicated efficiency. Fuel FR47334CVX was the most suitable for the standard Scania D13. Even though slightly better efficiency was achieved with fuel FR47338CVX than FR47334CVX in the Scania 12 running with the low compression ratio piston it was decided to estimate the brake parameters using the FR47334CVX because at high load it was producing 50% less soot than FR47338CVX. In Figure 92 the brake efficiency is presented for the Scania D12 running both with low and high compression ratio pistons and the standard Scania D13. When the load is higher than 10 bar gross IMEP all three engines have estimated brake efficiencies higher than 45%. A peak brake efficiency of 50% is achieved with both the low compression ratio D12 and the standard Scania D13 engines. Such a high efficiency is achieved while maintaining low brake emissions, see Figure 93; an exception is represented by the soot levels of the D12 because of the obsolete injection system. Up to the moment when this thesis was written no references have been found on the capability to achieve such high brake efficiency together with low emissions and acceptable maximum pressure rise rate using a heavy duty truck engine. Comparable values of efficiencies can be found in the big ship engines, for instance the biggest and most efficient engine in the world is the Wartsila-Sulzer RTA96-C that can reach a measured brake efficiency of 52% [79]. Higher brake efficiencies than 52% can be achieved by combining the standard combustion cycle with a waste heat recovery system. For instance the Wartsila-Sulzer 12RT-fl ex96C has a brake efficiency of 49.3% but if a waste heat recovery system is combined with this engine the efficiency increases up to 54.9% [80].

It might be argued that such high estimated brake efficiency was achieved because of the high boost and in real life applications there will never be enough exhaust enthalpy in order to achieve the desired inlet pressure when the exhaust pressure is only 0.25 bar higher than the inlet. Therefore, the exhaust enthalpy and the energy required to compress the inlet air to the desired pressure were calculated. The ratio between the compressor work and exhaust enthalpy represents the minimum efficiency that a turbo-system must have in order to achieve the desired inlet pressure; this ratio represents the product of the turbine, compressor and turbo mechanical efficiencies. The minimum turbocharger efficiency for the three load sweeps under examination is shown in Figure 94. In heavy duty engine applications usually this parameter is in the range of 45-55%. The boost values chosen for the standard D13 and low compression ratio D12 seem to be viable; on the other hand with the high compression ratio D12 at lower load the viability of the inlet pressure is on the border line.

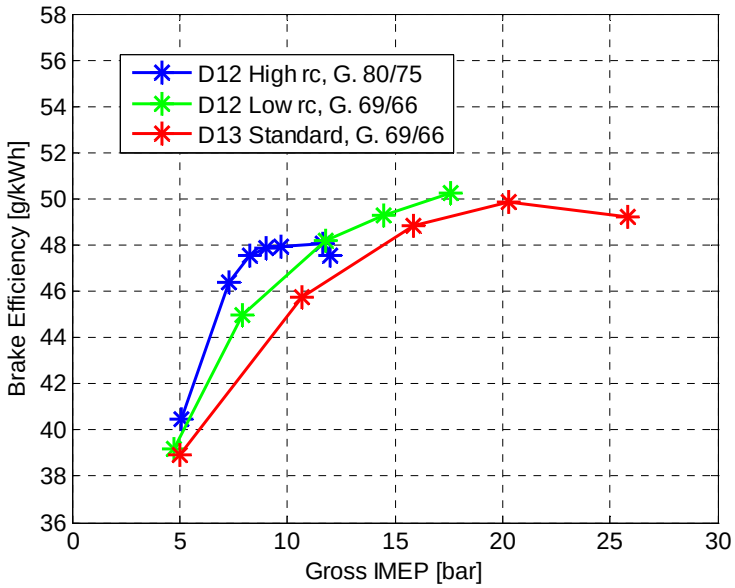


**Figure 92: Estimated brake efficiency using the modified D12 and the standard D13.**

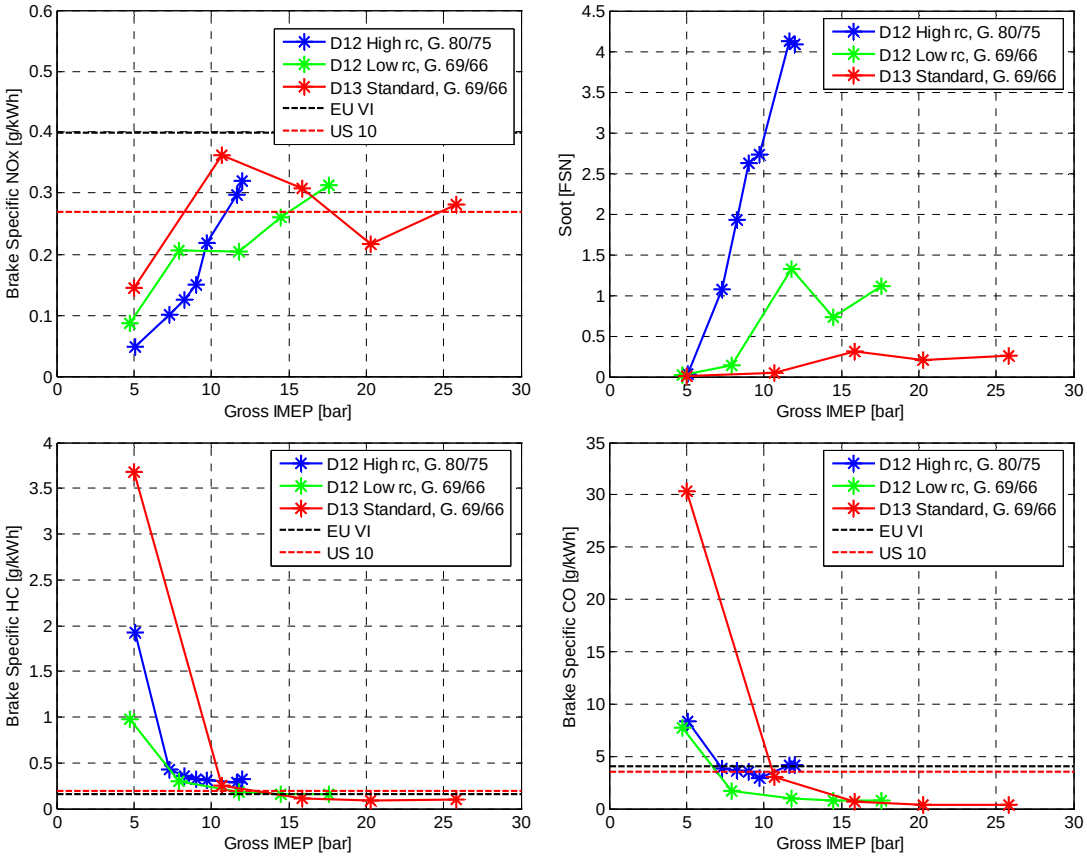


**Figure 93: Estimated brake NOx, CO, HC and soot using the modified D12 and the standard D13 engine.**

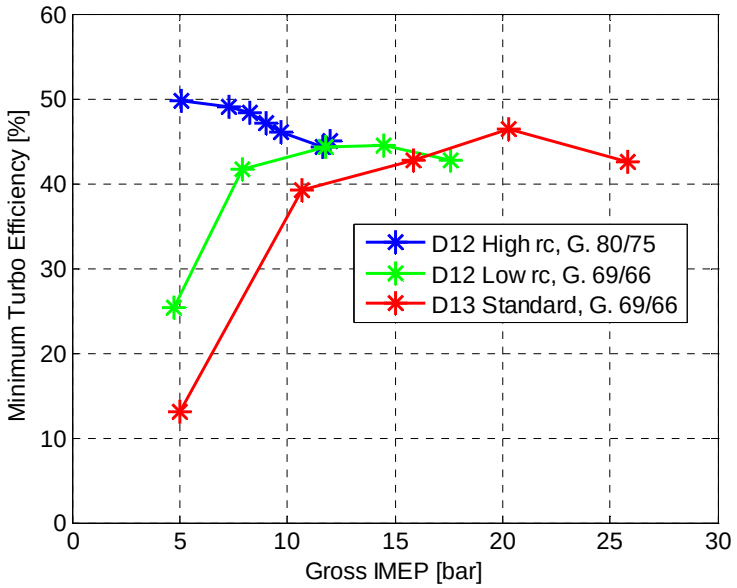


**Figure 94: Minimum turbo efficiency using the modified D12 and the standard D13.**

# 4 Summary and Conclusions

From an internal combustion engine perspective, environmental concerns such as global warming and increases in crude oil prices are highlighting the need for increasing the efficiency of the engines to reduce both global emission i.e. $CO_2$ and local pollutants i.e. NOx, soot, CO and HC. For classical internal combustion engine concepts such as diesel compression ignition and gasoline spark ignition, an unfixable trade-off exists between local and global emissions. HCCI was a part of the solution to this dilemma but low power density, acoustic noise and combustion control make this concept not so attractive for real life applications.

