

COMBUSTION SYSTEM DEVELOPMENT FOR THE NEXT GENERATION HD GAS ENGINES

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Combustion System Development for the Next Generation HD Gas Engines

COSTGAS

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Foreword

COSTGAS is a project in Heavy Duty gas engines in which the combustion system for the next generation gas engines is to be developed. The goal of the project is to increase the efficiency of the current gas engine platform by 10% and increase the torque by 20%. This is done while observing the boundary conditions of the current Euro VI emissions regulations.

The report has been produced by AVL Powertrain Scandinavia, Scania CV and the Royal Institute of Technology. The authors are Ludvig Adlercreutz (AVL).

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Sammanfattning

COSTGAS är ett projekt inom tung motorteknik för CNG och har som mål att förbättra de nuvarande gasmotorerna. I utvecklingsmålet ligger en förbättring av momentutvecklingen ur motorn med 20 procent samtidigt som verkningsgraden ökar med 10 procent.

Detta ska ske samtidigt som Euro VI emissionskrav bibehålls under dessa nya prestandamål. Den här rapporten kommer att täcka bakgrunden och provningen av olika kolvdesigner. Den kommer även att täcka in tidigare utveckling inom förbränningssystem samt delar av flödesmekaniken i det samma.

I projektet COSTGAS ligger fokus på utveckling av förbränningssystemet varför variationer av detta har undersökts på en forskningsmotor. Provmetoden som har använts har varit att utföra motorprover med varianter av kolvarna med bäst potential att nå dom uppsatta målen. Dessa prover har körts i en encylindrig forskningsmotor för att isolera egenskaper och utvärdera resultaten.

Målet har varit att genom variationer i förhållandet squish-to-bore nyttja den faktor som skapar en optimerad gasblandning för denna typ av swirl-assisterad motor. Den ökade turbulensen och hastigheten på strömningen ska också skapa bättre tålighet för EGR (Exhaust Gas Recirculation) för att reducera knack samt minska NO_x (Nitrous Oxides) och termisk last. Detta förbränningskoncept går också att använda för magerförbränning för att på så vis öka verkningsgraden ytterligare.

Summary

COSTGAS is a project within heavy duty CNG engine technology which is aimed to improve the current CNG engine. The goal of the project was to improve the torque output by 20% and at the same time increase the efficiency of the engine by 10% while observing boundary conditions set by the Euro VI emissions regulations.

This report covers the background and testing of different piston designs. This report will also cover the previous advancements in the field of combustion systems for CNG engines together with the theory involved in the flow mechanics in the combustion chamber.

In project COSTGAS the focus is on the development of the combustion system why variations of this have been tested on a research engine. The proposed method of investigation is to run variations of the pistons in order to conclude the greatest achievable potential to reach the goals mentioned above. By running the tests on a single cylinder research engine the effects of the design can be isolated and evaluated. Here the variations in squish-to-bore ratio in the piston will optimise the mixing of gas for swirl assisted pistons in a spark ignited engine. The added turbulence and flow velocity is also intended to create better EGR compatibility in order to mitigate knock, reduce NO_x and the thermal load in stoichiometric operations. This combustion concept can also be used for lean burn, enabling further efficiency gains.

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1 Introduction

COSTGAS is aimed to improve the torque output and efficiency of a HD CNG engine while maintaining the current emissions levels, Euro VI.

In the project COSTGAS, as part of the further development of the CNG engine, an initial literature study was performed to cover the background of the project. This development was intended to aid in making the CNG engine an attractive and viable option to diesel engines and thereby reducing the vehicles carbon footprint. The former generation concepts of CNG engines, i.e. lean gas combustion, have been available since the 90s and even then, it was promoted much in the same way they are today [1]. They claimed to deliver diesel-like performance but with much lower emissions levels, and virtually no particulate emissions. With time, more stringent emissions regulations have however affected the performance of all truck engines on the market, both CNG and diesel engines. For diesel engines, the reduced environmental impact has been shown in a real driving emissions test performed in 2016, comparing an old and a new, modern truck. From this comparison, the results showed reduced fuel consumption by 25% (8-9% in engine optimisation) and NO_x emissions by 95% over the last 24 years [2]. This improvement was driven by newer emissions regulations but would have decreased the margin CNG engines had to its benefit. This may be one factor why the CNG engine has not increased its market shares. The development in stringency for the Euro emissions regulations can be seen in Figure 1-1. The Euro V and VI emissions classes are closer explained in chapter 4.6.

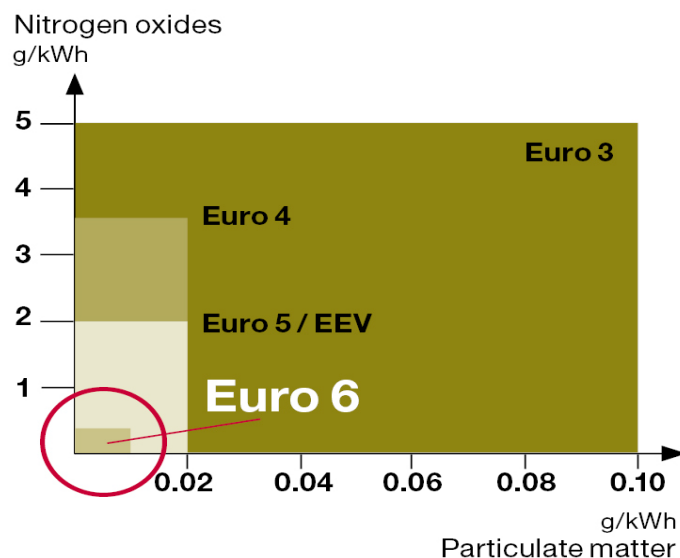


Figure 1-1, Evolution of Euro III-VI emissions regulation [3]

2 Project objectives

In this project, the main focus will be to investigate how to increase the torque output and fuel efficiency from a CNG engine in order to improve the benefits of choosing such an engine. One contributing factor linked to both these parameters is the in-cylinder flow which affects the charge mixing and combustion. In order to address this flow, the intake ports and piston shape will be further investigated in order to create more beneficial combustion conditions. These parameters are investigated in the background study, with a main focus on the piston shape.

3 Limitations

The scope of this project is to look at CNG combustion in a heavy duty single cylinder engine. A variation of the composition of the gas will not be investigated, nor will the turbo system. Since the experimental research engine is externally supercharged, the boost and exhaust backpressure will be adjusted according to existing data.

4 Background

Covering the background to CNG engines was a central part in the initial work, as well as choosing the research questions for this project.

CNG engines have traditionally been converted from diesel truck engines as this reduces development costs significantly and made these engines a viable option to introduce to the market. The main benefit for both the customer and manufacturer were the typically lower fuel price of CNG compared to diesel and also the lower cost of the exhaust aftertreatment system, EATS. These factors are still valid today, especially the EATS compared to diesel where the systems of today require oxidation catalyst, diesel particle filters and even dual NO_x catalysts and running with AdBlue [3]. All the equipment comes costly for the customer, both in total vehicle cost and in maintenance and operation. Comparing fuel prices, Figure 4-1, it is clear that the cost of CNG has been stable for more than 5 years and overall has been cheaper than diesel.

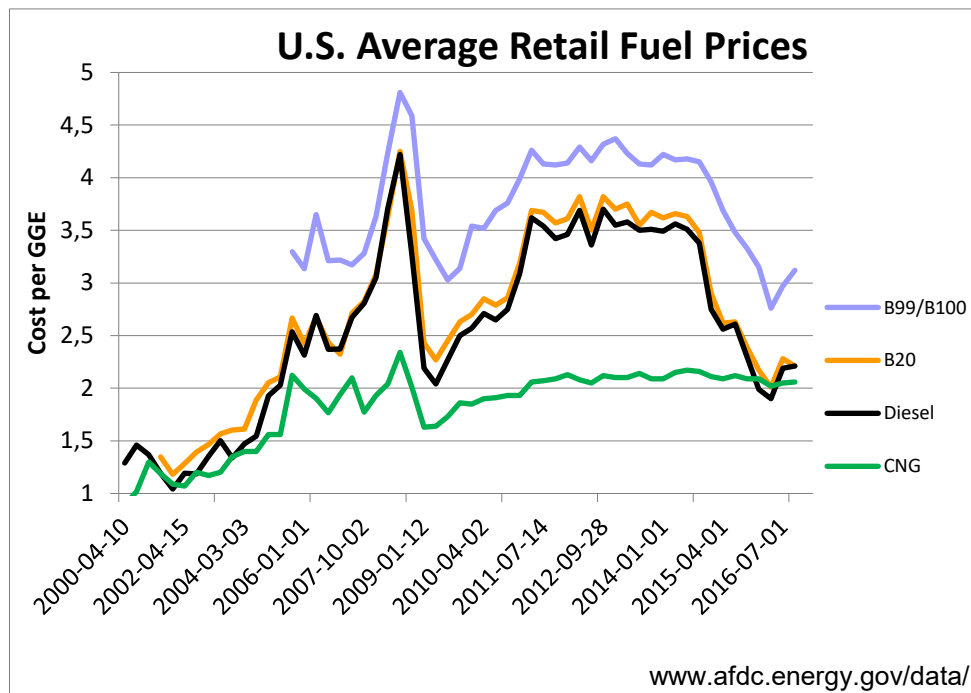


Figure 4-1, Comparison of fuel prices for biodiesels, diesel and CNG in the US market in gallon gasoline equivalent, GGE [4]

So even though there are large financial benefits of running on CNG for the customer it has not yet become the viable alternative it was first believed to be. Improving the efficiency and torque output from the engine would improve these chances and is the main investigative focus of this report.

As part of increasing the efficiency of a CNG engine the literature survey has identified the large-scale flow structures in the combustion chamber as a key feature to improve the flame propagation and total combustion. These flow structures can be divided into two major types: swirl and tumble which are describe in the following chapters 4.1.1 and 4.1.2.

4.1 FLOW STRUCTURES

Through the opening intake valve and intake motion of air into the combustion chamber the energy in the air motion can be used to promote air/fuel mixture and flame propagation. By inducing these large scale motions, the kinetic energy can be preserved during the compression stroke. The intake flow motion is defined along the axis of the combustion chamber and the motions are called swirl and tumble, described in Figure 4-2 and in section 4.1.1 and 4.1.2. These flow structures enter the combustion chamber at different velocities, which are mainly turbulent velocities with a high Reynolds number. This creates different size eddies which range from about a millimetre in size to about the size of the bore. As the piston moves up on its compression stroke the available space against the cylinder head grows smaller. This breaks down the eddies into smaller, more energetic, fractions. By maintaining the flow energy and breaking the flow structures down by fTDC the turbulence would be much higher than for a system with decayed turbulence from the inlet jet. This promotes higher flame propagation and more rapid combustion, which can be used to counteract the slower combustion which occurs when adding EGR [5].

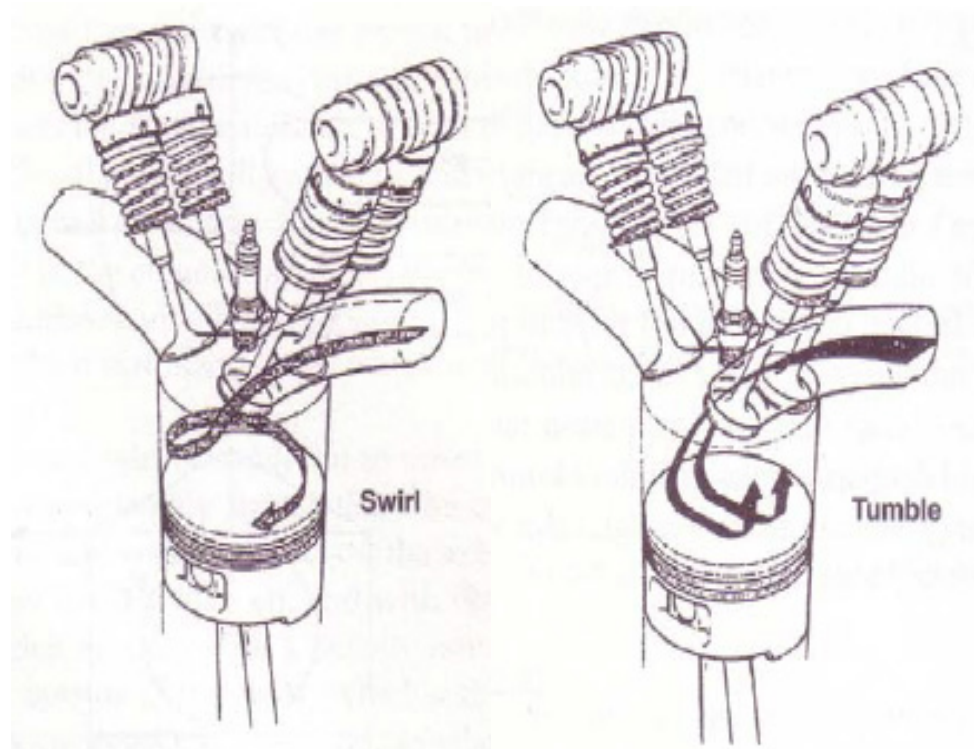


Figure 4-2, Schematic of swirl and tumble [5]

Turbulence is usually defined as the fluctuation ($\bar{u}'$) about the mean flow velocity [6]. TKE, Turbulent Kinetic Energy, is used as a quantification of the kinetic energy per unit mass of the turbulence in the flow. TKE has the unit $J/kg = m^2/s^2$ [7] and is represented mathematically as $TKE = 1/2(\overline{u'^2} + \overline{v'^2} + \overline{w'^2})$ for the fluctuation around all three axes. Söder reports in his doctoral thesis that there is no clear

definition of the in-cylinder turbulence as this value is highly dependent of the averaging procedure for the mean flow [6].

4.1.1 Swirl

As mentioned in the introduction, HD CNG engines have been developed from diesel engines which uses swirl supported combustion systems, though it should be pointed out that there are quiescent diesel combustion systems available as well [1, 8]. The swirl is used in diesel engines to promote post oxidation of soot [9] and can in the CNG engine be used to promote flame propagation if converted to turbulence created around TDC. However, the turbulence created in a swirl system has been reported, in the available investigations, to be limited to be created before firing TDC, fTDC, for CNG combustion systems [8].

The swirl motion is defined as the axial movement around the vertical axis in the combustion chamber. If the combustion chamber is of pancake type, with a flat cylinder head and piston, the swirl will continue to move axially until all kinetic energy is lost. This phenomenon is due to the conservation of the swirl in the chamber as nothing has caused it to break down into smaller vortices. Should the piston be of open bowl type, then the squish between the piston and cylinder head will cause the swirl to break up as the charge is forced down into the bowl. Squish is more thoroughly described in section 4.1.3. This turbulence break-down usually occurs too early before TDC for the combustion to be fully effective [10], hence the limitations of the swirl combustion system. Another drawback with swirl is the increase in heat transfer to the cylinder walls, which was narrated by Söder in 2013 [6]. Swirl is quantified by its swirl number, SN, which is defined as the rotational speed of the swirl flow normalised by the rotational speed of the engine, see equation 1 [11]. The angular solid-body rotational speed of the swirl is normally measured in steady state conditions with measured air flow through the cylinder head. A paddle wheel inside the replaced cylinder liner is spun up by the swirl flow and gives thereby the rotational speed of the swirl flow. The air flow through the cylinder head is the equivalent for the rotational speed of the engine and can be chosen accordingly. It was also found that the swirl flow scales closely to the rotational speed of the engine [11, 5], why the definition of swirl was adapted.

$$SN = \frac{\omega_{Swirl}}{\omega_{Engine}} \quad (1)$$

4.1.2 Tumble

Tumble is the axial motion around the axis perpendicular to the swirl axis [11]. This is promoted through the design of the intake ports and intake valves, but also of the shape of the combustion chamber in the cylinder head, i.e. penthouse combustion chamber and similar designs. Tumble in a penthouse combustion chamber has been proven in light duty engines to be very effective and could help to shift the promotion of the decay of turbulence to after fTDC. This is advantageous when increasing the compression ratio and still achieving 50% mass fuel burnt, MFB50, around maximum brake torque, MBT [8]. A higher tumble also decreases the cycle-to-cycle variations, as Söder reports from his literature survey. He pointed at two reports by Vermorel et al. [12] and Fogelman et al. [13] which

both showed these findings. Wheeler et al. [14] showed that increasing the tumble would provide relatively high EGR dilution tolerance without the use of a high energy ignition system. It was also shown that with the increased tumble it was possible to increase the compression ratio and add EGR while maintaining the same combustion phasing.

Tumble is, as opposed to swirl, not accelerated as the piston compresses the charge but broken down from its large scale flow motion into multiple, smaller, vortices and creates turbulence. This is due to the height in the combustion chamber where at TDC there is no room for the large scale to survive. For swirl, in order to create turbulence, the flow motion must be altered in the piston for open bowl type combustion system with a flat cylinder head.

Tumble is defined in the same way as swirl, with the rotational speed of the axial flow normalised by the rotational speed of the engine, see equation 2. As for the swirl, tumble number is also measured with laboratory equipment.

$$TN = \frac{\omega_{Tumble}}{\omega_{Engine}} \quad (2)$$

This measurement of swirl and tumble is not fully exact but is considered close enough under the circumstances the option of measuring inside a motoring engine is very difficult and also expensive.

4.1.3 Squish

Squish is an effect that occurs when the piston reaches Top Dead Center, TDC, and is the compression of the gas charge between the piston crown and cylinder head for a bowl-in-piston layout. The low clearance around the piston rim forces the charge down into the piston. Squish is not defined as one of the large-scale flow structures like swirl and tumble but is an important phenomenon that aids in advancing the swirl motion at TDC. Through interaction with the swirl it can convert the large scale gas velocities into small scale turbulent kinetic energy. In 1986 Arcoumanis et al. [15] in [11] showed through experiments on a research diesel engine that in an idealised situation the squish motion can increase the rotational kinetic energy of the charge, making it spin faster. This increase was by as much as the square of the cylinder-to-bowl ratio and is an important aspect of the interaction of squish and swirl.

Squish in a swirl-supported combustion system increases the swirl motion, as mentioned in the previous segment, this effect is called spin-up. In contrast for the tumble supported combustion system, the squish does not increase the gas motion velocity. Apart from the spin-up of the swirl motion, the squish interacts in the bowl of the piston and depending on the amount of swirl the charge motion behaves differently. The interaction is shown Figure 4-3, where the flow structure is pointed out with the white arrows. In the first case with a quiescent combustion system, the squish motion moves along the flat cylinder head and down into the bowl along the centre line of the piston. This results in the flow in the bowl to move in the direction of the squish flow. As swirl is increased this interaction changes the way the squish flow enters the piston bowl. As the swirl reaches $R_s=2.5$ the squish is forced down into the bowl instead of following the cylinder

head. This changes the flow structure in the piston and direction of the motion in the bowl.

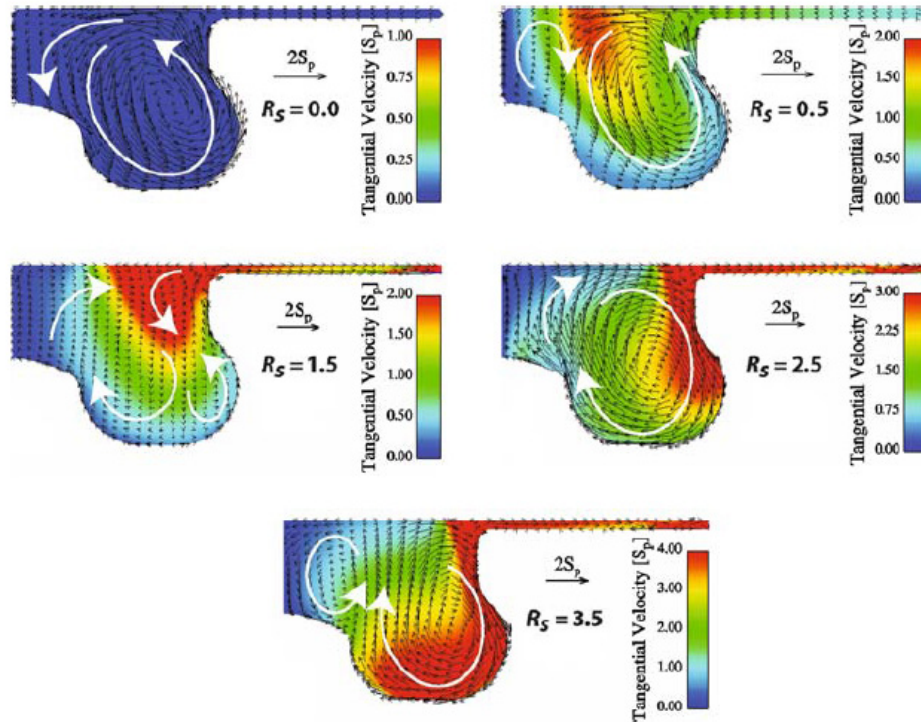


Figure 4-3, Swirl and squish interaction for different levels of swirl [11]

A factor which is affected by in-cylinder gas motion is the convective heat transfer that occurs due to shear stress of the turbulent flow motion [6]. This increases the temperature of the charge which reduces the efficiency and in worst case scenarios be a source of auto-ignition of the fuel. Simulations performed by Cummins on a re-entrant type piston [16] showed that the rim of the re-entrant type piston, which will be further presented in chapter 4.3, is exerted to a higher thermal stress which is needed to be accounted for in the design of the piston. Apart from a potential source of hot spots on the piston, the increase in temperature may also cause material failure in the piston. Figure 4-4 shows this stress as the temperature increase in the piston material. The report showed that this increased thermal stress could be handled through increased piston cooling by a larger cooling cavity in the piston. This gallery was expanded closer towards the rim in order for the oil to be able to cool it.

