SGC Rapport 2013:286

Extending the performance, fuel efficiency and stability of stoichiometric spark ignition natural gas engines – Gas engine research at KCFP 2007–2012

(Utökad prestanda, bränsleeffektivitet och stabilitet för stökiometriska naturgasdrivna tändstiftsmotorer – Gasmotorforskning på KCFP 2007–2012)

Mehrzad Kaiadi, Per Tunestål, Ashish Shah

"Catalyzing energygas development for sustainable solutions"

SGC Svenskt Gastekniskt Center AB

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Denna studie har finansierats av:
Energimyndigheten
E.ON Gas Sverige AB
Göteborgs Energi AB
Lunds Energikoncernen (publ) AB
Öresundskraft AB
Scania CV AB
Saab Automobile Powertrain
Volvo Car Corporation AB
Volvo Powertrain AB
AB Volvo Penta
Cargine Engineering AB
Caterpillar Inc.
Chevron Energy Technology Company, a division of Chevron U.S.A. Inc.
Finnveden Powertrain AB
Hoerbiger Control Systems AB
National Instruments Sverige AB
Toyota Motor Corporation
Wärtsilä Finland Oy

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Malmö 2013

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Malmö, Sweden 2013

Martin Ragnar
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Authors' foreword

The gas engine project at Lund University has previously explored extending the performance, fuel efficiency and stability of SI natural gas engines. The first NG engine activities started in the beginning of the 1990s. In the first and second phase of the natural gas engine project a lot of time was spent to measure turbulence and to investigate how the combustion chamber design influenced combustion parameters and emissions. Those experiments were applied on a single-cylinder Volvo engine TD102.

The third phase of the natural gas engine activities started from 2000 and ended in 2003. A 6-cylinder turbocharged Volvo engine (i.e. TG103) was used in this phase. Different experiments were performed including investigating different locations for fuel injection, investigation of cylinder-to-cylinder and cycle-to-cycle variations in a Lean Burn natural gas engine and a study to compare Lean Burn natural gas engine versus stoichiometric operation with EGR. The results from this phase showed that the stoichiometric operation is a better choice than lean operation since by using a three-way catalyst the emissions level can be kept at very low levels and differences in efficiency are not significant.

The fourth phase of the project started in the beginning of 2007. Based on the results from phase three, the engine operating concept was decided to be stoichiometric and the research activities focused mostly on the problems associated with this type of operation. Extending the dilution limit of the engine and developing closed-loop control to operate the engine at its dilution limit has been the main method to improve throttle losses. A new method for calculating cyclic variation was developed that significantly improved the transient capability of the engine control system. The method consequently applied on a closed-loop dilution limit control which resulted in improvement in specific fuel consumption with acceptable transient capability. Ion current signals were studied at different operation condition. Cyclic variation of ion current integral was found to be a robust combustion stability parameter which can be used as more economical alternative to cyclic variation derived from pressure sensors. Moreover, the key features to improve the engine performance are identified as, right amount of EGR at different operating regions, right compression ratio, Variable Geometry Turbocharger (VGT), high turbulent pistons, long route EGR system and model-based control. The started research activities from phase four prolonged and a part of that performed in phase five.

The existing phase started in the beginning of 2010 and it is expected to end in 2012. Since the beginning of 2011 the project mainly focuses on exploring alternative ignition techniques as after completion of all previous phases it was observed that the capability of conventional spark plug ignition system was the factor limiting the extent of dilution and hence emission reduction and efficiency improvement. Two most feasible alternative ignition systems were identified, namely diesel pilot injection and pre-chamber type ignition system but it was soon realized that the former has already received considerable attention as there are products in the market under different names like The Hardstaff OIGI® (Oil Ignition Gas Injection), Westport's High-Pressure Direct Injection (HPDI) applicable to a wide range of engines. Comparatively, however, the concept of pre-chamber ignition has received limited attention and is mainly applied to stationary or large bore marine engine which do not face as severe speed and load transients as experienced by a heavy duty engine for mobile application. Reasons behind this are believed to be limited knowledge about the mechanism of ignition resulting from a pre-chamber type ignition device and hence gaining deeper insight into this mechanism is currently the objective of the gas engine project.

This report gives a summary of the results from the first three phases of the gas engine activities at Lund University but the main focus of the report is on phase four and five.

To the project a reference group has been linked consisting of the following persons;
Mattias Svensson and Corfitz Nelsson, Swedish Gas Technology Centre
Ingemar Magnusson, Volvo Global Trucks Technology
Folke Fritzson, Scania
Kenth Svensson, Caterpillar
Jari Hyvönen, Wärtsilä

Summary

Most heavy-duty engines are diesel operated. Severe emission regulations, high fuel prices, high technology costs (e.g. catalysts, fuel injection systems) and unsustainable fuel supplies are enough reasons to convince engine developers to explore alternative technologies or fuels. Using natural gas/biogas can be a very good alternative due to the attractive fuel properties regarding emission reduction and engine operation. Heavy-duty diesel engines can be easily converted for natural gas operation which is a very cost effective process for producing gas engines. However, due to the high throttle losses and low expansion ratio the overall engine efficiency is lower than the corresponding diesel engines. Moreover the lower density of natural gas results in lower maximum power level.

In this project (phase 4) key features and strategies which result in improved efficiency, increased maximum power and improved transient capability of heavy-duty natural gas engines have been identified, validated and suggested. High EGR rates combined with turbocharging has been identified as a promising way to increase the maximum load and efficiency of heavy-duty gas engines. With stoichiometric conditions a three way catalyst can be used and thus regulated emissions can be kept at very low levels. Obtaining reliable spark ignition is difficult however with high dilution and there will be a limit to the amount of EGR that can be tolerated for each operating point. Extending the dilution limit of the engine and developing closed-loop control to operate the engine at its dilution limit has been the main method to reduce throttle losses.

Information about the combustion stability can be derived either from pressure or ion-current data measured inside the cylinder. The reliability and robustness of the combustion stability parameters derived from pressure and ion-current data were validated in separate studies. A new method for calculating cyclic variation was developed that significantly improved the transient capability of the engine control system. The method consequently applied on a closed-loop dilution limit control. Only applying closed-loop control to operate the engine at its dilution limit resulted in 4.5% to 8% improvement in specific fuel consumption at 1200 RPM.

The dilution limit can also be extended by replacing the combustion chambers with high turbulence pistons which enhances the combustion. New pistons were designed to increase the turbulence level to shorten the combustion duration.

The new piston modification resulted in changes in the exhaust gas characteristics. These changes, together with the fact that Variable Geometry Turbocharger (VGT) has big potentials to reduce further the throttle losses were the motivation to replace the turbocharger with a well-matched VGT. It was demonstrated that VGT can be used instead of throttle in more than 60% of the whole operating range of the engine meaning no throttle losses in this area.

Since reliable spark ignition is challenging at high boost and dilution levels pre-chambers have been considered as a solution in phase 5 of the project. Unfueled pre-chamber spark plugs have been evaluated and found to provide marginal improvement of the dilution limit. Problems with pre-ignition limit their usefulness at high load however. Ongoing work with fueled pre-chambers should provide better opportunities.

In summary the key features to improve the performance of a stoichiometrically operated natural gas engine are identified as: right amount of EGR at different op-

erating regions, right compression ratio, VGT, high turbulence pistons, powerful ignition, long route EGR system and model-based control. Applying the combination of these strategies resulted in up to 10% improvement in specific fuel consumption while preserving combustion stability.

Sammanfattning på svenska

Majoriteten av tunga motorer är idag dieselmotorer. Strikta emissionskrav, höga tillverkningskostnader (t.ex. katalysatorer, bränsleinsprutningssystem) och icke hållbar bränsleförsörjning är tillräckliga orsaker för att övertyga motorutvecklare att undersöka alternativa teknologier och bränslen. Naturgas/biogas kan utgöra ett mycket konkurrenskraftigt alternativ pga. gynnsamma bränsleegenskaper för motordrift och minskade emissioner. Det är relativt okomplicerat att konvertera tunga dieselmotorer till ottomotorer med naturgasdrift och därigenom är detta ett kostnadseffektivt sätt att producera gasmotorer. Pga. höga strypningsförluster och lågt expansionsförhållande blir emellertid energieffektiviteten lägre än för motsvarande dieselmotor. Dessutom ger naturgasmotorn lägre maximal effekt på grund av naturgasens låga densitet.

Projektet har identifierat och validerat nyckelområden för ökad energieffektivitet, effekt och transientprestanda för tunga gasmotorer. Hög halt av avgasåterföring (EGR) kombinerat med turboladdning kan höja såväl effekt som energieffektivitet. Med stökiometriskt bränsle-/luftförhållande kan dessutom en trevägskatalysator användas och därigenom når man mycket låga emissionsnivåer. Med hög utspädning blir det svårare att åstadkomma tillförlitlig antändning och därför finns det en utspädningsgräns för varje arbetspunkt som inte får överskridas. Höjning av utspädningsgränsen och utveckling av återkopplad reglering för att köra motorn på utspädningsgränsen har varit huvudspåret för att minska strypningsförlusterna.

Information om förbränningsstabiliteten kan erhållas antingen från tryckmätning eller från jonströmsmätning i cylindern. Tillförlitlighet och robusthet för stabilitetsparametrar baserade på båda dessa mätmetoder har validerats i separata studier. En ny metod för att beräkna cykel-cykelvariation under transient drift har utvecklats och ledde till en drastisk förbättring av regleringen. När återkopplad reglering av utspädningen infördes ledde detta till en minskning av bränsleförbrukningen med 4.5-8 % vid 1200 RPM.

Utspädningsgränsen kan utökas genom förändring av kolvgropen till en utformning som ger högre turbulens och därigenom snabbare förbränning. De nya kolvarna sänkte avgastemperaturen och ledde till lägre laddtryck. Detta motiverade ett byte av turboaggregat till en välmatchad VGT (variabel turbingeometri). En VGT kan dessutom ersätta trotteln för lastreglering i 60 % av arbetsområdet vilket gör att strypningsförlusterna försvinner helt i detta område.

Eftersom tillförlitlig antändning är utmanande vid högt laddtryck och kraftig utspädning har förkammartändning studerats i projektet. Förkammartändstift utan separat bränsletillförsel har provats och visat sig ha viss potential för utökad utspädningsgräns. Det som begränsar deras användbarhet är problem med glödtändning vid hög last. Pågående studier av förkammare med bränsletillförsel bör erbjuda bättre möjligheter.

Sammanfattningsvis har följande nyckelområden identifierats för att förbättra prestanda och energieffektivitet för gasmotorer: rätt EGR-halt för varje arbetspunkt, rätt kompressionsförhållande, VGT, högturbulent förbränningsrum, kraftfull antändning, lågtrycks-EGR och modellbaserad återkoppling för styrningen. När alla dessa åtgärder kombineras erhålls upp till 10 % minskning av bränsleförbrukningen med bibehållen förbränningsstabilitet.

Innehåll

1. Background

As it was described in the “foreword” this report focuses on the gas engine project activities from phase 4 and 5 however in order to keep a track on the previous gas engine activities at Lund university, the report is started with a summary of the three first phases.

1.1 Executive summary of phase 1, 2 and 3

1.1.1 Phase 1 (1994-1997) & phase 2 (1998-2000)

The main focus of these phases was to study the influence of different combustion chamber design on combustion parameters as well as emissions.

In-cylinder flow measurements are made on six different combustion chambers in a single-cylinder[1] (Volvo TD102) engine, using Laser Doppler Velocimetry (LDV) technique. Cylinder pressure and emissions are also measured and evaluated in terms of heat release and indicated specific emissions. The flow measurements show a large difference in mean velocity and turbulence between the six pistons. A high turbulence combustion chamber (Quartette) identified which showed fast main combustion, due to the high turbulence peak located close to top dead center. This piston showed to be more tolerant to diluted mixtures in terms of engine stability than the original combustion chamber (Turbine), thus more suitable for lean burn operation. The results from these studies resulted in several technical papers and scientific reports. Some of the results from these phases are reported in [2], [3] and [4].

1.1.2 Phase 3 (2001-2003)

The engine used in phase 1 and 2 of the gas engine project is based on a rather old diesel engine design originating from the 1960’s. The two valve cylinder head was not the optimum and also the crevice volumes between piston and cylinder liner were large. This means breathing problems and large amounts of unburned hydrocarbons. It was thus time to replace the engine with a newer design. A new engine was supplied by Volvo (i.e. 6-cylinder turbocharged Volvo engine TG103) for performing some research on a multi-cylinder engine. Some of the results from this phase of the gas engine activities can be found in [5], [6] and [7]. Summary of the different moments of this phase are presented as follow:

Multi-cylinder tests with various spark gaps and fuel injection locations

Tests are performed on a turbocharged multi-cylinder version of the TD102 engine (TG103). Pulsed single-point fuel-injection close to the cylinders (at the throttle) led to large cylinder to-cylinder variations in lambda, resulting in variations in IMEP between the cylinders. Cylinders with richer mixtures have higher NOX emissions, and cylinders with leaner mixtures have higher cycle-to-cycle variations. Placing the fuel injection before the turbo compressor led to much lower cylinder-to-cylinder and cycle-to-cycle variations, the maximum load was however reduced. Late ignition timing, high boost pressures and lean mixtures led to the need for a small spark gap to avoid misfires. A larger gap improves combustion quality at idle, due to the higher spark energy with a larger gap.

[1] 6-cylinder engine with one of the cylinders operational and the other 5 motored

Base engine performance and ion-current measurements

The original ignition system was replaced by a SAAB-TK4 ignition system. With this system the ion-current signal can be measured, using the spark plugs as sensors. Various air/fuel ratios and ignition timings are tested to see the influence on combustion and emissions, compared to the standard settings of lambda and ignition timing. Cycle-to-cycle variations increase rapidly with standard ignition as the mixture becomes leaner. MBT ignition is much more tolerant to highly diluted mixtures. HC and CO emissions are slightly lower with standard ignition timing because of more post-oxidation, due to higher expansion and exhaust temperatures. The NOX emissions are up to three times higher with MBT ignition since the maximum cylinder temperature increases. The maximum load is increased with MBT ignition, as expected. Also efficiency and main combustion speed increase with MBT ignition timing. The ion-current measurements show a strong correlation between variations in IMEP and variations in the current signal, averaged over 300 cycles.

New engine control system

A new engine control system was installed. The open-source code made it possible for us to implement new control features, such as closed-loop lambda control, engine dynamometer control, cylinder-individual fuel-injection duration and timing, cylinder balancing, and much more. A powerful ignition system and ion-current measurement/evaluation in all cylinders are also valuable features of this system. Idle quality and maximum load was improved compared to the original control/ignition system.

Combustion chamber effects

Quartette and Turbine combustion chambers are compared in the multi-cylinder engine, with closed-loop lambda control and fuel injection before the turbo compressor. Various air/fuel mixtures and ignition angles are tested at 12 bar BMEP, 1200 rpm. Quartette has much faster main combustion and less cycle-to-cycle variations than Turbine, the same results as in the single-cylinder tests. The HC emissions are slightly higher with Quartette piston; this was not the case in the single-cylinder experiments. One reason may be that those tests were made without a turbocharger; the boost was generated by an external compressor. This means that the post-oxidation of HC in the exhaust manifold before the exhaust turbine should be higher with the Turbine combustion chamber, since the temperature is higher with this slow burning combustion geometry. No clear trend in NOX emissions, at MBT ignition, can be seen between the two pistons. The efficiency is the same, or slightly lower, with Quartette.

Port fuel-injection and cylinder balancing

A slight variation in fuel flow between the cylinders can be a problem when operating close to the lean limit, since a small increase in air/fuel ratio will cause poor stability in that cylinder. Variations in airflow between the cylinders may also cause the same problem, even if the fuel flow is the same in all cylinders. One way to overcome this problem is to measure the combustion in all cylinders, using the ion-current signal. The results show that both cylinder to cylinder and cycle-to-cycle variations are decreased with ion-current cylinder balancing.

Stoichiometric operation with EGR versus lean burn operation

Port fuel injection, and Quartette combustion chamber, is used in these tests. Emissions, combustion and ion-current signal are evaluated, comparing lean burn vs. EGR operation. The raw emissions of HC and NOX are lower with EGR operation. CO emissions are higher due to stoichiometric combustion. HC emissions after the three-way catalyst are 10 to 30 times higher at lean burn operation than with EGR. The NOX emissions are up to 700 times higher after the catalyst with lean burn compared to EGR operation. The CO emissions are 10 times higher with EGR, after the catalyst compared to lean operation.

The early combustion duration (ignition to 5% burnt) is much longer for the EGR case, since EGR has a stronger influence on the laminar flame speed than excess air. The main combustion duration is however similar for both EGR and lean operation. The high turbulence peak close to TDC (with this combustion chamber) may explain the fast main combustion for both EGR and lean operation. The brake efficiency is slightly lower with EGR compared to a high-efficiency lean burn strategy, mostly because of the high CO emissions. Higher load can be achieved with EGR compared to the lean burn cases.

The ion-current signal is very weak at lean operation. A strong signal can be found at stoichiometric operation diluted with EGR though, which can be utilized for closed-loop control.

1.2 Phase 4 & 5 background (2007-2012)

Phase four (2007-2009) started based on the results from phase three, the engine operating concept was decided to be stoichiometric and the research activities focused mostly on the problems associated with this type of operation. The studies from this phase extended partly to the fifth phase of the project (2010-2012).

From phase three it was concluded that the stoichiometric operation of the natural gas engine should be prefered than the lean burn operation since with stoichiometric operation the cost effective 3-way catalyst can be used resulting in ultra-low engine-out emissions and the engine efficiency with diluted operation was comparable to lean burn operation. However due to several reasons the overall efficiency of the stoichiometric operated natural gas engines are significantly lower than the corresponding diesel engines.

Normally stoichiometric operated natural gas engines use a throttle to regulate demanded power. Use of a throttle introduces pumping losses which are particularly significant at low loads. Beyond that, the high octane number and high auto-ignition temperature of natural gas makes Compression Ignition (CI) an ill-suited concept. Normally for stoichiometric operation of natural gas Spark Ignition (SI) operation concept is used which results in use of lower compression ratio compared to diesel engines and thereby the lower efficiency achieves.

Furthermore, since diesel engines operate lean, the exhaust gas temperature is low and the construction materials do not tolerate high exhaust gas temperatures. The converted engines have more stringent limitations in terms of high exhaust gas temperatures than dedicated SI engines.

In summary, ***throttle losses***, ***lower compression ratio***, ***sensitivity to high exhaust gas temperature*** and ***lower fuel density*** are main parameters which results in lower heavy-duty natural gas engine performance compared to diesel engines.

1.2.1 Objectives of the project

The main objective of the project during 2007-2010 was to develop and apply new strategies to improve the overall engine *efficiency*; extend the *maximum power* level and improve the transient capability of the engine. Comparing a heavy-duty natural gas engine to a corresponding diesel engine shows that:

- Overall engine efficiency is lower due to throttle losses and lower compression ratio.
- Maximum power level of a natural gas engine is lower due to the lower gas density and knock phenomena.
- Transient capability of the engine is limited due to the diluted operation of the engine and the small lambda window.

1.2.2 Scope of the project

The scope of the project was as follow:

- Use stoichiometric operation with three-way catalyst
- Explore diluted operation with large EGR fraction in order to increase efficiency and suppress knock
- Employ in-cylinder sensing and control to limit cyclic, cylinder-to-cylinder and lambda variations
- Explore engine modification to improve efficiency and extend the maximum load level
- Explore Hythane as an alternative fuel to natural gas

1.3 Outline of the report

Chapter one supplies a background to the gas engine project at Lund University. This chapter starts with a summary of the previous work performed at the department and continues with presenting the objectives and the scopes of the latest phase of the project. In chapter two the experimental engine, measurement system and control system used during phase 4 and 5 of the project are discussed. In chapter three a dilution limit control was designed and evaluated and the importance of using model-based control strategy is highlighted. In chapter four the needed engine modification to improve engine performance is discussed. In chapter five the effect of Hythane on combustion is discussed. Chapter six covers experimental work with prechamber spark plugs for improved ignition quality.

The main conclusions from phase four and five are discussed in chapter seven. In chapter eight the planned work is listed followed by references.

2. Experimental setup

2.1 The test engine

The project has had an experimental nature. The experimental engine was originally a heavy-duty diesel engine from Volvo Trucks which has been converted for natural gas operation; see Table 1 for engine specification. The engine is equipped with a short-route cooled EGR system and a turbocharger with wastegate.