The ambitious objective of this work was to create a combustion concept with very high efficiency, low local emissions, acceptable acoustic noise, easy to control, high power density and viability in running the whole load and speed range. The path was not straight forward and it required a deep understanding on the best way to burn fuel, the interaction between combustion and cylinder environment, and most important of all the interaction between fuel and type of combustion. When gasoline partially premixed combustion started at Lund University, based on previous PPC experiences, it was thought that the best way to burn fuel is kinetically controlled combustion with a broad separation between end of the injection and start of the combustion, thus resulting in low NOx, low soot and high efficiency. It was soon realized that by burning gasoline in this way, the combustion was still hard to control, the acoustic noise was high and despite the fact that now it was possible to run high load with low emissions and relatively high efficiency, low load operations became a problem because of ignitibility issues. Roughly six months of research showed that the best way to burn fuel is in mixing or partially mixing controlled mode. With mixing controlled combustion the trade-off between local and global emissions can be overcome if the engine runs with the appropriate EGR-$\lambda$ combination, the correct combustion phasing and duration, and the right fuel.

By running the whole load range with ~50% of EGR and ~1.5 in $\lambda$, the maximum temperature during combustion is always between 1500 and 2000 K. CO and HC can be oxidized because the temperature is higher than 1500 K while NOx formation can be limited because combustion takes place below 2000 K. Unless a fuel contains $N_2$, the strategy seems to work with a wide range of fuels. Both gasoline type of fuels and diesel can run the whole load range with high combustion efficiency and low NOx emissions. In terms of emissions the main difference between diesel and gasoline is in the amount of soot they produce with the above mentioned EGR- $\lambda$ combination. Diesel fuel produces up to 2.8 FSN of soot at 26 bar gross IMEP while gasoline roughly 0.3 FSN. It was found that this paramount gap was due to the chemical composition of the two fuels rather than differences in octane number that can result in different stratification levels. It was discovered that if two fuels have the same ON but one has a boiling point range of gasoline and the other of diesel, this second one will produce a factor of 5-6 more soot. Because gasoline has lower tendency to soot, it was possible to run the engine at 26 bar IMEP with 50% of EGR, $\lambda$ 1.3 and still keep soot below 0.3 FSN without producing much NOx with fuel burning in mixing controlled mode. This means that gasoline partially premixed combustion has much higher power density as compared to classical diesel or diesel PPC combustions. In this manuscript it was also demonstrated that if the injection system is not up to date, lot of soot can be produced with gasoline fuels as well. With the gasoline partially premixed combustion concept presented in this thesis, very

high efficiency was obtained by the appropriate selection of combustion phasing and combustion duration, the high amount of boost, the low combustion temperature and the narrow nozzle umbrella angle. These five ingredients helped to simultaneously reduce the heat transfer and the exhaust loss thus resulting in higher gross indicated work. This result was fuel properties independent, both diesel and gasoline can achieve high efficiency by properly tailoring the combustion process. High boost level promoted low temperature reaction activities that helped to mitigate the maximum pressure rise rate in partially premixed combustion mode thus resulting in acceptable acoustic noise both with diesel and gasoline.

Gasoline partially premixed combustion was tested both in a single cylinder Scania D12 and in a six cylinder Scania D13 modified to run only with one cylinder active using a variety of fuels in the gasoline boiling point range but very different octane number. The D12 achieved a gross indicated efficiency in the range of 55% between 5 and 18 bar gross IMEP, while the D13 52-53% between 5 and 26 bar IMEP. The brake efficiency was estimated using the engine manufacturer data for both pump and friction losses, it was found that 50% brake efficiency is viable in both engines. With both engines NOx is very low and in the majority of the load points it is below 0.40 g/kWh. Very low soot was obtained using the D13. As previously mentioned at 26 bar IMEP soot was in the range of 0.25 FSN. Values in the range of 1 FSN or higher were measured with the D12 as a result of the obsolete injection system. High efficiency and low emissions were achieved keeping the relative maximum pressure rise rate below 8 bar/CAD with both set-ups.

The analysis of different type of fuels demonstrated that the best fuel for this combustion concept is a fuel in the boiling point range of gasoline but with an octane number in the range of 70. Low octane number is necessary in order to properly run the engine from idle to maximum load without the need of high inlet temperature (e.g. 283-293 K are enough to start the engine even with ambient coolant temperature) while a short carbon chain and absence of poli-aromatics and poli-naphthenes content is necessary to limit soot production.

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# 6 Abbreviations and Symbols

| | |
|---|---|
| **$\lambda$:** | Relative excess of air |
| **$\eta_b$:** | Brake efficiency |
| **$\eta_{comb}$:** | Combustion efficiency |
| **$\eta_{comb\_ineff}$:** | Combustion inefficiency |
| **$\eta_g$:** | Gas exchange efficiency |
| **$\eta_{ge}$:** | Gas exchange efficiency |
| **$\eta_{gross}$:** | Gross indicated efficiency |
| **$\eta_m$:** | Mechanical efficiency |
| **$\eta_n$:** | Net indicated efficiency |
| **$\eta_{net}$:** | Net indicated efficiency |
| **$\eta_{ther}$:** | Thermal efficiency |
| **AFst:** | Stoichiometric air fuel ratio |
| **Ag:** | Argon |
| **aTDC:** | After top death center |
| **ATDC:** | After top death center |
| **BMEP:** | Brake mean effective pressure |
| **BTDC:** | Before top death center |
| **CA10:** | Crank angle at 10% total fuel energy released |
| **CA5:** | Crank angle at 5% total fuel energy released |
| **CA50:** | Crank angle at 50% total fuel energy released |
| **CA9010:** | Combustion duration |
| **CAD:** | Crank angle degree |
| **CI:** | Compression ignition |
| **CN:** | Cetane number |
| **CO:** | Carbon monoxide |
| **$CO_2$:** | Carbon dioxide |
| **cp:** | Specific heat at constant pressure |
| **cv:** | Specific heat at constant volume |
| **D.:** | Diesel |
| **dP:** | Cylinder pressure derivative |
| **E.:** | Ethanol |
| **EGR:** | Exhaust gas recirculation |
| **EOI:** | End of injection |
| **EPA:** | Environmental protection agency |
| **Ex:** | Exhaust energy |
| **FSN:** | Filtered smoke number |
| **FT:** | Fischer Tropsch |
| **FTIR:** | Fourier transformed infrared spectroscopy |
| **Fuel MEP:** | Fuel mean effective pressure |
| **G.:** | Gasoline |
| **GDI:** | Gasoline direct injection |
| **$H_2$:** | Hydrogen |
| **HCCI:** | Homogeneous charge compression ignition |
| **HD:** | Heavy duty |