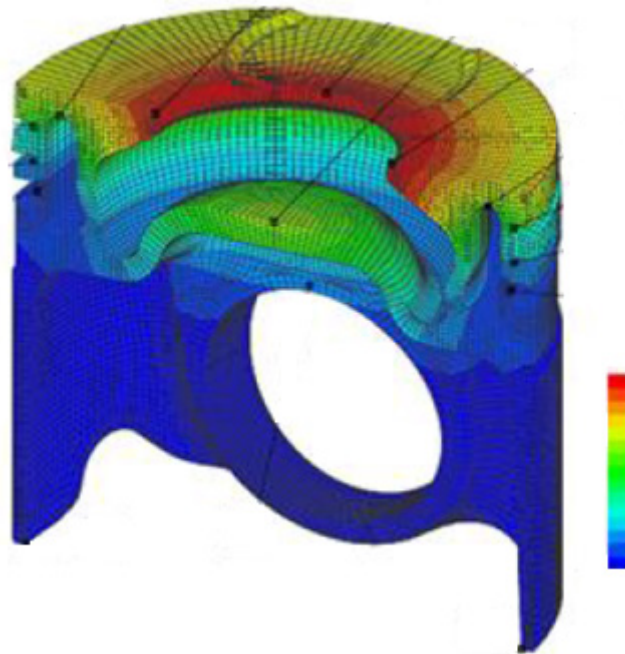


Figure 4-4, Temperature distribution at high squish velocities for re-entrant piston. Red is hotter temperature and scales down to blue according to the scale in the picture [16]

4.2 AUTO-IGNITION AND KNOCK

Auto-ignition, knock, together with pre-ignition are different types of uncontrolled combustion of the fuel in a SI-engine that may cause mechanical damage to the engine [17]. Auto-ignition is the combustion that occurs when the temperature of the charge supplies sufficient activation energy to the fuel in order for it to self-ignite before the flame front arrives. Auto-ignition is activated by factors in the combustion chamber such as temperature and pressure, but may be circumvented by fast main combustion, making time a factor as well. Auto-ignition is called knock due to the oscillating eigenfrequencies that occur in this process which causes a metallic pinging sound. The knocking behaviour is normally audible and also visible on the cylinder pressure trace should a pressure transducer been installed. Pre-ignition can occur at local hot spots, at rapid increases of temperature in the combustion chamber or by glowing soot from the previous combustion cycle. Due to the rapid increase in pressure and temperature by these abnormal combustion phenomenon, the engine performance must be reduced due to the safety measures taken to save the engine from mechanical failures. The most common way to avoid knock and pre-ignition is to retard the spark advance which results in lower efficiency in the engine. [17]

4.3 ASPECTS OF RUNNING CNG

4.3.1 Lean Burn

A CNG engine can run under lean burn or stoichiometric operation and the lean burn engines were favoured since the 90s [18] as converted diesel engines used for heavy duty operations [1]. The benefits have already been covered in the report, but as a review Posada [18] covered some more aspects of running the lean burn engine in 2009. The author mentions that emissions control in the exhaust aftertreatment system in a lean burn need an oxi-catalyst to handle the carbon monoxides and hydrocarbons to reach Euro IV and Euro V. This required a closed loop engine management system and tuning of the engine. In 2003 most heavy duty CNG engines ran lean burn combustion and managed to meet the regulations without exhaust aftertreatment, according to Posada [18]. Downsides to the lean burn combustion system include cycle-to-cycle variation which lead to increased HC due to unburned fuel passing through the combustion chamber. The slower flame speed of the diluted charge [19] has already been mentioned, as has the method to increase the speed of flame propagation prior to firing TDC, fTDC

4.3.2 Stoichiometric

For the stoichiometric engine, the three-way catalyst is used to cover the stringent regulations and since the introduction of Euro VI emissions legislation this has been the main approach for gas engines. The drawback of lower efficiency was chosen for the three-way catalysts efficiency of converting NO_x. Other drawbacks of the stoichiometric engine are the higher in-cylinder temperature and a higher tendency to knock which requires lower compression ratio. This also gives a lower brake efficiency [18]. Whelan et al. [20] reported on the risks of catalyst degradation due to methane slippage over the engine. To curb these drawbacks, EGR is added to dilute the charge which brings down the temperature in the combustion chamber, which in turn reduces the NO_x emissions. Furthermore, EGR also reduces the degree of dissociation in the high-temperature burned gases. This increases conversion of chemical energy into useful energy around TDC [17].

4.4 PREVIOUS REPORTING ON PISTON TESTING

During the literature study, the following reports were found. Most of the reports were written for lean burn engines, pre-Euro VI, but reports on combustion concepts that are valid for stoichiometric operation as well.

Lean-burn engines were favoured in Euro V due to their high efficiency, operating at $\lambda > 1.6$, where also the emitted NO_x was reduced by late combustion, limiting the need for NO_x aftertreatment of the exhaust gases. Figure 4-5 shows the effect of air excess ratio on the emissions when going lean. Drawbacks of lean operation include a lower flame speed in the lean flame and increased cycle-to-cycle variations, together with increased HC and CO emissions. The development of an open chamber piston design with the focus of creating turbulence around the spark plug at moment of ignition to promotes faster flame propagation through the end-gas. This approach is also applicable on stoichiometric operation, but will equate to

higher thermal load on the engine and ancillaries. To mitigate this problem, cooled EGR is used to reduce the combustion temperature. This does however reduce the flame speed which is why the results from lean burn engine research is still relevant. The advantage of stoichiometric operation is the possibility to use a three-way catalyst to reduce CO, HC and NO_x emissions given that the engine control system is robust enough. [21, 22] Other drawbacks of the stoichiometric combustion system are the lower efficiency, increased pumping losses at low loads and increased knocking due to increased thermal load. This is however mitigated with EGR, except for the lower overall efficiency which still limits the stoichiometric engine in relation to the lean burn engine and the diesel. [21]

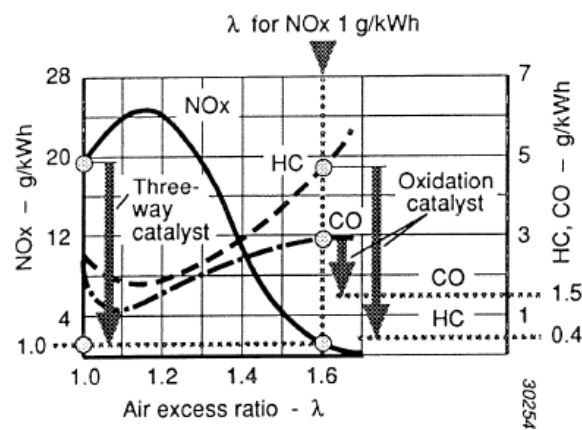


Figure 4-5, Effect of λ on emissions from open chamber SI engine [1]

In 93, Sakurai et al. at Tokyo gas company reported their testing from development of CNG genset (generator and engine). The report presented only one operating point which was set at low load, just under 10 bar BMEP (40 kW) at 1500 rpm [23]. Six pistons were tested during their investigations, Figure 4-6. The bathtub, bowl-in-piston and re-entrant piston all similar theoretical approach, to use squish to accelerate the swirl motion and create turbulence. These represent different levels of squish lengths. The bathtub piston was the baseline piston which was reported to have been used in several gas engines previously. The Nebula and TG piston rely on the same idea to split up the swirl motion in order for it to collide in the centre of the combustion chamber. The turbulence created was then thought to promote fast flame propagation. The TG piston was developed by the authors and named after the company, Tokyo Gas. The chamber in head combustion chamber was thought to enable short flame propagation and deliver a squish motion that would give more turbulence. The shape is likely to leave exhaust gases left in the combustion chamber at each gas exchange, resulting in internal EGR. On the other hand, the exhaust valve can be left open longer without hitting the piston.

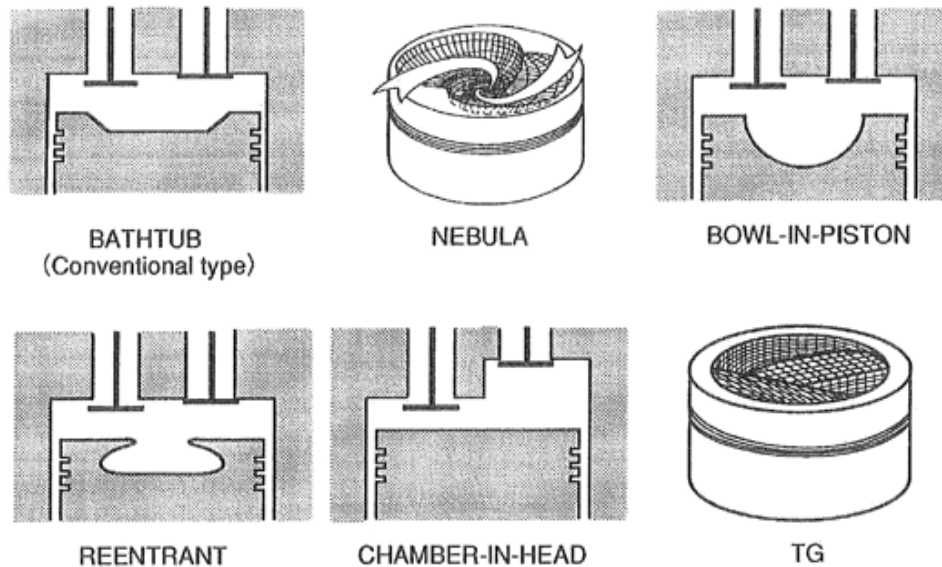


Figure 4-6, Piston shapes presented from investigations by Sakurai et al. [23]

Sakurai et al. also investigated the turbulence intensity and mean velocity in the combustion chamber through hot wire measurements. These measurements showed a large peak in turbulence intensity for the re-entrant piston, compared to the other pistons. It is noteworthy that the presented data of turbulence intensity might have been beneficial to the re-entrant piston due to the placement of the hot-wire probe. However, the results differ substantially from the results from the results reported by Einewall and Johansson [10, 24]. This discrepancy may be related to the testing method used where Einewall and Johansson used Laser Doppler Velocimetry instead of the hot-wire probe. Sakurai et al. concluded their work with stating that the TG, Nebula and re-entrant piston all showed low NO_x levels but that the re-entrant piston was lacking somewhat in combustion stability compared to the other two. This was believed by the authors to be related to the larger turbulence intensity causing delayed ignition and slightly slower flame propagation.

In 1994 Cartellieri et al. at AVL presented in a conference proceeding their development process of the AVL TRIFLOW gas piston. This piston was designed using the early versions of the AVL FIRE CFD simulations program. The design theory was to control the mean flow velocity at the spark plug gap and to promote turbulence in the other parts of the combustion chamber. This theory was believed to avoid quenching of the spark and early flame propagation but to maximise the flame propagation through the turbulent kinetic energy, TKE. The turbulence and higher intensity flow field in the rest of the combustion chamber would promote a fast and complete combustion which would improve the engine efficiency. These design parameters were in line with previous research on flame kernel development performed by others in that period of time [1]. The derived piston shape is presented Figure 4-7, where the left is the early rendition of the TRIFLOW piston used in AVL FIRE. The right was presented in a conference presentation on

gas engines from 2013 and seemingly presents the turbulence level in the piston bowl [25]. This is simulated in later version of AVL FIRE.

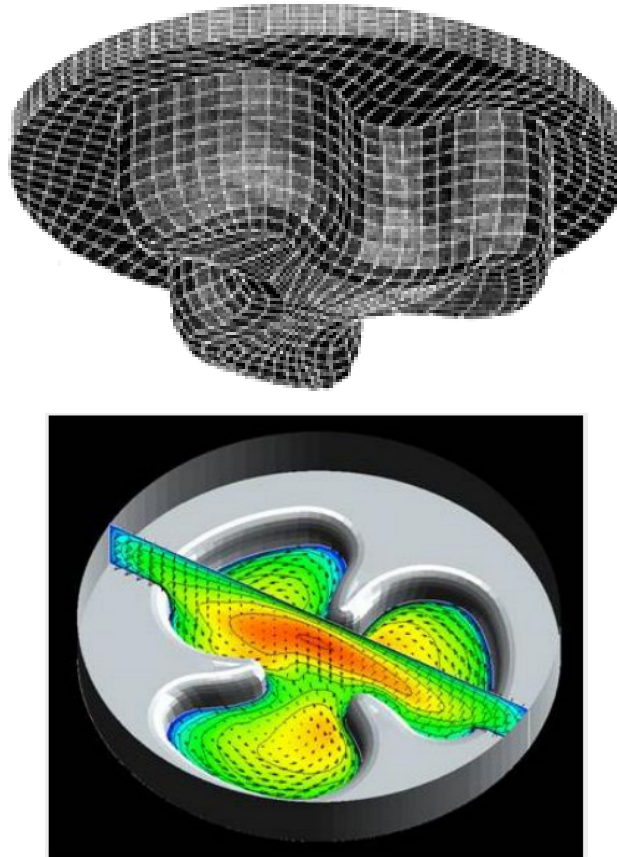


Figure 4-7, AVL TRIFLOW piston. Top: Original iteration by Cartellieri et al [1]. Bottom: Newer iteration, Herdin [25]

The simulations from AVL FIRE presented by Cartellieri et al. show in Figure 4-8 an increase in heat release rate over the conventional piston, which in this case was an off-centre open bowl piston of fairly conventional straight wall design. TRIFLOW 1 is another iteration of the TRIFLOW piston with only two wings similar to the wings in the TRIFLOW piston in Figure 4-7. The rate of heat release for the TRIFLOW piston is higher than for the other pistons, which is in line with the theory on high flow velocity to favour a fast heat release rate. The accumulated heat release in the same figure shows this as well. The simulated mean flow velocity and turbulent kinetic energy is presented in Figure 4-9 and shows a large increase in TKE after TDC which should provide good conditions for the flame propagation upon ignition. These simulation results do however not line up with simulations performed later in 2009 by de Castro Viana et al [26]. In that report they simulated the different effects of swirl levels on the turbulence in the combustion chamber using the TRIFLOW piston. The baseline used in their simulations was a more modern equivalent of the open bowl piston compared to the one in the simulations by Cartellieri et al. This open bowl piston was a centrally aligned wide bowl, in comparison to the off-centre small bowl by Cartellieri et al. The baseline used a swirl number of 2.6 and the swirl for the other simulations

varied the swirl number between 2.2, 2.6 and 3.2. The compression ratio of these test was 14:1. For Cartillieri et al. the numbers were 2.6 in swirl and 12:1 in compression ratio. Despite these similarities in input data, the results are different in Turbulent Kinetic Energy in Figure 4-9 and Figure 4-10. The turbulence peak occurs at different timing around TDC, and for different peak values. The shape of the curves is also different. If the swirl is tilted in the combustion chamber, this would affect the development of the flow motion. Since no data is given for this, it is difficult to know whether this is the reason for the different results.

These simulations were presented for TKE close to the spark plug, for both reports. The discrepancy continues when comparing the mean turbulent energy in Figure 4-11, not only in amplitude, but in timing of the peak value as well. The differences in peak value is peculiar as the in-data values should be similar Figure 4-9 and Figure 4-10 are in the same range, but that the mean value differs in Figure 4-11. This could instead be related to updates in the computational models used, or that the initial input data differed substantially. For instance, the bore was 30 mm larger and the stroke 25 mm longer for the simulations by Castro Viana et al. Both the simulations of TKE between the two reports, Figure 4-9 and Figure 4-10, show different results in the turbulence variation during the last 30 CAD of compression BTDC. This may indicate some errors in the simulations setup done by de Castro Viana et al.

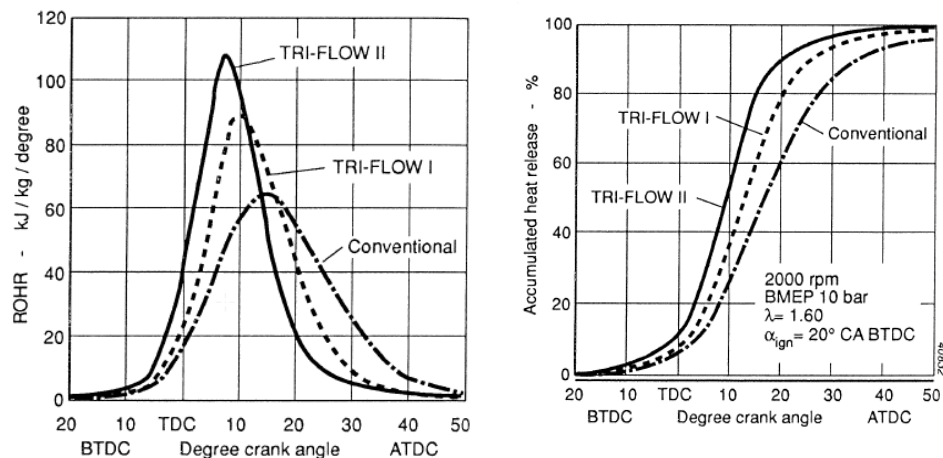


Figure 4-8, Rate of heat release and accumulated heat release from AVL FIRE simulations [1]

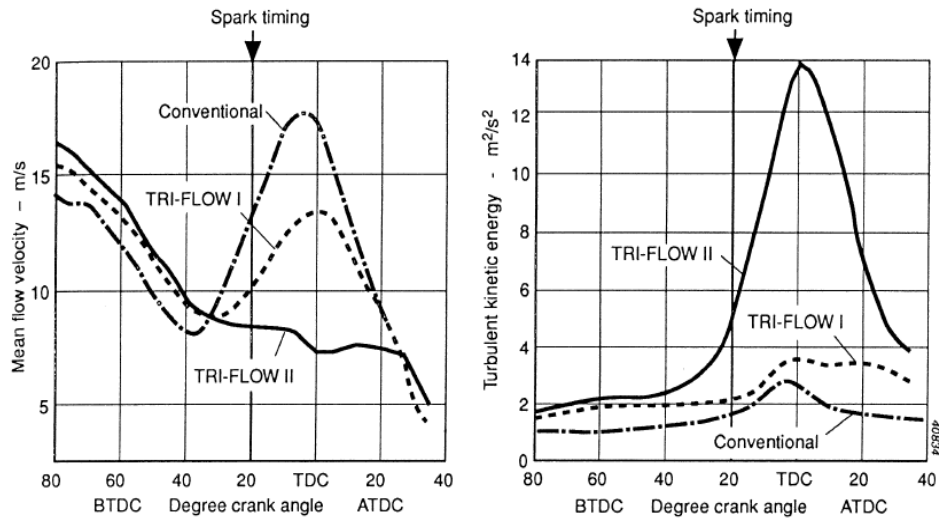


Figure 4-9, Mean flow velocity and TKE in the combustion chamber at spark around fTDC [1]

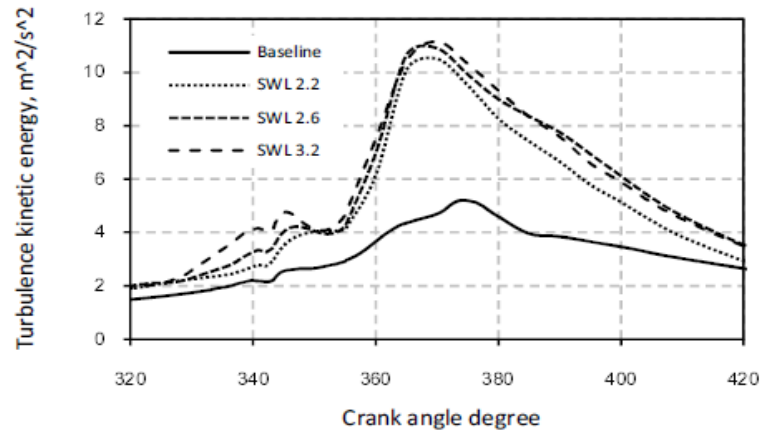


Figure 4-10, TKE simulation results from varying swirl levels in combustion chamber [26]

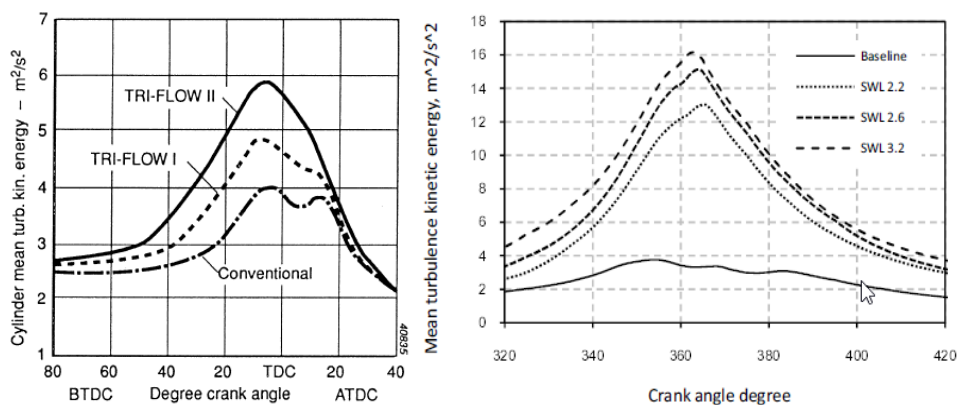


Figure 4-11, Mean TKE from simulations, Left: Cartellieri et al. [1] Right: de Castro Viana et al. [26]

Johansson and Olsson [24] at Lund University tested different piston shapes in 1995 in a single cylinder research engine. This was done in order to investigate cycle to cycle variations and early flame propagation and the main focus was not on the optimisation of the gas engine. The simple piston shapes investigated does however provide a basic understanding on how the pistons perform. The different

pistons presented in the report are shown in Figure 4-13. Of the ten different pistons tested, two were tested with the same shape, but different compression ratios. Square16 and Hemi16 hold the same shape as pistons Square and Hemi, as can be seen in Figure 4-13. For these tests, the turbulence in each combustion chamber was measured with Laser Doppler Velocimetry through a quartz glass in the cylinder head.