Table 1. Specification of the engine

Number of Cylinder	6
Displacement	9,4 Liter
Bore	120 mm
Stroke	138 mm
Compression ratio	10,5 :1
Fuel	Natural gas

Originally the engine is equipped with a single-point injection system, with four injectors at the fuel injector assembly. The single-point injection system was replaced by a multi-port injection system. The main reasons for using a multi-port injection system are the possibility of adjusting injection for each cylinder individually (i.e. cylinder balancing) and faster response to throttle changes (i.e. better transient capability). The original fuel pressure system supplies gas at a pressure of approximately 10 bar. The test-bench engine is supplied with natural gas at a pressure of 4 bar, thus the port injection system is equipped with 12 injectors (2 per cylinder) to compensate the lower pressure and cover the whole load range, see. This also makes it possible to operate the engine with two different gaseous fuels simultaneously, in case it is desired.

Cylinder-individual control of fuel injection and ignition is possible with a new platform developed by Hoerbiger Control Systems. Three Cylinder-Control-Modules (CCM) are designed especially for cylinder-individual control of ignition and fuel injection as well as ion current measurements. The modules use the well-known message-based protocol Control Area Network (CAN) for communication. The engine setup is shown in Figure 1. The three CCMs can be seen below the inlet part and the three boxes on top are ion current modules.

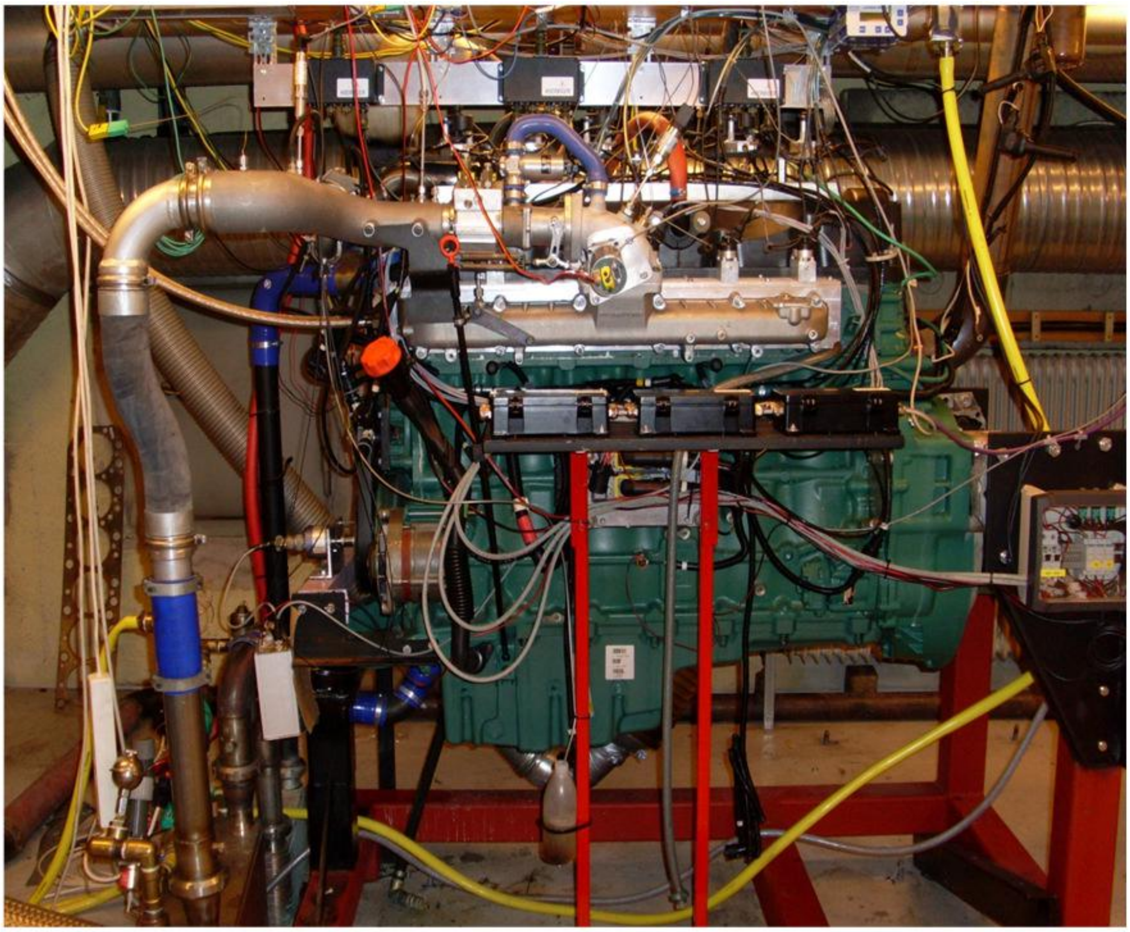

Figure 1. The experimental engine is connected to a dynamometer.

2.2 The engine control system

A master PC based on GNU/Linux operating system is used as a control system. The main program is written in C++ which communicates with the three CCMs for cylinder-individual control of ignition and fuel injection via CAN communication, see Figure 2. Crank and cam information are used to synchronize the CCMs with the crank rotation. Flexible controller implementation is achieved using Simulink and C-code is generated using the automatic code generation tool of Real Time Workshop. The C-code is then compiled to an executable program which communicates with the main control program. The program fullfills the requirements for realtime application.

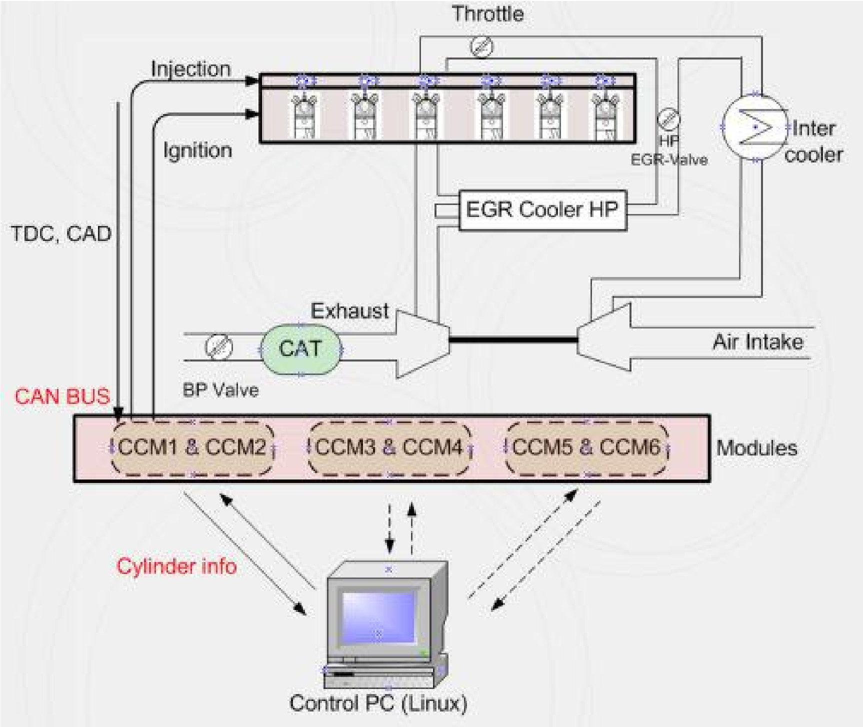

Figure 2. Engine setup structure. A master PC based on GNU/Linux operating system is used as a control system

2.3 Measurement instruments

Different parameters were measured from the engine. Some of them such as crank and cam information and throttle position were needed to operate the control system and the engine. Lots of parameters such as in-cylinder pressure, ion-current, emissions, torque, different flows and temperatures etc. were measured to analyze and evaluate the performance of the engine. Data was sampled with different units, data acquisition cards and with different sample resolution. The master control PC communicates with two other PCs which sample emissions and some other parameters (i.e. slow sampled data) via TCP/IP communication protocol. Figure 3 shows the data flow and communications between PCs and the engine units.

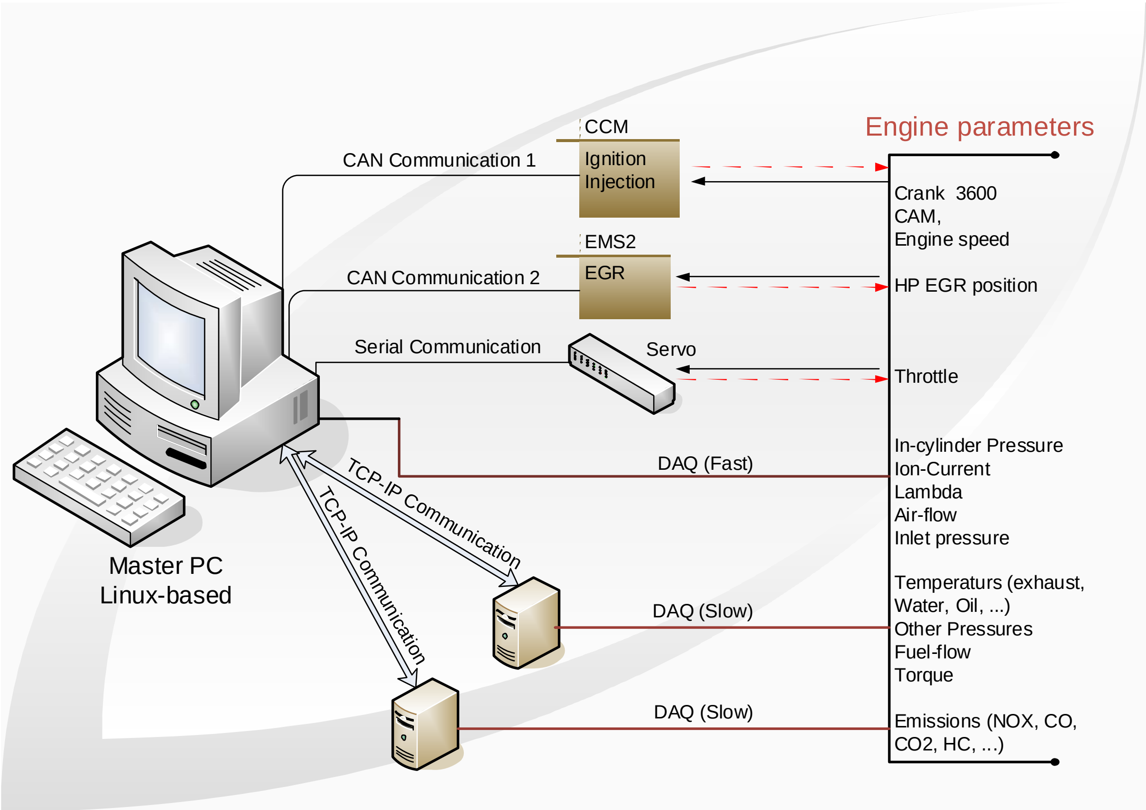

Figure 3 Different parameters were measured from the experimental engine. These parameters are such as in-cylinder pressure, cam and crank position, EGR valve position, throttle position, emission, fuel and air flow etc.

The cylinder head is equipped with piezo electric pressure transducers of type Kistler 7061B to monitor cylinder pressures for heat release calculations. The ion-currents are sampled by CCMs using the spark plugs as sensors. In-cylinder pressure and ion current data are sampled by a Microstar 5400A (DAP 5400A) data acquisition processor which is able to measure 1.25 MS/second (with 8 channels) [8]. To determine the crank shaft position a Leine & Linde encoder is used which provides five pulses per crank angle degree giving a resolution of 0.2 CAD. The in-cylinder pressure and ion current data were sampled with a resolution of 0.2 CAD. Three other parameters, lambda, inlet pressure and air flow were also sampled with DAP 5400A but with lower resolution. Controlling lambda is one of the objectives of this work; thus fast lambda measurement and fuel calculation is essential. To measure lambda a broadband sensor from ETAS is used to provide a faster response than the lambda calculated from exhaust analysis. To measure air flow a Bronkhorst F-107A is used to provide accurate and fast response data for lambda control.

All temperatures were measured by Pentronic Type K thermocouples. Torque is measured from a load cell and fuel flow was measured by F type Bronkhorst flowmeter. All these data were sampled by a HP 3852A data acquisition unit which gives a sample rate of 0.5 Hz. Emissions such as HC, CO, CO_2, NO_2, NO_X and O_2 are measured before and after catalyst. Different analyzers were used to measure the emissions. A summary of the measurement techniques are presented in Table 2.

Table 2. Measurements technique for different emissions

Emission	Measurement technique	Range
NOX	Chemiluminescence Detector (CLD)	0.10000 ppm
HC	Flame Ionization Detector (FID)	0-10000 ppm
CO_2, CO	Non-Dispersive Infrared Detectors (NDIR)	0-10%,0.16%
O_2	Paramagnetic Detector (PMA)	0-25%

EGR was calculated by measuring CO_2 at inlet and exhaust. The latter a NO_X/Oxygen sensor is used to measure oxygen content and the amount of EGR was estimated based on that.

3. Dilution Limit Control

In this chapter an introduction about throttle losses and the effects on part/low load efficiency is presented. Operating an engine at its maximum dilution limit is suggested as a practical solution to decrease the throttle losses and improve the efficiency. To ensure the combustion stability at these operating conditions robust control is necessary. Details about the control structure, experiments and the evaluation of the control performance at steady-state and transient conditions are reported. Moreover a new method to calculate the combustion stability is developed and discussed in this chapter.

3.1 Improving Efficiency at Low/Part load

By operating natural gas engines stoichiometrically a three way catalyst can be used which results in very low emissions however these types of engines suffer from poor gas-exchange efficiency at part or low loads due to high throttling losses. EGR is a well-known practice to improve engine fuel economy, decrease knock tendency and reduce raw NO_X emissions in certain operating regimes. Using EGR results in improved fuel economy due to the following facts:

- *Reduced throttling losses* (low/part loads): The addition of inert exhaust gas into the intake system means that for a given power output, the throttle plate must be opened further, resulting in increased inlet manifold pressure and reduced throttling losses (see Figure 4).

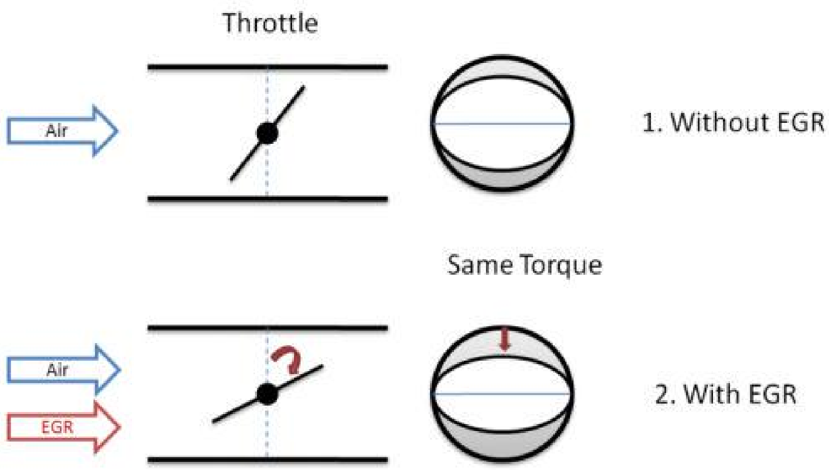

Figure 4. Adding EGR means that for a given power output, the throttle plate must be opened further which results in lower throttle losses

- *Reduced heat rejection*: Lowered peak combustion temperatures not only reduce NOX formation, it also reduces the loss of thermal energy to combustion chamber surfaces, leaving more energy available for conversion to mechanical work during the expansion stroke. Increased EGR rate makes the combustion colder and the combustion duration longer but it can be compensated somewhat through advanced ignition timing.

- Reduced chemical dissociation: The lower peak temperature result in more of the released energy remaining as sensible energy near Top Dead Center (TDC), rather than being expended (early in the expansion stroke) on the dissociation of combustion products. This effect is relatively minor compared to the first two.

The experimental engine is a standard production engine and is equipped with a short route EGR system which is mainly used to suppress knock and not for improving fuel efficiency. When taking into account the advantages of EGR in improving efficiency it is desired to operate the engine at its dilution limit at low/part load operation regions. The dilution limit is imposed by increased cyclic variation of the combustion intensity that reduces the drivability. Thus, there will be a limit to the amount of EGR that can be tolerated for each operating point. However, closed loop control of EGR based on combustion stability parameters can be a good means to improve the efficiency and preserve the engines stability at the same time. On the way to reach this goal different methodologies and combustion stability parameters are used. Pressure/Ion current based dilution limit control is applied on the EGR separately in order to maximize EGR rate as while preserving combustion stability. Furthermore, standard closed loop lambda control for controlling the overall air/fuel ratio is applied in order to keep the catalyst efficiency at its highest level all the time.

3.2 Combustion Stability

Different combustion stability parameters can be used to measure the roughness of the engine operation. In the following subsections a new method for calculating cyclic variation and combustion stability parameters derived from pressure signals and Ion current signals are discussed.

3.2.1 Combustion Stability Parameter Based on In-Cylinder Pressure

One important and well-known measure of cyclic variability and combustion stability, derived from pressure data, is Coefficient of Variation (COV) of the Indicated Mean Effective Pressure (IMEP) [9], [10]. It is defined as the standard deviation of IMEP divided by the mean value based on e.g. 100 cycles. The following equations express cyclic variations where N is the number of the cycles.

$$COV(IMEP) = \frac{\sigma_{IMEP}}{\overline{\overline{IMEP}}} \times 100 \qquad (1)$$

$$COV(IMEP) = \left(\frac{\sqrt{\dfrac{\sum (IMEP_{net} - \overline{IMEP})^2}{N}}}{(\overline{IMEP})} \right) \times 100 \qquad (2)$$

3.2.2 New Method for Calculation of Cyclic Variations

Using the traditional method for calculating COV(IMEP) during transient operation produces erroneous results. A gradual deterministic change in IMEP results in a mean value that is not representative for the entire evaluation interval and the change in operating point will be interpreted as cyclic variation. This means that COV calculations based on mean values are not suitable for transient operations.

It is desired to calculate and update the cyclic variations continuously and smoothly over a fixed number of cycles i.e. 100 cycles. It is also desired to calculate COV(IMEP) in a way that transient operation of the engine does not affect the calculations too much. To achieve this goal a new method is suggested which update the dataset continuously and uses a sort of low pass filter instead of mean value in order to eliminate the deterministic errors.

It is assumed that the mean value of IMEP is based on a dataset of 100 cycles (3). The dataset is desired to be updated continuously. This means that, the vector of IMEP updates each cycle by replacing the oldest value by the newest one i.e. $IMEP_1$ is replaced by $IMEP_{101}$ (4).

$$\overline{IMEP} = \frac{[IMEP_1 + IMEP_2 + IMEP_3 + \ldots + IMEP_{100}]}{100} \tag{3}$$

$$\overline{IMEP_{New}} = \frac{[IMEP_2 + IMEP_3 + \ldots + IMEP_{100} + IMEP_{101}]}{100} \tag{4}$$

During step changes and transients there are still risks to get deterministic changes. The definition of Low-Pass Filter was helpful to calculate a filter set of COV(IMEP). A new variable is defined and named $IMEP_{filter}$ and is calculated according to (5) where *k* is the cycle number and λ is a predefined weight that can be selected between [0 1]. Selecting λ is a trade-off between accuracy and transient performance. λ was chosen equal to 0.3 in this study. The final expression for calculating the cyclic variation of IMEP is expressed (6).

$$IMEP_{filtered}(k+1) = \lambda_m IMEP_{filtered}(k) + (1-\lambda_m) IMEP_{net}(k) \tag{5}$$

$$COV(IMEP) = \left(\frac{\sqrt{\frac{\sum (IMEP_{net} - IMEP_{filtered})^2}{N}}}{IMEP_{filtered}} \right) \times 100 \tag{6}$$

To evaluate the performance of the new method a simulation is performed in Simulink environment and the filter-based calculation was compared to the mean based calculation. The comparison is performed in two stages, first under steady-state condition where the IMEP data are not varied and in the second step a transient condition is provided by varying IMEP between 5 and 15 bars. The results for stead-state and transient cases are presented in Figure 5 and Figure 6 respectively. Figure 5 shows that under the steady-state conditions the methods work well and COV(IMEP) calculations are smooth and reliable. Figure 6 demonstrates comparison between the two methods under simulated transient conditions. The figure shows that the filter-based method can easily catch the transient and remove the deterministic changes due to the lower weighting on the older IMEP values. The mean based method is, however, unreliable and cannot be used during transients.