**HEDGE:** High efficiency diluted gasoline engine
**Hlv:** Lower heating value
**HT:** Heat transfer
**HTR:** High temperature reactions
**ICE:** Internal combustion engines
**IMEP:** Indicated mean effective pressure
**ISFC:** Indicated specific fuel consumption
**LHV:** Lower heating value
**LTR:** Low temperature reactions
**MBT:** Maximum brake torque
**$m_{cyl}$:** In cylinder mass
**MeOH:** Methanol
**MEP:** Mean effective pressure
**$m_f$:** Fuel mass
**MON:** Motor octane number
**MPRR:** Maximum pressure rise rate
**n.a.:** not available
**NOx:** Nitrogen monoxide and dioxide
**$O_2$:** Oxygen
**PPC:** Partially premixed combustion
**PRF:** Primary reference fuels
**Qhr:** Chemical heat release
**rc:** Compression ratio
**RoHR:** Rate of heat release
**RON:** Research octane number
**SACI:** Spark assisted compression ignition
**SI:** Spark ignition
**SOIp:** Start of pilot injection
**T:** Temperature
**T10:** Fuel 10% distillation temperature
**T50:** Fuel 50% distillation temperature
**T90:** Fuel 90% distillation temperature
**TDC:** Top death center
**Tex:** Exhaust temperature
**THE:** Towards high efficiency
**Tmax:** Maximum temperature
**Tref:** Reference temperature
**Tsoc:** Temperature at the start of the combustion
**UA:** Umbrella angle
**UHC:** Unburned hydrocarbon
**VCR:** Variable compression ratio
**$V_d$:** Displacement volume
**$W_g$:** Gross indicated work
**$W_n$:** Net indicated work
**WOT:** Wide opened throttle
**Zn:** Zinc

# 7 Appendix

## 7.1 Gasoline Partially Premixed Combustion in a Light Duty Engine

At the time this research was performed it was still believed that kinetically controlled combustion was the best way to burn the fuel-air mixture in order to achieve high efficiency and low emissions of NOx and soot. In addition at the time [107] was written, the author still did not fully understand the relationship between injection pressure, injection duration and engine speed in order to achieve the optimum level of stratification when combustion starts. It was believed that at every load-speed point the injection pressure was suppose to be as high as possible in order to eliminate soot. Thus the results obtained in [107] are not as good as the achievement in [48] and [49]. Using a VOLVO D5 with a standard piston but narrower nozzle umbrella angle (see Table 9) a load sweep was performed between 6 and 17 bar gross IMEP at 2000 rpm. The fuel used was regular Swedish pump gasoline; see Table 6. It was impossible to work with loads below 6 bar IMEP because of ignitibility issues.

**Table 9: Geometrical properties of the VOLVO D5**

| | |
|---|---|
| Displaced Volume | 2378 cc |
| Stroke | 92.3 mm |
| Bore | 81 mm |
| Geom. comp. ratio | 16.15 :1 |
| Injection Cone Angle | 120° |
| Injector Type | Solenoid |
| Nozzle (number x diameter) | 7 x 0.14 mm |

The injection strategy developed for the Scania D12 was slightly modified. The load sweep was performed using three injections placed at -90, -55 and -25 [aTDC]; their position was held constant during the whole sweep. The load was mainly controlled using the first injection, small variations in the third injection were done at high load while the amount of fuel in the second injection was unchanged; see Figure 95. This injection strategy was empirically developed and the main target was to keep the maximum pressure rise rate within acceptable levels. At 17 bar IMEP a maximum value of pressure rise rate of 12 bar/CAD was measured while at minimum load 8 bar/CAD. During the sweep the injection pressure was held constant at 1400 bar.

The gross indicated efficiency is shown in Figure 96. A flat value between 46 and 48% is visible in the load range under examination. This number is lower than what was achieved in the Scania D13 and D12 both because of lower thermal and combustion efficiencies. It is thought that lower thermal efficiency is caused by an increase in heat transfer due to the higher area to volume ratio of the D5 as compared to the D12 and D13. The combustion efficiency spans from 90 to 95% between 6 and 17 bar gross IMEP; this parameter is between 4 and 9 percentage points lower as compared to the D12 and D13. Poor combustion efficiency was due to the high levels of CO and HC emissions; see Figure 97. It is believed that the low maximum cycle temperature combined with the long

mixing period (more fuel in the squish area) and controlling of the load by varying the amount of fuel in the first injection contributed to the high levels of CO and HC. The mixing period and maximum cycle temperature as a function of the load are shown in Figure 98. Low combustion temperature was responsible for the low NOx production while the use of a good injection system combined with low fuel-air mixture stratification (because of the long ignition delay) resulted in very low soot in the load range under examination; see Figure 99.

It is believed that these results can be substantially improved to the levels of the two Scania engines by applying the knowledge gained in the latest phases of the gasoline partially premixed combustion concept applied to the two heavy duty engines.