In their introduction they report on two different strategies to how the combustion chamber should be designed. The first was that with minimum turbulence and flow velocity the heat losses are minimised and emitted NO_x would be reduced due to slower combustion speed. This would however not work very well in a lean burn engine but perhaps more suitable in a stoichiometric engine. The other strategy was to achieve a fast combustion with low cycle to cycle variations. The lean operation would be advantageous from a thermodynamic point of view due to the higher ratio of specific heats for the expansion stroke as well as being able to run higher compression ratio due to less knock. The high compression ratio lean engine would suffer from increase in HC-emissions though, as flame quenching and partial burn could be expected. They also report that Southwest Research Institute had conducted experiments between combustion systems with fast or moderate burn rates. They concluded that no differences in level of NO_x or in thermal efficiency could be found between the two systems, [27] in [24].

The results from the measurements done by Johansson and Olsson show that the turbulence scaled well with engine speed in their experiments, except for the Hemi piston where the turbulence stayed almost the same for the engine speeds tested. Further, they plotted their turbulence data against the squish ratio and showed good correlation between the two, Figure 4-12. Also shown in Figure 4-12 is the data on the rate of heat release plotted against the turbulence. The authors expected a correlation between these parameters but commented on the two different groups. They reasoned that this was the effect of available flame area due to the combustion volume located at different radii from the spark plug. For both these plots, the turbulence data was taken for the crank angle degrees corresponding to the heat release, 10-90%.

Johansson and Olsson conclude their report saying that their results show that the design of the piston crown greatly affects the flow field in the cylinder. They also state that the timing of the turbulence around TDC is of greater importance than the peak value of the turbulence.

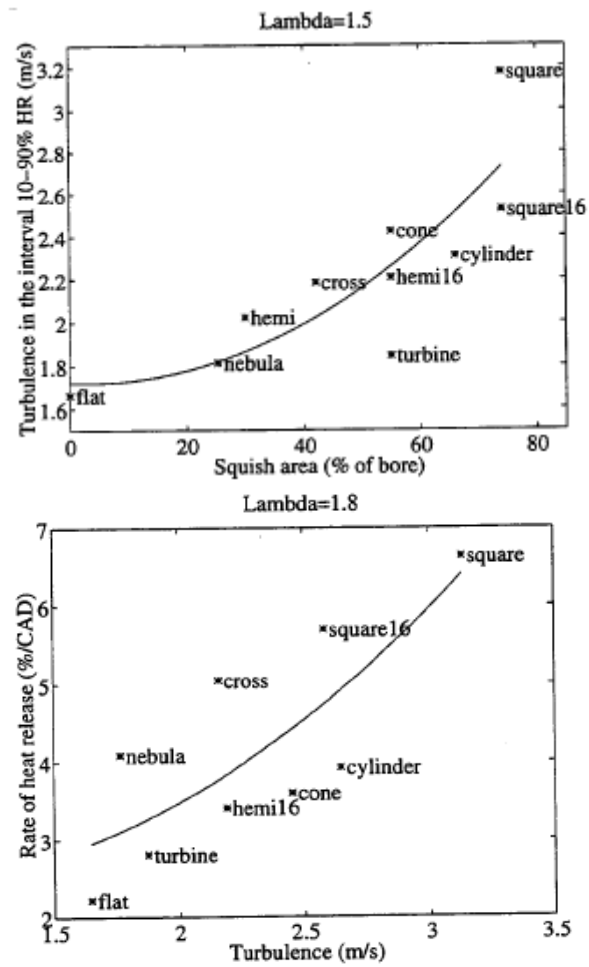


Figure 4-12, Left: Turbulence at 10-90% heat release as a function of squish area.
 Right: Rate of heat release as a function of turbulence [24]

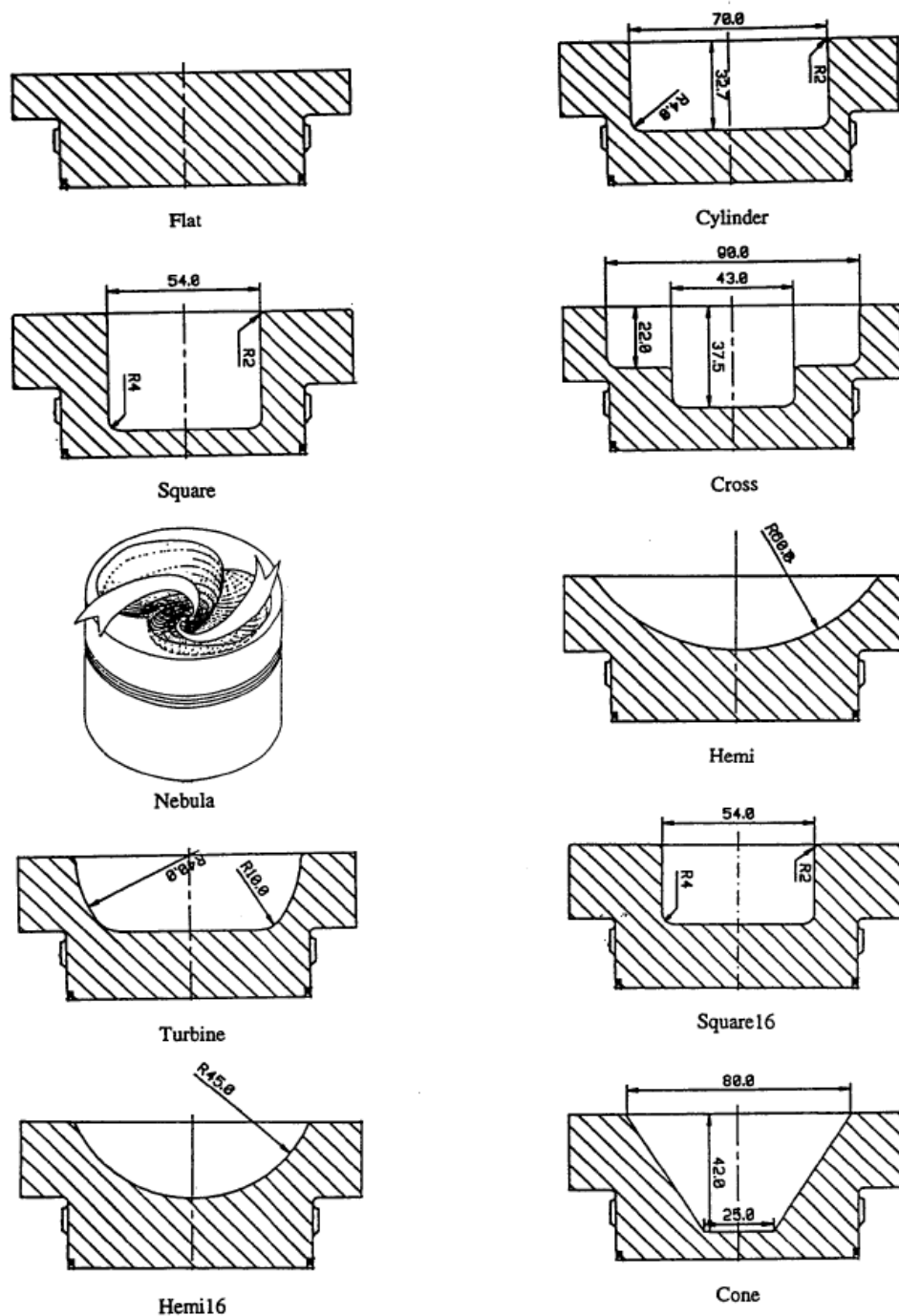


Figure 4-13, Piston shapes investigated by Johansson and Olsson at Lund University [24]

These investigations were later continued by Einewall and Johansson, also at Lund University. In their research they looked into more advanced piston shapes and more in-depth investigation on the combustion of natural gas in order to increase efficiency and reduce emissions [22]. The piston crown shapes were chosen in order to promote further breaking-up of the swirl in the piston in order to create turbulence. The turbulence was targeted to increase the combustion speed with a

timing towards TDC [10]. The different pistons chosen are presented in Figure 4-14 with Turbine open bowl piston and Nebula piston chosen yet again to be included in the investigation. Added to the experiment are the re-entrant type pistons with a square and a round hole which are the extension of squish area from the open bow piston. The square hole is used to promote further turbulence as the swirl flow enters the piston bowl. The Quartette piston was based on the design from the AVL TRIFLOW piston presented earlier but with an added wing to the design. This was design to increase the swirling motion in the combustion chamber and use the squish areas to break it down into turbulence late in the compression stroke. The last piston, the Fair Top was developed from the cross section piston used by Johansson and Olsson. It was designed to perform in the same way as the Nebula and Quartette piston and make the swirl collide to create turbulence.

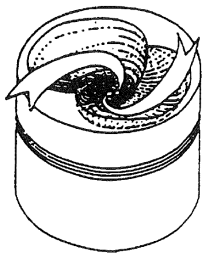


Figure 1: The Nebula combustion chamber from Ricardo.

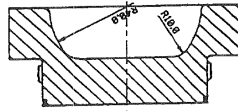


Figure 2: The turbine combustion chamber.

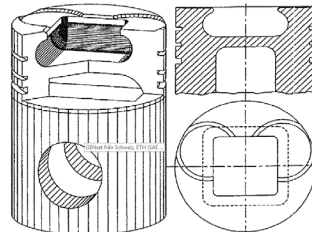


Figure 3: The Reentrant Square piston geometry.

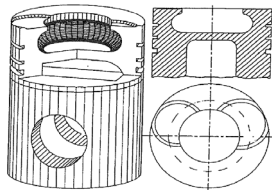


Figure 4: The Reentrant Cylindric piston geometry.

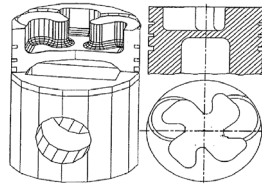


Figure 5: The Quartette piston geometry.

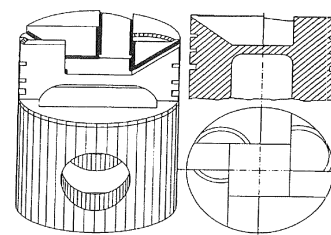


Figure 6: The Fair Top piston geometry.

Figure 4-14, Piston shapes investigated by Einewall and Johansson at Lund University [10]

The measurements on turbulence were performed in the same way as the previous experiments, through Laser Doppler Velocimetry. Despite the seemingly identical experimental setup, the turbulence level measurements were not identical for the two pistons from the previous measurement.

Additionally, two different swirl levels were tested for their effect on turbulence for two different pistons. The different swirl numbers were 2 and 2.8 and another camshaft with higher lift was also tested. The results are shown in Figure 4-15 and show that added swirl does add to the peak turbulence at TDC. However, the magnitude is much dependent of the piston shape used. This resembles the simulations by de Castro Viana et al. [26] mentioned above. However, the y-axis is in different unit then for Figure 4-9 and Figure 4-10 covering the reports from Cartillieri et al. and Castro Viana et al and thereby not comparable. Figure 4-15 present the turbulence measured on the engine,

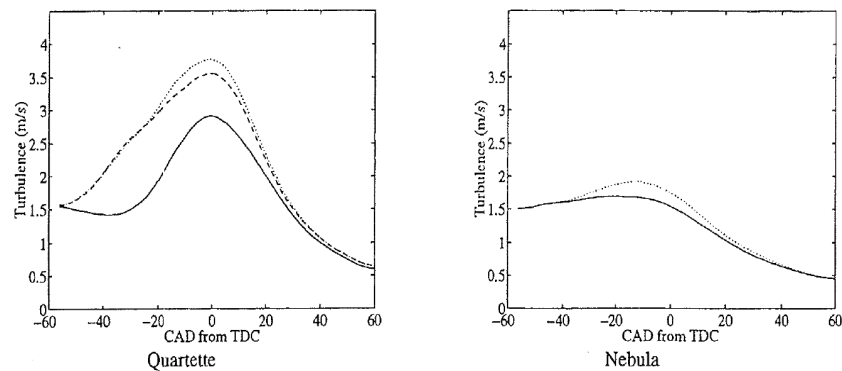


Figure 13: The turbulence in Quartette and Nebula, with two kinds of camshafts and cylinder heads. The engine was operated at 1200 rpm and the window was located in the center.

— $z = -5$ mm, low-swirling cylinder head and natural gas camshaft
 ---- $z = -5$ mm, high-swirling cylinder head and natural gas camshaft
 $z = -5$ mm, high-swirling cylinder head and diesel camshaft

Figure 4-15, Comparison of turbulence with added swirl [10]

Einewall and Johansson conclude their report by stating that the Quartette piston and re-entrant piston showed the best overall results from the operating points tested. The Quartette piston showed the largest turbulence peak and also had the highest lean limit of the tested pistons. The heat release rate shows on a rapid combustion with this piston. The turbine piston which is quite common in modern CNG engines showed quite slow combustion, timed late, after TDC. Despite this, it is still frequently used in modern engines. This might be due to the reduced heat transfer to the piston and cylinder walls and lower NO_x-levels, together with fairly high efficiency in the tested operating points. It also produced the lowest levels of HC. This lies in line with the results mentioned by Johansson and Olsson [24] that there was no or little difference in efficiency or NO_x levels for moderate or fast burn systems.

Roethlisberger et al. [28] and Nellen and Boulouchos [29] showed experiments on a genset machine with both a re-entrant piston and an open bowl piston. The re-entrant piston was reported to deliver a much more stable combustion and wider flammability limit both for lean burn as well as for stoichiometric combustion with EGR. These tests did however also show an increase of CO emissions which could originate from incomplete combustion. Other drawbacks with the re-entrant piston was the reported lower thermal efficiency which was believed to be related to the heat transfer to the cylinder walls and piston.

Wu et al. [30] presented in 2016 CFD simulations on three different piston shapes, see Figure 4-16, for a large bore SI engine. Piston #1 is a common open bowl shape. Piston #2 is a re-entrant type piston with the same bowl depth as piston #1 and only marginally larger squish to bore ratio. The last piston, piston #3, is a cross rib design with small wings in four locations along the otherwise open bowl design. This reason for this design might be to interact with the swirl flow and cause turbulence in the outer rim of the piston. This gives a more stable flow around the spark plug, albeit with a higher rotational speed. The large bore of the engine may be the enabler to this slightly unconventional approach.

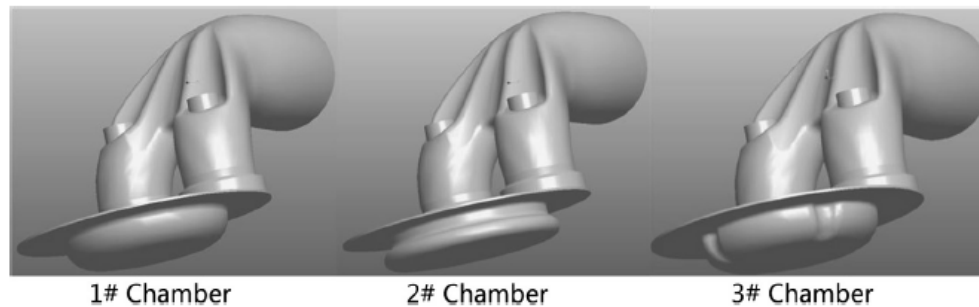


Figure 4-16, Simulated piston shapes by Wu et al. [30]

The results from these simulations are remarkable though, as can be seen in Figure 4-17 where the mean TKE shows much larger peak values than has been reported by others. Also, the jump in the curve just before 600 CAD may indicate that there are errors in their simulation results.

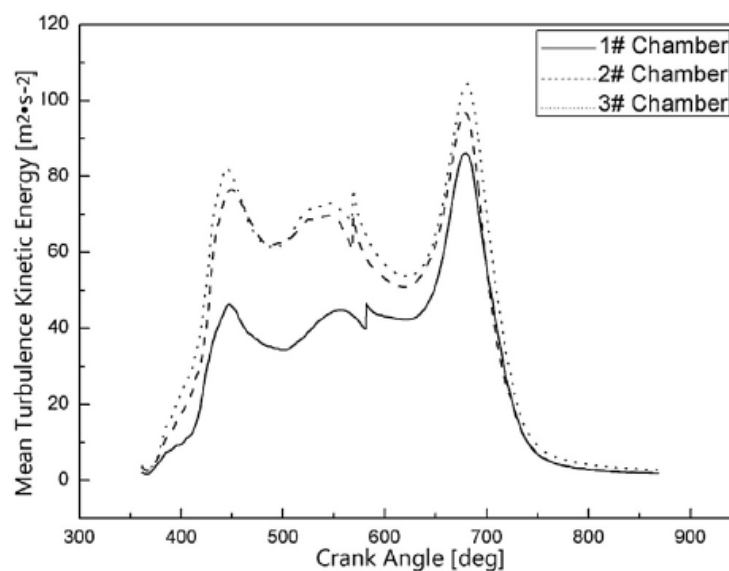


Figure 4-17, Results from simulations by Wu et al. [30]

Arnberger et al. [8] from AVL performed CFD simulations as part of the development of combustion concepts for commercial gas engines with high efficiency. In the simulation two different piston shapes were compared, an open bowl piston and a re-entrant piston. The development process aimed to increase the BMEP, Brake Mean Effective Pressure, in order to better match that of the Diesel engine. The mean effective pressure of a gas engine is commonly 20-21 bar BMEP, whereas diesel engines are currently running more than 24-25 bar BMEP. They also pointed out the reduced potential to reduce CO₂ emissions compared to a modern diesel engine running on B7 (7% biodiesel). Comparing the mass fraction carbon content of the fuel and net calorific value of the two fuels, they claim a maximum CO₂ saving potential of 24-25%, per energy input. This aligns with similar calculations performed as well by Adlercreutz et al. [31]. Further

mentioned by Arnberger et al. was the difference in fuel efficiency which reduced the CO₂ benefit to 15-17%. Adding the limitations of throttling losses on part load and lower cylinder mass further decreases the CO₂ benefit. Arnberger et al. also pointed to the limitations of the current gas engines and their relationship to the diesel engines they are developed from. The inherited cylinder head from the diesel engine use in CNG engines gives a swirling flow and the design process of the combustion chamber is thereby focused on the piston bowl to improve combustion efficiency.

The baseline engine in their simulations was a research engine with ~2 litre/cylinder and had a swirl number of SN=1.6, with a compression ratio of 12:1. The simulation results are shown in Figure 4-18 and visualise the theory of swirl and squish length in the two right figures. The swirl is seen as the blue zone “going into the paper” and the red zone “coming out from the paper” creating a swirl around the centre axis. The swirl is spun up by the squish in the open bowl, and the increase in squish length increase the flow field velocity. The theory presented by Cartellieri et al. [1] stated the benefit to have a symmetrical flow at the start of ignition with slow flow by the spark. Figure 4-18 shows this behaviour. To the left in the same figure the turbulence is simulated and it is clear that the re-entrant piston creates higher level of turbulence kinetic energy in the bowl. This is beneficial to the combustion speed of the fuel charge as the flame propagation speed is increased.

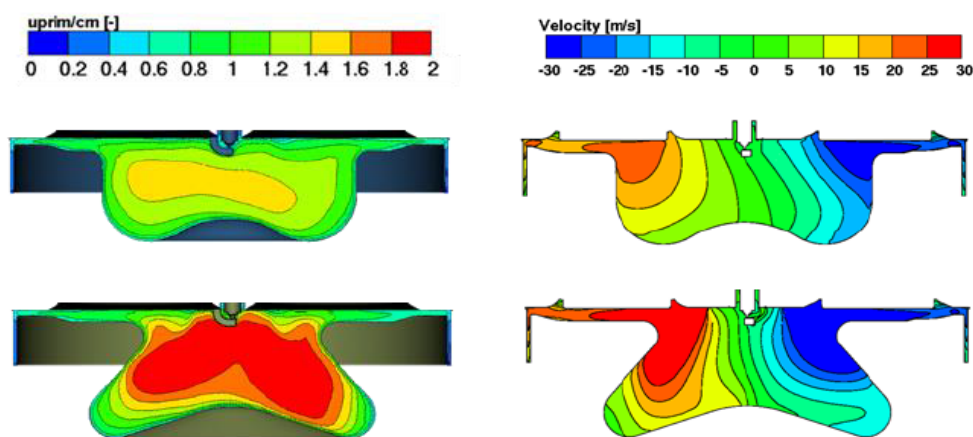


Figure 4-18, Simulations of turbulence and flow field velocity in the open bowl and re-entrant piston. Left: Turbulence [U'/cm]. Right: Flow field velocity

The simulations also show a benefit in the rate of heat release, with a steeper increase in release rate for the re-entrant piston over the open bowl piston. The re-entrant piston also showed a higher EGR compatibility, being able to withstand higher EGR rates which could be explained by the increased turbulence and the more rapid flame propagation. From the simulations on EGR compatibility both pistons showed a sudden increase in instability, albeit at different levels of EGR. This gives a clear indicator when calibrating the engine to know when the EGR-limit is reached. The benefits of EGR on the combustion is that it can enable for de-throttling at part load which lowers the pumping losses. Higher EGR rate also

lowers the exhaust temperatures at high engine loads and a capability to mitigate knock through lower combustion temperature.