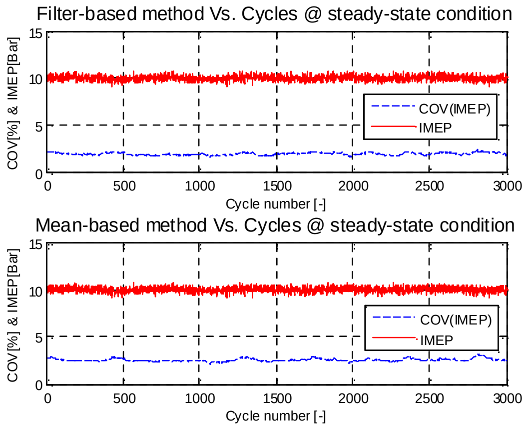

Figure 5. COV(IMEP) calculations with different methods at steady-state conditions

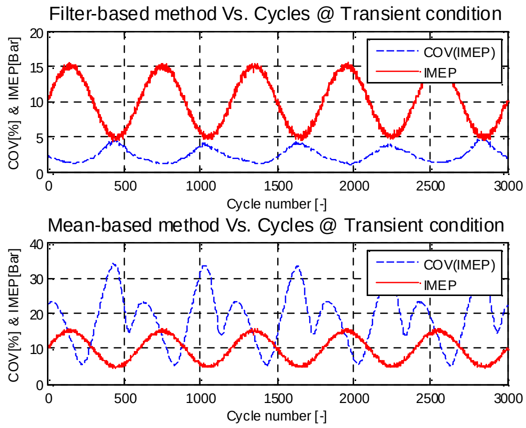

Figure 6. COV(IMEP) calculations with different methods under transient conditions

3.2.3 Combustion Stability Parameter Based on Ion current Signals

COV(IMEP) is a combustion stability parameter derived from pressure signals. Direct measurement of in-cylinder pressure can be implemented with pressure sensors although their use in production vehicles is very expensive, not only in their capital cost but in their required precision fitting and machining procedures. The ion current technique is a method of measuring in-cylinder combustion information in a non-intrusive and economical manner. A lot of researchers have shown interest in ion-sensing in recent years concerning measurement techniques and its possible applications [11], [12], [13], [15], [16]. The following subsections try to explain very briefly the basics of ion current and finally aim to find a compatible ion current based combustion stability parameter to COV(IMEP). The parameter should be robust enough to be used for diagnostic purposes.

Chemical reactions during the combustion process produce ions and electrons and the motion of these charged particles can be measured by applying a voltage (≈ 100 V DC) over the spark plug which is used as an ion sensing probe. One proposal is to divide the ion current into three parts: the ignition phase, the chemical-ionization phase and the thermal-ionization phase [17]. Figure 7 shows a typical ion current trace and a pressure signal of an average of 400 cycles from the test engine. The ignition phase starts with charging the ignition coil and ends with the coil ringing after the spark. The chemical-ionization phase reflects the early flame development in the spark gap and thermal-ionization phase appears in the burned gases behind the flame front. The peak position often appears close to the position of maximum cylinder pressure.

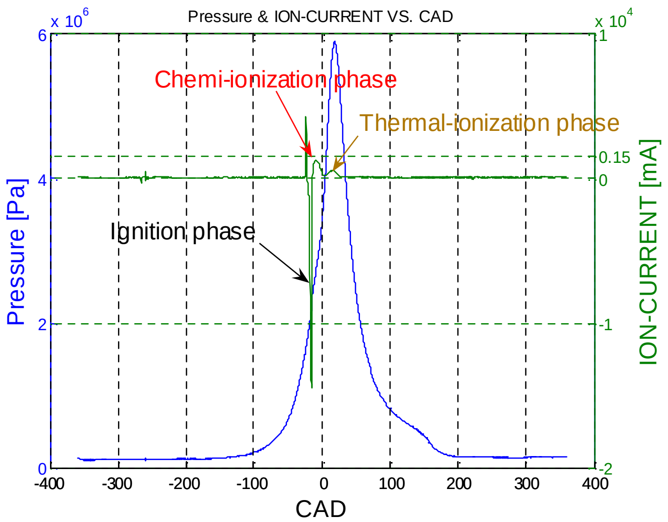

Figure 7. Typical Pressure and Ion current signals Vs. Crank Angle Degree (CAD)

In order to find a proper ion current based combustion stability parameter, the behavior of the ion current signal at different operating conditions was investigated. The engine was operated with different air/fuel ratios and different EGR rates.

Figure 8 shows ion current signals with different air/fuel ratios. According to the figure the strongest ion current signal is achieved when operating the engine somewhat lean (i.e. Lambda =1.1) but as the engine operates leaner or richer the ion current amplitude decreases. Figure 9 shows the ion current signal behavior with different amounts of EGR. By increasing the EGR rate the amplitudes of the first and the second peaks decrease. This effect is strongest on the second peak which almost disappears at highly diluted operating conditions. The ion current signal is highly depended on the combustion temperature i.e. the colder the combustion the weaker the ion current signal.

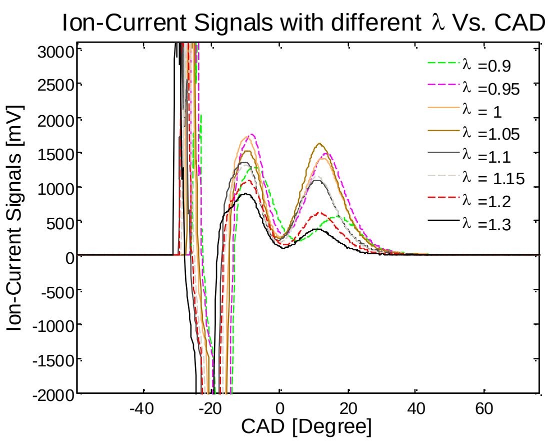

Figure 8. Ion current signal decreases when operating the engine too rich or lean

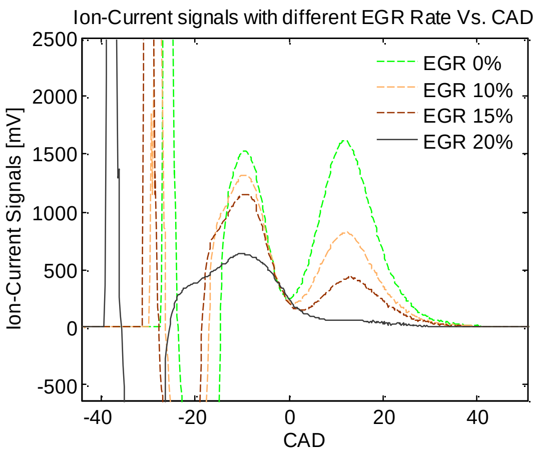

Figure 9. Ion current signal decreases when increasing the amount of EGR

For combustion diagnostic purposes, reliable signals and parameters are requirements. It is demonstrated in Figure 8 and Figure 9 that the signals from the first and the second peak (especially second peak) becomes very weak as the combustion becomes colder. For this reason, the information from the peaks may not be robust enough for diagnostic purposes especially during very lean or diluted operation. However, the area created by the first and the second peak contains useful and reliable information even at low combustion temperature that can be used for combustion diagnostic and control purposes. The area under the first and second peak can be expressed as a new parameter as in (7) where $U_{ion}(\theta)$ is the voltage produced by the ion current interface. The ion-integral limits θ_1 and θ_2 must be chosen so that the ignition phase is not a part of the integral and also such that it includes the entire first and the second peaks (see Figure 10).

$$Ion_{Integral} = \int_{\theta_1}^{\theta_2} U_{ion}(\theta) d\theta \qquad (7)$$

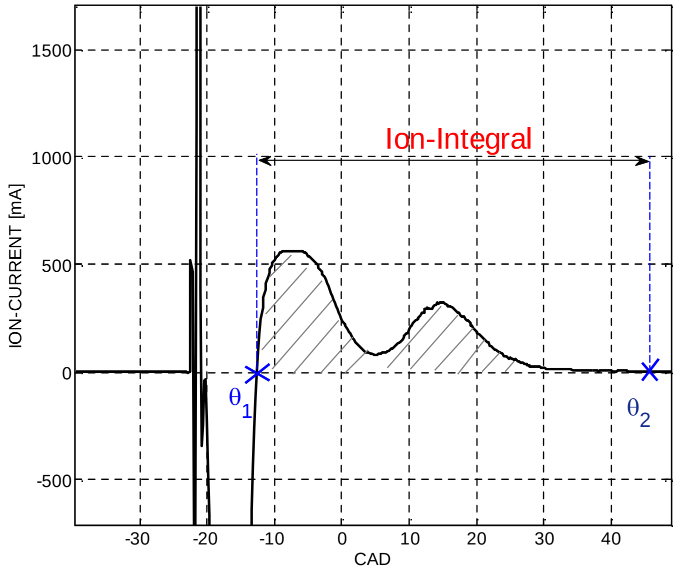

Figure 10. Ion-integral includes both the first and the second peak of the ion current signals

An experiment according to Table 3 is performed. The aim of this experiment is to find out if there is any correlation between COV(ion-integral) and COV(IMEP).

Table 3. Test matrix to capture the correlation between COV(Ion-Integral) and COV(IMEP)

Speed (RPM)	EGR Rate (%)
800	0-4-8-10-12-15
1000	0-4-8-10-12-15-20
1200	0-4-8-10-12-15-20
1400	0-4-8-10-12-15-20

To compute the cyclic variation 100 cycles of the data were used. Figure 11 shows COV(ion-integral) correlation with COV(IMEP). It can be seen that COV(ion-integral) depends linearly on COV(IMEP) with different slopes at different engine speeds. Figure 11 also shows that the level of COV(ion-integral) is much higher than the COV(IMEP). The slopes indicate that with increasing EGR the COV increases due to the colder and longer combustion. At low engine speed, COV(ion-integral) is much higher than at high engine speed. Figure 12 shows mean values of ion current signals for 400 cycles at different engine speeds. It shows that the chemical-ionization phase of the ion current signal becomes stronger with higher engine speed. One possible explanation is the better establishment of the early flame with higher engine speed. The thermal-ionization phase remains unchanged as engine speed varies. Better signal establishment at higher engine speed results in lower COV(ion-integral) level.

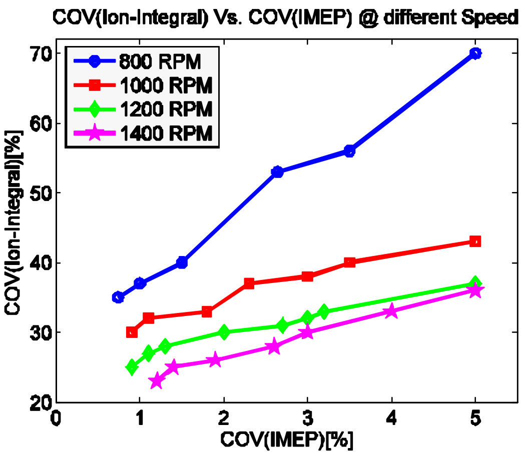

Figure 11. COV(Ion-integral) seems to be a function of speed

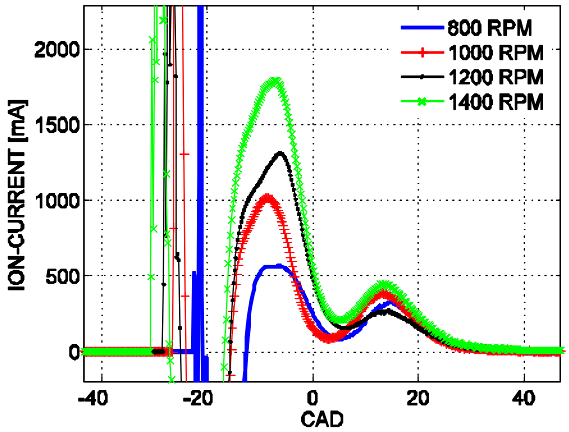

Figure 12. Ion current signals become stronger with higher speed

A new parameter named COV(INDEX) is introduced as a combustion stability parameter which is based on COV(ion-integral) and the product of engine speed and COV(ion-integral). This parameter is introduced as a compatible parameter to COV(IMEP). A multiple regression is performed which takes into account both effects from COV(Ion-integral) and the product of engine speed and COV(ion-integral) and calculates the statistics for a line that best fits the data. Expression (8) is derived which describes the correlation line.

$$\begin{aligned}\mathrm{COV}(\mathrm{INDEX}) &= 0.000238\times(\mathrm{Speed}\times\mathrm{COV(Ion\text{-}integral)})-\\ &(0.007\times\mathrm{COV(Ion\text{-}integral)})-5.97\end{aligned} \tag{8}$$

COV(INDEX) is calculated for the experimental data. Figure 13 shows how COV(INDEX) correlates with COV(IMEP) of the experimental data. The standard deviation for the residuals was calculated to 0.42 which assure the accuracy of the estimation equation.

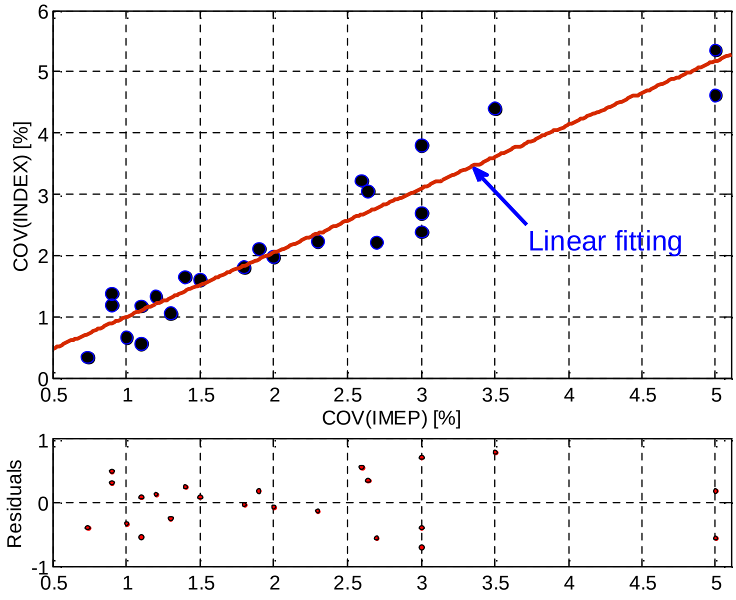

Figure 13. COV(INDEX) is derived based on a multiple regression from ion-integral and engine speed data and correlates well with COV(IMEP)

3.3 Closed-Loop dilution limit control

One of the objectives of this work is to develop a tool for mapping the best positions of the throttle and EGR valve where the engine has the lowest pumping losses and the combustion stability index is still less than 5%. The combustion stability indicator can be either COV(IMEP) or COV(INDEX), both give the same information and the only difference between them is that COV(IMEP) is derived from pressure signals and COV(INDEX) is derived from ion current signals. The solution applied here is to develop separate controllers for load (throttle), combustion stability (EGR) and lambda (fuel injection). The control structure is shown in Figure 14.

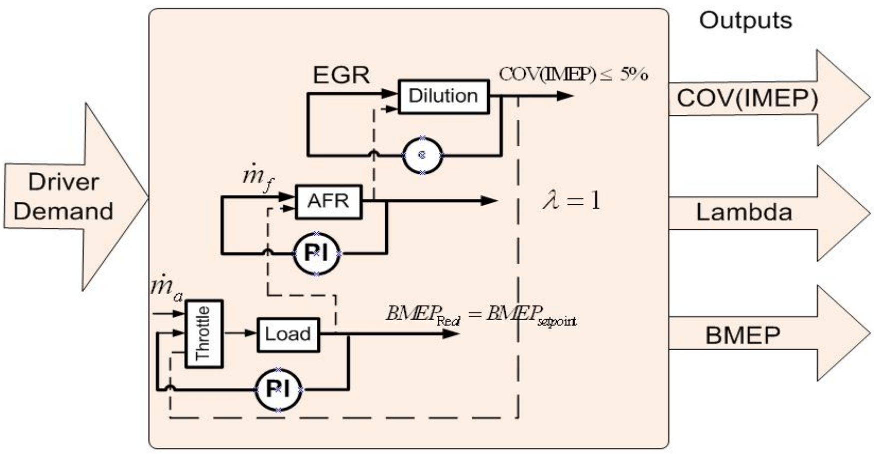

Figure 14. Closed-Loop maximum dilution limit control

3.3.1 Closed Loop Lambda Control

To keep the 3-way catalyst efficiency at its highest possible level a feedback controller is needed. Closed loop lambda control evaluates the signal from the broadband lambda sensor. The sensor measures the oxygen content in the exhaust gas, and thus provides information about the mixture composition. The closed-loop lambda control strategy uses the injected fuel quantity as the manipulated variable and compensates for the lambda error. The error signal is based on the difference between the measured lambda and a desired set-point lambda and a Proportional Integral (PI) controller was used to generate a fuel-offset based on the error.

3.3.2 Closed Loop EGR Control

The regulator always attempts to operate the engine at the maximum dilution limit. COV of IMEP or “INDEX” is set as the limitation indicator and the level is set to 5 percent. The closed loop EGR control evaluates the calculated combustion stability parameter to control the EGR valve. The error signal is based on the differences between the calculated COV(IMEP) and a set-point COV(IMEP) for 5%. The EGR valve opens more as long as the COV(IMEP) is less than 5%, and if COV(IMEP) exceeds 5% the regulator starts to close the EGR valve.

3.3.3 Closed Loop Load Control

The increase in EGR ratio that follows from activating the combustion stability controller decreases the amount of air and thus, with fixed lambda, the load. Thus, a Load controller is designed to adjust the throttle position to keep the load at a predefined level. It was also desired to have an automatic tool to find the best positions of throttle and EGR based on the drivers load demand. The load controller fulfills this requirement. The engine is connected to an electric dynamometer, and the torque is measured with a load cell. Brake Mean Effective Pressure (BMEP) is calculated from the measured torque according to equation 3.11 [18].

Closed loop load control evaluates the signal from the load cell. The error signal is based on the difference between the measured BMEP and the demanded BMEP and a throttle offset is generated from that. The throttle is adjusted by the regulator to keep the measured BMEP at the same level as the desired BMEP. A PI controller with Bump-less transfer and Anti-Windup algorithms was selected for the task.

3.3.4 Closed loop Ignition Timing Control

For each operating condition optimal spark timing can be obtained. The optimal spark timing is called Maximum Brake Torque (MBT) timing that yields to maximum output load, highest efficiency and thereby lowest fuel consumption for each operating point. A common rule says that MBT timing results if 50% of the fuel is burned at about 10 CAD ATDC (i.e. CA50=10 ATDC) [1].

CA50 is almost unique for each operating condition, meaning different ignition timing is needed to obtain MBT at different loads and engine speeds. The ignitions timing needed to achieve MBT at different load and engine speeds are presented in Figure 15. Traditionally MBT timing is implemented as an open-loop control where the ignition timing is found by using a static lookup tables. In this study a feed-forward map together with a PI controller is designed and used to achieve MBT for each cylinder individually. The error signal is based on the differences between the calculated CA50 and the set-point value. The ignition timing is subsequently adjusted.

In multi-cylinder engines there are always variations in the performance of different cylinders which has negative impacts on the overall performance of the engine. The ignition timing was controlled individually for each cylinder. Figure 16 shows that using a feedback controller improve the combustion phasing balance in all cylinders. The engine was operated at engine speed 1200 RPM and 8 bar BMEP in this experiment.

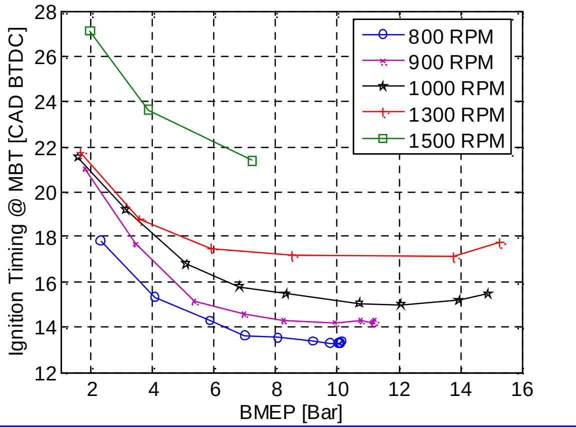

Figure 15. MBT timing for different loads and speeds

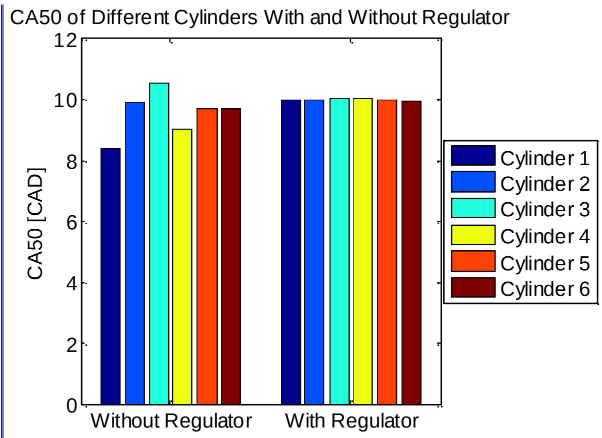

Figure 16. CA50 of all 6 cylinders with and without MBT controller

3.3.5 Experimental Results

The engine was tested for a variety of speed/loads at steady-state condition. Since the throttling losses are more critical at lower loads, it was decided to operate the engine in this region. Three different loads (i.e. 2.5, 4 and 5.5 bar BMEP) are chosen to be operated with different engine speed levels i.e. 800, 1000, and 1200 RPM (see Table 4). The start of injection was fixed for all cases. To provide a basis for fair comparison, experiments were conducted in two steps, first without adding EGR and no regulator was activated. In the second step by activating the controllers the engine was operated at its maximum dilution limit, at the demanded load with MBT timing and λ=1. The results are evaluated in terms of Brake Efficiency, pumping losses, fuel consumption and stability. Engine runs at stoichiometric operating condition with a 3-way catalyst. Since the results for different engine speeds were similar, only the results from one of the engine speed (1200 RPM) tests at 3 different loads are shown here (results from all engine speeds and loads can be found in [19] and [20]).