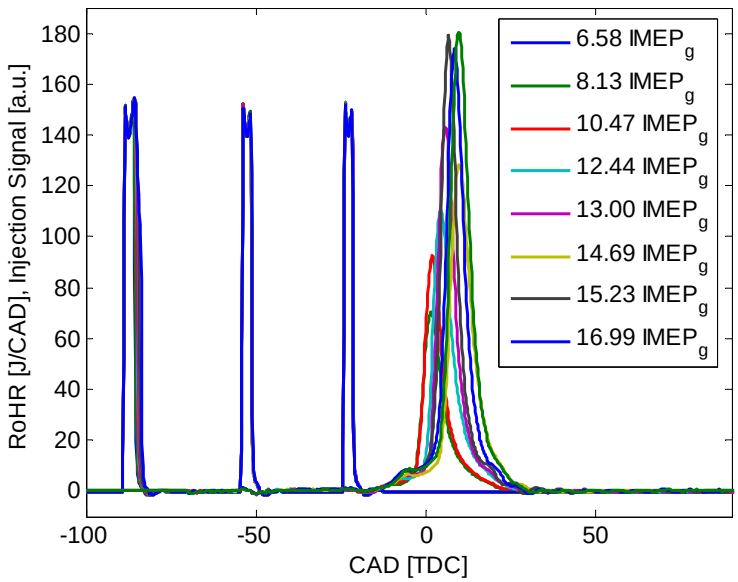


**Figure 95: Rate of heat release and injection pulses as a function of load.**

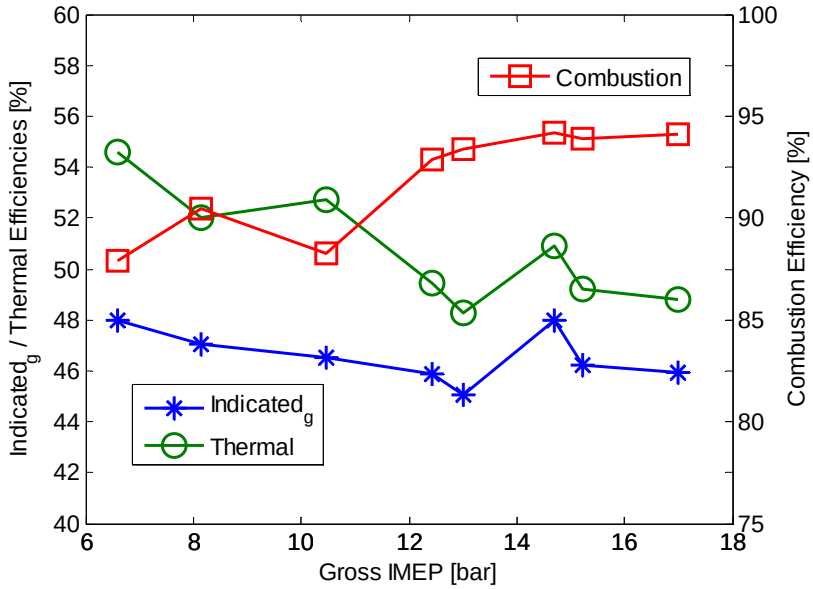


**Figure 96: Combustion, thermal and gross indicated efficiencies as a function of load.**

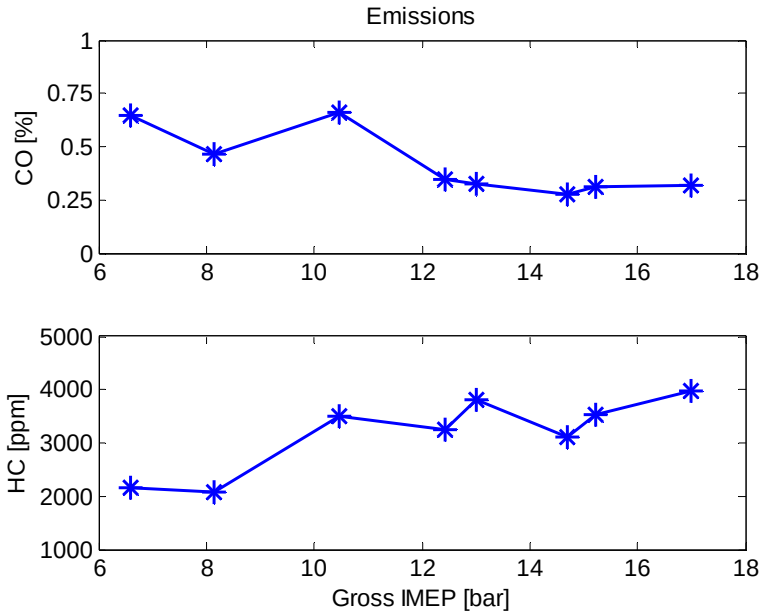


**Figure 97: CO and HC as a function of load.**

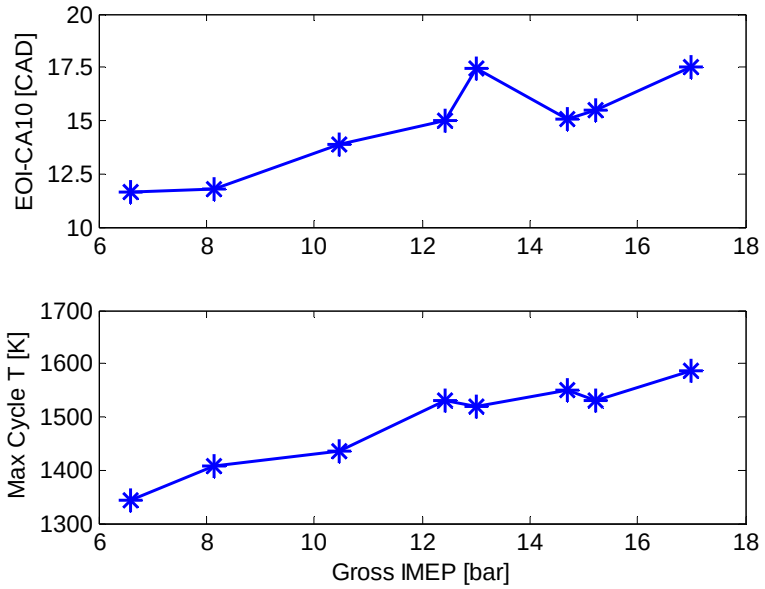


**Figure 98: Mixing period and maximum mass averaged cylinder temperature as a function of load.**

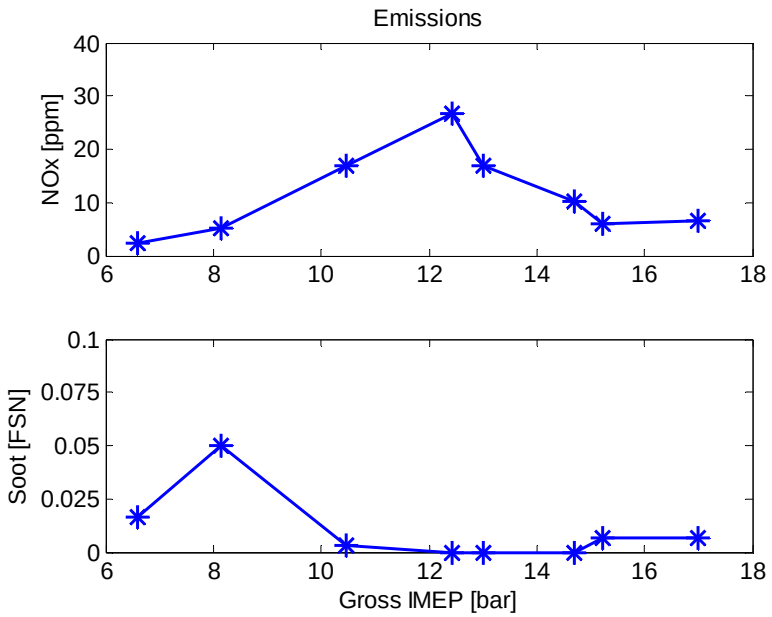


**Figure 99: NOx and soot as a function of load.**

## 7.2 Boosted Conditions, Maximum Pressure Rise Rate Evaluation

When the engine load increases, for conventional combustion conditions, the boost increases proportionally to the IMEP. At constant combustion duration and CA50 this results in a linear increase of the maximum pressure rise rate. In Figure 101 the maximum pressure rise rate (MPRR) has been plotted as a function of the load and it has been assumed that the boost was linearly proportional to the IMEP.

This increase in MPRR with load does not mean that the engine acoustic noise is increasing. For instance when performing a boost sweep while motoring an engine it will result in higher MPRR, see Figure 100, but the acoustic noise does not change.

In order to take into account the fact that the maximum pressure rise rate can not be directly linked to the acoustic noise, GM introduced the Ringing Intensity [111]. In the GM equation the pressure wave amplitude inside the combustion chamber was correlated to the MPRR through a factor β. For the engine used in that paper β was found equal to 0.05. This index has been developed for homogenous mixtures and it works quite well with HCCI type of combustion. In the case of partially premixed combustion the fuel-air mixture distribution is from homogeneous and in addition the stratification level increases with load. This means that if the Ringing Intensity has to be used in gasoline PPC as a measure of the acoustic noise, the factor β has to be mapped as a function of the stratification. This option is obviously not practical. To overcome this problem and have a measure of the acoustic noise at different loads, it has been thought to normalize the maximum pressure rise rate with the absolute inlet pressure. For instance in the case of Figure 100 the normalized pressure rise rate is boost independent; this correlates well with the fact that the acoustic noise does not change when increasing the inlet pressure while motoring the engine. For the measurements carried out in [48] and [49] the absolute and relative maximum pressure rise rates are shown in Figure 102. A threshold value of 8 bar/CAD has been assumed.