As a result of the simulations the authors also simulated lean operation on the same two pistons due to the higher combustion speed of the re-entrant piston. The main benefits are the increased efficiency with a lower fuel consumption and reduced methane slip over the engine. Drawbacks include the increased NO_x emissions, which was shown in Figure 4-5, which requires the use of an SCR-system in order to realise Euro VI legislations. The results showed a potential to use the faster combustion rate of the re-entrant piston to increase the lean-limit of the engine.

4.5 CNG FUEL QUALITY

The following chapters, 4.5-4.7, was originally published in [32] and has been rewritten to fit this report. Much of the use of CNG is valid both for light duty as well as for heavy duty. Specific HD use has been added to this report.

In the CNG market there not yet been decided on a standardisation of CNG for vehicles, but several quality specifications are used as a compositional guide. The intent of the standardisation of CNG is to have specified properties for engines to properly use the fuel and reduce emissions while maintaining safety and satisfactory performance under all climate conditions and driving demands. For CNG the quality specifications are in the composition of the gas and the levels of variations can be accepted from both manufacturers and end-users, i.e. the engine and car manufacturer. The specification is also meant to provide for a safe operation of the vehicle and associated equipment and protect the fuel system from corrosion, poisoning and liquid or solid deposition. It is also an important aspect in fully using the potentials of the resource. From the predictable behaviour of a fuel the optimisation of the combustion can be performed and thereby deliver greater driveability and range. CNG does however vary in its composition depending on the different production sites which have led to the categorisation in which levels the variations can still have a viable fuel and low emission levels. To this there are also seasonal variations that cause the composition to vary; in the latter case the variation is mainly in the hydrocarbon content [33].

In 2002 the International Association for Natural Gas Vehicles, IANGV, posed in a report the need to harmonise the certification fuel specification, commercial fuel specification and engine manufacturers' specification to have the end emissions to be lowered and performance increased, while the gas quality varies from production field to production field [33]. Previous fuel quality specification reports do however only propose the set levels of the gas composition. In the upcoming standardisation from Svensk Standard the composition will be more strictly defined and contain requirements rather than suggestions, a line that will be used in the upcoming EU- standardisation of CNG for ICE as well. The new standardisation will include specified testing methods in order to regulate the set levels.

The above mentioned report from IANGV was based on previous investigations on the variations and how they may influence the operation of the vehicle. In this report, the author Ly [33] reports from a report by Liss and Thrasher, who conducted a nationwide survey in 1992 in the US on produce variations in CNG [34]. They found a 14% variation in heating value, 14% variation in density, 20% variation in Wobbe index and 25% in the air-fuel ratio. These large variations affect the emission levels and the performance of the engine together with the knock resistance of the fuel, this was one the factors behind the proposition from IANGV.

Ly further references to a report from 1997 where the Gas Research Institute, U.S., published the results from a test program carried out by the Clean Air Vehicle Technology Centre. It investigated the impact of how the variations in gas composition affected the engine operations for light-duty engines. The gas used was determined from the previously mentioned investigation by Liss and Thrasher. The study found that overall even dramatic changes would not dramatically affect the performance or emissions for a vehicle designed to run on natural gas. It was also added that if the vehicle operator would experience any problem, this was due to a widely fluctuating fuel and the experience would only be transitory. In the study it was also found that the dedicated NGV consistently performed better than the bi-fuel NGV as the latter compromises. The emission levels of the bi-fuel NGVs were also higher than the dedicated NGVs, but none of these displayed large changes in emission levels as the gas composition was varied. This was thought to lead to changes in the fuel specifications without drastic changes in emission levels [35].

A European study in 2001 done by Schollmeyer and Wegener, again referenced by Ly, investigated the different natural gases distributed in Europe. The results from the study was a proposed classification of H(>41MJ/kg) and L group($\leq$ 41MJ/kg), which stands for high heating value and low heating value respectively [36]. The grouping would allow the engine's ignition map to be switched between the two settings and thereby adapt more easily between the different properties of the two groups. [37] This has been followed by the Svensk Standards classification of Grade A and B, which sets the requirements for the quality of the gas at the fuel filling station. The grades include specifications for the Wobbe-index and methane number.

4.5.1 Details of the gas composition

In the quality specification not only the composition of the parts mentioned by Liss and Thrasher are specified, but also levels of other substances are defined. These are water content, sulphur and sulphur bindings, CNG compressor oil carry-over, carbon dioxide, oxygen, particulates, methanol and hydrocarbon. In more recent standardisation reports the levels of siloxanes and hydrogen are also defined.

The water content, or actually water dew point in the gas, describes the water condensation behaviour of natural gas. It is regulated as it in liquid form is a precursor to the formation to corrosive compounds together with carbon dioxide, hydrogen sulphide or oxygen which causes corrosion in storage tanks or fuel lines. It could also, as the gas varies in pressure, cause ice particles that may disrupt fuel

flow or even engine stoppage. Gas driers are available for this application and are used to avoid this problem, [38] according to [33].

The sulphur in CNG does to some extent occur naturally in the CNG, but is also added for leak detection. The sulphur causes problems in the aftertreatment system as it causes catalyst poisoning which impairs the functionality of the catalyst. There are also some sulphur components that are corrosive and may cause fractures in the fuel system. Removing the sulphur all together leads to the need for another leak detection system which is more expensive to implement, [38] according to [33].

The oil carry-over is related to the pressure increase at the fuelling station with compressors leaving traces of oil from its operation. This oil may add to the emissions of the combustion engine and should therefore be kept to a minimum. In an investigation on oil carry-over from 1995 [39], according to Ly [33], changing from mineral to synthetic oil reduced the carry-over, as well as adding filters to the piping which catches the droplet oil in the fuel. Through the pressure increase, the oil can also be heated and vaporised. It is therefore advised to have another filter further down the fuelling lines in order for the oil vapour to condensate and be caught by this filter. This carry-over does have an advantage though, as it may protect the containers and fuel lines as well as lubricating the injectors [38].

There are no limits to levels of carbon dioxide in CNG, given that there is a limitation to the water content which limits the risk of corrosion on the fuel system. There is however a recommended limit under SAE J1616 of 3.0% to maintain stoichiometry. [33] Carbon dioxide can act as a source of cooled EGR to the fuel, limiting the available oxygen in the combustion chamber. Since this cannot be regulated by the engine control unit, the engine behaviour could be erratic in between fuelling.

Oxygen content is limited in SAE J1616 within the flammability limits of the natural gas, as it may add to the acceleration of the combustion. Oxygen could also add to the corrosion of the fuel system if the water content was above the allowed levels. [33]

Particulates or other foreign matter should technically not be contained in the CNG. If dust is present, clogging and abrasion might be caused to the fuel system components.

Methanol is used to control the pressure water dew point and to avoid freezing. Methanol in the storage tanks causes corrosion and therefore no methanol is allowed in the gas. [33] Testing on methanol content is normally not done, but there are standardized tests available [38].

The variation of heavier hydrocarbons in the gas composition is related to seasonal changes in the natural gas as well as covering peak demand loads. As heavier hydrocarbons have higher heating value than methane, the Wobbe index is used as a measure on how the gas can vary in its composition with constant heating value. With a changing Wobbe index, the air to fuel ratio changes as well and it is carbon dioxide is widely used to reduce the heating value as well as air to fuel ratio. In modern engines with closed loop lambda control the effect of the Wobbe index is

however not of great importance as the lambda is constantly adjusted in the ECU [33].

Siloxanes are mostly present in Biogas where it comes from the biodegraded material present in landfills, such as shampoo bottles and other containers. Siloxane may form Silicon dioxide upon combustion which collects in deposits on valve, cylinder walls and liners which cause abrasion and blockages.

Hydrogen may be added to either hydrogen-enriched CNG or through changes in the gas composition, see Wobbe index above. Hydrogen could create corrosive compounds with sulphur or oxygen which may cause failure to the gas tanks.

4.5.2 Knock resistance

The common way to compare the knock resistance of fuels is using the octane rating which compares a fuel against iso-octane. This is created for liquid fuels, but it is possible to use it for gaseous fuels as well though this takes time and is an expensive process. Instead methane and hydrogen can be used as the reference fuels and a Methane Number, MN, scale is calculated. Basically the MN of 100 exhibits a knocking behaviour similar to that of 100% methane whereas a MN of 0 is similar to that of 100% hydrogen. This method is known as the AVL Methane number, but there is also the CARB method available. Neither of these methods is however considered to be practical enough and there are attempts being done to refine the calculation methods. The AVL MN is considered to be good, but complicated as special software is needed to calculate the methane number. The CARB method is considered easy to use but gives to large variations for different gas compositions.

From the methane number scale there are conversion scales to the octane rating for easier comparison of different types of ICE fuels. The corresponding octane number for CNG is from approximately 115 to over 130. The range is due to the composition of the gas and its variations. The adding of heavier hydrocarbons as mentioned above also lowers the knocking resistance, hence the range of the octane number, [40] according to [33]. The new limit values for the produced gas for natural gas vehicles is set to between a minimum MN 65 for CNG grade A and a minimum for MN 80 for Grade B, though the levels are not fully set yet.

4.6 EMISSIONS

One of the main selling points for CNG engines have previously been that the exhaust gasses are virtually particle free [1] as the fuel is gaseous [41, 42]. As particulate measuring equipment became better it became possible to quantify these emissions and legislate on the levels, hence the introduction of PN and PM levels in Euro VI. The main contribution of particulates in a CNG engine is believed to come from the engine oil, but may also be introduced from poor fuel quality or A/F-ratio in the combustion. The particulate matter from CNG combustion was reported by Minutolo et al. [41] to be less than 1 μm in size and has filterable and condensable fractions. They continue in reporting that this matter is usually larger molecular weight hydrocarbon that are not fully combusted. It has also been shown that in locally slightly rich mixtures, particles with a mean size of 2-3 nm could be detected in high numbers. These particles were said to be "clusters of high-molecular-mass aromatic compounds, both pericondensed and incompletely condensed structures formed at local slightly

fuel-rich conditions and not oxidized in the post-oxidation zones of the system” [41].

Hydrocarbon emission are related to the methane slip that may occur during valve overlap [43]. They can also occur due to trapping in the crevice volumes. This may also occur due to cycle-to-cycle variation, misfires in the engine, and unburned fuel in the combustion chamber due to quenching. This problem is most common in the crevice areas and as the compression ratio is increased in the engine the density in the crevice area is unavoidably also increased. As the flame propagation in the combustion chamber has limited possibility to combust the trapped charge in the crevices, this leads to increased levels of HC-emissions as this charge is left unburnt. This is the source of 50-70% of the emitted methane hydrocarbons, as Königsson reported for lean burn Dual Fuel in 2013 [44]. This leads to higher levels of hydrocarbons in the exhaust system and catalyst which are already difficult to catalyze [17].

Königsson also reported that swirl could be used to improve the oxidation of HC that returned from the crevice area. At SN around 3 the HC emissions were reduced by approximately 20% compared to a quiescent combustion system. It was concluded that for lean burn, $\lambda > 1.8$, quenching became more the dominant source of HC emissions.

4.6.1 Emission regulations of CNG

Emissions from Heavy Duty CNG truck are regulated with the same levels as for Heavy Duty diesel trucks, but with the exception that the HD CNG engines are not tested in stationary conditions. The emission standards regulate the emitted hydrocarbons (HC), nitrous oxide formations (NO_x), carbon monoxide (CO) and particulate matter (PM) and number (PN). The HC emissions are split up for the CNG engine into non-methane hydrocarbons (NMHC) and pure methane emissions (CH_4). The diesel emissions are regulated to the total hydrocarbon level. The allowed levels of these emissions are presented in Table 1 for both the previous level, Euro V and for the future Euro VI. Both are presented for reference of the quantification of the levels. There are currently no legislative CO_2 -emission limits on truck engines.

Table 1, EU emission standards for Heavy-Duty CNG engines [45, 46]

Emission stage	Date	CO	NMHC	CH_4^a	NO_x	PM	PN
				[g/kWh]			[1 / kWh]
Euro V, ETC	2008.10	4.0	0.55	1.1	2.0	0.03	-
Euro VI, WHTC	2013.01	4.0	0.16 ^b	0.5	0.46	0.01	6.0x10 ¹¹

a – for NG engines in Euro V and NG and LPG engine for Euro VI

b – THC for diesel engines

There is also an ammonia concentration limit of 10 ppm which is applicable for engines running with a SCR-catalyst. It is also reported by dieselnets.com that a maximum limit for NO_2 in the NO_x emission may be defined in a later stage [47].

4.6.2 Volume and range of CNG

Even though CNG has many benefits as a fuel and has a high energy density, MJ/kg, the major downside of CNG is the gaseous state. The volume of gas needed to give equivalent energy is due to the density larger than for liquid fuels which in turn results in larger fuel tanks. This can be seen in Figure 4-19 below, where the net calorific value of CNG is higher than for diesel, 46-49 MJ/kg compared to 41-42 MJ/kg for diesel. The density of CNG is approximately 0.80 kg/m^3 which bring down the effective range of the vehicle [36]. The fuel tanks are designed to cope with the large pressures of the CNG, which does increase the weight as to ensure adequate stress fracture resistance in the tanks. Further problems with the tanks include collision safety and overall weight of the truck. Natural gas can also be stored as liquefied gas, but only at very cold temperatures (-162°C). This can be done in cold-resistant tanks but much energy is used to maintain the liquid state. When liquefied the fuel is called LNG, liquid natural gas. [36]

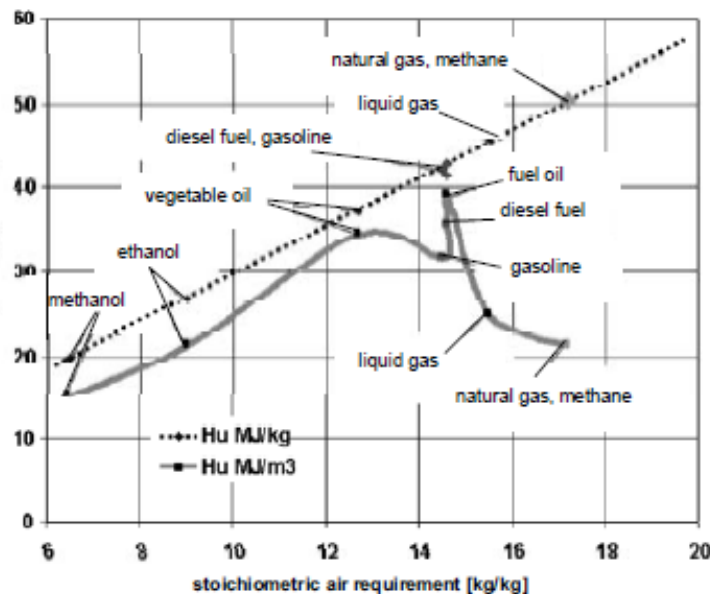


Figure 4-19, Energy content for different fuels [48]. Values for lower heating value, H_u

4.6.3 CO₂ equivalence

In 2014, Nylund et al. [49] reported in their comparison of performance for diesel and CNG buses. They claim that taking in the benefit of the specific CO₂ intensity for both fuels (55 g CO₂/MJ for methane and 73 g CO₂/MJ for diesel [50]), the benefits of the CNG bus is nullified due to a lower efficiency of the engine according to the authors. Including HC-slip in their equivalence calculations there was very little difference between the two fuels. They concluded that the gas engine needed to be improved vastly in order to deliver the benefits in greenhouse gas emissions. This comparison was based on CNG but if Biogas is used instead, the environmental impact can be reduced. Noteworthy here is the discrepancy between the numbers on efficiency reported by Arnberger et al. [8] and Nylund et al., where Arnberger et al. reports a higher efficiency on the engine output. One explanation may be the nature of the application of the transit buses which suffers more from the pumping losses at part load, depending on the cycle tested.

4.7 MIXING OF AIR AND FUEL

Mixing of gases is difficult to perform as both medias are in gaseous state. Chiodi et al. stated in 2011 [51] the difficulties to achieve homogenisation especially at high gas velocities. This was due to the injector nozzles caused low fuel penetration. The authors stressed that the homogenisation was more controlled by the charge motion than for liquid fuels.

Yu et al. [52] performed PLIF-imaging (Planar Laser-Induced Fluorescence) in experiments on low-pressure gas jets to investigate the structure and mixing of fuel and air. They presented the conceptual profile of the jet in Figure 4-20 which shows the main mechanism for entrainment through engulfment of ambient air along the upstream edge of the large-scale structure. It was also shown in the report that the injection pressure affects the spray formation and that a higher pressure results in a longer and wider jet spray. This aligns theory and practice and can confirm the model. The injection pressures used in the test ranged from 3 bar to 7 bar. These experiments do not align with what Chiodi et al. claims above which may indicate that the theoretical model and experiments by Yu et al. is applied to an experimental bomb with no flowing charge motion that affects it.

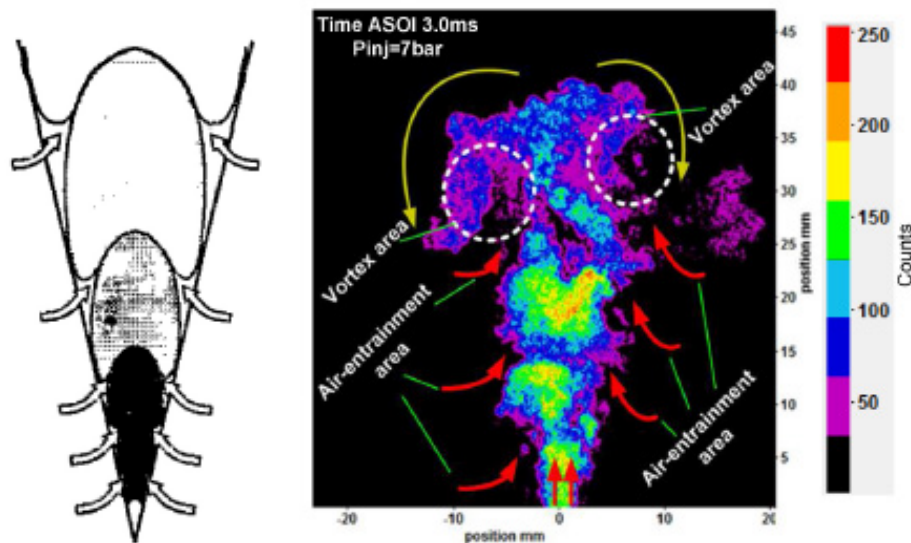


Figure 4-20, Left: A conceptual profile of the air-entrainment and mixing process of gaseous fuel injection, [53], source from [52] . Right: Concentration field of gas jet from PLIF imaging [52]

5 Planned research questions

In order to design a piston bowl that enhances the combustion by increasing the turbulence level post fTDC the theory indicates that there should be a pre-existing level of turbulence surrounding the spark plug when firing to aid initial flame propagation. This gas movement is also necessary in order to mix the air and gas prior to igniting the mixture. Since the cylinder head for the planned experiments is based on a swirling system, this flow field is formed in the piston bowl on the intake stroke and can be maintained during the piston strokes. This is due to the piston being of circular type and there is only friction against the surface that slows down this movement in the baseline type of piston. The baseline engine uses a turbine type piston which is a common piston design due to a conventional manufacturing process. Compared to more difficult designs, such as the Quartette or Triflow piston mentioned in chapter 4.4, the turbine is cheaper to manufacture which keeps the unit price down. Another factor that should be pointed out is that a circular piston shape seems to aid the flame propagation after TDC as the swirl speed motion is not stopped like in a split chamber piston. However, the split chamber piston like the TRIFLOW and Quartette piston indicate that there is plenty of turbulence to create a rapid combustion. This is used to burn the charge rapidly and avoid heat transfer to the piston after the combustion. Another issue with the split chamber piston is that they may cause too much heat loss from the combustion due to the increased surface area in the piston. These types of pistons should also slow down the swirl motion causing the system to lose energy. The chosen research question is how the altering of the squish and swirl parameters in the piston and cylinder head influence the combustion stability. By adding more turbulence, the intended benefit is to be able to increase the level of EGR in the combustion which can help to mitigate knock and to deliver more torque from the engine.

The research goal set in the project was to increase torque by 20% and decrease fuel consumption by 10% while maintaining Euro VI emissions standards. By creating a higher level of turbulence in the combustion, more EGR can be introduced into the air and fuel mixture which can be used to mitigate knock. By doing this, an increase in torque can be observed. To further increase efficiency, lean operation is to be tested together with EGR to reduce NO_x under lean conditions.

6 Technical approach

The testing is done on a Single Cylinder Research Engine, SCRE, at AVL Motortestcenter, in Södertälje, Sweden. The engine is a Scania engine with a 130 mm bore cylinder with the possibility of changing the cylinder liner for other bores if needed. The combustion system is taken from the production setup and adapted for the SCRE setup. As it is a SCRE only stationary operating points will be investigated. These operating points are chosen from the World Harmonized Stationary Cycle, WHSC, and based on the full-size performance of a Scania OC13 CNG engine. This is done even though gas engines are not run for this cycle under EURO VI compliance testing but is adapted here in order to have valid operating points during SCRE testing. The SCRE is then run at these operating point and data is gathered and then further analysed.