Table 4. Test matrix of the operating conditions

Engine Speed (RPM)	**BMEP (Bar)**	**Strategies**
800	2,5	NO EGR
	2,5	With regulator
	4	NO EGR
	4	With regulator
	5,5	NO EGR
	5,5	With regulator

1000	2,5	NO EGR
	2,5	With regulator
	4	NO EGR
	4	With regulator
	5,5	NO EGR
	5,5	With regulator
1200	2,5	NO EGR
	2,5	With regulator
	4	NO EGR
	4	With regulator
	5,5	NO EGR
	5,5	With regulator

Figure 17 shows slightly higher gross-indicated efficiency in the cases with activated controller. By increasing EGR rate the combustion is somewhat colder and combustion duration will be longer but it can be compensated somewhat by advancing the ignition timing. The combustion efficiency can increase somewhat however since the exhaust gas has a second chance to be combusted in the cylinder. The net result shows a slight increase in gross indicated efficiency. The figure shows that the gas-exchange efficiency is higher when the controllers are activated since using EGR lets the throttle open even further to keep the load at the same level. More open throttle means higher inlet pressure which results in higher gas-exchange efficiency. The effect is stronger at higher loads since the turbulence is higher and the engine tolerates more EGR than at lower loads. The effect on mechanical efficiency is neglectable which was expected.

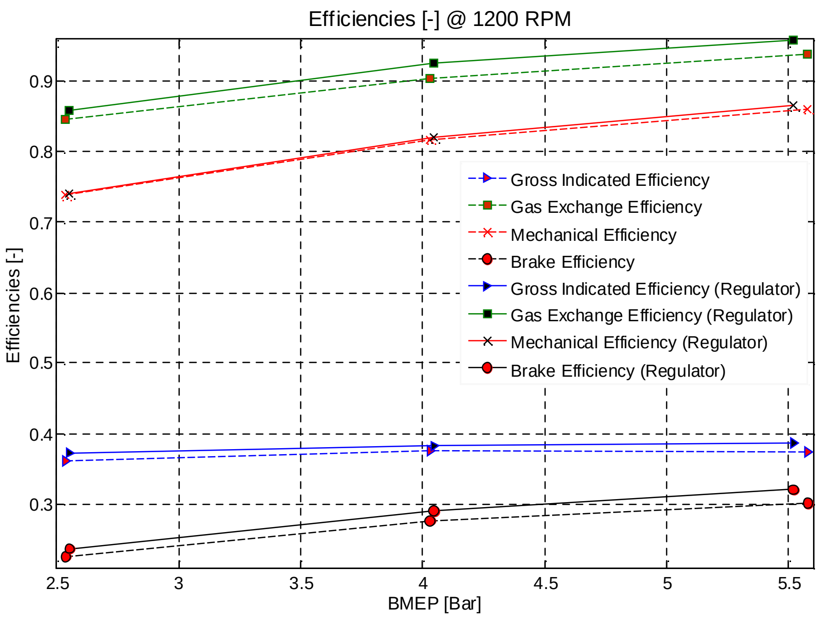

Figure 17. Due to the use of the regulators gas-exchange efficiency will be improved which results in improvement of the total efficiency

The dilution limit and the stable operating region (i.e. region where COV(IMEP)is lower than 5%) for different loads at 1200 RPM are plotted in Figure 18. X-axis shows BMEP in bar, Y-axis shows the rate of EGR in percentage and the colored region shows the level of COV(IMEP). As load and speed increases, more EGR can be tolerated in the engine because of lower residual fraction and higher turbulence level. The figure also verifies the effect of EGR on cyclic variations.

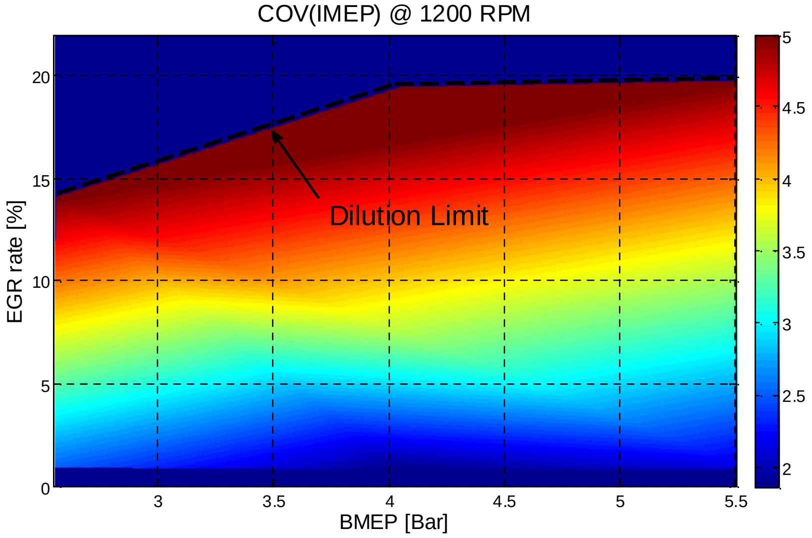

Figure 18. The maximum dilution limit is specified at 1200 RPM. In this study the stable operating region is defined as the operating region where COV(IMEP) is lower than five percent

Pumping losses are calculated and presented in form of PMEP. Figure 19 shows PMEP for the three evaluated loads (2.5, 4, and 5.5 bar BMEP) at 1200 RPM. The X-axis shows the EGR valve opening position in percentage and the Y-axis shows the throttle opening position in percentage. The triangle shows the stable region for the three loads. A certain throttle opening is needed to achieve a certain load. As the EGR valve opens more the throttle must be opened more to achieve the same amount of load. As an example in Figure 19, the case at 5.5 bar can be considered. When the EGR valve is closed the throttle is 34% open but as the EGR valve opens more the throttle opens up to 42% to keep the load at the same level. The throttle opening results in almost 0.15 bar reduction in PMEP which corresponds to over 6% reduction of the ***Specific Fuel Consumption*** (SFC) (see Figure 20). Figure 20 shows the same type of plot as Figure 19, but it shows SFC as a function of throttle and EGR valve position for the three loads at 1200 RPM.

The same experiment was performed by using COV(INDEX) instead of COV(IMEP) as combustion stability indicator in the EGR regulator. The results and the control performance were as good as with COV(IMEP).

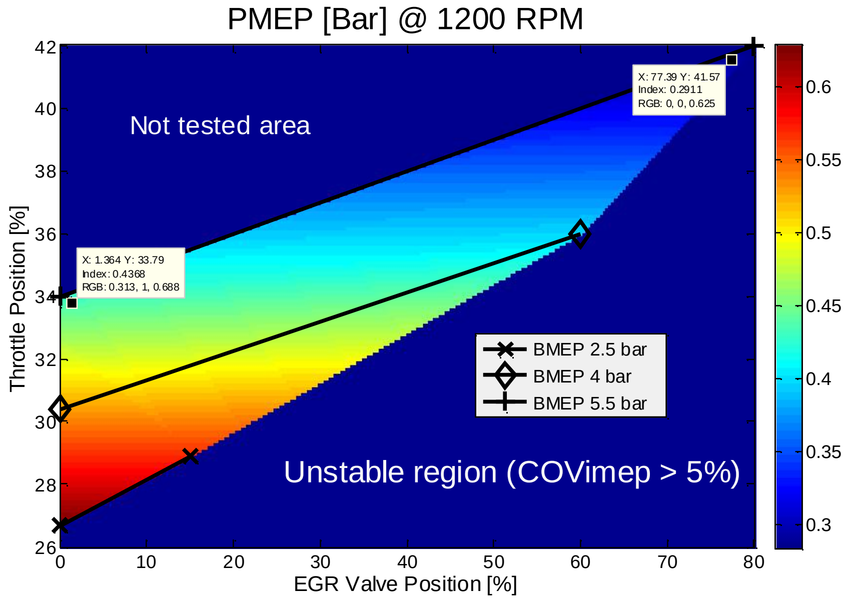

Figure 19. PMEP [bar] decreases as the throttle opens more.

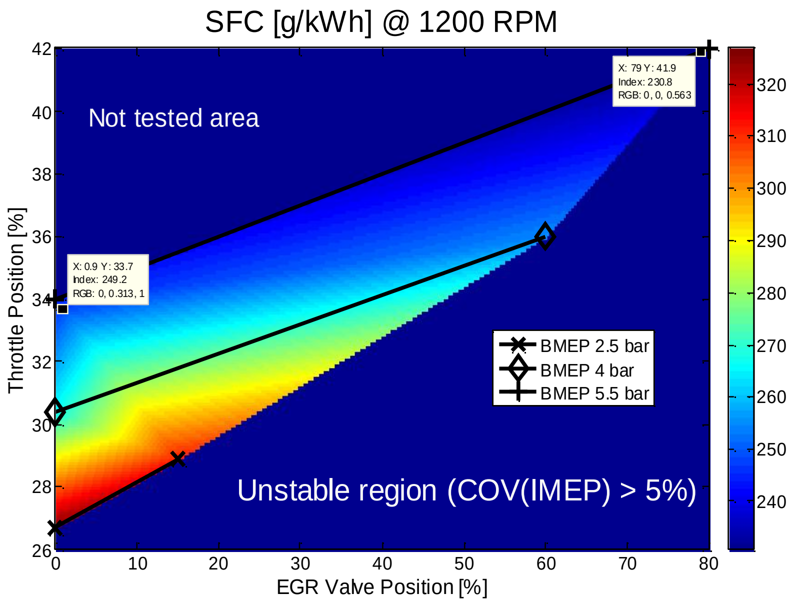

Figure 20. SFC reduces as a result of more throttle opening

Figure 21 illustrates a typical performance of the different controllers. At the top of the figure CA50 is plotted as a function of cycle number. CA50 is tuned around 10 CAD ATDC by adjusting the ignition timing which is plotted below CA50 in the figure. Lambda is adjusted by controlling the injection duration and the cyclic variation is controlled by regulating the amount of EGR.

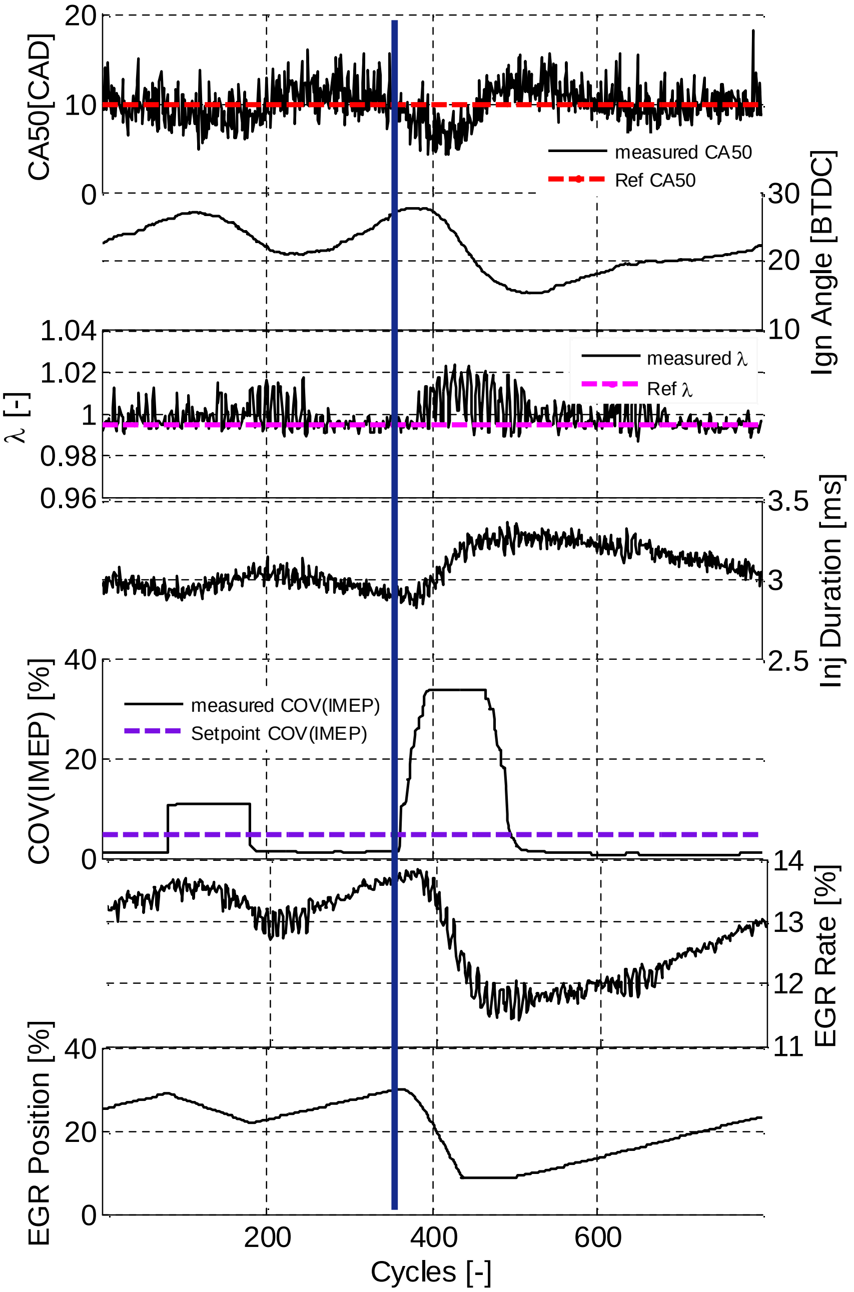

Figure 21. Controllers performance. As COV(IMEP) increases from the predefined limit (i.e. 5%) EGR, Lambda and ignition timing are adjusted

A disturbance resulted in increase in cyclic variation, as it increases over the predefined value (i.e. 5%) the EGR valve starts closing and EGR rates decreases. As result of this change, injection duration and ignition timing should be adjusted to achieve high catalyst efficiency and MBT. The overall steady-state performance was very good however the transient performance of the lambda controller was limited. However it should be mentioned that the developed dilution limit control should be applied for steady-state operation or light transients and not sharp transient operations.

Only PI feedback controllers were used in this study but a combination of PI controllers and feed-forward maps or model-based controllers will further improve the performance. PID type controllers do not perform well when applied to systems with time delays. Model-based controller overcomes the problem of delayed feedback by using predicted future states of the output for control. A model-based controller was developed, implemented and evaluated successfully in the project.

4. Engine Modification to Improve Efficiency and Extend the Maximum Load limit

As already mentioned in the introduction there are mainly two reasons for natural gas engines to have lower efficiency than the corresponding diesel engines: first the lower compression ratio which is limited due to knock and second, the use of a throttle for load control which causes severe pumping losses. A fuel's octane number is a measure of its knock resistance [22]. Normally natural gas engines use the same combustion technology as gasoline engines due to the similarity in fuel properties. However natural gas has a higher octane number than gasoline and thus natural gas operated engines can have higher compression ratio without experiencing severe knock problems. The experimental engine has, originally, a conservative compression ratio at 10.5 which is quite low and can be increased to higher levels (e.g. 12).

Another reason mentioned for the lower efficiency was throttle losses[2]. It was established in chapter 4 that by operating the engine at its dilution limit, throttle losses will be reduced drastically. It means that the extension of the dilution limit can be another strategy to improve the fuel economy. As the engine operates more diluted the combustion will be colder and as a result the combustion duration becomes longer. Increasing the turbulence level is one of the strategies that can enhance the combustion process and shorten the combustion duration. Thus, high turbulence level during the combustion is favorable especially with highly diluted mixture. One parameter which highly affects the turbulence level in internal combustion engines is the shape of the combustion chambers.

The main objective of this chapter is to discuss the details of some engine modifications which resulted in improving the overall engine efficiency and extending the maximum load limit of the engine. The compression ratio of the engine was increased and the piston shape was designed to reduce the combustion duration. The new piston modification resulted in some changes in the exhaust gas characteristics. This was the motivation to replace the turbocharger with a well-matched VGT to adjust the boost pressure level and extend the maximum load. The engine's EGR system is modified in a way to deliver more EGR and also to control it in a faster and more robust way. Each of the performed modifications and the following results in terms of engine performance are discussed in detail in the following subsections.

4.1 Combustion chamber

The first modification to the engine is to redesign the combustion chamber to achieve higher compression ratio and to increase the turbulence level. The compression ratio can be increased by reducing the compression volume, however to increase the turbulence level the bowl shape should be designed in different way. In [21], [23] and [24] the effects of different combustion chambers designs on gas flow, combustion and emissions have been studied. One of the designs, named "Quartette", that offers the highest turbulence level with good performance has been chosen from [21].

[2] Natural-gas engines can be operated stoichiometric or lean. Some of the lean operated natural-gas engines does not use throttle. The statement is only valid for the engines which uses throttle.

Figure 22 illustrates the measured turbulence level, mean velocity and the rate of heat release together with the shape of the original piston and Figure 23 contains the same information for the Quartette piston. The turbulence measurements are not from this study but since the shapes of the pistons and the engine configurations are very similar, it can be assumed that the results with the experimental engine will be also very similar. In the figures the green lines represents the mean velocity data, the red lines represent the turbulence in the cylinder and the blue lines represent the rate of heat release. Laser Doppler Velocimetry (LDV) was used to measure the turbulence and mean velocity in two directions and 5 mm below the spark plug. Figure 23 shows that the turbulence is much higher with Quartette and peaks close to top dead center which result in much faster combustion than the original piston. A set of new pistons were machined according to the "Quartette" design. The compression volume was calculated to achieve a compression ratio of 12.

The results with the new pistons are presented in the following subsections.

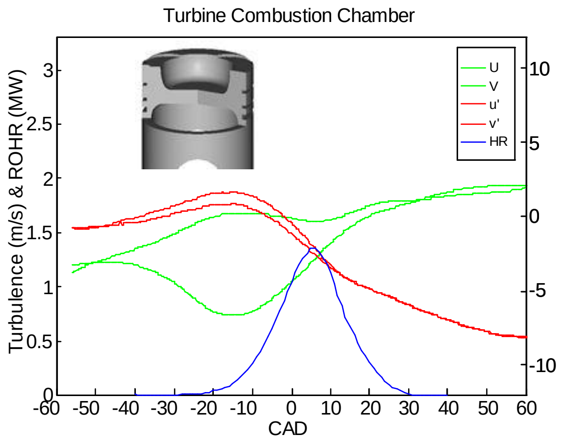

Figure 22.The original piston shape, turbulence measurements, mean velocity and heat release rate [21]

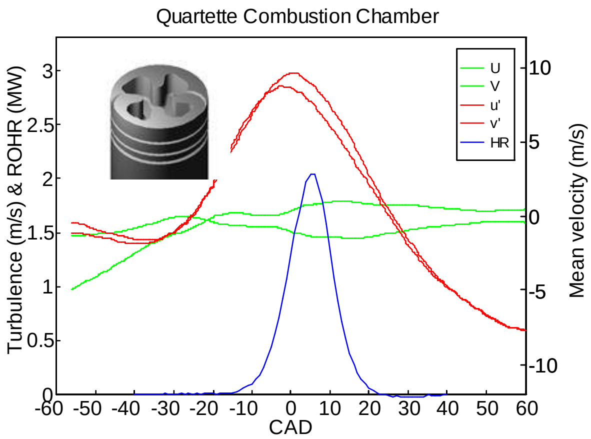

Figure 23.The Quartette piston shape, turbulence measurements, mean velocity and heat release rate [21]

4.1.1 Combustion duration

Combustion duration is calculated as the crank angle difference between 10% and 90% mass fraction burned. Figure 24 shows the combustion duration when the engine was operated at different engine speeds with Wide Open Throttle (WOT). Due to the higher turbulence level generated by the Quartette piston, the combustion duration is shortened by almost 40% which is a significant reduction. Figure 24 confirm the results reported in [21] about the high turbulence generation with the Quartette design.