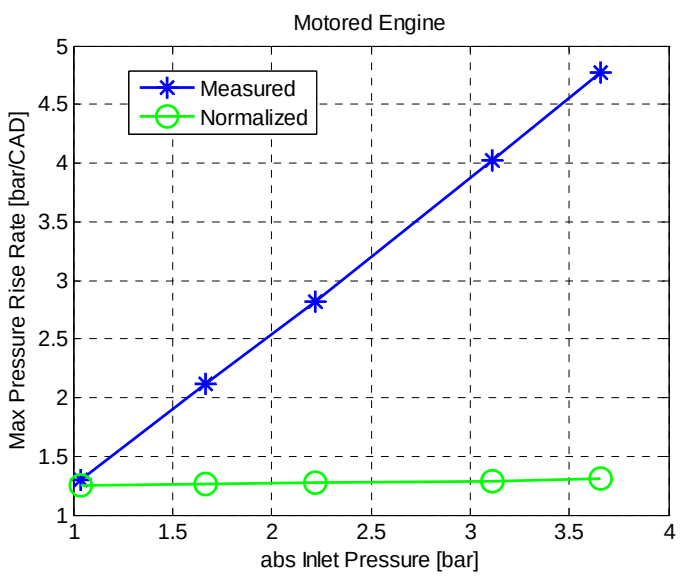


**Figure 100: Normalized and measured maximum pressure rise rate for motored conditions as a function of the inlet pressure.**

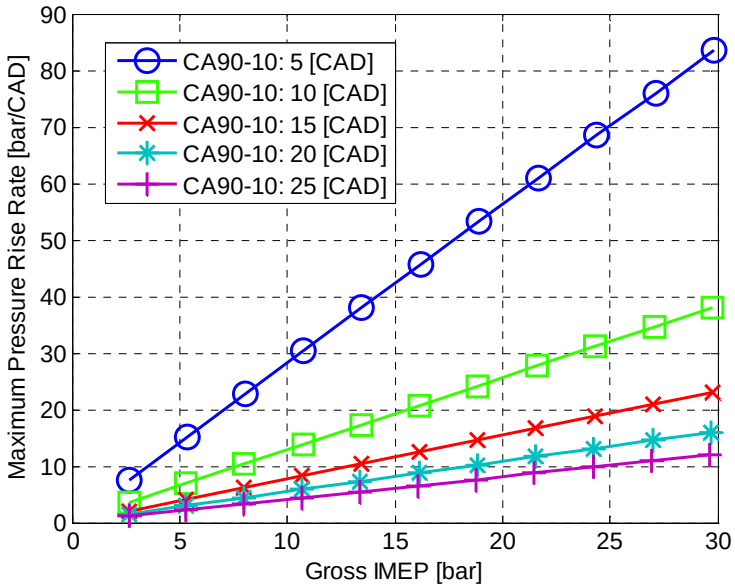


**Figure 101: Maximum pressure rise rate computed using the model described in 3.1.1 as a function of load and combustion duration assuming CA50 at 8 [aTDC].**

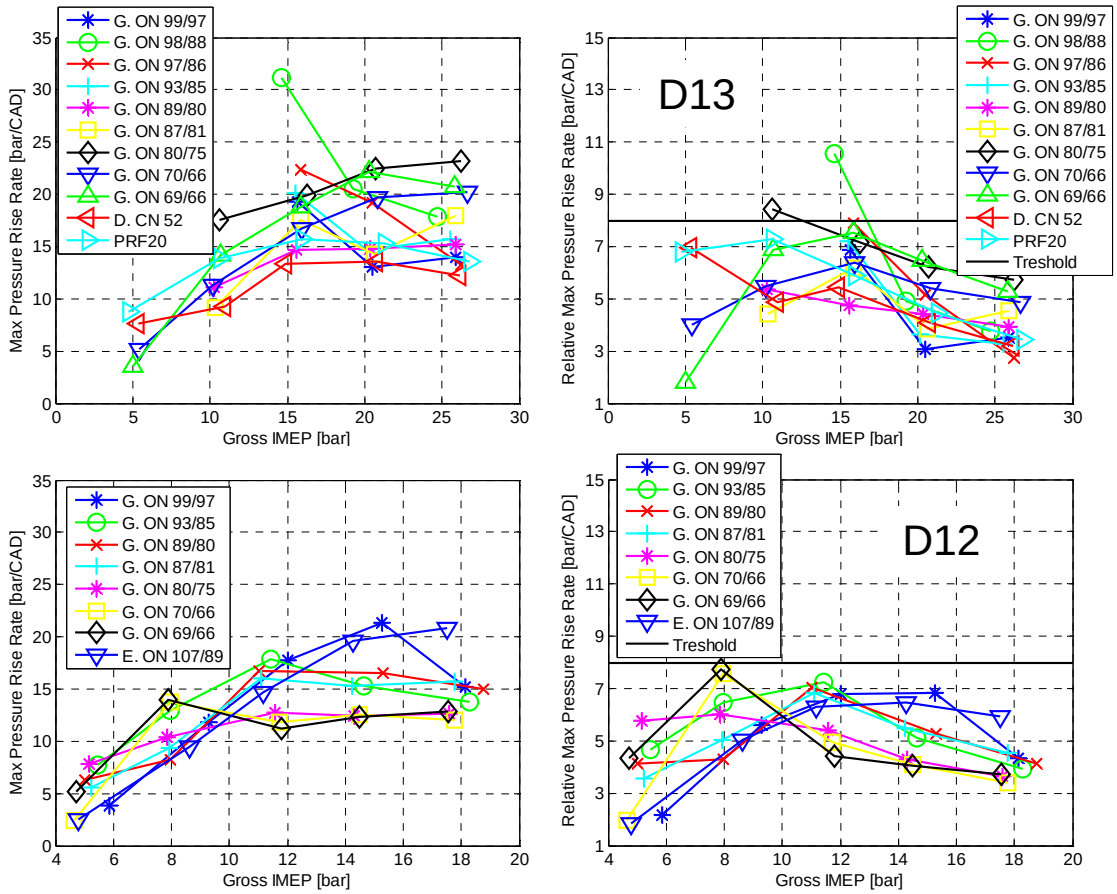


**Figure 102: Normalized (right) and absolute (left) maximum pressure rise rate as a function of load using the Scania D13 and D12.**

## 7.3 Mean Effective Values and Efficiencies Definitions

Figure 103 shows the energy flow in internal combustion engines from the input energy per cycle down to the output work. In order to compare engines of different sizes it is necessary to use normalized quantities. The most common way is to divide the energy under examination with the displacement volume thus resulting in mean effective pressure parameters conventionally measured in [bar]. The definitions of the parameters presented in Figure 103 can be found in Equation 5.

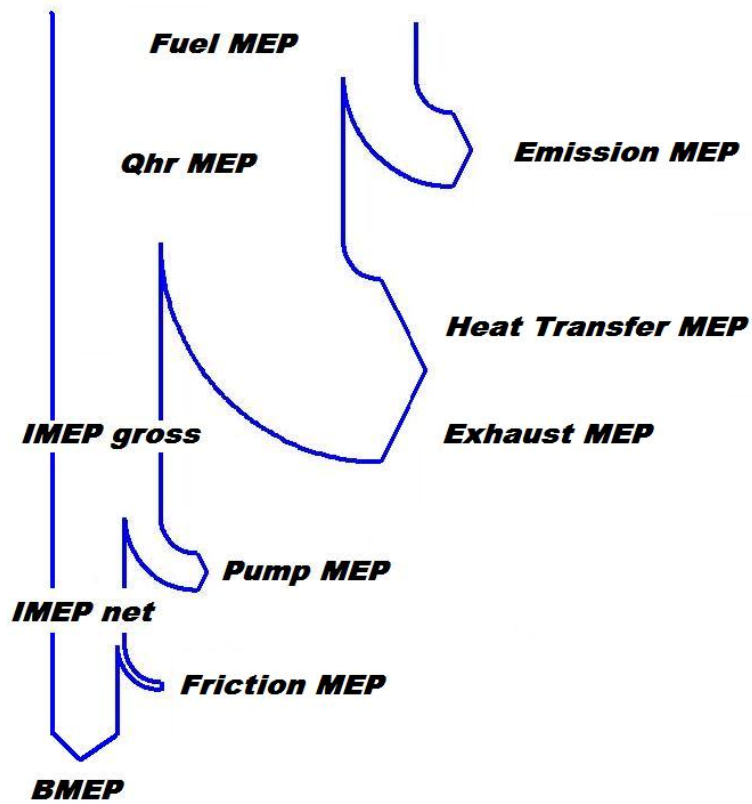


**Figure 103: Sankey diagram of the energy flow in internal combustion engines.**

**Equation 5:**

$$\begin{cases} FuelMEP = \dfrac{m_f \cdot LHV}{V_d}; \quad Q_{hr}MEP = \dfrac{Q_{hr}}{V_d}; \quad IMEP_g = \dfrac{W_g}{V_d}; \quad IMEP_n = \dfrac{W_n}{V_d}; \\ BMEP = \dfrac{4 \cdot \pi \cdot T}{V_d} \end{cases}$$

$$\begin{cases} EmissionMEP = FuelMEP - Q_{hr}MEP; \quad ExhaustMEP = \dfrac{m_{cyl} \cdot c_p \cdot (T_{ex} - T_{ref})}{V_d}; \\ HeatTransferMEP = Q_{hr}MEP - IMEP_g - ExhaustMEP; \\ PumpMEP = IMEP_g - IMEP_n; \quad FrictionMEP = IMEP_n - BMEP \end{cases}$$

To allow a comparison of different load and speed points, the parameters presented in Equation 5 can be normalized with the displacement relative input energy per cycle; see Equation 6.