In order for the SCRE to run at all operating points, the engine can be charged through a compressor providing intercooled charged air to the engine setup. A backpressure valve is used to simulate the backpressure of a turbocharger. The cooled EGR used for the recirculation is taken upstream of the backpressure valve and then further compressed through a supercharger that is used to create the positive drive pressure to the intake side of the engine. A fully programmable ECU is used to control the fuel to the engine. The operating points from the WHSC cycle were chosen from multi cylinder engine operations and the operating parameters transferred to the SCRE control unit.

6.1 TESTING CYCLE DATA POINTS

The testing cycle is based on the World Harmonized Stationary Cycle used for engine emissions regulations testing for heavy duty engines. The operating points, presented in chapter 6.1.1, are defined from the full size base engine, the Scania OC13. The data is calculated in accordance to the definition of the cycle. These values are in turn scaled for a single cylinder engine and used as operating data for each testing point.

The WHSC is not used for CNG engines in emissions regulations as these engines are only regulated for transient cycles. However, as the single cylinder engine cannot run under transient conditions, the WHSC was chosen despite the cycles non-relevance for CNG engines. The steady state cycle was also chosen in order to have operating points that are central for engine testing in emissions regulations in order to evaluate further data. These operating points are weighted which is also beneficial in the post analysis of the gathered data.

6.1.1 WHSC operating points

The cycle data points are graphically presented in Appendix A: Figure 11-1 in WHSC data, and numerically Table 8 in Appendix B: WHSC Numerical Data. This cycle includes 13 operating points with P1 repeated, so in effect 12 operating points. 2 operating points are added to the testing table in order to cover the maximum power point as well as high load at low engine speed.

6.2 HARDWARE TESTING

The hardware tested included three different types of cylinder heads, with different levels of swirl. The purpose of the testing was to investigate how the swirl and squish interacts in stoichiometric combustion. It was also intended for investigation on the potential of mixing in EGR in order to mitigate knock and also, for some cases reduce the negative pressure in the inlet manifold during throttling. Enabling the use of more EGR in the combustion process is a key feature to enable higher engines loads and reaching the target of increased torque. These tests are then extended with three different levels of squish lengths for a turbine type piston, in steps of 4 mm, presented in Table 2. The degrees of freedom for the design of the piston squish length was limited by the bore size, the piston height and minimum material thickness possible to withstand the peak cylinder pressure in the engine.

Table 2. Cylinder head and piston variation

HW variation			
Cylinder head, swirl	1,25	1,7	3,02
	Low Swirl - LS	Medium Swirl -MS	High Swirl – HS
Piston squish length	16,1 mm	20,1 mm	24,1 mm
	Low Squish - LS	Medium Squish - MS	High squish - LS

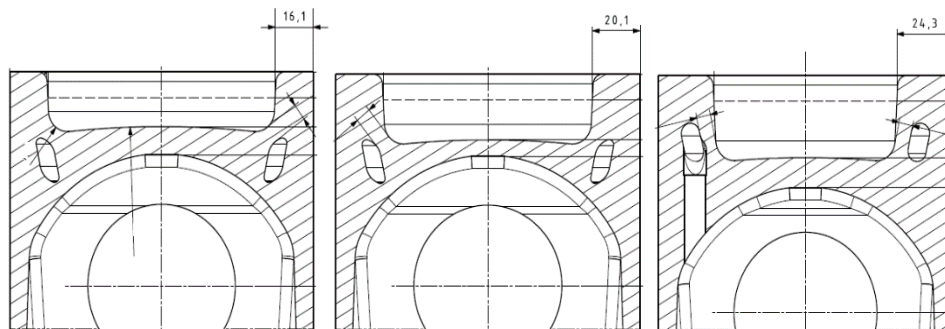


Figure 6-1. Piston cutouts for each of the three different squish variation

From these hardware variations the measured data is then analysed from a number of data points selected in Table 4, in chapter 6.2.1. The annotations for the different swirl and squish levels are used for an easier representation of the different combinations, i.e. MS/HS indicates Medium Swirl, High Squish and so forth.

The baseline engine, in single-cylinder format, has the specifications shown in Table 3.

Table 3. SCRE specification

Bore x stroke	130 x 160
Geometric Compression ratio	11,7
CNG	Natural gas from LNG. MN(AVL) = 80
Engine speed	According to Table 8
Valve train	Push rod, 4 valve
Intake and fuel injection system	Standard intake manifold with standard port fuel injection, adapted for SCRE
EGR	Cooled, externally compressed for additional levels of freedom regarding amount of EGR
Charging system	Stand-alone external supercharger. Backpressure valve to simulate turbo backpressure.
Lambda	Stoichiometric
Load	Part and full, idle
Cylinder pressure measurement	AVL Indicom with AVL GU21 pressure transducer

6.2.1 Testing Procedure

The engine was set up to run at the same conditions as the full size engine and measurements were taken accordingly at each data point. 5 data points were chosen as comparative points for the different hardware setups. P1 chosen at idle, P3 and P2 at low and high load and medium speed, P10 at higher engine speed and load and MaxPWR at maximum power. Additionally, the EGR capacity of the engine is investigated at high load to analyse the effect of EGR mixing at these conditions. EGR is intended to be used in order to mitigate knock and increase the torque output from the engine. This is presented in Table 4.

Table 4 - Hardware Testing points

Test	Operating conditions
WHSC operating points selection	P1, P3, P2, P10, MaxPWR
EGR Variation, high load	P2: 5% and 15%
EGR Variation, low load	P3: EGR 0-20% in steps of 5%

The EGR variation for high load was 5% and 15% run at 300, 333 and 350 Nm load. For these operating points the engine was first run at knock limit and spark was then retarded in steps, each step represented by an increase in exhaust temperature of 20°C until a maximum exhaust temperature of 800°C, or operational instability, was reached. One point at maximum load and each EGR rate was also run. This gives a map over how the engine can be operated under different spark advances. The reference exhaust back pressure was kept at reference value.

For EGR on low load the rate ranged from 0-20% EGR, where at each step of 5% an MFB50 sweep was performed. The variation in MFB50 was performed from 4-20 CAD aTDC with steps of 4 CAD per each step.

By altering the spark advance at high loads, pushing the limits of the engine it is possible to investigate these limits of the combustion system. These limits at high

load are knock, instability and/or exhaust temperature. By visualising this in a plot, see Figure 6-2, it is possible to create an easy understanding of the limitations of different hardware setups. Running the engine like this is possible due to the stand-alone supercharger which creates large degrees of boost freedom when analysing the combustion system. Using this layout when comparing the different hardware setups, it is possible to view the limitations side by side.

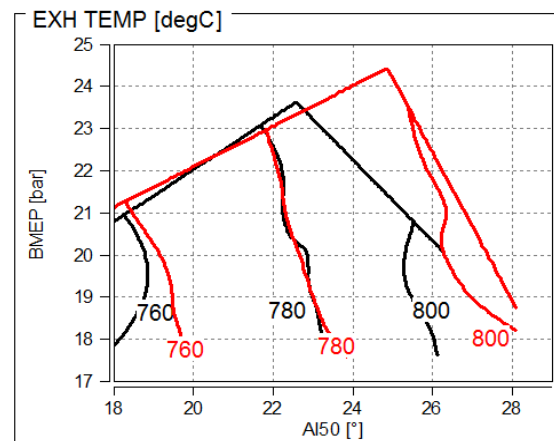


Figure 6-2. Example of limitation of knock and exhaust temperature

7 Results and Discussion

The initial testing of the hardware gave, as mentioned in chapter 6.2.1 - Testing Procedure, a set of operating points that span the entire speed and load range. These points were chosen as they created an array of engine conditions in which the squish and swirl should break apart factors which could affect air and fuel mixing in a CNG engine.

The data from the first testing points in Table 4 will be compared for swirl and squish separate. MS/LS is the same for both variations and used as baseline, where the abbreviation follows the same structure set in Table 2. The grey and yellow boxes in, Figure 7-1 for instance, is used to categorise the operating points where load and rpm is varied. In the grey box the load is varied between mid and maximum load and in the yellow box the rpm is varied from 55% to 80% of max rpm. The emissions will be presented in two ways to investigate patterns between the operating points, and between the hardware variation.

Beginning with the emissions from the initial testing points, the NO_x -emissions in Figure 7-1 all peak at P3 and followed closely by P2, which is the same speed, but different torque. This indicates that NO_x for these hardware tests is speed related rather than load related as the load increase from P3 to P2 is percentage wise larger than the difference in engine speed. According to Heywood [17], the NO_x emissions are governed by the timing of the combustion in relation to TDC which does not explain the similar levels in NO_x between p3 and P2. The MFB50 is earlier for P3 than P2, but with the higher load in P2 comes a higher peak pressure which increases the thermal creation of NO_x . As the speed increases, both sets of hardware shows a decrease in NO_x as the time for NO_x formation is limited.

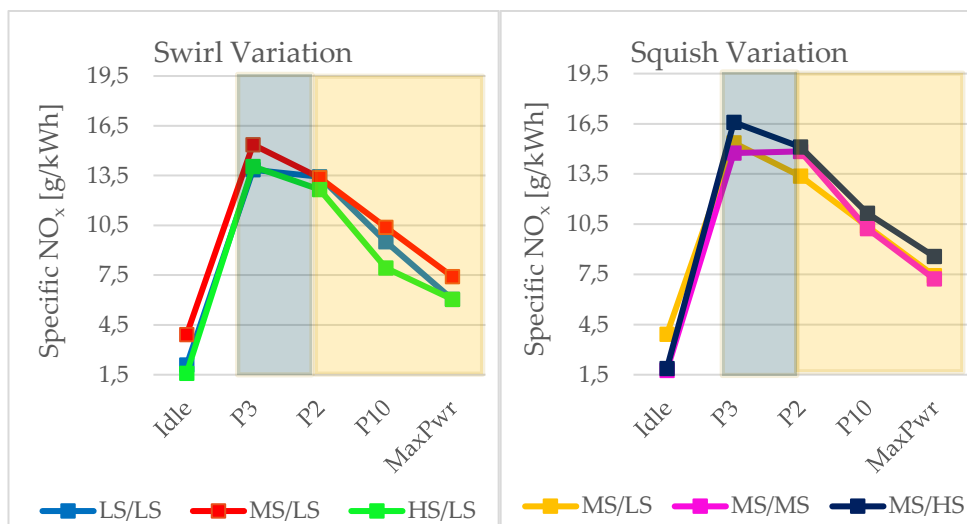


Figure 7-1. Specific NO_x as function of load point

Plotting the hardware variations instead shows in Figure 7-2 that for all operating points the MS/LS piston shows the highest value, apart from P2. In P2 the engine

was run against knock limit, whereas the other operating points are run against mapped value. This gives a more uniform and quick combustion duration, which could be the explanation for the deviation of the pattern of peak emissions levels for MS/LS. Overall, both LS/LS and HS/LS produce similar combustion durations, why MS/LS shows increased levels of NO_x. The lower NO_x for HS/LS is due to the maximum cylinder pressure being lower for this operating point, according to Table 5. Varying the squish length indicates low difference between the low and mid squish, except for idle where the low squish length stands out with higher NO_x for MS/LS. Also here it is possible that the combustion duration is the explanation for the variance in hardware behavior.

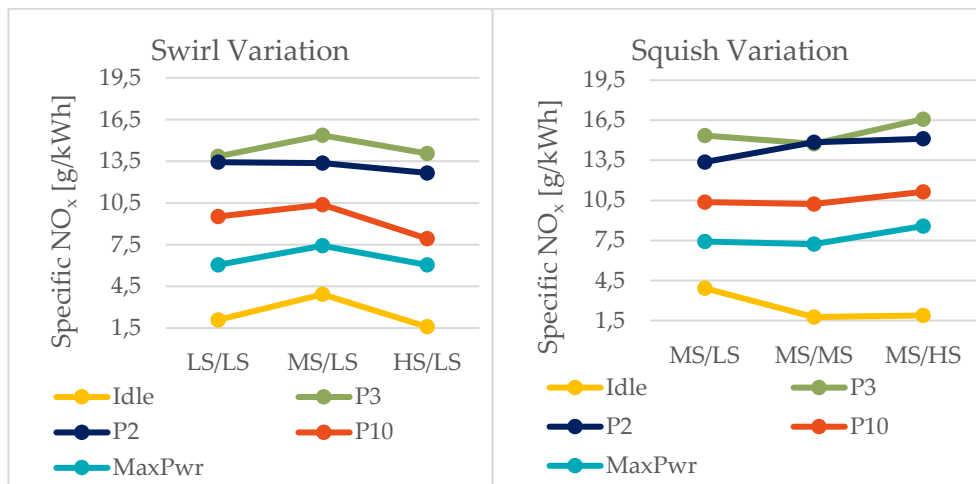


Figure 7-2. Specific NO_x as function of hardware variation

Table 5. Maximum Cylinder Pressure for the different hardware combinations

P _{max}	LS/LS	MS/LS	HS/LS	MS/LS	MS/MS	MS/HS
	Swirl			Squish		
Idle	13,41	17,24	12,89	17,24	13,93	13,08
P3	41,73	40,04	40,22	40,04	41,76	41,9
P2	99,62	99,52	101,41	99,52	100,73	105,28
P10	106,17	105,97	99,73	105,97	105,25	102,89
MaxPWR	96,89	94,4	96,43	94,4	96,17	95,37

The CO emissions show instead a minimum value for points P3 and P2, Figure 7-3. Heywood [17] describes the emission of CO as a function related to the equivalence ratio, which in this case should be the mixing inside the combustion chamber as the global equivalence ratio is the same for all operating points. These local variations are present through the cycle-to-cycle variation, but as the compared value is an averaged value over the same measuring time, a measuring point richer in CO should indicate fuel rich areas in the combustion. For the swirl variation the low swirling head shows benefits of increasing speed and load, as CO decreases substantially. Increasing the engine speed shows a repeating trend of a peak for

P10 but lower for the lower and higher engine speeds. This pattern repeats for the squish variation.

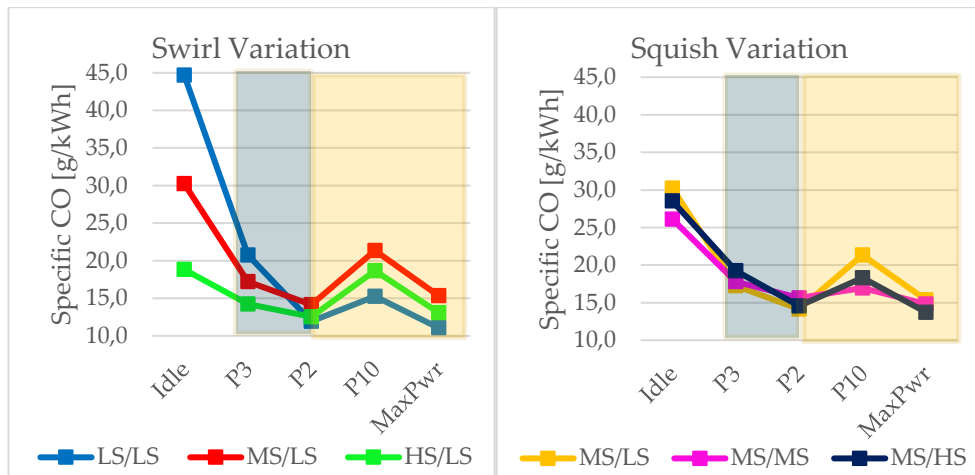


Figure 7-3. Specific CO emissions as function of load point

Comparing the hardware variation in Figure 7-4 does not show any clear patterns between the operating points, however, the emissions value aligns in the same order regardless of swirl or squish variation.

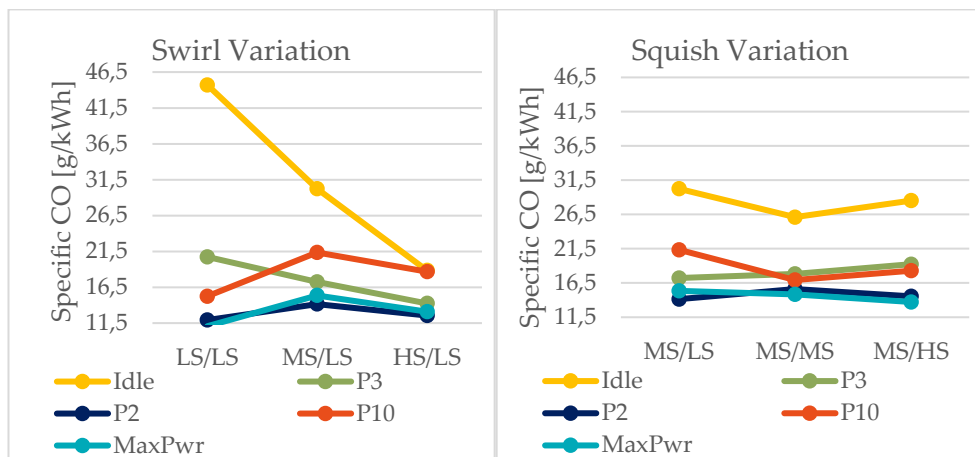


Figure 7-4. Specific CO emissions as function of hardware variation

The HC emissions in Figure 7-5 behave fairly similar for all the operating points, regardless of the hardware configuration. For the swirl variation a slight increase is visible for the higher engine speeds of P10 and MaxPwr for MS/LS. At these points the combustion duration for MS/LS was shorter meaning the residual fuel from crevices or flame quenching could not be post combusted in the same extent. With a later combustion it is still possible to combust some of these residuals.

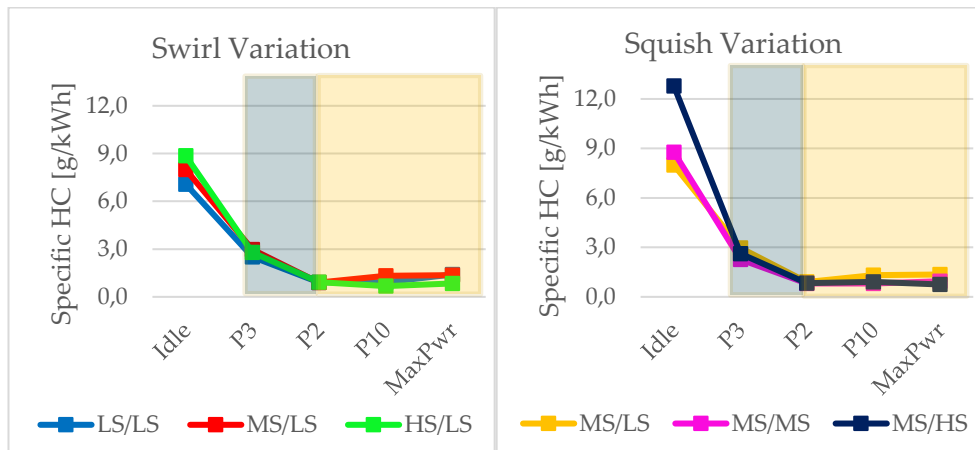


Figure 7-5. Specific HC emissions as function of load point

The higher HC emissions for both squish and swirl variations in Figure 7-6 for idle and P3 is most likely due to the low MAP through throttling at these points. The other operating points, which are all boosted operating points, show very little difference in-between the different configurations. The lower MAP value causes oil to be drawn up into the combustion chamber and emitted when the exhaust valves open. The high value for MS/HS at idle is possibly due to the new piston rings which had not yet been worn in at the time of testing. However, this may as well not be likely as P3 followed with lower specific HC emissions. Another explanation may be the thermal expansion of the piston which at low loads cause larger crevices between the piston and cylinder liner. The hotter piston in P3 expands and narrows the gap.

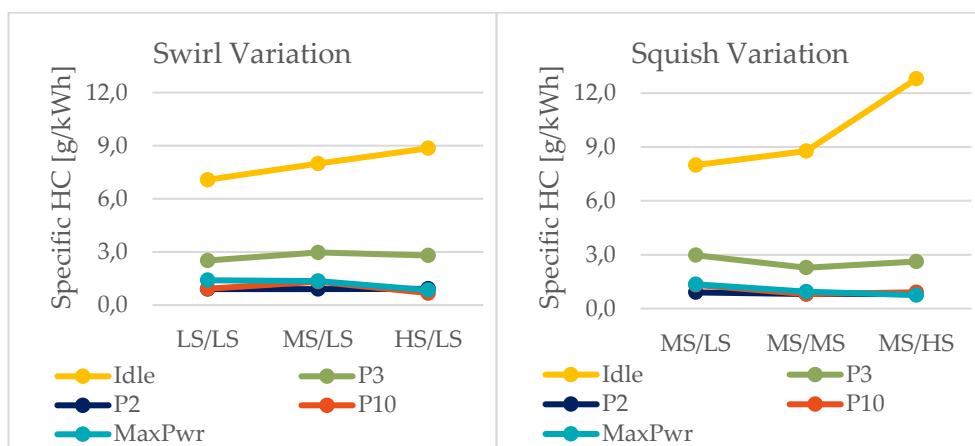


Figure 7-6. Specific HC emissions as function of hardware variation

CO₂ emissions are directly related to the fuel consumption and is thereby a measure on the efficiency. This indicates that the Low swirl head shows a higher efficiency, except for MaxPwr where the High swirl head shows slightly better values. For the squish variation the mid squish piston gives the higher efficiency for every point except for P2 where the efficiency is reduced. The reason for this is

not entirely clear when analyzing the data but Figure 7-1 and Figure 7-3 both indicate an increase in emissions why the anomaly should be correct.