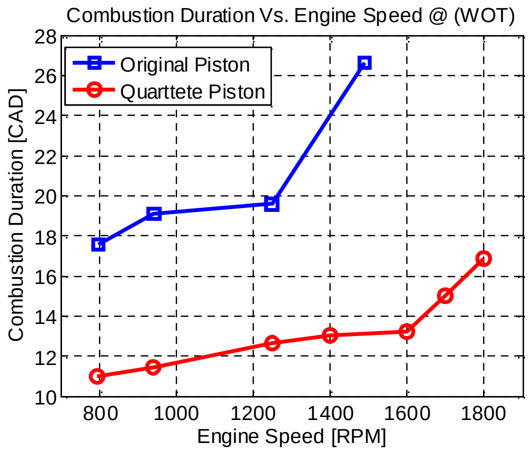

Figure 24. Combustion duration (CA10-CA90) at different engine speed (WOT)

4.1.2 Efficiency

The gross-indicated efficiency data for the same operating points and for the two pistons are compared in Figure 25. The gross-indicated efficiency is improved by at least two percentage points with the Quartette pistons due to faster burn and higher compression ratio. The new pistons' shape resulted in increasing the turbulence level which consequently enhances the burn rate. The shorter combustion duration minimizes the heat transfer losses to the cylinder walls which improves the efficiency. Furthermore, increasing the compression ratio results in a higher effective expansion ratio meaning more thermal energy is used in the cylinder rather than left in the exhaust (i.e. lower exhaust loss). At these operating points (WOT) the changes in gas-exchange efficiency and the mechanical efficiency were negligible and thereby no data are presented. At lower loads the gas-exchange efficiency can be improved. Since the throttle is not fully open, EGR can be used to reduce the throttle losses and since, with the Quartette piston, more EGR can be tolerated the throttle can be more open, for a certain load, and with more open throttle the pumping losses are lower.

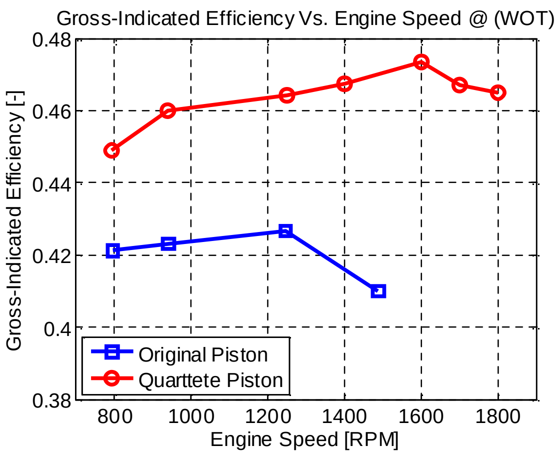

Figure 25. Gross-indicated efficiency at different engine speed (WOT)

4.1.3 Maximum load

Apart from efficiency improvement, extending the maximum load limit of the engine was another objective of this study. In order to find out the influence of the piston modification on the maximum load of the engine, maximum BMEP at different engine speeds are plotted in Figure 26. It shows that the maximum BMEP achieved by Quartette is somewhat lower than with the original pistons.

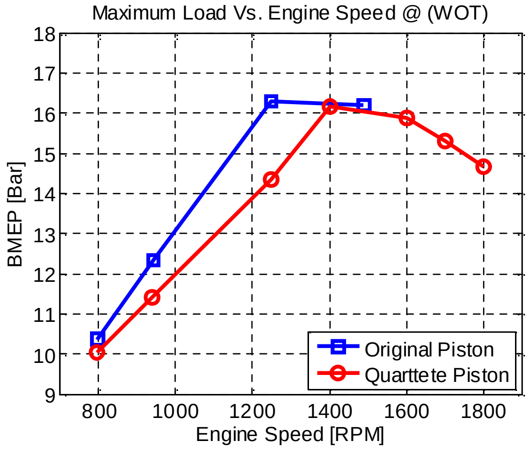

Figure 26. Maximum Load at different engine speed (WOT)

The new piston modification results in some changes in the exhaust gas characteristics and since the engine is equipped with a turbocharger with wastegate, the changes in exhaust gas characteristics may have direct influence on the boost pressure. The pressure after the compressor is plotted in Figure 27 for the same operating points as in Figure 26. The absolute boost pressure with Quartette pistons is lower which results in lower maximum BMEP. Turbine work is performed by the flow which turns the turbine and the shaft. From the conservation of energy, the turbine work per mass of airflow, *W,* is equal to the change in the specific enthalpy, *h,* of the flow from the entrance to the exit of the turbine as expressed in (9). Specific enthalpy means enthalpy per mass of airflow. By expressing the definition for enthalpy and taking into account parameters such as efficiency, η, pressure ratio, P_r, and specific heat ratio, a new expression for the turbine work is derived according to (10) [27].

$$W = h_{in} - h_{out} \tag{9}$$

$$W = (c_p \times \eta \times T_{in}) \times \left[1 - P_r^{\left(\frac{\gamma-1}{\gamma}\right)} \right] \tag{10}$$

From (10) it is clear that the inlet temperature of the turbine will affect the amount of work done by the turbine. Figure 28 shows the exhaust gas temperature which is the same as the inlet temperature of the turbine. With the Quartette pistons the exhaust gas temperature is lower than with the original pistons. This is due to the higher compression ratio of the Quartette pistons which results in more expansion. More expansion means that more heat is converted into mechanical work in the cylinder and less energy in the form of exhaust gas enthalpy is available. Lower exhaust temperature means lower exhaust energy and thereby less charging from the turbocharger.

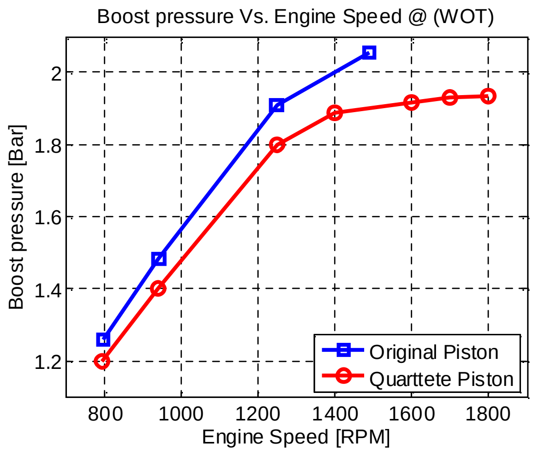

Figure 27. Pressure after compressor at different engine speed (WOT)

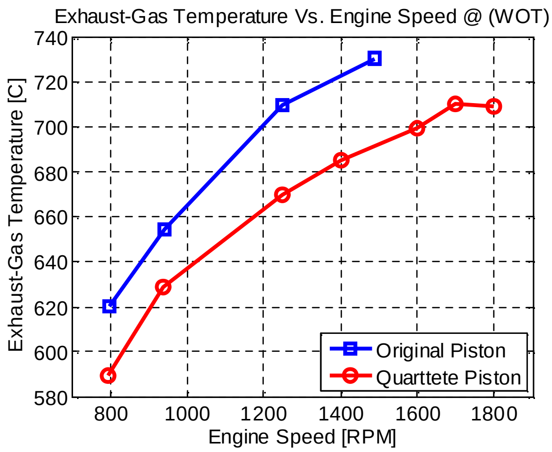

Figure 28. Exhaust gas temperature at different engine speed (WOT)

As noted in the introduction, the exhaust gas temperature from this type of natural gas engines (i.e. diesel converted engines) must be kept low. According to the manufacturer the exhaust gas temperature should be lower than 760^{0}C. The lower exhaust gas temperature obtained with higher compression ratio is beneficial in two ways. First, there is a higher margin to the highest allowable exhaust gas temperature and second, there is potential to extend the maximum load limit by replacing the turbocharger with a better matched unit.

4.2 Variable Geometry Turbocharger

A Variable Geometry Turbocharger (VGT) has the ability to change the turbine geometry in order to obtain the desired boost pressure. VGT offers attractive properties such as high flexibility, minimal amount of lag and wide operating range. VGT technology is extensively used in diesel engines; however the use of VGT is very much ignored in gasoline engines due to their relatively high exhaust temperatures. Ordinary VGT materials and designs cannot withstand temperatures over 890°C [26]. The exhaust gas temperature of gasoline engines could, however, reach up to 1000°C, versus 650°C in diesel engines [25]. Normally natural gas engines use the same combustion technology as gasoline engines due to similar fuel properties. In heavy-duty applications, since the engine speed does not exceed 2000 RPM, the exhaust gas temperature is not very high. As an example, the maximum allowable exhaust gas temperature for the experimental engine that operates stoichiometrically is 760^{0}C which is tolerable for VGT material. Traditionally heavy-duty natural gas engines use the same turbocharging technology as gasoline engines. They are equipped with a turbocharger with wastegate but it is quite simple and advantageous to replace the by-pass turbocharger with a well-matched VGT to achieve the required boost pressure.

4.2.1 Extending the maximum load

After consulting with the R&D group at Volvo a VGT was designed and mounted on the engine. The engine is operated at the same engine speeds with WOT and by altering the geometry of the turbine housing the boost was increased until either knock occurred or the pumping losses started to increase. To suppress knock, EGR was added and the ignition timing was adjusted accordingly. Figure 29 shows the boost pressure achieved by the VGT in comparison with the by-pass turbocharger with original and Quartette pistons. The boost pressure was increased up to much higher levels with VGT without sacrificing gas-exchange efficiency. The increase in boost pressure resulted in increased maximum load as presented in Figure 30 compared to previous configurations. The peak load increased by 18 percent from 16 to 19 bar BMEP with maintained gas-exchange efficiency and exhaust gas temperature limitation.

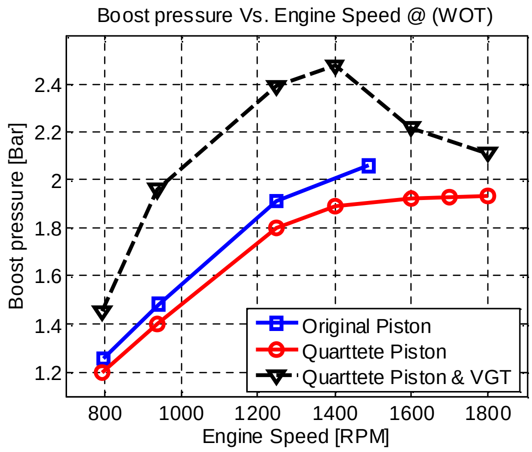

Figure 29. Boost pressure achieved after replacing the turbocharger with a VGT in compared to the other configurations.

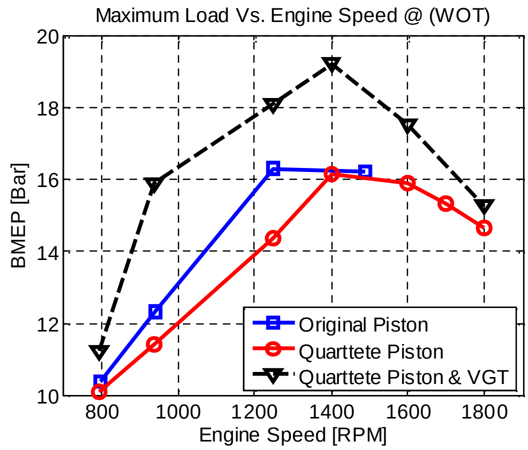

Figure 30. Maximum load achieved after replacing the turbocharger with a VGT compared to the other configurations

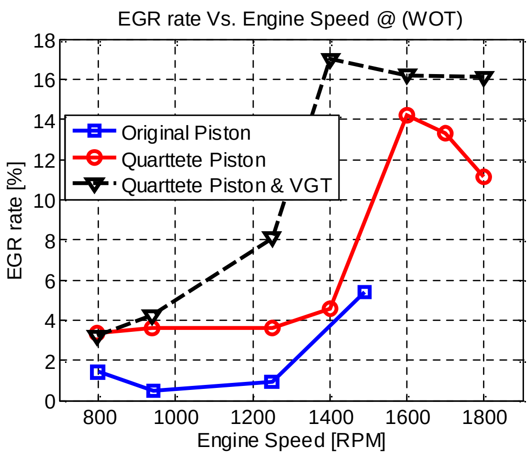

Figure 31. Amount of EGR used to suppress the knock with different configuration viz. Quartette piston and VGT, Quartette/original piston with by-pass turbocharger

By increasing the compression ratio the knock tendency of the engine increases. The knock tendency increases even more when the boost pressure increases. EGR has been used as a remedy to suppress knock. When adding EGR, the ignition timing was adjusted to achieve MBT; of course, MBT was not achieved in some operating points. Figure 31 shows the amount of EGR needed with different engine configurations. The needed EGR rate is obtainable in the entire operating region with all configurations.

4.2.2 Reducing throttle losses by means of VGT

Using a throttle always leads to pumping losses. The gas-exchange efficiency for the whole operating regime of the experimental engine is presented in Figure 32. As BMEP decreases at each engine speed, the gas-exchange efficiency also decreases due to the more closed throttle which results in more pumping losses. Some strategies have been reported to reduce the throttling losses. In [28],[19] and [20] EGR is used together with closed loop dilution limit control to reduce the throttling losses. With this strategy the benefits are limited to the dilution limit of the engine; the transient performance of the engine is also limited. In [29] a technology called Waste Energy Driven Air Conditioning System (WEDACS) is developed to recover throttling losses. In [30], [31] and [32] the new and efficient strategy viz. Variable Valve Timing (VVT) is introduced which results in higher efficiency in a wider operating range. However these strategies are associated with complexity, more components and higher cost.

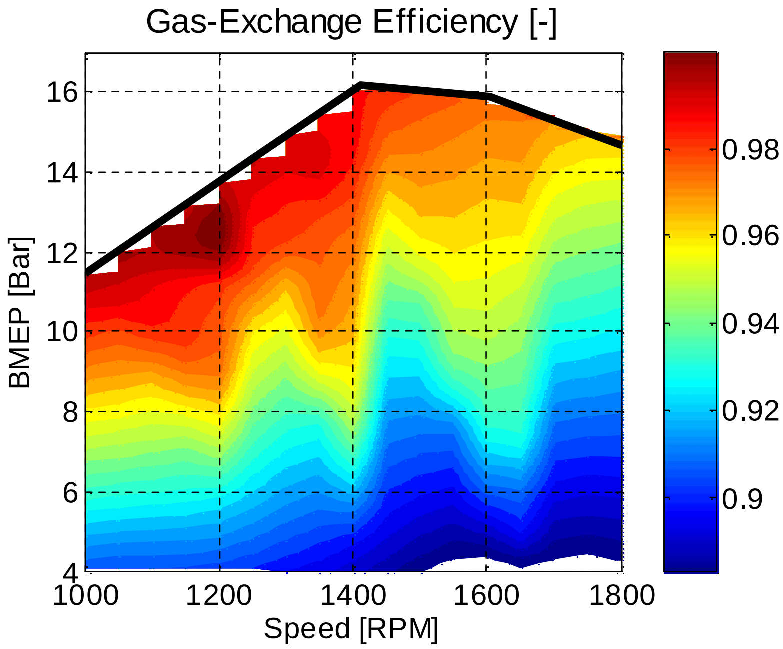

Figure 32. Gas-exchange efficiency as a function of Load and Engine speed for a HDNG engine

This subsection presents an innovative strategy where a VGT is used to reduce the throttling losses in large operating region. In [33] the feasibility of using VGT for the heavy-duty natural gas engine is established. Normally a throttle is used to control the desired torque in the engines, but it is also possible to use a VGT instead of the throttle in a large operating range to control the desired torque. This is possible due to the flexibility of the VGT to adjust the inlet pressure by altering the geometry of the turbine housing. In this operation region the throttle is kept fully open and the VGT is used to adjust the inlet pressure and finally control the demanded torque. This means that the throttle will not be used in a large part of the operating region and no throttle use means any throttle losses.

The boost threshold for the experimental engine is specified in Figure 33. VGT start producing boost only with enough amount of exhaust mass flow rate. Figure

33 shows the inlet pressure as a function of engine speed and BMEP after installing a VGT on the engine. The VGT operating region is above the boost threshold and is specified with the dashed lines. The covered operating region is roughly 60% of the total operating region of the engine.

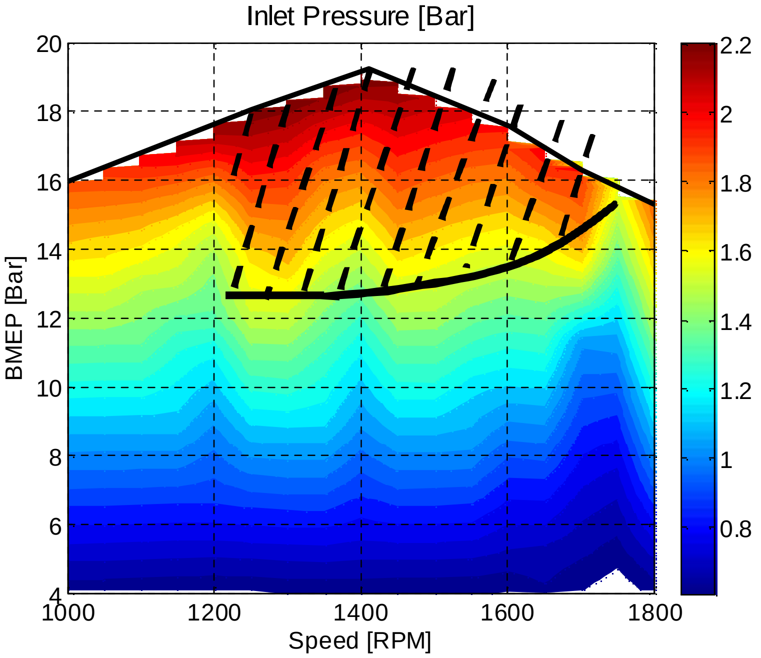

Figure 33. Inlet pressure as a function of Load and Engine speed for the HDNG engine equipped with VGT. Operating range with VGT is specified with the dashed lines

The feasibility of reducing losses by means of VGT is studied and quantified in detail and is reported in the following three subsections. In the first subsection the possible gains with VGT in terms of gas-exchange efficiency is discussed. The time constants of load transients when using VGT or throttle are calculated and compared in the second subsection and in the third subsection the achieved results are validated by performing the same experiments at other operation points.

Improvement in Gas-Exchange Efficiency

To quantify the gain in gas-exchange efficiency, IMEP was altered between 10 and 14 bar at 1000 RPM once by using the throttle and once by using the VGT. First the throttle is used to change the load. In this case the VGT position is fixed (i.e. 60% closed) and the throttle was altered between 40 and 100% opening to achieve the desired load. The results are evaluated in terms of gas-exchange efficiency (see Figure 34). The Y-axis to the left shows the throttle position and the Y-axis to right illustrate the gas-exchange efficiency. The X-Axis indicates the number of cycles. Once the throttle was fully opened, the gas-exchange efficiency increased from roughly 98 percent to almost 100 percent. It confirms that the throttling losses results in roughly 2 percent unit losses in gas-exchange efficiency at this operating point. In the second experiment the throttle was kept fully open and the VGT is used to alter the desired load at the same engine speed. The VGT position is altered between 0 and 60% to adjust the desired boost and subsequently to achieve the desired IMEP. The load was altered in the same range as it was altered by the throttle. Figure 35 shows the gas-exchange efficiency as the VGT is used to control the load. The Y-axis to the left shows the VGT position and the Y-axis to right illustrates the gas-exchange efficiency. Since no throttling has occurred to adjust the desired load, no losses are introduced and the gas-exchange efficiency is kept at its highest level all the time i.e. almost 100%. During the transient (i.e. step responses) some wave pulses are generated which result in overshoots in the opposite direction (see Figure 35). For instance when opening the VGT, exhaust backpressure decreases, but it takes a while before the inlet pressure falls, thus lower pumping loss and higher gas-exchange efficiency for some cycles can be achieved. In the same way higher pumping loss is achieved for some cycles when closing the VGT. The duration of these overshoots corresponds to the turbochargers lag at this operating point.