**Equation 6:**

$$\begin{cases} \eta_{comb} = \dfrac{QhrMEP}{FuelMEP}; \quad \eta_{ther} = \dfrac{QhrMEP - IMEP_g}{FuelMEP}; \quad \eta_g = \dfrac{IMEP_g}{FuelMEP}; \\ \eta_n = \dfrac{IMEP_n}{FuelMEP}; \quad \eta_b = \dfrac{BMEP}{FuelMEP} \end{cases}$$

$$\begin{cases} \eta_{comb\_ineff} = 1 - \dfrac{QhrMEP}{FuelMEP}; \quad \text{RelativeExhaustLoss} = \dfrac{ExhaustMEP}{FuelMEP}; \\ \text{RelativeHeatTransfer} = \dfrac{HeatTransferMEP}{FuelMEP}; \quad \eta_{ge} = \dfrac{PumpMEP}{FuelMEP}; \quad \eta_m = \dfrac{FrictionMEP}{FuelMEP} \end{cases}$$

## 7.4 Minor Projects between 2006 and 2008

During the time span between 2006 and 2010, three main projects were carried out by the author of this thesis:

1. The VIMPA project, 2006-2007.
2. The HCCI fuel rating index project, second half of 2008.
3. Gasoline partially premixed combustion, 2008-2010.

Though the VIMPA project was not extremely relevant to the scientific community it deserves to be mentioned because it enabled me to gain the knowledge and the necessary skills to develop the gasoline partially premixed combustion concept examined in this manuscript. When the VIMPA project was terminated, I started to work on the third project though I have simultaneously spent few months trying to develop a HCCI fuel rating index. This intermediate work deserves to be mentioned as well mainly because it gave me the opportunity to learn how HCCI combustion is affected by the fuel composition. Also this second project was vital for the development of gasoline partially premixed combustion.

### 7.4.1 VIMPA

VIMPA is an acronym that stands for vibrating microengine for power generation and microsystem actuation. The goal of the project was to develop a very small internal combustion engine with the aim to produce electricity and replace the electrochemical batteries. The idea behind the project is the following: the best electrochemical battery (Zn-Ag) has an energy density of roughly 1.5 MJ/kg (see Figure 7) on the other hand a hydrocarbon or organic fuel has an energy density of roughly 45 and 29 MJ/kg respectively. By building a micro internal combustion engine with a minimum overall brake efficiency between 4 and 6 %, it should be in theory possible to replace the electrochemical batteries. In the VIMPA project the reference weight has always been considered the fuel weight and not the engine one; this is valid with the assumption that the fuel weight is much higher than the engine weight.

The VIMPA project was a consortium formed by three universities: Scuola Superiore Sant Anna (SSSA), Berlin Technical University (TUB) and Lund University (LU). SSSA developed the idea and was in charge of manufacturing and designing the engine, TUB was responsible for the design of a piezoelectric injection system and LU had the task to study combustion in the milli domain. The first engine design proposal and working principle are shown in Figure 104. Similar to a two stroke engine, the original VIMPA consisted of two main chambers: the combustion chamber and some sort of crankcase where fuel and air mixes. Reed valves were supposed to be used to exhaust combusted gasses and to intake the fresh fuel-air mixture. It might be noted the absence of a conventional piston-combustion chamber system. Two membranes were supposed to function both as combustion chamber and piston. The volume variation due to the elastic behavior of the membranes was meant to supply the fuel-air mixture with the compression energy to achieve ignition. HCCI was supposed to be the combustion type of this exotic engine and the device was meant to be fueled with ethanol. The advantages of this first design was that the only friction comes from the hysteretic behavior of the membranes then the use of HCCI combustion would have led to relatively high efficiency and low emissions. After more than a year of research, SSSA realized that it is impossible

to build a membrane with a diameter of 40 mm, 20 μm thick that is capable of moving up and down ± 2.43 mm at 100-1000 Hz and in addition to be enough resistant to the thermal and mechanical stresses of the combustion event. The sealing and timing of the reed valves was discovered to be practically unfeasible. In addition the research performed by LU demonstrated that if HCCI combustion has to be used ethanol is not the right fuel. It was found that when HCCI combustion takes place in the milli domain a fuel very easy to vaporize and ignite is preferable. The fuel for the VIMPA engine was switched from ethanol to diethyl ether.

Due to the practical experience gained in studying HCCI combustion in the milli domain, in the middle of 2007 LU proposed a second VIMPA design that is mainly a miniature free piston engine; see Figure 105. At the end of the project (end of 2007), the VIMPA second design was manufactured, coupled to the TUB injection system and tested; see Figure 106. The power shaft was connected to a crank mechanism connected to an electric motor. Because of lack of time, VIMPA could be tested only once and the viability of a self sustaining combustion process (engine not coupled to the electric motor) could not be studied. The only test showed that VIMPA could run at 5800 rpm producing 2.28 bar IMEP with an indicated efficiency of 8.84 %.