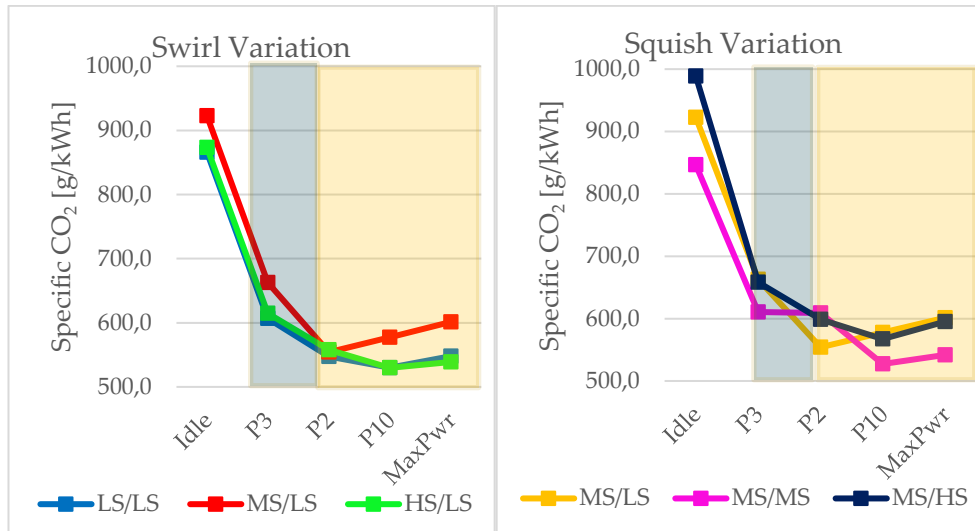


Figure 7-7. Specific CO₂ emissions as function of load point

Comparing the hardware variation, Figure 7-8, all setups show the same trend, except for P2 where the squish variation indicate that the engine may not have been running as efficiently for this operating point as it had for the other points. According to these five operating points it is clear that both altering the swirl and squish would improve the emissions. The best swirl level is the High swirling head, as it lowers NO_x and CO the most. For HC and CO₂, the low swirl head indicates the lowest emissions, however the high swirl head is well in par with those levels. The best squish levels of these three is the mid squish piston from an emissions standpoint.

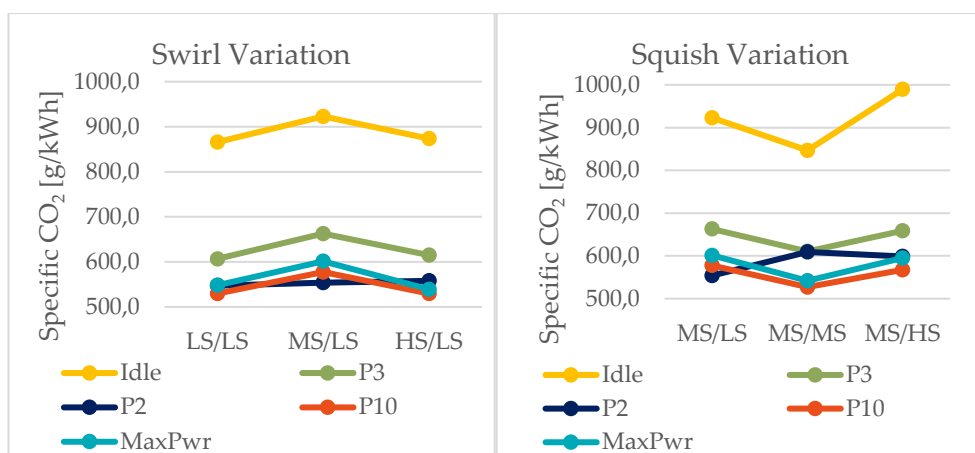


Figure 7-8. Specific CO₂ emissions as a function of hardware variation

The five operating points are not only run to investigate the emissions from the engine, but to investigate how the hardware combination may affect the efficiency of the engine. In Table 6 the values for the BSFC is presented together with the

difference in fuel consumption for each variation. All these operating points, except for P2 were run in accordance to the reference data set. P2 was instead run against knock. This should indicate that there could be room for improvement if the engine is calibrated further for each hardware setup. Seeing as the fuel consumption is quite bad for the different setups at idle, in comparison to the baseline, it should be possible to improve the results for this operating point.

Table 6. BSFC comparison for the different hardware combinations. Percentage value compared to baseline

BSFC	LS/LS	MS/LS	HS/LS			
	Swirl			MS/LS	MS/MS	MS/HS
				Squish		
Idle	380,6 (+21,6%)	312,9	365,6 (+16,8%)	312,9	356 (+13,8%)	466 (+48,9%)
P3	225,2 (0,5%)	224,1	223,1 (-0,4%)	224,1	222 (-0,9%)	234,9 (+4,8%)
P2	99,62 (+0,1%)	99,52	101,41 (+1,9%)	99,52	100,73 (+1,2%)	105,28 (+5,8%)
P10	185,2 (-1,5%)	188,1	188,2 (+0,1%)	188,1	186,1 (-1,1%)	187,7 (-0,2%)
MaxPWR	190,5 (-3,1%)	196,5	193,9 (-1,3%)	196,5	191,9 (-2,3%)	194,7 (-0,9%)

The thermal efficiency shows how much of the fuels energy has been converted to mechanical energy. Overall, LS/LS shows together with MS/MS the best energy conversion for these operating points, except for idle.

Table 7. Thermal efficiency

η_{th}	LS/LS	MS/LS	HS/LS			
	Swirl			MS/LS	MS/MS	MS/HS
				Squish		
Idle	25,3%	26,1%	26,4%	26,1%	26,9%	25,2%
P3	36,3%	37,0%	36,6%	37,0%	37,0%	37,0%
P2	40,1%	39,1%	39,4%	39,1%	39,1%	39,8%
P10	41,1%	40,6%	40,4%	40,6%	41,4%	41,6%
MaxPwr	40,4%	39,6%	39,9%	39,6%	40,5%	40,6%

The combustion duration for these hardware variations is presented in Figure 7-9 and in detail in Figure 7-10 for P3. For the swirl variation the pattern is repeated for all five different operating points, though with a shallower angle between the points for P2. The lower combustion duration for baseline is intriguing but requires more investigations to why this phenomenon occurs, as this is not in line with the theory on swirl. For the squish variation the pattern is not as clear as two points for MS/HS lies outside this pattern. This could be due to the piston rings that had not been worn in and the vacuum in these two operating points caused a higher blow-by into the combustion chamber from the crank house. For these variations P2 shows a pattern which does not resemble any of the other operating points.

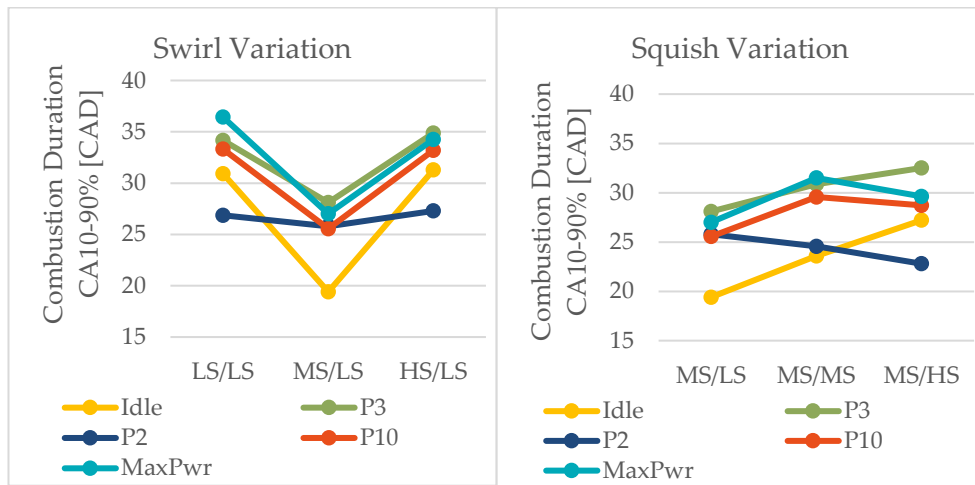


Figure 7-9. Combustion duration as a function of hardware variation

The combustion duration is shown in detail for P3 in Figure 7-10, where the Heat Release Rate is presented. The MFB50 for these hardware setups align as this operating point was run against the same MFB50 for each case. Looking at the swirl variations they show a faster initial burn up till MFB10 but then burns slower compared to baseline at 10-50% burnt. After MFB50 the combustion speed is similar to baseline, with low swirl(LS/LS) being slightly slower still. IN the figure MS/LS and HS/LS align making it difficult to separate the two.

According to the theory, the higher swirl should initiate a faster combustion, which it initially does. However, the two swirl variations align for this part which is not in accordance to the theory This could be related to the inlet manifold and the throttled conditions in this operating point.

Increasing the squish, MS/MS, gives an initial burn time which aligns with the baseline, but increases in speed after MFB30 and burns faster than baseline. Increasing the squish further to MS/HS, the initial burn is slower to develop before MFB10. The combustion time MFB10-90 is then increasingly faster compared to baseline.

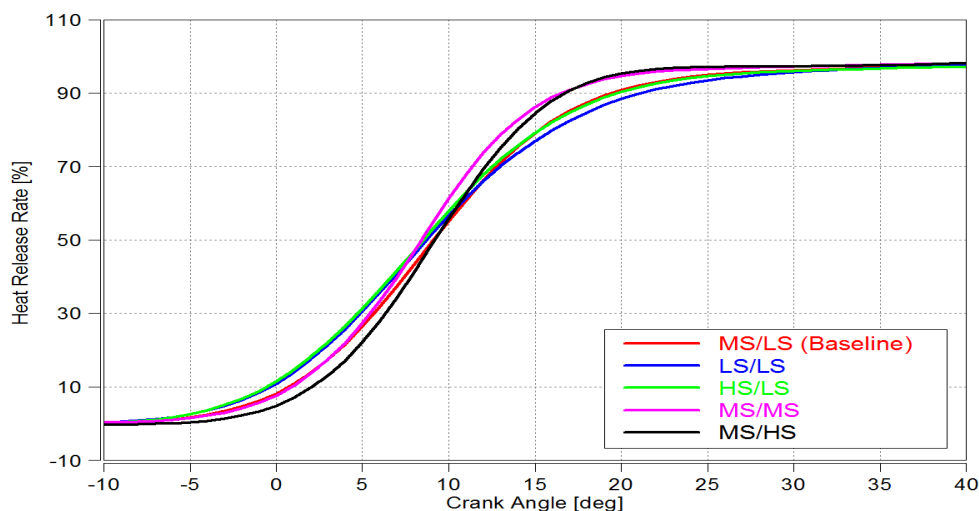


Figure 7-10. Averaged Heat Release Rate, 1292 RPM, 83Nm

7.1.1.1 Lean operation with EGR

In an attempt to further improve the efficiency of the engine, it was run under lean operation together with 10% EGR for P2. This is done as a way to reduce the NO_x emissions which rise for lean conditions. Reducing the NO_x opens up the possibility to run lean in a mixed mode combustion and thereby increase efficiency. To evaluate the results, the engine was also run without EGR so that it would be possible to compare the effect of the added EGR. The results for the increased efficiency in Figure 7-11, where the reduction in BSFC is presented compared to the stoichiometric operating point without EGR. Running the engine lean together with EGR results in a total reduction of BSFC of 7,3%, at 1292 RPM and 333 Nm full load. The difference in fuel consumption of adding EGR is due to two things. As the exhaust back pressure is kept at its reference value, the added boost needed to compensate for the added EGR does give a lower BSFC as the pressure balance over the engine is shifted. The other effect is thermodynamic where the higher specific heat ratio for EGR compared to air increases the work done on the piston [54]. The pumping losses mentioned previously that may reduce the fuel consumption has no affect here as the operating point is boosted.

A downside to lean operation is the initial increase in NO_x which is lowered for $\lambda > 1,2-1,3$. Increasing the NO_x increases the load on the EATS to be able to catalyse the emissions under lean operation. If these emissions were to be too high, the concept may not be possible to implement. Figure 7-12 presents the specific NO_x emissions for both these cases and the results show that at $\lambda = 1,3$ the NO_x is similar to the levels at stoich. In the figure it is also possible to view the mentioned increase of NO_x for $\lambda > 1,2-1,3$. Adding the EGR shows that it is possible to reach the same levels of NO_x at less lean conditions compared to without the EGR. This results in a wider operating area for the engine before reaching its instability limit.

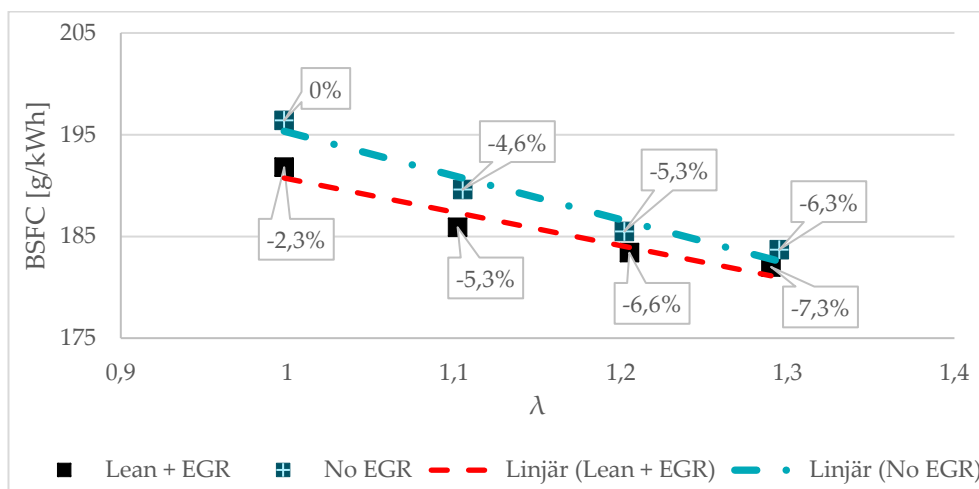


Figure 7-11. BSFC benefit from running lean with and without EGR, 1292 RPM, 333 Nm full load

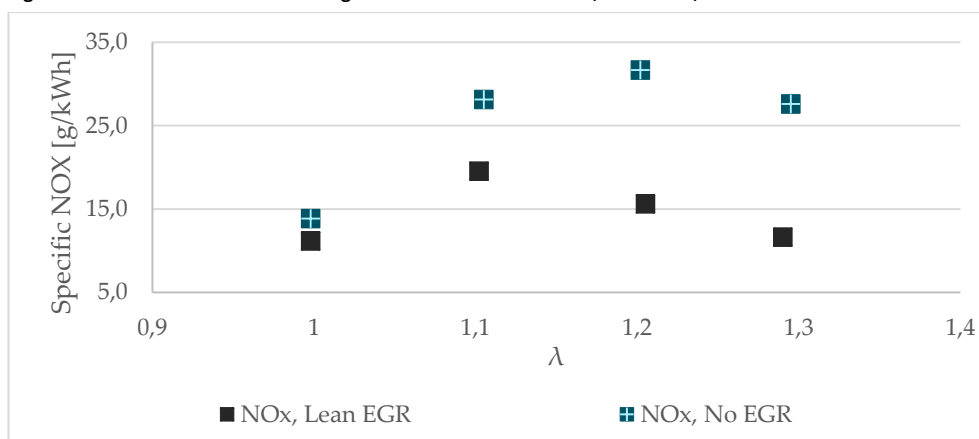


Figure 7-12. Specific NOx under lean operation, with and without EGR, 1292 RPM, 333 Nm full load

Analyzing the Heat Release Rate in Figure 7-13 gives for both cases that leaning out the charge did improve on the initial part of the combustion, but the main combustion is faster for stoichiometric combustion. Adding EGR to the lean operation slows down the later part of the combustion, but maintains the initial part. Adding EGR to the opposite combination, at stoich, gives a faster initial combustion which for the later part slows down.

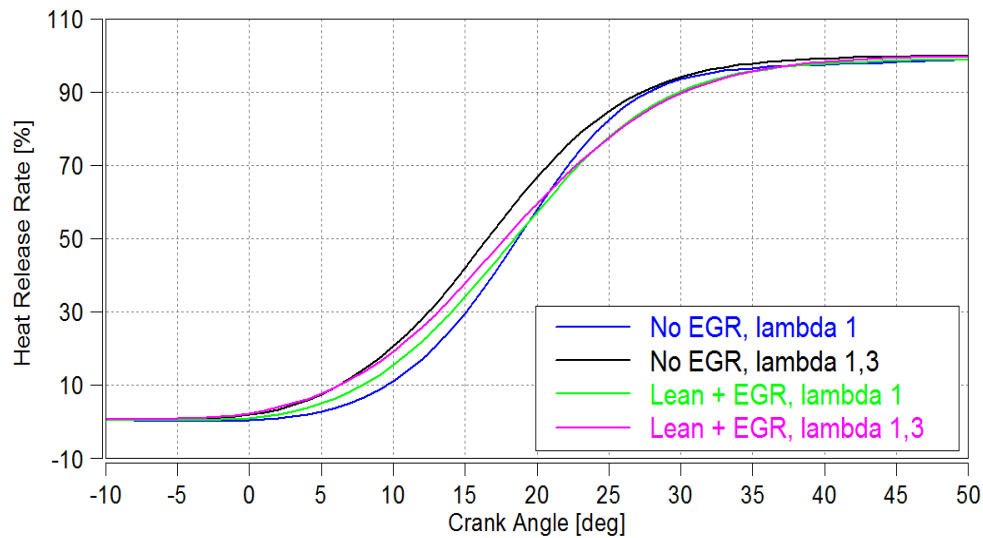


Figure 7-13. Averaged Heat Release Rate for 1292 RPM, 333Nm. $\lambda=1,3$

7.1.2 EGR Variation, High load

5% EGR

For the EGR analysis performed, the engine was run at three different loads, and the settings for MFB50 was then analysed as described in Figure 6-2. Adding more EGR to the combustion process is a key feature in order deliver more torque from the engine. This is due to the mitigation of knock in the combustion process and being able to advance the spark further in order to transform more of the combustion energy into mechanical energy.

In Figure 7-14 the combustion data from the low swirl head tests are presented in comparison with the standard setup. MAP is the Manifold Absolute Pressure and shows if the setup suffers from flow losses. COV IMEP is the measurement of instability, where the limit is 3% before the engine runs too unstable for proper operation. Plotting BMEP against spark advance when shifting MFB50 between knock limit and max exhaust temperature gives the outer limits of the combustion system. The triangles in Figure 7-14 shows these limitations in effect. The further graphs in this figure show BSFC and exhaust temperature which are self-explanatory and are used to compare the fuel consumption and how the exhaust temperature varies between hardware setups. From the figure it can be seen that the LS/LS setup runs cooler closer to the knocking limit and also by the instability limit. The latter does allow for a larger area of operation before reaching the limits of the system. For the middle of the temperature range, where the engine is not run against any limit the two setups align. The COV limits shows about the same operational stability for those areas where the graphs overlap. For the standard hardware it was possible to run beyond 3% COV for 20 Bar BMEP load, but not for LS/LS. The BSFC show approximately the same fuel consumption for both hardware setups, with a slight improvement for unstable conditions for retarded spark timing. The maximum load point was run higher at LS/LS than for the

standard setup but with a retarded spark. However, there was still some margin for more load from the MS/LS setup.

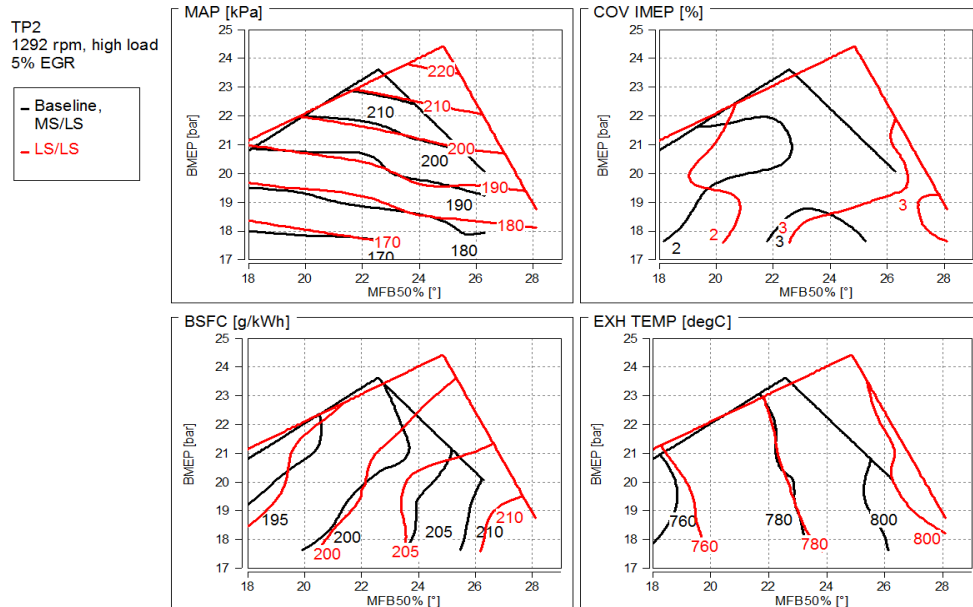


Figure 7-14. Combustion data for 5% EGR for LS/LS compared with Baseline

The combustion data analysis for the High Swirl/Low Squish piston variation in Figure 7-15 show that this setup runs cooler in exhaust temp at high loads, compared to the baseline, MS/LS. The maximum load from this setup is lower than for LS/LS but the higher swirl does allow for a wider range of spark timings at high loads. At 20 Bar BMEP, the fuel consumption is higher for HS/LS, but is improved as the load increased. The turbocharger needs to add more boost earlier in the load curve, which is in line with the previous analysis that the higher swirl requires more boost due to the flow losses. Exhaust temperature does go down at high loads with late MFB50, which decreases the thermal load on the engine components.