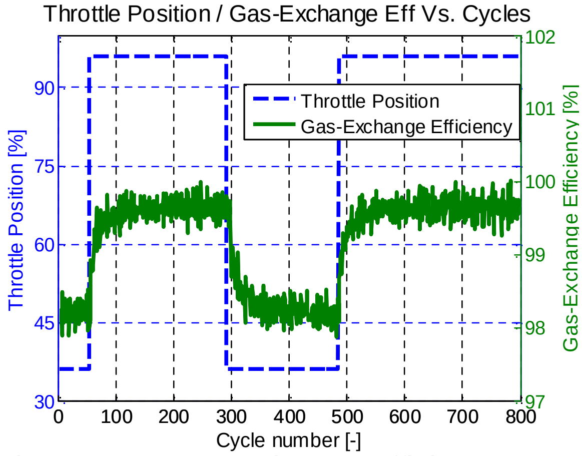

Figure 34. Gas-Exchange efficiency as throttle used to control the desired load at 1000 RPM

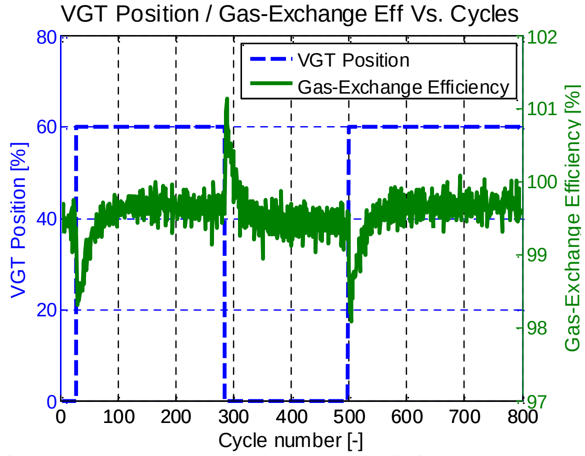

Figure 35. Gas-exchange efficiency as VGT used to control the desired load at 1000 RPM

Suggested control strategy for lowest possible throttle losses (Fuel-efficient driving)

To avoid the throttle losses as much as possible a control strategy is suggested. It gives two options to the driver to control the desired load. The driver will be able to drive either in the conventional way i.e. with throttle or in a fuel-efficient mode. With the fuel-efficient strategy, the throttle is avoided as much as possible. With this mode, to control the load mainly VGT, partly EGR and partly a combination of throttle and EGR is used.

The whole operating region of the engine is divided into three main regions. The biggest operating region, located above the boost threshold, can be covered by the VGT. In this region, as already discussed in the previous subsection, the throttle is fully open and the load is totally controlled by altering the geometry of the turbine housing. This results in adjusting the boost pressure which consequently decides the torque. This region is specified in Figure 36. 60 percent of the total operating region can be covered by VGT. The second region is located under the boost threshold. The throttle can still be fully open and by adding EGR the amount of load is decreased. This means that in this region the EGR valve is the main actuator for controlling the desired load. Since the throttle is still fully open in this operating region there will not be any throttle losses. There is a limit for the amount of EGR with the fully open throttle. Increasing cyclic variation in IMEP can be a good indication of this limit. The throttle is used combined with the optimal amount of EGR in the third region. Figure 36 demonstrates the three discussed operating regions and Figure 37 describes the suggested control strategy.

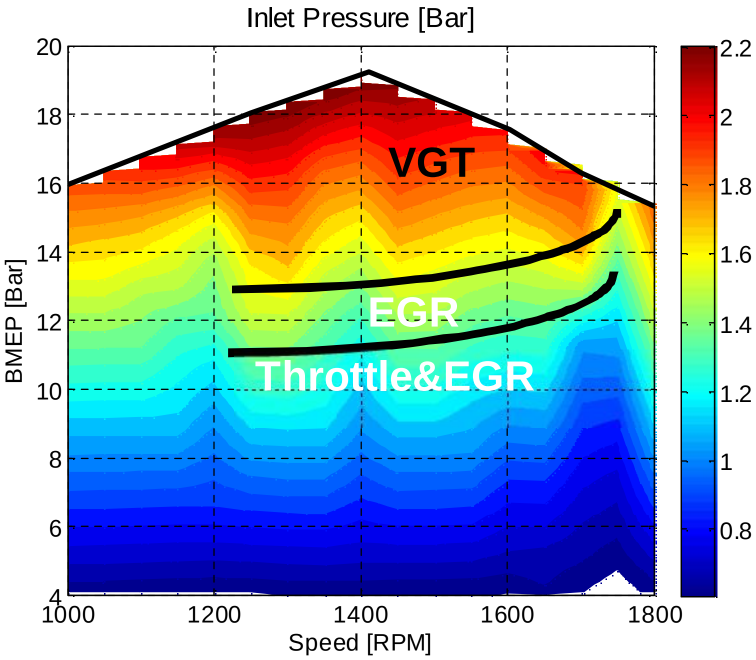

Figure 36. The economical driving requires different strategies in different operating regions.

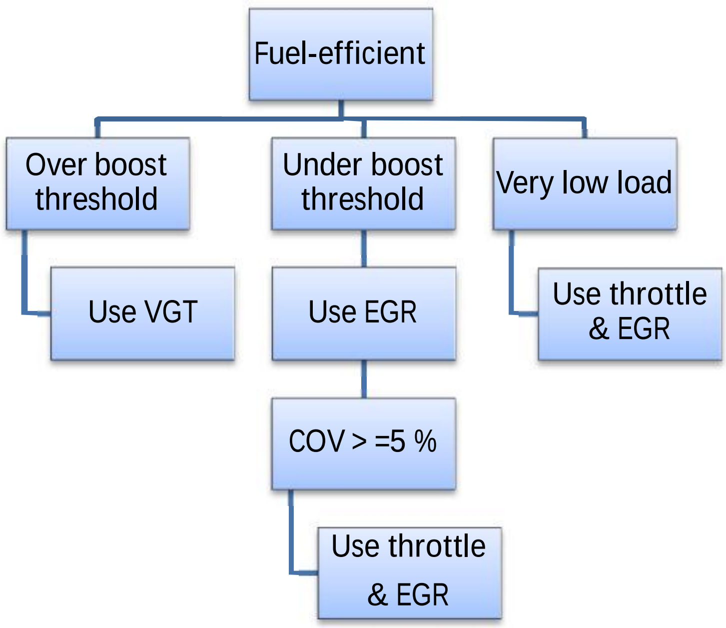

Figure 37. The suggested control strategy for fuel-efficient driving results in the lowest possible throttling losses.

4.3 EGR system

The engine is equipped originally with a short route EGR system also called High Pressure (HP) EGR system. It was established in chapter 3 that high dilution is very advantageous at part load. Since the engines' pistons are replaced with high turbulence pistons the engine will be more tolerant to higher EGR rates. However the HP EGR configuration can deliver only up to 20% EGR. To increase the engine's ability to deliver more EGR and also to control the EGR rate in a faster and more robust way a long route EGR system, so called Low Pressure (LP), is added to the engine (see Figure 38). Moreover a back pressure valve was installed after the catalyst to increase the exhaust pressure when needed in order to deliver the desired EGR rate. This type of double EGR configuration (HP + LP) is previously presented in [34].

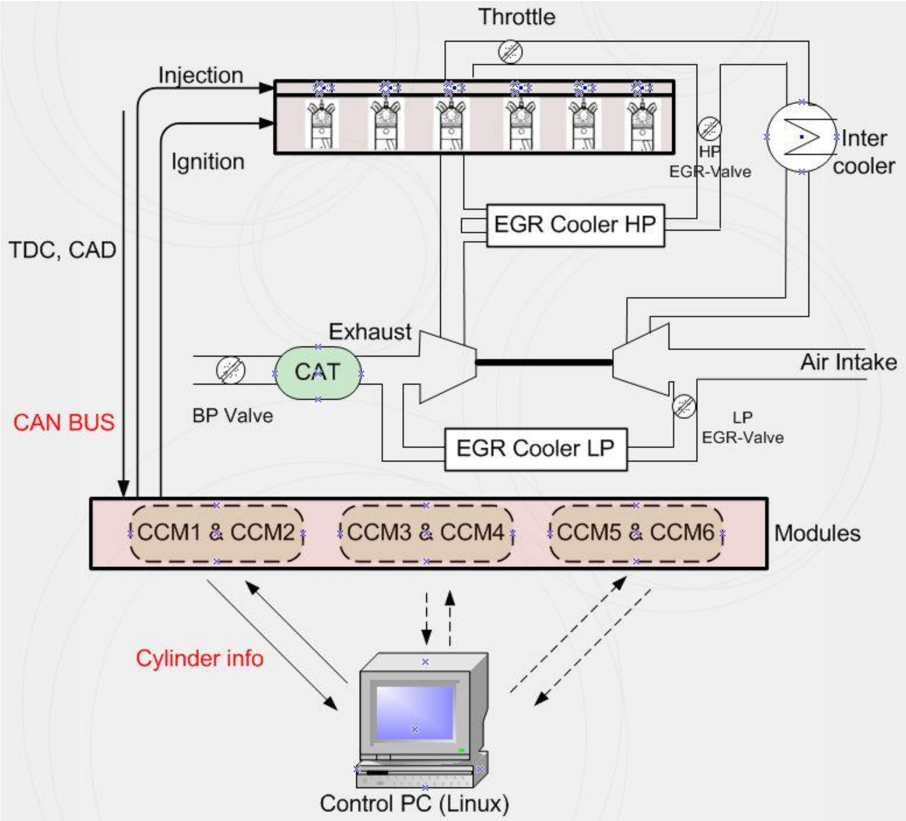

Figure 38. The engine setup with both short and long-route EGR systems.

4.3.1 Short route versus Long route EGR System

In order to compare the dynamics of the two EGR configurations, a step response was made to vary the rate of EGR between roughly 1-2% and 15% at 1000 RPM. This experiment was done once with the short route EGR position and once with the long route EGR position. The results from these experiments are presented in Figure 39 and Figure 40 for the short route and long route EGR system respectively. The Y-axis to the left belongs to the EGR valve position and the Y-axis to the right shows the measured EGR rate over 800 combustion cycles. The operating conditions such as load, engine speed etc. were the same for both experiments.

Surprisingly the dynamics with the long route EGR system are much faster than with the short route EGR system which was in contrast to our hypothesis. The dynamics of the short route EGR system relies on the dynamics of the turbocharger and its lag. The other reason for the faster dynamics with the long route EGR system rather than the short route is the better pressure difference.

Apart from the faster dynamics, the long route EGR system is advantageous in several other ways. First of all, engine operation is more stable due to better distribution and mixing of the EGR with air and fuel (i.e. the longer distance the more time for mixing). Secondly, the EGR goes through an intercooler and the mixture becomes colder and consequently the engine becomes more knock resistant. Third, due to the higher pressure difference more EGR can be delivered than with the short route EGR system. The only drawback with the long route EGR system is that more water will be produced in the intake which should be systematically drained.

By taking into account all the discussed facts, it can be easily concluded that mid-range control of EGR is not needed since by only using the long route EGR system, the requirements for robust EGR control can be fulfilled (i.e. the short route EGR system is not needed at all).

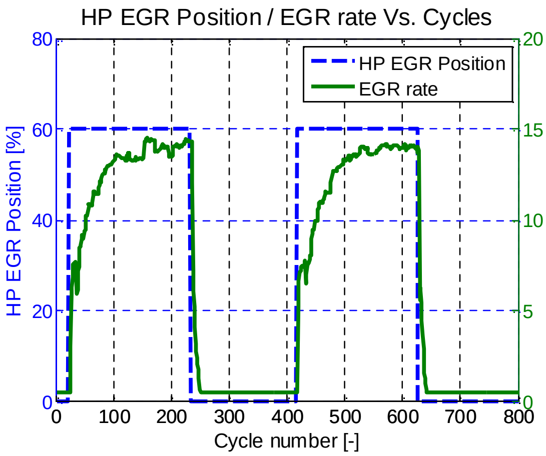

Figure 39. The dynamics of the short route (i.e. HP) EGR system.

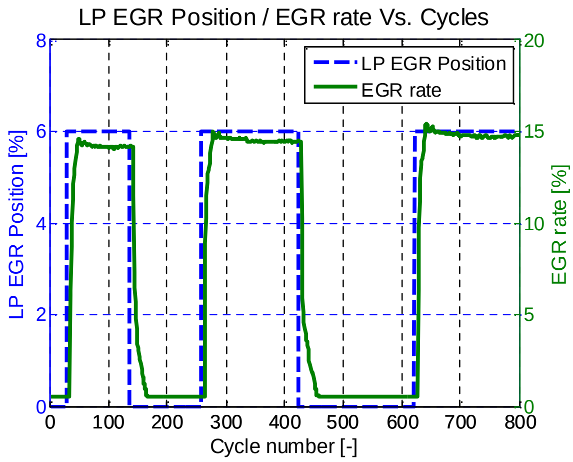

Figure 40. The dynamics of the long route (i.e. LP) EGR system.

5. Extending the Dilution Limit and Studying the Effect of Hythane on Combustion

Extending the dilution limit has been identified as a beneficial strategy to improve efficiency and decrease emissions in stoichiometrically operated natural gas engines. However the dilution limit is limited mainly due to the lower burn rate of natural gas. One way to extend the dilution limit in a natural gas engine is to operate the engine with a natural gas which is enriched with hydrogen.

Hydrogen and methane are complimentary fuels in some ways. Natural gas consists mainly of methane (~90%) which has a relatively narrow flammability range. This is especially a problem when a natural gas engine is operated close to its lean or dilution limit. Methane has a slow laminar flame speed, especially in lean or diluted operation, while hydrogen has a much faster laminar flame speed [36] and [37]. Figure 41 shows flame propagation velocity for methane and hydrogen versus air-fuel ratio. Methane is a fairly stable molecule that can be difficult to ignite, but hydrogen has an ignition energy requirement much lower than methane. In [39], [40], [41] and [42] some benefits of using Hythane are reported. In [43] benefits of hydrogen addition to a lean burn natural gas engine are investigated. In the work the hydrogen addition was varied from 5 to 20% (by volume) while the engine was operated lean. Some small improvements in efficiency and unburned hydrocarbon emissions are reported in the work. In [44] a study was performed by the author of this thesis to investigate the effect of Hythane on a stoichiometrically operated multi-cylinder natural gas engine.

The main objective of the work was to investigate the knock sensitivity and possible gain in dilution limit, lean limit, emissions and efficiency of the engine by enriching the natural gas with 10% hydrogen. No significant differences in terms of knock margin, efficiency and emissions levels between the blended natural gas and the pure natural gas were observed in the study. From the results it was concluded that when the engine is operated stoichiometrically the combustion characteristics are already good with short enough combustion duration and the Hythane shows no improvement compared to natural gas. Since no significant results were achieved with 10% hydrogen addition, a complementary study was performed with 25% hydrogen addition. The main objective of this work was to investigate the effects of the hydrogen addition on knock margin, efficiencies, emissions and dilution limit of a stoichiometrically operated natural gas engine. It was also desired to compare the possible advantages with Hythane in diluted operation versus the lean operation natural gas engines.

One important parameter which should be addressed and kept in mind during the reading of this section is the importance of the recent piston modification. The pistons of the engine were replaced by highly turbulent pistons which speed up the combustion rate. This characteristic of the piston has similar effects as Hythane on the combustion process. This means that the effects of the Hythane could be more obvious with the original pistons rather than the new pistons. It should also be mentioned that during these experiments VGT was not mounted on the engine.

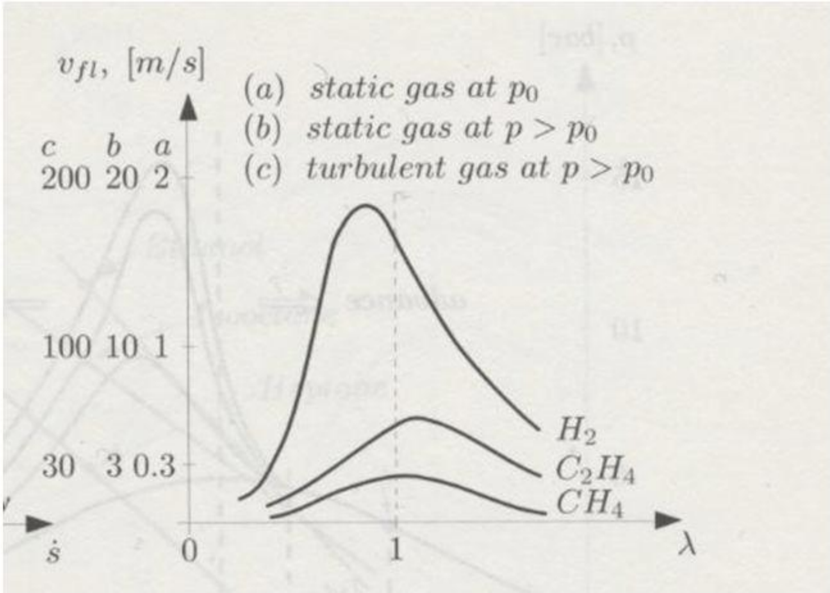

Figure 41. flame propagation velocity over air-fuel ratio [38]

5.1 Gas data

The compositions of the natural gas and the Hythane are presented in Table 5 and Table 6 respectively. The main change in the gas properties is the percentage of Methane which is reduced by 25% and replaced by hydrogen. All the percentages reported in the tables are volume based. Some important properties of the fuels such as H/C ratio, stoichiometric AFR value, lower heating value and density of the Hythane and natural gas are presented in Table 7.

Table 5. Natural Gas composition

Composition	%	Structure
Methane	89.8	CH4
Ethane	5.87	C2H6
Propane	2.23	C3H8
I-Butane	0.37	C4H10
N-Butane	0.51	C4H10
I-Pentane	0.13	C5H12
N-Pentane	0.08	C5H12
Hexane	0.06	C6H14
Nitrogen	0.3	N2
CO2	0.63	CO2

Table 6. Hythane composition

Compositions	%	Structure
Methane	67.4	CH4
Ethane	4.3	C2H6
Propane	1.7	C3H8
I-Butane	0.2	C4H10
N-Butane	0.4	C4H10
I-Pentane	0.11	C5H12
N-Pentane	0.09	C5H12
Hexane	0	C6H14
Nitrogen	0.3	N2
CO2	0.5	CO2
Hydrogen	25	H2

Table 7. More properties of the fuels

Property	Natural Gas	Hythane
H/C Ratio	3.7	4.32
Stoichiometric AFR	16.9	17.56
LHV (MJ/kg)	48.4	50.76
Density (Kg/Nm^{3})	0.82	0.638

5.2 Experiments

The experiments were performed in three different stages. Since the objective was to compare the effects of the natural gas and the Hythane respectively on engine operating parameters; the engine is operated twice at each stage, once with natural gas and once with Hythane. Details about the experiments are as follows:

- **Lambda response**: in this experiment the engine operated at 1000 RPM. The sweep is started at lambda=0.8 and lambda was subsequently increased until the lean limit was reached. In this experiment the air flow is constant and the fuel flow is varied to achieve the desired lambda value. The ignition timing was adjusted to achieve Maximum Brake Torque (MBT) timing.
- **EGR response**: In this experiment the engine was operated stoichiometrically at 1000 RPM. The EGR ratio was increased until the dilution limit is reached.
- **Map**: The engine was operated stoichiometrically at different loads and engine speeds to study the overall performance of the engine in the entire operating region. It was of particular interest to investigate the knock margin of the engine. It should be kept in mind that in this experiment EGR was only used to suppress knock and not to increase efficiency[3].

Since the stoichiometric value is not the same for both fuels, this value was calculated for each fuel and the lambda sensor was calibrated once the fuel was changed. For each operating point measurements from 300 cycles were collected.

5.3 Results

In the first experiment the lean limit of the engine was investigated. The lean limit of the engine at 1000 RPM was extended from 1.6 to 1.8 by using the Hythane as fuel (see Figure 42). The sudden increase in cyclic variation of IMEP due to misfire in some cycles is used as an indication to identify the lean limit of the engine. The combustion duration becomes longer as the engine operates leaner. Extension of the lean limit is due to faster burn rate and lower energy requirement for ignition of Hythane. For comparison it can be mentioned that the lean limit with the original pistons was 1.4 and was extended to 1.6 with the new pistons. Average of combustion duration for all 6 cylinders is calculated and plotted over a lambda range for both fuels in Figure 43. Combustion duration is defined as the difference between the Crank Angle (CA) where 10% of the total heat is released (CA10) and the CA where 90% of the total heat is released (CA90). Combustion duration is

[3] EGR can also be used to improve the efficiency by reducing the throttling losses

shorter with Hythane due to the faster flame speed. The difference is not much as the engine operates close to stoichiometric but as the engine operates leaner, the combustion duration increases and the difference is more obvious. Due to the higher burn rate of Hythane, the ignition timing is retarded to achieve MBT timing.