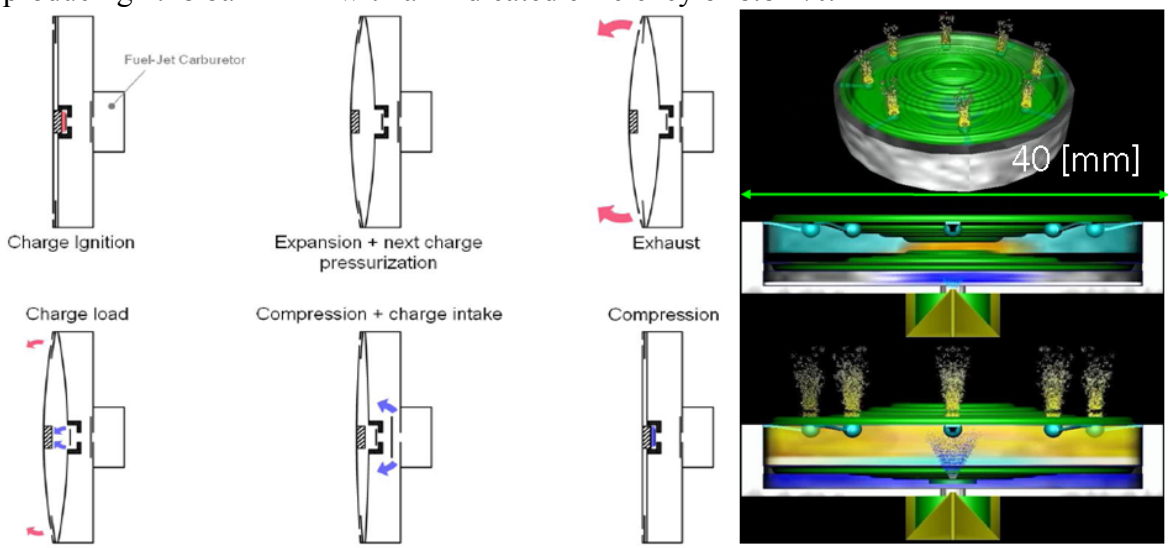


**Figure 104: VIMPA first design (right) and working principle (left).**

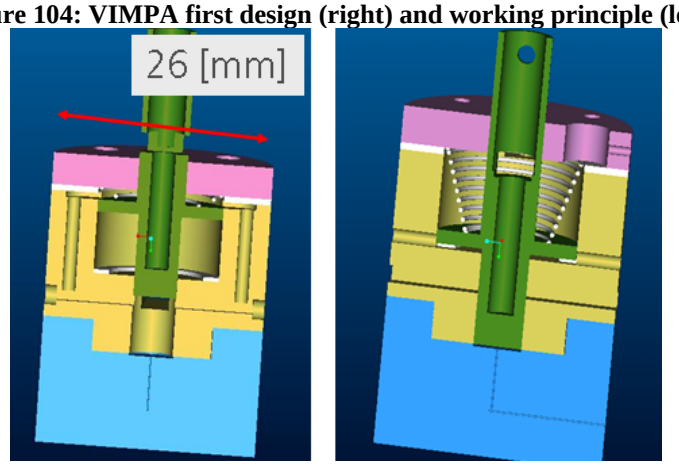


**Figure 105: VIMPA second design.**

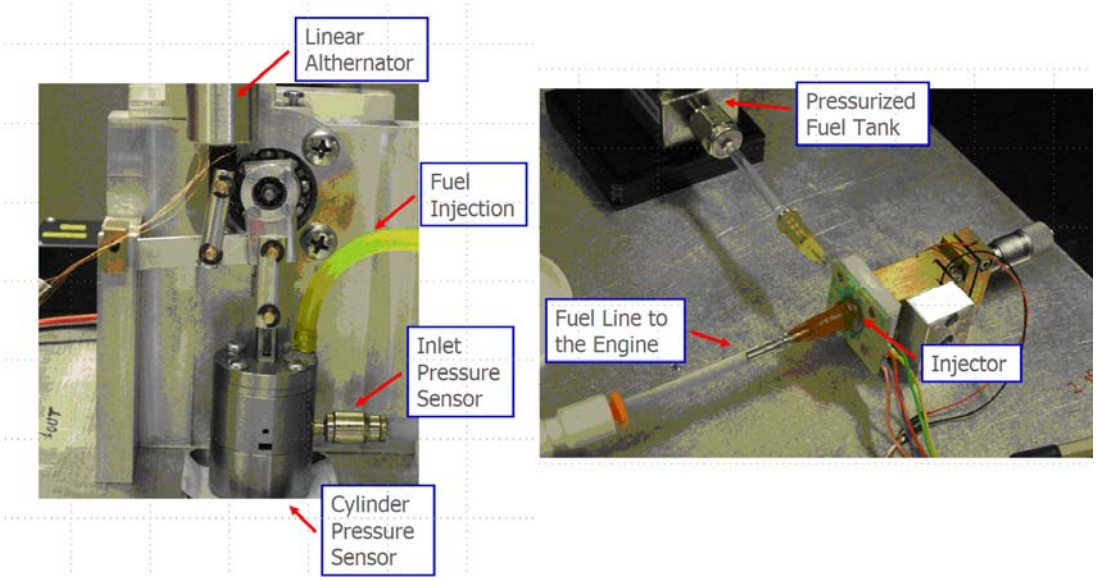


**Figure 106: VIMPA experimental set-up (left) and injection system (right).**

In order to study combustion in the milli domain it was decided that the optimum bench test would have been a small model airplane engine. The two years of research were carried out using an OS 25 with a displacement volume of 4.11 $cm^3$. The research was organized as follow:

1. Glow plug assisted combustion was studied using both a metal engine and optical diagnostic. The fuel used was both ethanol and the standard fuel for this type of engines that is a blend of methanol, nitromethane and lubricant oil.
2. Studies of HCCI combustion using both a metal and an optical engine. The fuel used was diethyl ether.
3. Understanding the influence of combustion chamber, wall temperature and compression ratio on HCCI combustion in the milli domain.

In [98] the main target was to understand how efficient a glow plug engine is. Experiments were carried out between 15000 and 17000 rpm at wide opened throttle conditions. The IMEP was in the range of 2-3 bar while the efficiency was spanning between 5.5 and 8.5%. Optical diagnostic was performed in [99] at 9600 and 13400 rpm. It was found that reactions start in the hotter part of the combustion chamber (i.e. near the glow plug) and combustion proceeds apparently as in SI combustion. The analysis of the speed of this propagating reaction front was found to be in the range of 86 – 104 m/s. This finding underlined that in a glow plug engine combustion is characterized by a propagating autoignition mechanism rather than flame propagation. The next step was to transform this small glow plug engine in a HCCI engine using like fuel ethanol. The use of ethanol was abandoned after few weeks of research because it needs more than 473 K inlet temperature and then when combustion starts it is too violent to a point that after roughly 500 cycles the connecting rod breaks into pieces! It was decided to switch ethanol with diethyl ether since ambient temperature is enough to achieve HCCI combustion when the effective compression ration is ca 7:1. In [100] a speed sweep was performed between 7500 and 17500 rpm at WOT conditions. The IMEP was between 1 and 2.5 bar while the efficiency between 2 and 5 %. It was found that such low indicated efficiency was due to the combustion efficiency that was in the range of 20%, on the

other hand the thermal efficiency was roughly 25%. In this paper optical diagnostic was performed on this small HCCI engine at 6500 and 13400 rpm. It was found that combustion starts and evolves the same way as in conventional size HCCI engines. The analysis of the boundary layer showed that in the radial direction this parameter is in the range of 2 mm while in glow plug combustion only 0.7 mm.
During these two years of research a model was developed to compute the trapping efficiency and residual gas fraction in two stroke engines [101].

### 7.4.2 Activation Energy[15]

It is well known that knock in SI engine and the autoignition process in HCCI combustion are governed by the same reactions [102] [103]. This means that both RON and MON are relevant in this advanced combustion concept. Unfortunately these two indexes can not describe alone the autoignition quality of gasoline like HCCI fuels [104] because their chemistry is different at different running conditions. In 2005 Kalghatgi proposed the octane index (OI) concept in which RON and MON are linked together through the K value: $OI = (1-K)\cdot RON + K\cdot MON$ [84]. K is a function of the temperature and pressure history during the compression stroke. In order to use this equation Kalghatgi introduced the parameter Tcomp 15 (the temperature at 15 [bar] during the compression stroke) and K was expressed as a function of this temperature. For a given set of running conditions the OI is able to classify how prone a fuel is to autoignite. In 2007 Shibata developed a HCCI fuel rating index based on the normal paraffin (n-P), iso-paraffin (i-P), olefin (O), aromatic (A), and oxygenate (OX) content of a fuel and its MON [105]. The index is expressed according to Equation 7. The coefficients m, a, b, c, d, e and Y are tabulated and are a function of the temperature of the mixture when the pressure in the compression stroke is 15 [bar]. The higher the number, the more resistant this fuel is to ignite.

**Equation 7:** $$HCCI\ Index\,(abs) = m\cdot MON + a\cdot(n-P) + b\cdot(i-P) + c\cdot O + d\cdot A + e\cdot OX + Y$$

The idea behind the work reported in this thesis is to propose a HCCI fuel rating index based on the energy which is supplied to the fuel air mixture from the intake stroke to the start of the combustion. Since the start of the combustion, CA0, is very difficult to determine, CA5 was chosen instead. The energy balance can be written as:

**Equation 8:** $$E_a = (m_a + m_f)\cdot c_p\cdot(T_{IVC} - T_{ref}) + \int_{IVC}^{CA_5} P\cdot dV - Losses$$

The Losses term was modeled according to the Woschni's equation [106] while the temperature at intake valves close, $T_{IVC}$, was expressed as a function of the inlet and exhaust temperature as in [56] minus the temperature drop due to the heat of vaporization of the fuel (the intake temperature was measured prior the injection event). To account for inlet pressure and engine size variations the absolute activation energy, $E_a$, was normalized with the absolute boost pressure and displacement volume: $E_{a\_relative} = \frac{E_a}{V_d\cdot P_{in}}$.