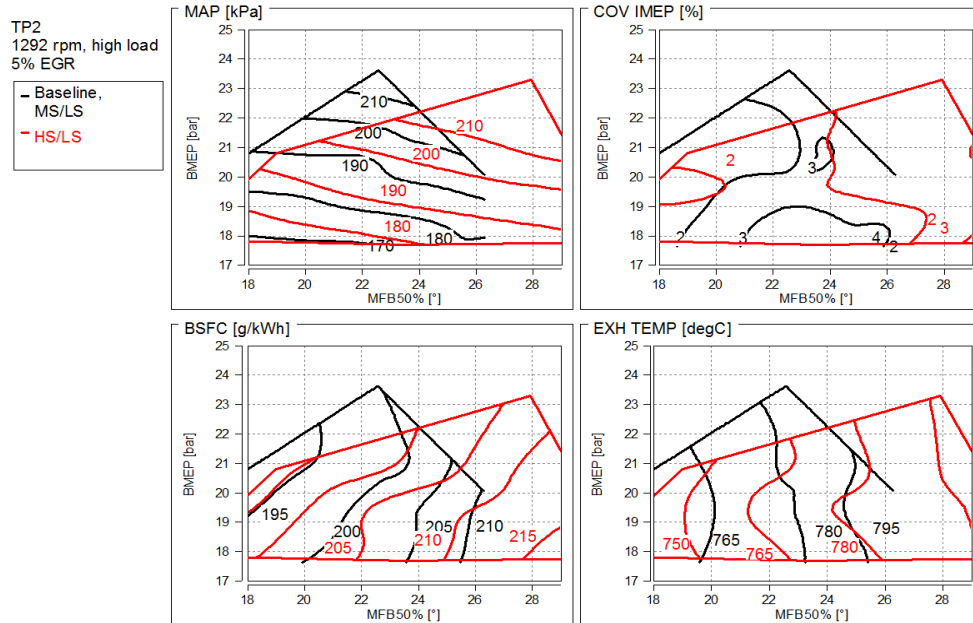


Figure 7-15. Combustion data for 5% EGR for HS/LS compared with Baseline

For the mid squish piston, Figure 7-16, the data is almost completely overlapping the baseline data. The engine requires pretty much similar levels of boost for the same load, but there is a benefit in the increased spark advance possible to achieve. As for exhaust temperature, the MS/MS setup runs slightly hotter closer to knock limit, but aligns along the instability line in the graph. Operational stability in COV shows that the runs similar to baseline, with a better stability around peak BMEP.

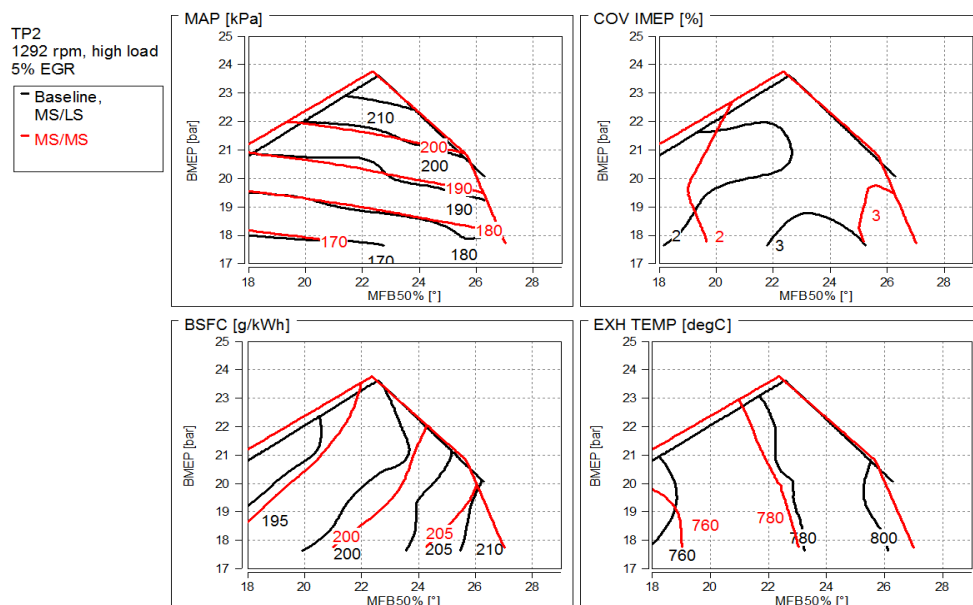


Figure 7-16. Combustion data for 5% EGR for MS/MS compared with Baseline

The final hardware variation, MS/HS, with longer squish length in the piston show a lower peak load possible before running into the limitations of the engine. The running parameters are otherwise similar to the baseline, but a better stability along large areas of operation.

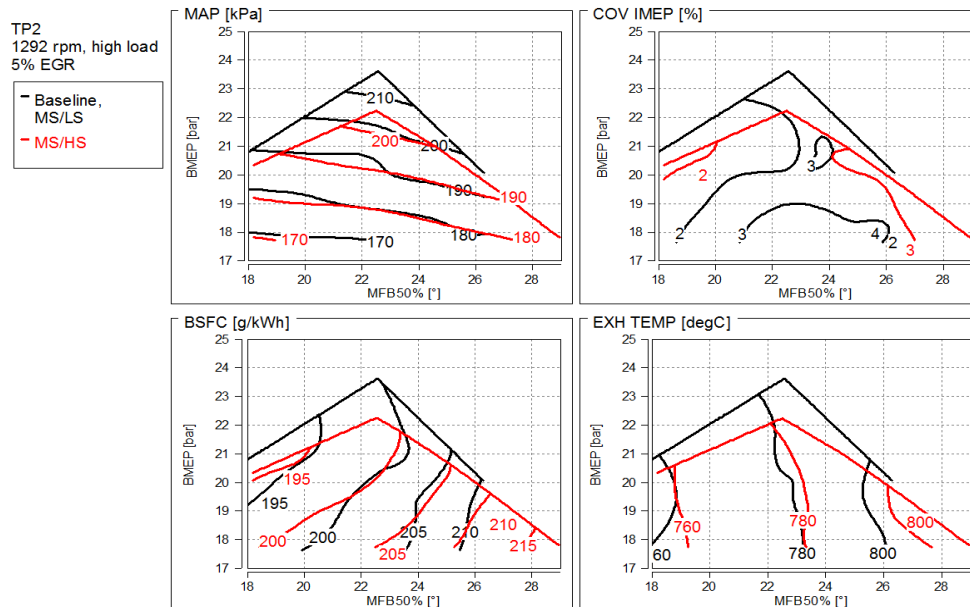


Figure 7-17. Combustion data for 5% EGR for MS/HS compared with Baseline

15% EGR

Adding more EGR pushes the boundaries for the combustion system further, but also shows that the benefit of more EGR in respect to the ignition limit and the BSFC. This becomes clear in Figure 7-18 where both 5 and 15% EGR is compared. The higher EGR requires higher manifold pressure but gives in turn the same load output at lower fuel consumption. This also lowers the exhaust temperature which reduces the thermal load on the engine. This is a following effect to the theory on EGR.

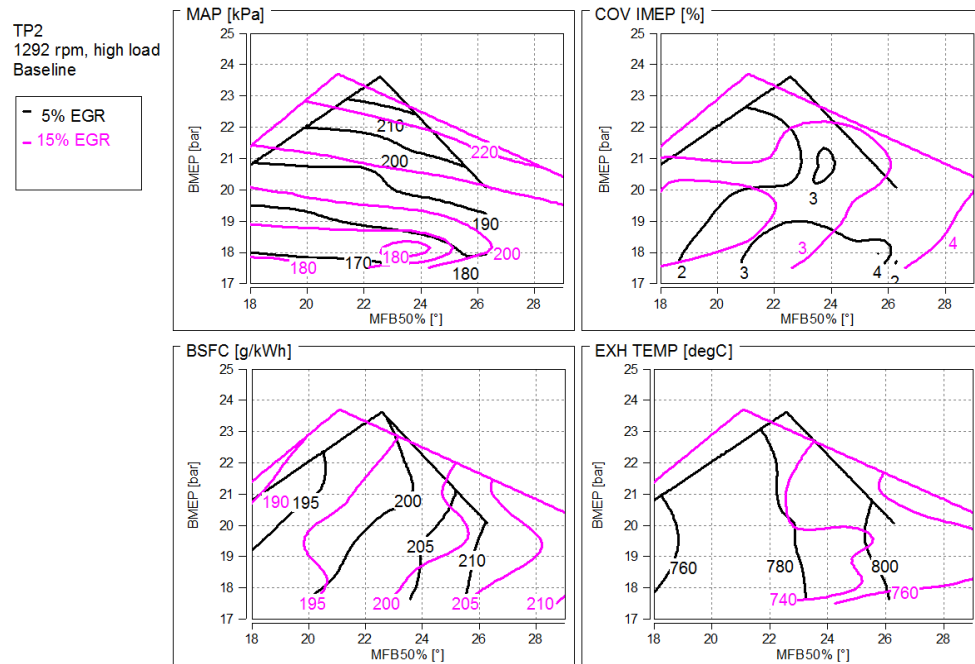


Figure 7-18. Comparing effect of 5% and 15% EGR, baseline setup

Making the same comparison for emissions gives Figure 7-19, showing how the emissions are affected by the added EGR in the baseline setup.

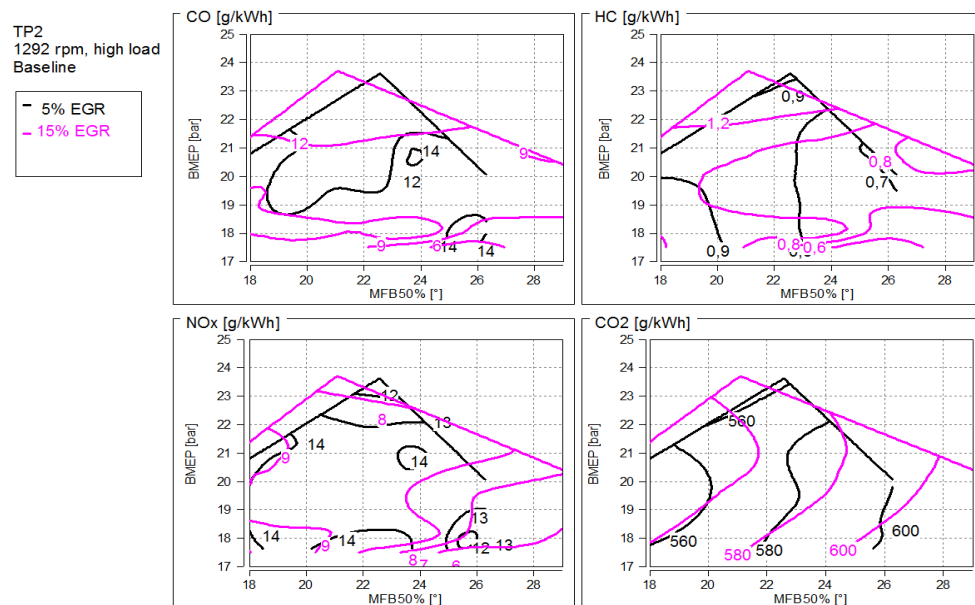


Figure 7-19. Emissions comparison for 5% and 15% EGR at high load

Adding more EGR did improve the CO emissions. According to Heywood [17] this is related to the equivalence ratio, as mentioned previously. The added boost level for the higher level of EGR is the reason for this improvement, as well as the effect of the inert gas in the combustion chamber. The HC emissions showed a reduction

when retarding the spark. This behavior is due to the later flame development with a retarded spark which can combust the residual hydrocarbons. Looking at the NO_x emissions, the levels are overall lower for higher EGR, which is again in accordance with the theory on adding EGR, that it lowers the combustion temperature and thereby also the nitrous emissions, despite the added boost pressure. The levels of CO_2 are comparable for both levels of EGR, though the fuel consumption is lower for the higher EGR level.

Running the different hardware setups for 15% EGR further shows the limitations of the combustion system and the EGR-stability. LS/LS does deliver similar peak torque output as the standard setup, see Figure 7-20.

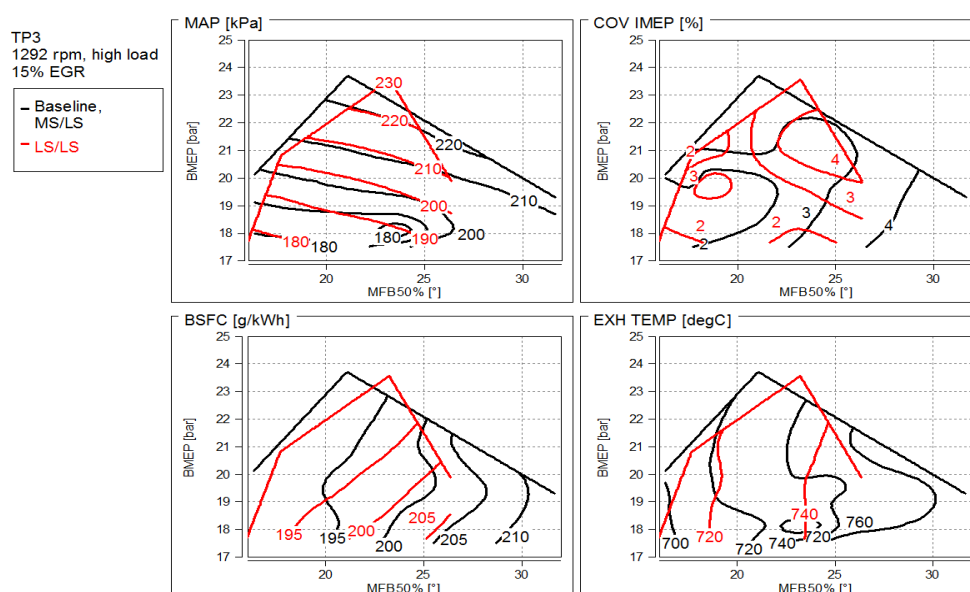


Figure 7-20. Combustion data for 15% EGR for LS/LS compared with Baseline

As was shown for 5% EGR, LS/LS requires similar boost levels as for the baseline, though the flow losses should be lower for a low swirling head according to the theory. This indicates that there are other losses in the system counteracting the benefit of reduced flow losses that should be a result from the LS/LS setup. The need for boost with 230 kPa is due to the retarded spark timing for LS/MS. The fuel consumption is also reduced under high load conditions, but not for the later spark advance and loads below 20 bar BMEP, where the fuel consumption is slightly higher.

Increasing the swirl level for HS/LS resulted in decreased peak torque level by almost 4 bar BMEP, compared to the baseline, Figure 7-21. This BMEP level is however still higher than for the maximum BMEP delivered by the standard engine described in Appendix A: Figure 11-1. This setup is not on par with the other hardware setups. The figure shows a higher need for boost, but also a higher instability in COV, compared to the baseline. Fuel consumption is maintained though, with less thermal load on the engine.

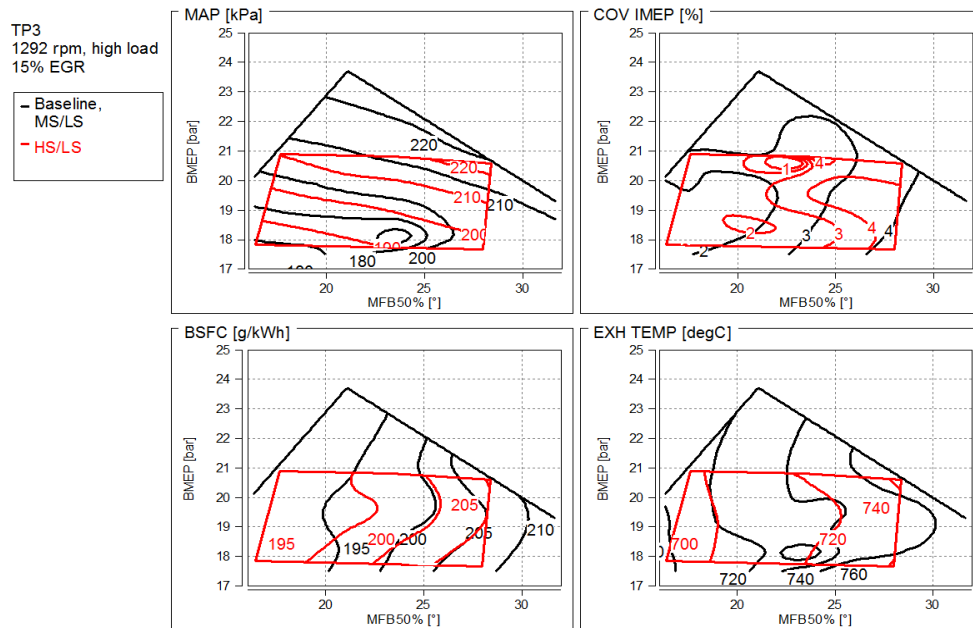


Figure 7-21. Combustion data for 15% EGR for HS/LS compared with Baseline

Changing setup to the mid squish piston rendered lower peak performance compared to the LS/LS setup, Figure 7-22, but still slightly higher compared to HS/LS. The engine under this configuration needed less boost for the equivalent load and also ran smoother in regard to the COV. Lower thermal load at these high torque outputs was also noticed, though quite small, together with a reduced fuel consumption.

For each of these hardware variations the combustion duration was extended as the spark timing was retarded to reach the measuring values. At knock limit the combustion duration did not change much, despite the higher load.

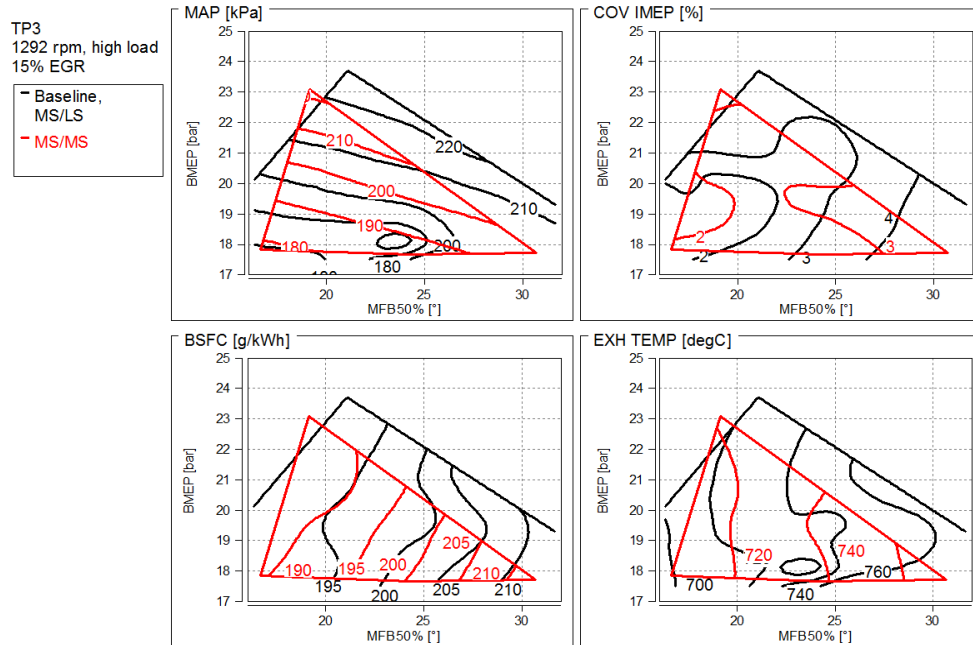


Figure 7-22. Combustion data for 15% EGR for MS/MS compared with Baseline

Continuing onto the last piston variation of the MS/HS, the data shows that this setup is not capable of increasing the load over the baseline. The higher loads from this setup is also far into the realm of instability, Figure 7-23. The need for boost is quite similar to the other standard swirl combination, but despite this, the fuel consumption is higher than for those combination.