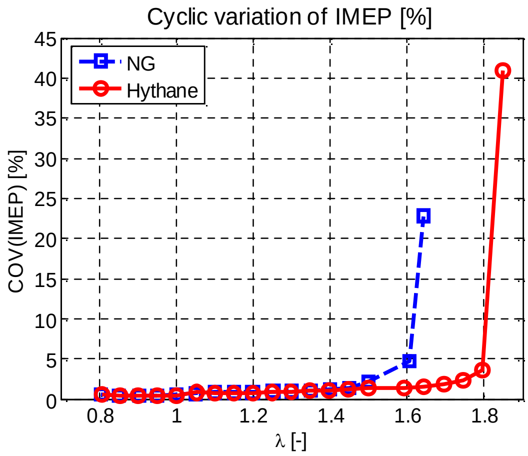

Figure 42. COV(IMEP) versus lambda

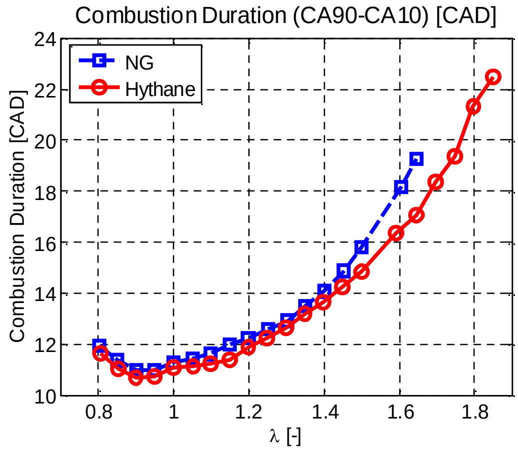

Figure 43. Comb duration versus lambda

CA10 and CA90 for both fuels over the lambda range are plotted in Figure 44 and Figure 45. CA10 is retarded as a consequence of the retarded ignition timing but CA90 is the same for both fuels.

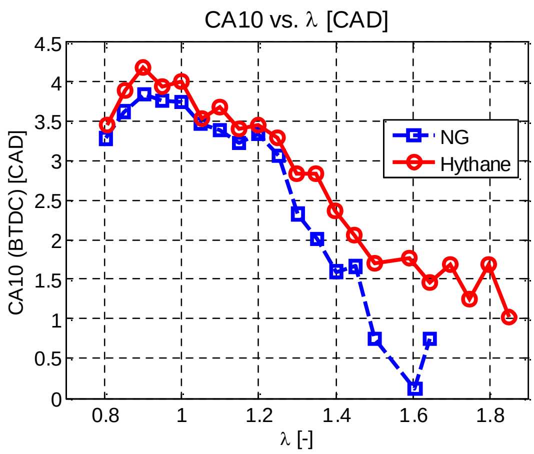

Figure 44. CA10 over lambda range for Hythane and natural gas

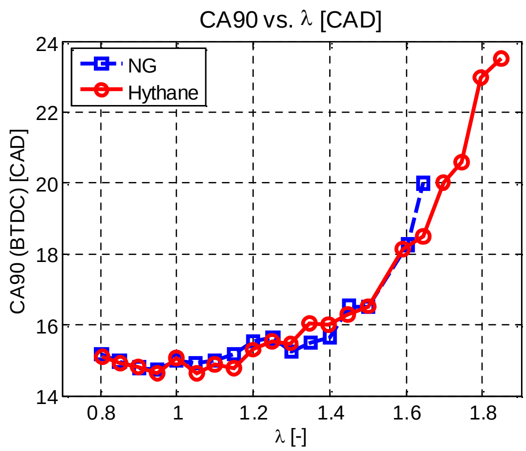

Figure 45. CA90 over lambda range for Hythane and natural gas

Unburned Hydro Carbon (HC) and Nitrogen Oxides (NO_X) are measured and calculated in form of specific emissions.

5.3.1 Specific NO_X versus Lambda

Figure 46 shows the specific NO_X over the lambda range. As the engine operates close to stoichiometric and goes up to lambda equal to 1.2, the in-cylinder temperature increases and as the engine operates leaner and leaner the in-cylinder temperature decreases. The fact that points to the proportional relation between temperature and NO_X generation can explain the trend in NO_X generation in Figure 46 . The peak temperature in this experiment is close to lambda 1.2 which is the peak for NO_X generation. Due to the higher burn rate of Hythane, there will be higher peak in-cylinder temperature and consequently somewhat higher NO_X generation. NO_X emission reduces drastically as the engine operates leaner and as the lean limit extends from 1.6 to 1.8 the NO_X will be reduced to very low levels. The region close to the lean limit of the engine is zoomed in and showed in the same figure to the right. It shows that by operating the engine on its lean limit the NO_X emissions can be reduced by 90% (i.e. to 0.33 g/Kwh) with hythane compared to natural gas which is significant. This factor is especially important in lean operated natural gas engines where a Selective Catalytic Reduction (SCR) system is needed to reduce the NO_X emissions to lower levels. By operating the engine on Hythane it seems that it may not be necessary to use SCR. It is still a trade-off between NO_X emissions, HC emissions and efficiency, since by extending the lean limit the combustion becomes colder which reduces the efficiency. A study should be performed to quantify the gains and the losses.

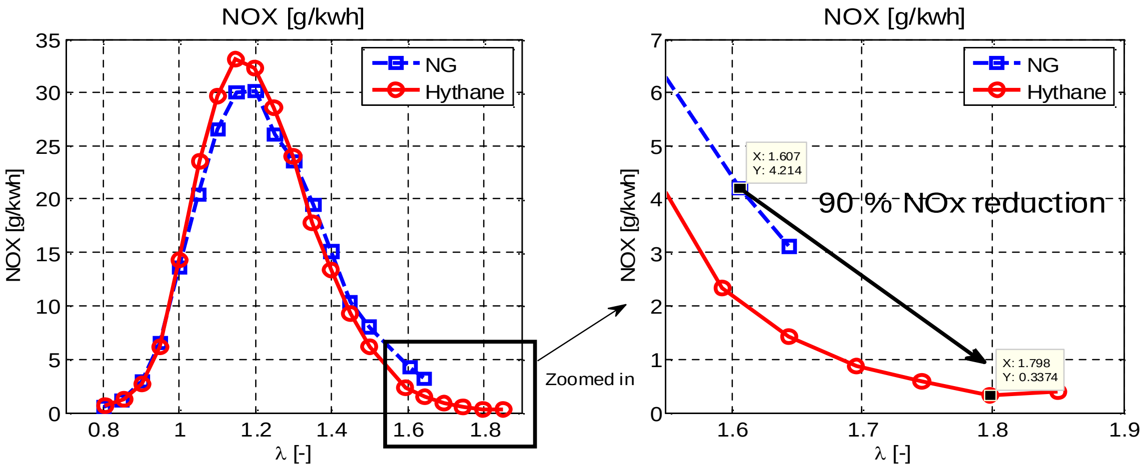

Figure 46. Specific NO_X emissions over lambda range for Hythane and NG are compared. NO_X for operating points close to the lean limit is zoomed in, to clarify the benefits in NO_X reduction

5.3.2 Specific HC versus Lambda

As it is stated in Table 7 the H/C ratio of the natural gas is about 16% lower than for Hythane. Specific HC emissions for natural gas and Hythane are plotted over a lambda range see Figure 47. The figure shows almost 15% reduction in HC emissions which corresponds well to the higher H/C ratio of Hythane.

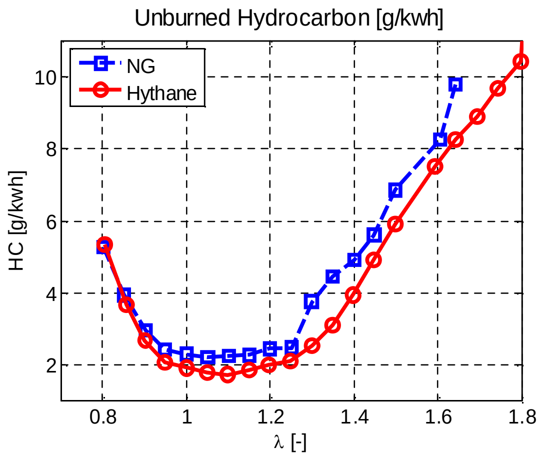

Figure 47. Specific HC over lambda range for Hythane and natural gas are compared.

5.3.3 Specific CO versus Lambda

Specific CO over a lambda range is presented in Figure 48 which shows almost a constant CO reduction with Hythane. The CO reduction is expected since Hythane contains lower carbon atoms than the natural gas.

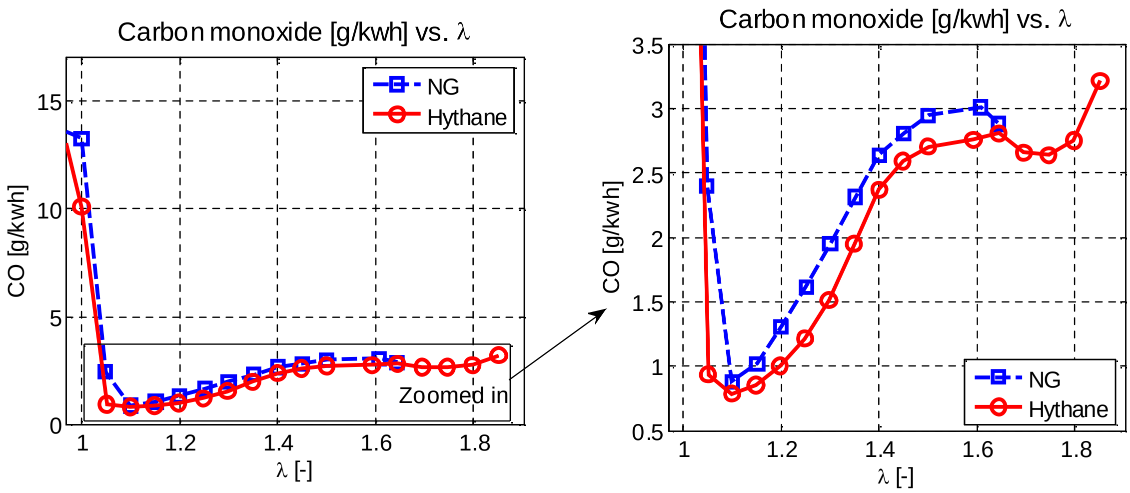

Figure 48. Specific CO emissions over lambda range for Hythane and natural gas are compared. New plot is generated to the right which shows the zoomed in area.

In the second stage of the experiment, the dilution limit of the engine is investigated. By adding EGR the combustion becomes colder and combustion duration longer. With very high EGR rate, it gets difficult to ignite the mixture which results in misfire in some cycles and consequently a sudden increase in cyclic variation. Due to the lower ignition requirement of Hythane the dilution limit of the engine is extended from 26 to 30 percent compared to natural gas at 1000 RPM.

5.3.4 Specific NO_X versus EGR rate

By adding EGR the combustion temperature and oxygen concentration are reduced which results in lower NO_X generation. Since the dilution limit is extended by Hythane, the reduction in NO_X emissions at the dilution limit will be higher than with natural gas. The NO_X emissions over the EGR range is plotted in Figure 49. The operating region close to the dilution limit region in Figure 49 is zoomed in and presented in the same figure to the right. By extending the dilution limit from 26 to almost 30 percent EGR at this operating point, the NO_X emissions can be reduced by 34% which is a remarkable reduction. Normally it is beneficial to dilute stoichiometrically operated engines to improve efficiency and control knock. The engine uses a three-way catalyst to reduce all the three emissions i.e. NO_X, HC and CO simultaneously, so reduced engine-out NO_X emissions may result in using a cheaper 3-way catalyst. The NO_X reduction by extending the dilution limit can be compared to the NO_X reduction at the lean limit. The NO_X reduction by extending the lean limit was much greater but it should be kept in mind that stoichiometric operation allows the use of a 3-way catalyst.

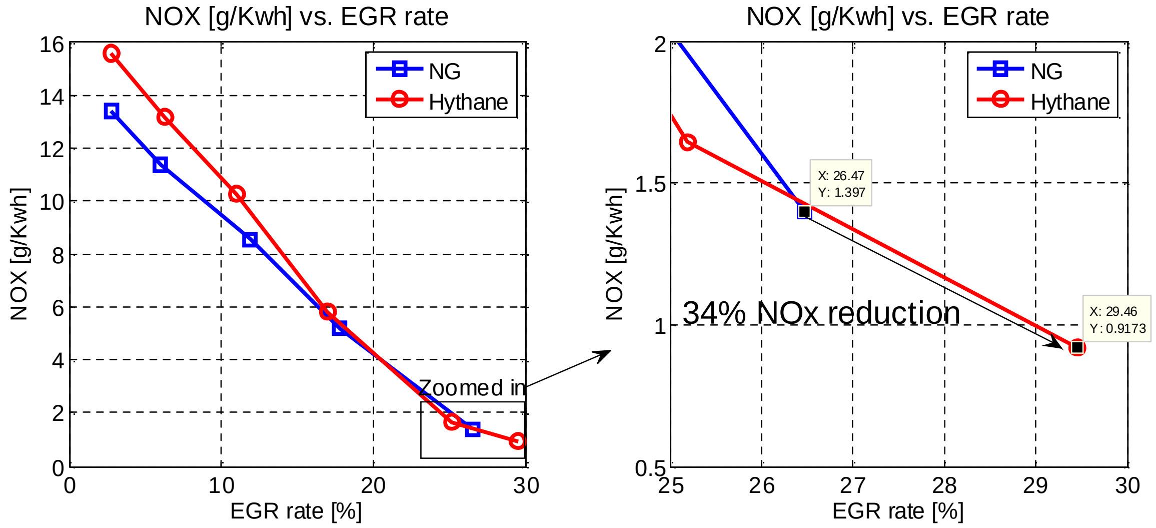

Figure 49. Specific NOX emissions over EGR range for Hythane and natural gas are compared. The operating region close to dilution limit is zoomed in to the right, to clarify the benefits in NOX reduction.

The same trend for HC was observed as reported in the Lambda response section. HC emissions were reduced as Hythane was used since the H/C ratio is higher for Hythane.

5.3.5 Efficiencies and knock margins

In the third stage of the experiments, the engine was operated from low to high load for a range of engine speeds. During this experiment the lambda was kept constant and equal to one i.e. stoichiometric operation. The ignition timing was adjusted to achieve MBT timing and EGR is only used to suppress knock when needed.

Small changes are observed in knock margin of the engine due to the low ignition energy of Hythane. The gas-exchange efficiency of the engine is improved somewhat due to the lower density of Hythane. To achieve the same load level with Hythane the throttle should be opened more which results in lower throttling losses and thereby higher gas-exchange efficiency. The gas-exchange efficiency will be improved further with Hythane if EGR is used in the operating points with high pumping losses since the dilution limit is extended with Hythane.

No remarkable changes in the other efficiencies i.e. combustion and mechanical efficiency were observed. These results together with some other important results such as combustion duration, exhaust gas temperature, CA50 and boost pressure can be studied in detail in [45].

In summary, by increasing the percentage of Hydrogen the gains in reduce HC and CO emissions and losses in terms of increased NO_X emissions are more obvious. By extending the lean and dilution limits of the engine the NO_X emissions will be reduced dramatically to very low levels. This is especially very advantageous for the lean operated natural gas engines due to the fact that they may not need to use any SCR.

6. Alternative ignition systems

The current phase of this project focuses on exploring alternative ignition techniques as after completion of all previous phases it was observed that the capability of conventional spark plug ignition system was the factor limiting the extent of dilution and hence emission reduction and efficiency improvement.

Two most feasible alternative ignition systems were identified, namely diesel pilot injection [46-48] and pre-chamber type ignition system [49-51] but it was soon realized that the former has already received considerable attention as there are products in the market under different names like The Hardstaff OIGI® (Oil Ignition Gas Injection), Westport's High-Pressure Direct Injection (HPDI) applicable to a wide range of engines. Comparatively, however, the concept of pre-chamber ignition has received limited attention and is mainly applied to stationary or large bore marine engine which do not face as severe speed and load transients as experienced by a heavy duty engine for mobile application. Reasons behind this are believed to be limited knowledge about the mechanism of ignition resulting from a pre-chamber type ignition device and hence gaining deeper insight into this mechanism is currently the objective of the gas engine project.

6.1 Summary of the results

As a first step, the performance and emission characteristics of a 6 cylinder gas engine were compared for operation with conventional and a pre-chamber spark plug [52] which was bought from the market [53]. It was observed that the dilution limit extended slightly with pre-chamber spark plug as seen in Figure 50. The main observation, however, was the notable reduction in flame development angle with pre-chamber spark plug (Figure 51) which fortifies the belief that the jets from pre-chamber offer much higher ignition energy than a single point spark. It was also observed that the pre-chamber spark plug cause charge pre-ignition at load exceeding 10 bar IMEPg mandating the requirement of some form of dilution to restrict the in cylinder temperature which also restricted the maximum load achievable with pre-chamber spark plugs as shown in Figure 52. Even so, the operation with pre-chamber spark plug was more stable as can be seen in Figure 53.

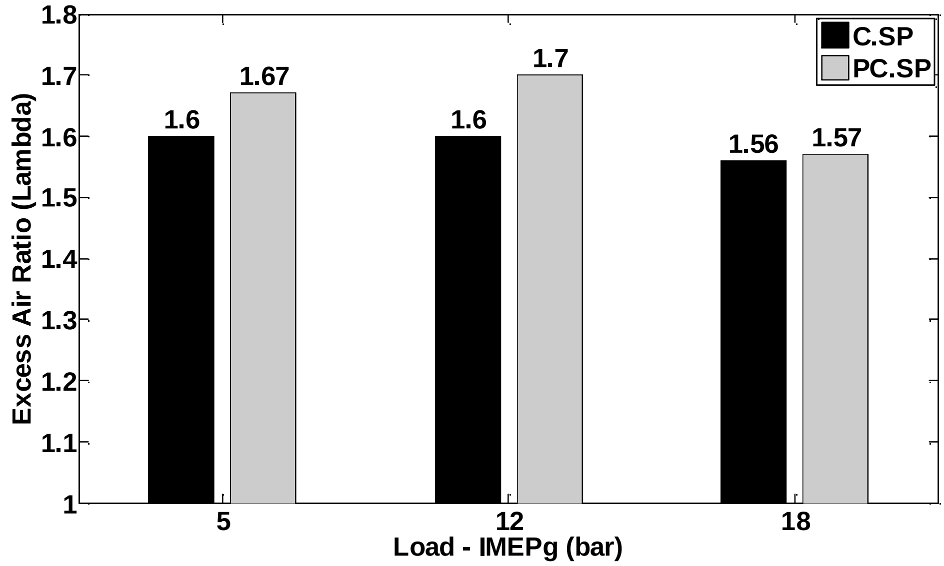

Figure 50. Comparison of dilution limit with excess air at 1500 rpm and various operating load

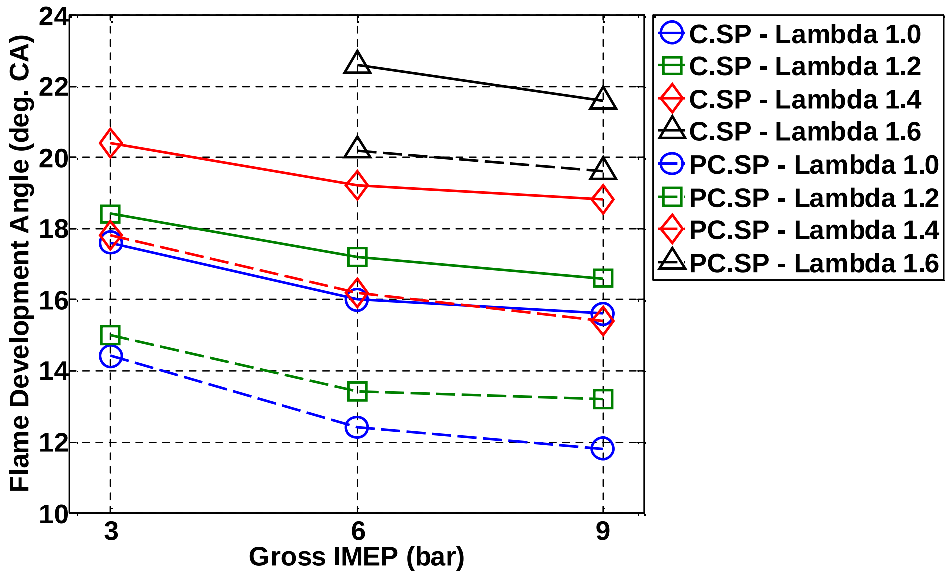

Figure 51. Flame development angle at 1500 rpm and under various load and dilution levels