[15] This research has never been published and it is a part of a joint work between Lund University and Chevron. Because public references are missing, more details will be given on this concept and how the experiments were carried out.

By using the standard Scania D12 (Table 4) fuels, in the boiling point range of gasoline, were tested; these included primary reference fuels, PRF, ethanol and real gasoline components supplied by Chevron shown in Table 6. Each fuel was tested without EGR, at 830, 950 and 1070 rpm and for each speed point three boost pressures were used 1, 1.5 and 2 bar absolute. The inlet temperature was adjusted to phase combustion at TDC. The input energy per cycle was roughly 10.5 bar fuel MEP.

It was found that fuels can be divided in three main categories:

1. Those that do not exhibit low temperature heat release under any pressure, temperature and speed combinations. This is the case of pure ethanol and its blends.
2. Those in which the energy released by the low temperature reactions is a function of pressure, temperature and speed. This is the case of gasoline fuels not blended with ethanol with an octane number roughly above 70-80 depending on the n-paraffin content.
3. Those in which the percentage of the total energy released by the low temperature reactions is independent on pressure, temperature, and speed. This is the case of fuels with an octane number below 80 containing lot of n-paraffin.

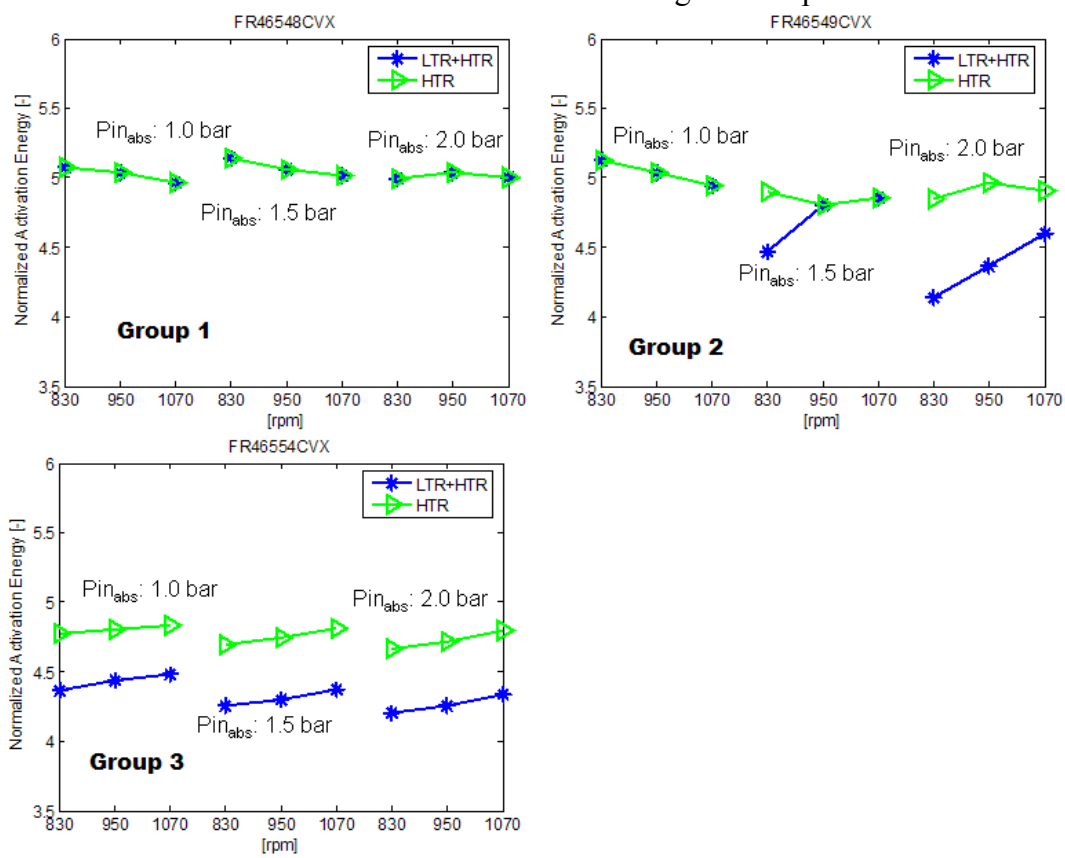


**Figure 107: Normalized activation energy for the three fuel groups. FR46548CVX: RON 98 – MON 88, 20% of ethanol. FR46549CVX: RON 99 – MON 97, FR46554CVX: RON 69 – MON 66.**

Figure 107 shows the normalized activation energy of practical gasoline fuels for each of the above mentioned categories. In Figure 107 the relative activation energy was computed both using CA5 belonging to the whole heat release (LTR+HTR) and CA5

calculated considering only the high temperature heat release (HTR). If the normalized activation energy is calculated excluding the low temperature reaction activities, for all the fuels this parameter is roughly independent on speed, pressure and temperature. On the other hand if the LTR are considered when CA5 is calculated, the normalized activation energy becomes fuel composition dependent for fuels with an ON between 100 and 70/80 and not containing ethanol.

Because the current trend is increase the power density of the engine using lot of boost, I believe that the activation energy must be calculated using CA5 belonging to the whole heat release. Unfortunately with the concept proposed so far it is not possible to use a single value of normalized activation energy for fuels belonging to the second group and thus to create an absolute HCCI fuel rating index. More full time work is needed in order to overcome this issue and thus create an absolute HCCI index based on the activation energy concept. One thing is for sure, RON and MON are not sufficient to describe the ignition properties of fuels in HCCI combustion and this is demonstrated in Figure 108; two fuels with the same RON/MON, but different chemical composition, need to be supplied with a different amount of energy in order to achieve autoignition.

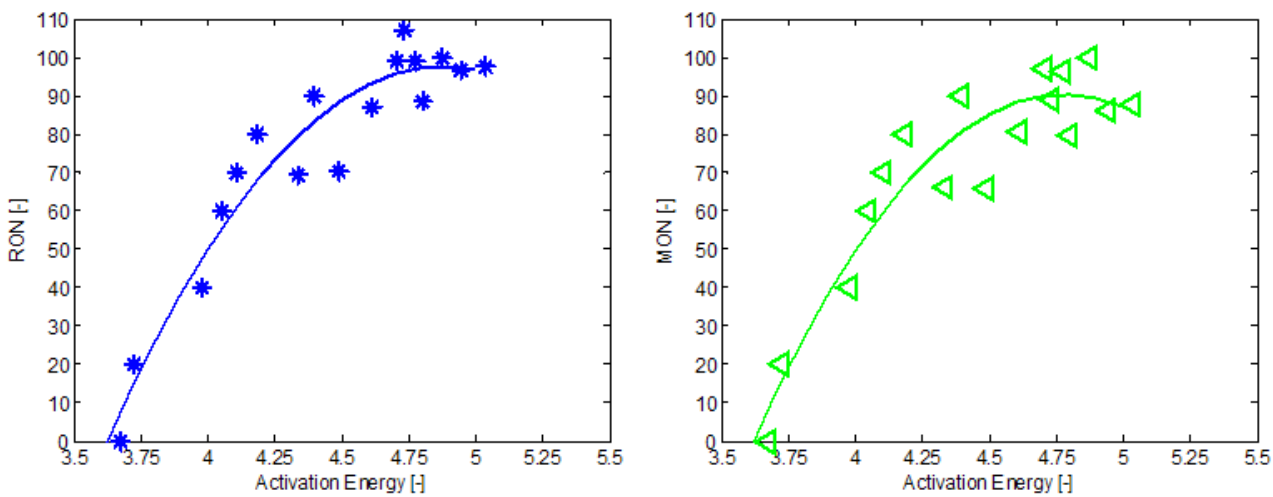


**Figure 108: RON and MON as a function of the averaged normalized activation energy for different type of fuels.**