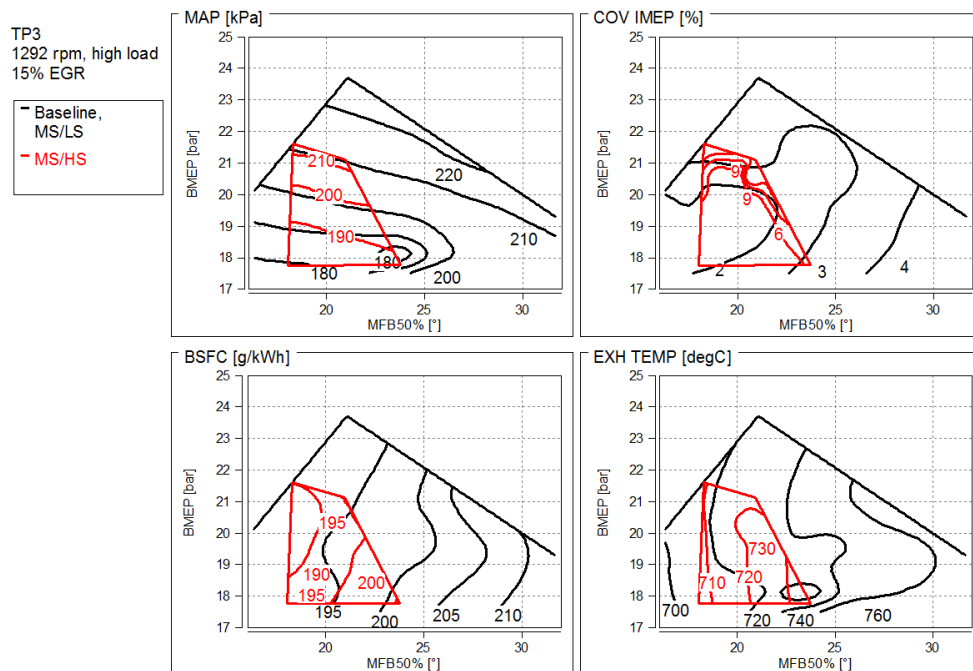


Figure 7-23. Combustion data for 15% EGR for MS/HS compared with Baseline

7.1.3 EGR Variation, low load

By running variations of both MFB50 and EGR gave the mapping in Figure 7-24 to Figure 7-27. From this it is possible to investigate the combustion stability in regard to EGR capability in the combustion system. It also shows how the engine runs for those operating conditions that require spark retardation, should there be a need to heat up the catalyst to reach light-off. The mapping also shows how much this would cost in respect to fuel consumption under these load conditions.

Running the engine with the low swirling head did prove beneficial NO_x, HC and CO₂ for the P3 operating point. This point is also used for this EGR test. The results in Figure 7-24 show that the low swirling head requires higher MAP, which reduces the pumping losses in the engine since this is a throttles operating point. Adding more EGR does however reduce some of the fuel consumption benefit of the lower swirling head. Adding more EGR also lowers the exhaust temperature slightly, as did retarding the spark.

Increasing the swirl showed a larger benefit compared to the standard configuration, both in fuel consumption and engine stability, Figure 7-25. Since the MAP did increase a lot, the reduced pumping losses is a probable cause for this improved fuel consumption. The exhaust temperature did decrease compared to the standard setup, when adding more EGR, as well as lowered as the spark was retarded.

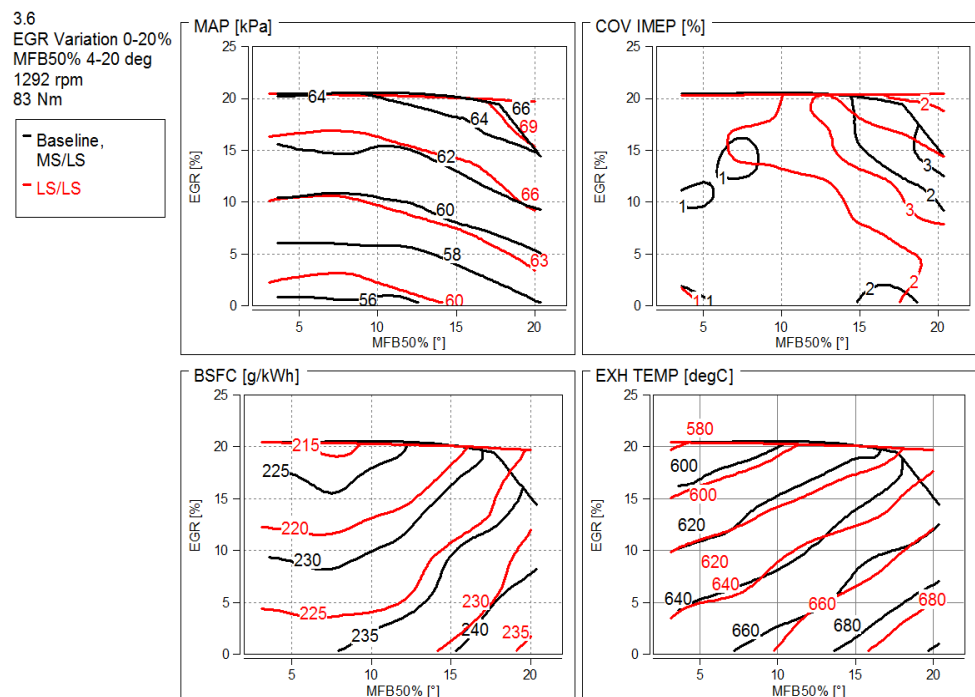


Figure 7-24. EGR variation 0-20% EGR, low load. LS/LS

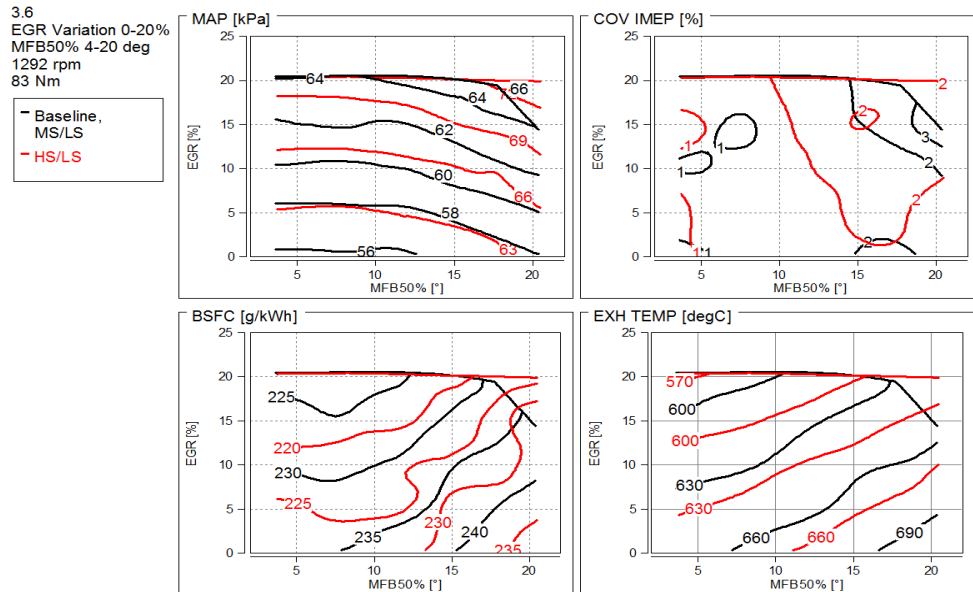


Figure 7-25. EGR variation 0-20% EGR, low load. HS/LS

Moving on to the MS/MS setup Figure 7-26 show that the increase in MAP pressure is not increased as much as in the swirl variation, but did still have a comparable impact on the fuel consumption. The engine did however not run as stable when adding more EGR and spark retardation. The exhaust temperature did not change much compared to the standard setup.

For the last variation of MS/HS the data show that MAP increases by 3-4 kPa over the whole range, as it did for MS/MS. However, it didn't not lower the specific fuel consumption as much as for this setup. A reason for this may be the increased squish area requiring extra force to move the air stream further into the center axis of the piston. The COV is similar to that of the MS/MS setup and the exhaust temperature follows to great extent that pattern as well.

3.6
EGR Variation 0-20%
MFB50% 4-20 deg
1292 rpm
83 Nm

— Baseline,
MS/LS
— MS/MS

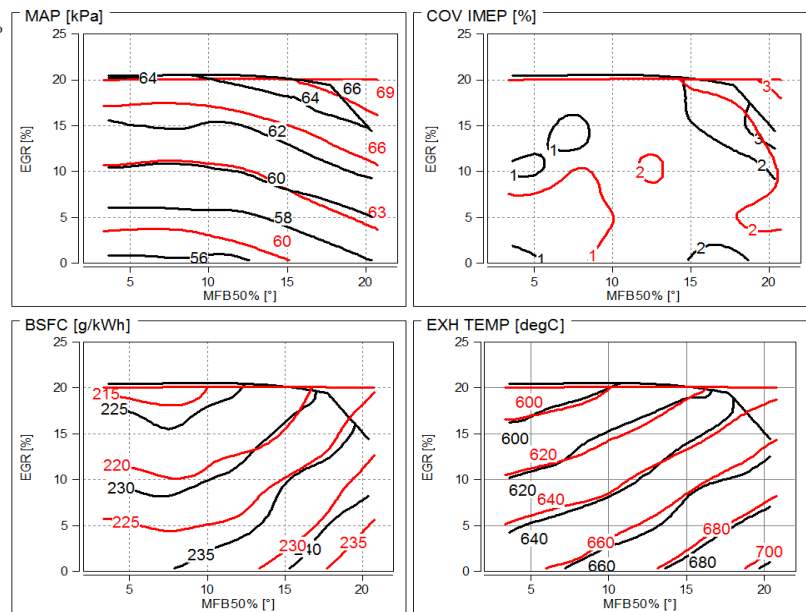


Figure 7-26. EGR variation 0-20% EGR, low load. MS/MS

3.6
EGR Variation 0-20%
MFB50% 4-20 deg
1292 rpm
83 Nm

— Baseline,
MS/LS
— MS/HS

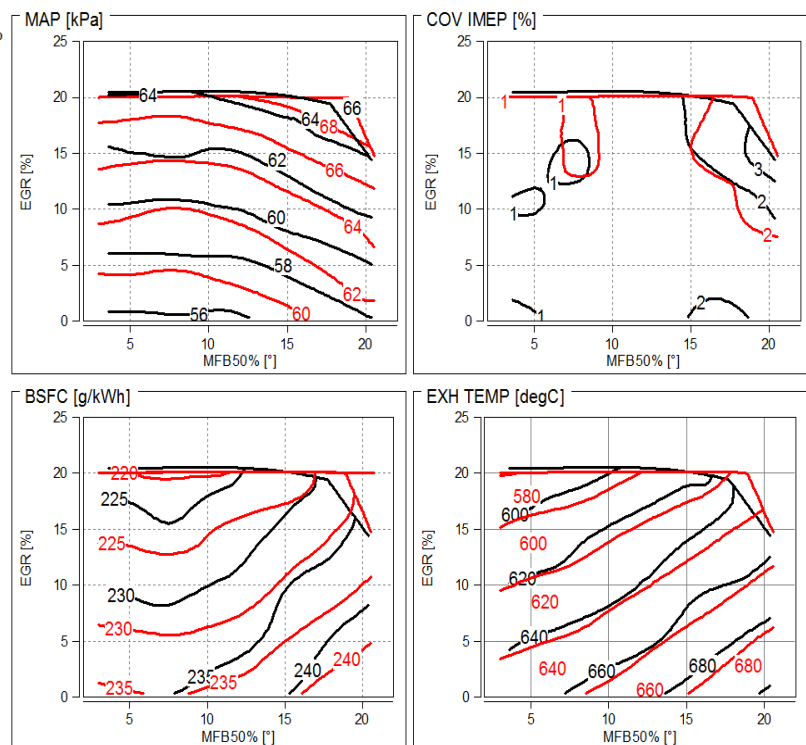


Figure 7-27. EGR variation 0-20% EGR, low load. MS/HS

8 Conclusion

The literature study showed a link to previous lean burn concepts and which piston bowl designs have been tested previously over the years. From the lessons learned in these reports it is clear that there are two theories on which combustion model gives the best results. The first model advocates a slower combustion process with low turbulence and flow velocities in order to reduce the heat losses in the system. The other theory is to increase the turbulence and flow velocities to increase the combustion speed. This reduces cycle-to-cycle variations and avoids the risk of knocking by combusting the fuel prior to the knock. However, the NO_x emissions were shown not to be substantially different between these two theories, which should be the case for the creation of thermal NO_x at high temperature and pressures. The swirl assisted combustion system which was used in the baseline has an open bowl piston intended to investigate these two strategies by altering swirl and squish in order to improve stoichiometric combustion. As a goal for the project, the engine was set out to deliver an increase in torque output by 20% while increasing the efficiency by 10%, compared to the baseline engine. To reach this goal, a test matrix with varied squish-to-bore ratio and swirl variation was introduced to evaluate the performance of the engine in regard to efficiency, emissions, performance and EGR compatibility. The latter is crucial as the increased EGR capability in the combustion system is used to mitigate knock as well as suppression the creation of thermal NO_x.

The results from the engine testing show that for the initial base operating points the hardware variations indicate that there is room for more improvement if the engine were to be calibrated for these new operating conditions. Even without calibration, running the engine against the same reference data points, there were improvements. The BSFC was shown to improve by up to 3,1% by the hardware change alone.

To further improve the efficiency of the engine and reach the target of 10% increase in efficiency the engine was run lean with EGR to reduce the increase in NO_x. This showed an increase in efficiency of 7,3% at high load. Adding the hardware improvement in an extrapolation gives a potential of reaching the 10% efficiency increase at MaxPwr for hardware combination LS/LS, Low Swirl/Low Squish.

It can be concluded that the effect of the hardware on the emissions was visible, even without any calibration of the engine at the operating points. Altering the swirl level had an effect on NO_x, CO, CO₂ and to some extent HC as well. Both lowering and increasing the swirl showed on lower emissions compared to the mid swirl of the baseline. Altering the squish parameter showed that the lowest data recordings came from the MS/MS, Mid Swirl/Mid Squish, setup. All the operating points followed similar pattern for the different hardware configuration, except for P2. The results differed greatly from the patterns and the reason behind this needs further investigation to why this is.

Investigating the high load performance for 5% EGR showed that the alterations in swirl had greater effect on the fuel consumption than the alterations in squish. Overall, the LS/LS hardware setup, together with MS/MS showed the best results.

Adding more EGR, increasing up to 15%, erased this increased efficiency and all the setups delivered similar or worse data, due to the increased boost needed for the added EGR. LS/LS and MS/MS still deliver torque levels similar to the hardware baseline operation, whereas the two other hardware setups delivered data lower than baseline, but still higher than the reference data. One reason for why the MS/HS, Mid Swirl/ High Squish, setup did not perform as it was believed to do could be due to the deeper piston bowl. The bowl depth does allow larger size tumble motion from the squish to form which is not beneficial for the overall mixture. This tumble should be broken down into smaller vortices to aid gas mixing. A future research topic could be to visually simulate these results through CFD and verify the results again.

The low load EGR variation was intended to investigate the possibility of increasing EGR in order to reduce pumping losses as well as having a combustion system that can perform with high levels of EGR. This variation shows that there is a possibility to increase the efficiency of the engine by 4% when adding up to 20% EGR with maintained operational stability. However, it is possible that this high level of EGR may not be possible to deliver from the peripheral systems but that is not a question to be answered in this report. The sweep in both MFB50 and EGR gave an operational map of how the engine could be operated. The sweep in MFB50 showed that it is possible to use a late spark to heat up the catalytic converter while still maintaining operational stability, giving more degrees of freedom when calibrating the engine.

To reach the project goal, the best setup which reaches the target has been the MS/LS and LS/LS setup which performed well under the different tests. The goal of increasing torque by 20% is reached by these two hardware configurations. It is also possible to reach the target with the baseline engine configuration but as the tests have shown this would come at a cost for emissions and fuel consumption.

9 Nomenclature

AdBlue	Ammonia additive for Selective Catalytic Reduction	ICE	Internal Combustion Engine
BMEP	Brake Mean Effective Pressure		
BSFC	Brake Specific Fuel Consumption	LPG/LNG	Liquefied Petroleum Gas Liquid Natural Gas
CAD	Crank Angle Degree	MN	Methane Number
CFD	Computational Fluid Dynamics	PLIF	Planar Laser-Induced Fluorescence
CNG	Compressed Natural Gas	SCR	Selective Catalytic Reduction
COSTGAS	Combustion System Development for Next Generation HD Gas Engines	SI	Spark Ignition
CO HC NO _x PN/M	Carbon Oxide Hydro Carbon Nitrous Oxides Particle Number/Matter	(f)TDC	(Firing) Top Dead Centre
EATS	Exhaust gas After-Treatment System	TN SN	Tumble Number Swirl Number
ECU	Electronic Control Unit	TKE	Turbulent Kinetic Energy
EGR	Exhaust Gas Recirculation	TG	Tokyo Gas
Euro V/VI	European emissions regulation class V/VI	WHSC	World Harmonised Stationary Cycle
IANGV	International Association for Natural Gas Vehicles	WHTC	World Harmonised Transient Cycle

10 Contact information

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Appendix A: WHSC Operating data

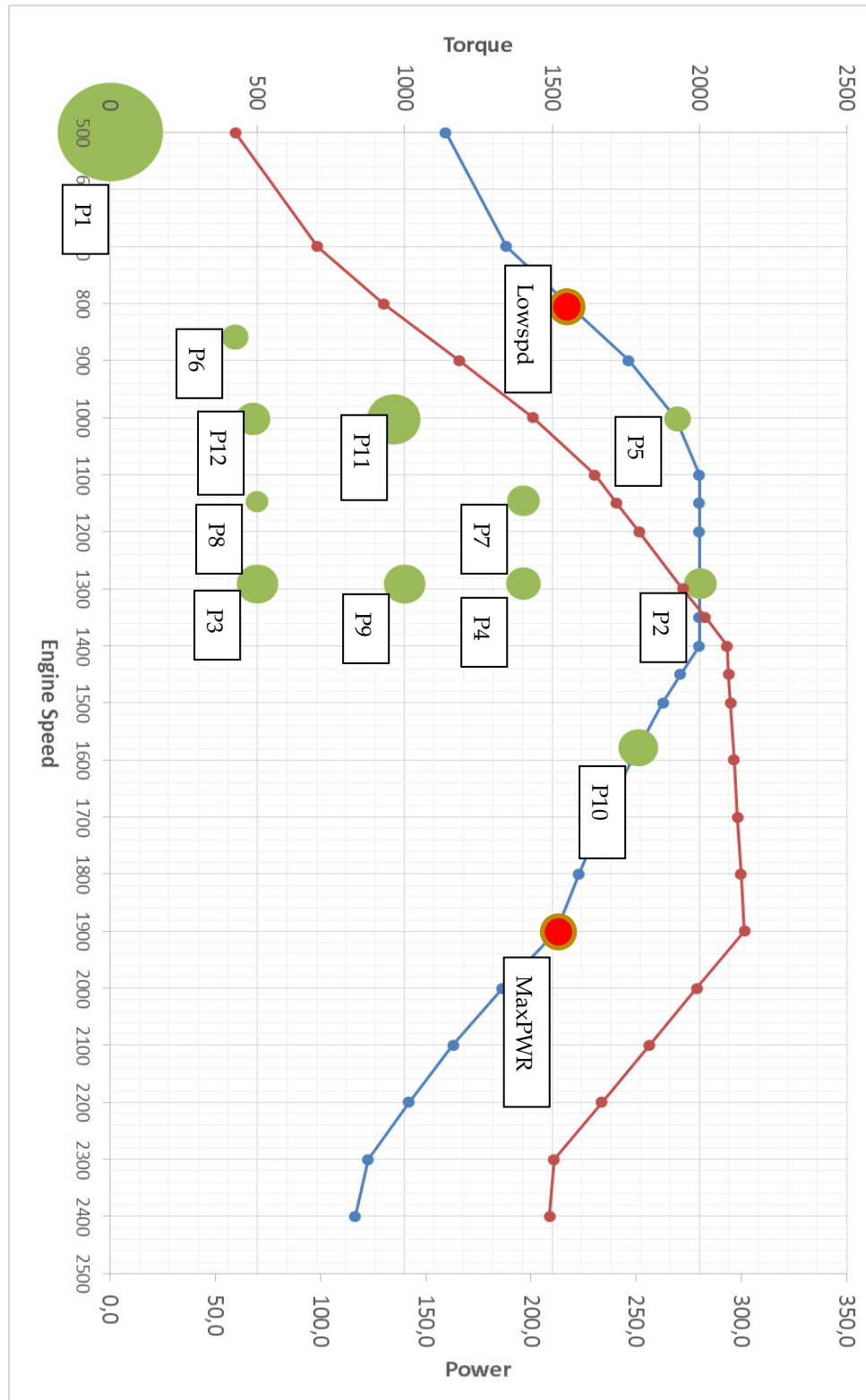


Figure 11-1. Full engine WHSC operating points

Appendix B: WHSC Numerical Data

Table 8. WHSC cycle data for SCRE engine

	Torque, 6-cyl	Torque, 1 cyl SCRE	Speed
Operating points	[Nm]	[Nm]	[RPM]
P1	0	0	500
P2	2000	333	1292
P3	500	83	1292
P4	1400	233	1292
P5	1923	321	1004
P6	421	70	860
P7	1400	233	1148
P8	500	83	1148
P9	1000	167	1292
P10	1790	298	1580
P11	962	160	1004
P12	481	80	1004
MaxPWR	1516	253	1900
Lowsped	1553	259	800

COMBUSTION SYSTEM DEVELOPMENT FOR THE NEXT GENERATION HD GAS ENGINES

COSTGAS is a project in Heavy Duty gas engines in which the combustion system for the next generation gas engines is to be developed. The goal is to increase the efficiency of the current gas engine platform by 10 percent and increase the torque by 20 percent. This is done while observing the boundary conditions of the current Euro VI emissions regulations.

The research was performed on a single cylinder research engine on which different hardware combinations with different variations to the piston and cylinder head was investigated. The results showed an increase in efficiency by up to 10 percent at high load and a potential of 5 percent increase for lower loads. Altering the hardware combinations showed a potential of lowering the emissions from the engine as well.

The large increase in efficiency was reached by using both lean operation methods as well as EGR. The latter not only increased efficiency but reduced the initial NO_x that comes under lean operation. The increase in torque was shown to be possible to deliver for certain hardware combinations.

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