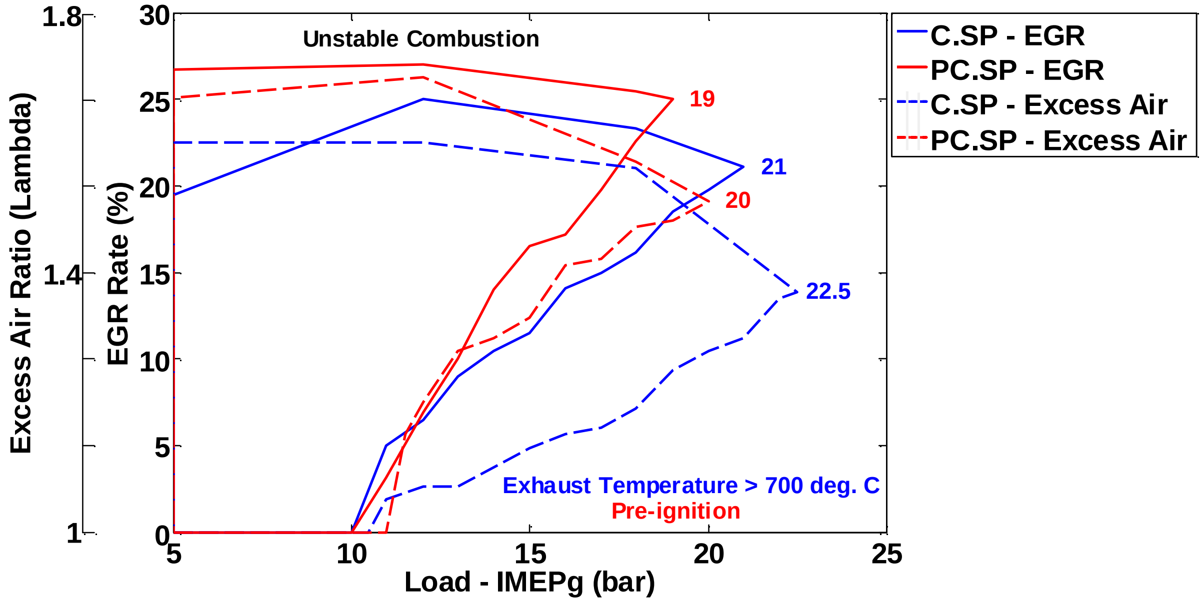

Figure 52. Comparison of minimum dilution required and available operating window at 1500 rpm

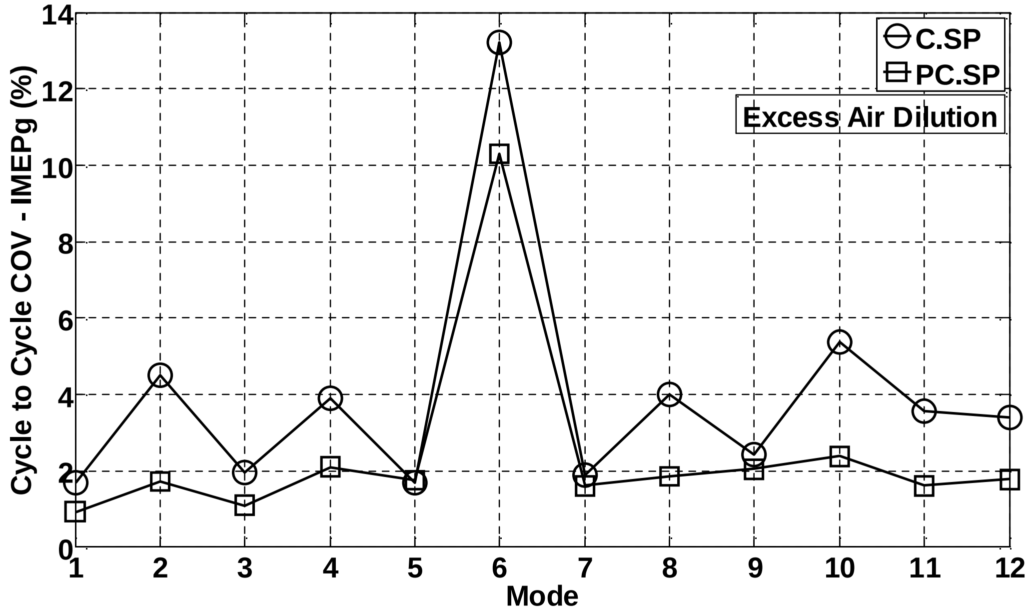

Figure 53. Coefficient of Cycle to Cycle variations in IMEPg during ESC-like12 mode test cycle under maximum excess air dilution

The applicability of ionization current sensing technique when operating with pre-chamber spark plug was also studied [54]. The main observation was the absence of second peak in ion current which is known to coincide with the time of peak cylinder pressure. Also, the strength of first peak was observed to be less affected by dilution and hence contributing to a higher ion current integral value even at dilution limit of operation. Figure 54 and Figure 55 present the ion current signal characteristics under various dilutions for convention and pre-chamber spark plug case respectively.

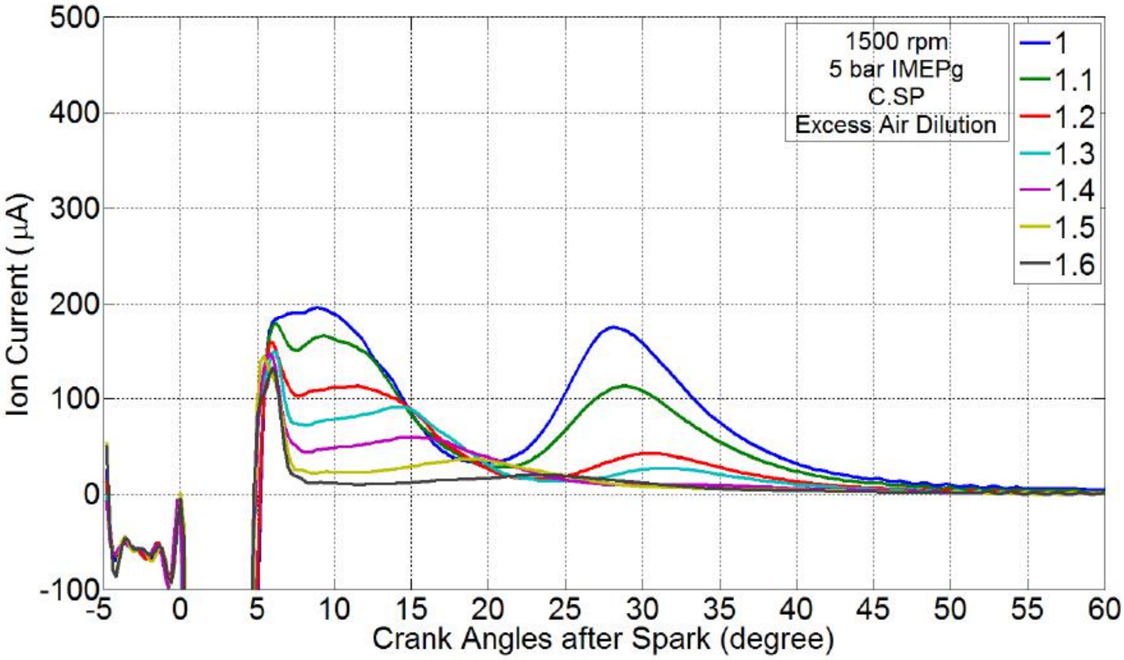

Figure 54. Ion Current signal for different excess air dilution conditions at 5 bar IMEPg for conventional spark plug

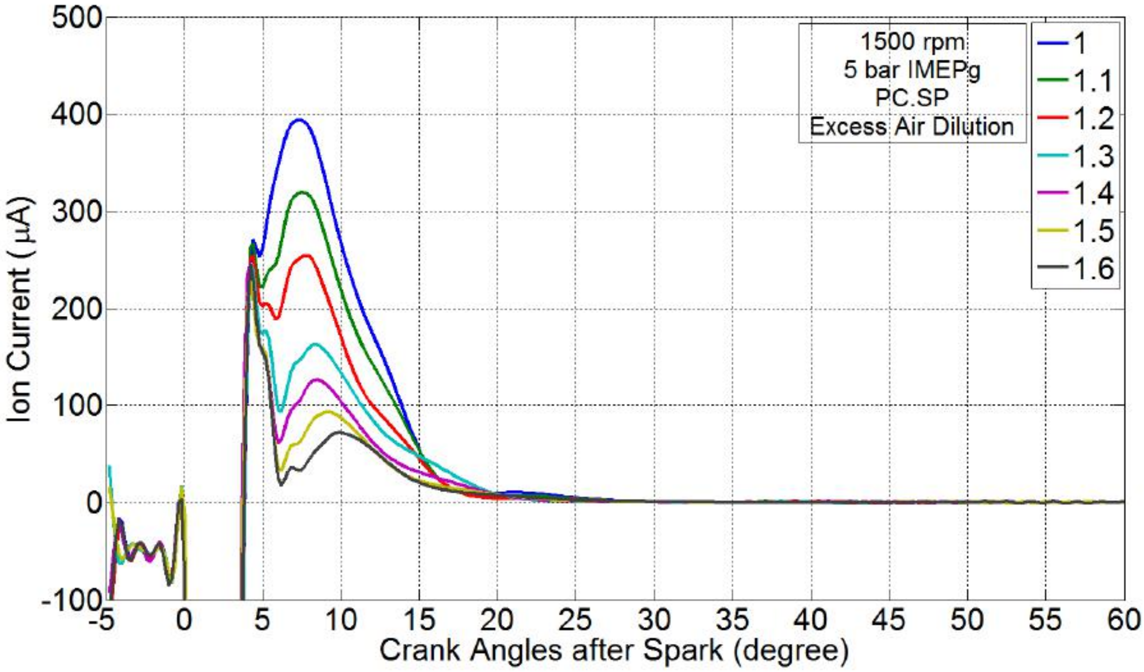

Figure 55. Ion Current signal for different excess air dilution conditions at 5 bar IMEPg for pre-chamber spark plug

Overall, it was observed that the mechanism of ignition resulting from pre-chamber type ignition device was very complex and it was difficult to identify a pattern from the results obtained so far. Several factors like the relative volume of two chambers, design and orientation of connecting nozzle, relative equivalence ratios of two chambers, and location of spark ignition inside the pre-chamber etc. largely affect the resulting ignition in main chamber and hence there is a need to understand the effect of each of these factors by conducting further detailed and controlled experiments.

6.1.1 Literature Survey

The concept of pre-chamber is not new and has been researched for nearly a century. Hence, it was decided to conduct a survey of literature available on different pre-chamber concepts and related research so as to better understan dthe concept and also avoid unnecessary repetition. Following the literature survey, pre-chamber system was identified as one of the divided chamber stratified charge concepts for SI engines which was proposed first by Sir Harry Ricardo in 1918 (US 1,271,942). Since then, several concepts (US 1,568,638 – C. E. Summers, US 2,121,920 – Marion Mallory, US 2,065,419 – A. Bagnulo, US 2,615,437 – N. O. Broderson) with either similar or slightly different objective have been patented and almost all were based on concept of having a readily combustible (stoichiometric mixture) in vicinity of ignition source in a pre-chamber with a very lean mixture in main chamber with the nozzle connecting these chambers so designed as to have a high velocity jet. Thus, the ignition in main chamber mainly relied on the high temperature of jet coupled with its turbulent mixing with main chamber charge.

A major deviation from this concept occurred in 1968 when a Russian scientist, L. A. Gussak, proposed a concept called Lavinia Aktyvatsia Gorenia (Avalanche activated combustion in English), popularly known as LAG-process [55,56] which was based on the chain branching theory developed by N. N. Semyonov [57]. This concept proposed the use of very rich mixture in pre-chamber (equivalence ratio around 2) which when ignited will result in incomplete combustion forming active species and ‘chain carriers’ which were then injected into the main chamber rapidly advancing the chain branching reaction. Available literature offers a very good insight into the mechanism of LAG-process of ignition but also highlights its sensitivity to fuel properties and other design factors. Hence it is necessary to study the behavior of this ignition mechanism with natural gas and at high operating loads as those reported by Gussak are only upto 8 bar IMEPg.

6.1.2 Future work

In light of the literature survey and the conclusions thereof, it is planned to conduct single cylinder experiments with specifically designed pre-chambers to study the effect of relative equivalence ratio between two chambers, effect of operating load (inlet pressure) and the effect of pre-chamber volume and nozzle area on the quality of ignition in main chamber. Gussak’s recommendations of prechamber geometry are chosen as a starting point which will later be altered following the response of this technique with natural gas. It is expected that this ignition system will enable operation with high dilution and hence possibility of employing a higher compression ratio (diesel like) which will further improve the efficiency. This would also enable direct comparison of this combustion concept with advanced conventional fuel combustion concepts like partially pre-mixed combustion (PPC), reactivity controlled compression ignition (RCCI) etc. in terms of operating efficiency and emissions. In future, optical studies with pre-chamber ignition system may also be carried out as a visual aid to further understand the metal engine experimental results.

7. Conclusions

A number of studies have been performed mainly to improve the performance of a heavy-duty 6-cylinder natural gas engine that operates stoichiometrically. To reduce throttling losses at part load operation regions, the concept of closed-loop dilution limit control was developed. A new method to compute cyclic variation was developed which is well suited for transient operation. The dilution limit control coupled with the new method to compute cyclic variation resulted in very simple and cheap strategy to save fuel. Excellent steady-state performance was achieved as well as limited transient functionality.

A Model-based control strategy was suggested to improve the transient performance. State of the art System Identification was used to obtain an empirical model which subsequently was used in a Model Predictive Control structure. Model Predictive Control was shown to be a suitable method for controlling lambda as long as appropriate input variables are chosen for the model.

Since operating the engine at its dilution limit (at part load operation regions) reduces fuel consumption it is advantageous to extend the dilution limit. Increasing turbulence by replacing the combustion chamber resulted in much more turbulence and thereby faster combustion.

The compression ratio used in this type of engine is very conservatively chosen and can be substantially increased without risk of knocking. By increasing the compression ratio from 10.5 to 12, more energy is used in the cylinder rather than left in the exhaust which results in higher efficiency. The higher compression ratio results in lower exhaust gas temperature and more margins for boost pressure and to the highest tolerable temperature by the engine. The heavy-duty natural gas engines have higher exhaust gas temperature than diesel engines but it is still low enough to be tolerable for a VGT. A well-matched VGT will provide desired boost pressure and extend the maximum achievable output power. Load control using the VGT instead of the throttle was also successfully applied in a large operating region (due to its low boost threshold). This resulted in reduced throttle losses and thus increased efficiency.

With higher compression ratio and high boost pressure the knock tendency of the engine increases and there is need for fast and robust EGR control. A comparison between short route and long route EGR was performed and long route showed much better performance in terms of EGR rate, dynamics, mixing, and knock tolerance of engine.

These results show that by mixing different technologies and applying relatively small modifications, significant improvement in engine performance can be gained.

In summary the key features to improve the natural gas engine performance are identified as: Right amount of EGR at different operating regions, Right compression ratio, Variable Geometry Turbocharger, fast burn combustion chambers, Long route EGR system and model-based control.

Moreover, different experiments were designed and performed to identify and quantify the gains and possible drawbacks with Hythane. From the results it was concluded that Hythane is a more appropriate fuel for lean burn operation engines. By extension of the lean limit, NO_X emissions will decrease significantly which could make the NO_X aftertreatment redundant.

As with the alternative ignition technique, it was identified that pre-chamber type ignition system can enable operation with more dilution and hence further improve operating efficiency and reduce emission. However, the design of pre-chamber spark plug used for initial experimentation did not have any additional fueling to the pre-chamber and hence the extension in dilution limit was marginal. Moreover, they caused charge pre-ignition above about 10 bar IMEPg and this is believed to be associated with the spark plug's design and material. Planned future experiments with fueled pre-chamber are believed to extend the dilution limit substantially and also enable operation with higher compression ratio (diesel like) further improving the efficiency.

8. References

[1] J.B. Heywood, ”Internal Combustion Engine Fundamentals”, International edition, McGraw-Hill, New York, 1988

[2] B. Johansson, K. Olsson: "Combustion Chambers for Natural Gas SI Engines, Part 1: Fluid Flow and Combustion", SAE paper 950469, 1995

[3] K. Olsson, B. Johansson: "Combustion Chambers for Natural Gas SI Engines, Part 2: Combustion and Emissions", SAE paper 950517, 1995

[4] Patrik Einewall and Bengt Johansson “Combustion chamber for supercharged Natural Gas Engines “ SAE Paper 970221

[5] Patrik Einewall and Bengt Johansson “Cylinder to Cylinder and Cycle to Cycle variations in a Dix Cylinder Lean Burn Natural Gas Engine” SAE Paper 2000-01-1941

[6] Patrik Einewall, Per Tunestål and Bengt Johansson “Lean Burn Natural Gas Operation vs. Stoichiometric Operation with EGR and a Three Way Catalyst” Lund Institute of Technology, SAE Paper 2005-01-0250

[7] Patrik Einewall, Doctoral Thesis “Study and Development of Techniques to Improve Engine Stability and Reduce Emissions from Natural Gas Engines” Sweden, Lund, 2003

[8] Internet source, Microstar Laboratories “http://www.mstarlabs.com/dataacquisition/5400/5400spec.html” 20101207

[9] Younis, A. F. and Raine R.R., (2000), “Application of a New Technique for the Evaluation of Cycle-by-Cycle Variation of Completeness of Combustion to Changes of Compression Ratio”, SAE Paper 2000-01-1213.

[10] Han, S.B. and Chung, Y.J.,(1999), “The Influence of Air-Fuel Ratio on Combustion Stability of a Gasoline Engine at Idle”, SAE Paper 1999-01-1488.

[11] Jürgen Förster, Achim Gunther, Markus Ketterer, and Klaus-Jurgen Wald, “Ion current sensing for spark ignition engines”, SAE Paper 1999-01-0204.

[12] André Saitzkoff, Raymond Reinmann, and Fabian Mauss, “In cylinder pressure measurements using the spark plug as an ionization sensor”. SAE paper No. 970857, pages 187– 197

[13] M. Hellring, T. Munther, T. Rögnvaldsson, N. Wickström, C. Carlsson, M. Larsson, and J. Nytomt, “Robust AFR estimation using the ion current and neural networks”, SAE 1999-01-1161

[14] A. Abdel-Rahim, N. Henein, E. Vandyne, “Ion current in a spark ignition engine using negative polarity on center electrode”, SAE 2007-01-0646

[15] I. Andersson, L. Eriksson, “Ion sensing for combustion stability control of a spark ignited direct injected engine”, SAE 2000-01-0552

[16] I. Haskara, G. Zhu, Ch. Daniels, J. Winkelman. “Closed loop maximum dilution limit control using in-cylinder ionization signal”, SAE 2005-01-3751

[17] Lars Eriksson, Lars Nielsen, and Jan Nytomt, “Ignition control by ionization current interpretation”, SAE SP 1149, (SAE paper No. 960045):73–79, 1996.

[18] Johansson, Bengt. “Förbränningsmotorer”, Book, Lund, Sweden 2004

[19] M Kaiadi, P Tunestål, B Johansson: “Closed-Loop Combustion Control for a 6-Cylinder Port-Injected Natural gas Engine” SAE Technical Paper 2008-01-1722

[20] M Kaiadi, P Tunestål, B Johansson: “Closed-Loop Combustion Control Using Ion current Signals in a 6-Cylinder Port-Injected Natural gas Engine” SAE Technical Paper 2008-01-2453

[21] P. Einewall, B. Johansson “Combustion Chambers for Supercharged Natural Gas Engine” SAE Technical Paper 970221

[22] C. A. Amann “History of the Internal Combustion Engine” page 36, ICE--volume 8, New York, ASME 1989

[23] B. Johansson, K. Olsson “Combustion Chambers for Natural Gas SI Engines Part 1: Fluid Flow and combustion” SAE Technical Paper 950469

[24] K. Olsson, B. Johansson “Combustion Chambers for Natural Gas SI Engines Part 2: Combustion and Emissions” SAE Technical Paper 950517

[25] R. Bosch GmbH “Automotive Handbook” Book, 6th edition, 2004

[26] K. Miller “Turbo: Real-World High-Performance Turbocharger Systems (S-A Design)”, 2004

[27] HIH. Saravanamuttoo, GFC. Rogers, H. Cohen “Gas Turbine Theory” Book, 5th edition, 2001

[28] I. Haskara, G. Zhu, Ch. Daniels, J. Winkelman. Closed loop maximum dilution limit control using in-cylinder ionization signal, SAE 2005-01-3751

[29] R.H.L. Eichhorn, M.D. Boot, C.C.M. Luijten “Throttle Loss Recovery using a Variable Geometry Turbine” SAE 2010-01-1441

[30] Alvon C. Elrod, Michael T. Nelson “Development of a Variable Valve Timed Engine to Eliminate the Pumping Losses Associated with Throttled Operation” SAE 860537

[31] Allen J., Law D “Production Electro-Hydraulic Variable Valve-Train for a new Generation of I.C. Engines”, SAE Paper Number 2002-01-1109, 2002.

[32] D. Nehmer, L. Flueckiger, S. Bohac, C. Cowland, “Development of a Fully Flexible Hydraulic Valve Actuation Engine Part I: Hydraulic Valve Actuation System Development”, Global Powertrain Congress, 2002

[33] M. Kaiadi, P. Tunestal, B. Johansson “Improving Efficiency, Extending the Maximum Load Limit and Characterizing the Control-related Problems Associated with Higher Loads in a 6-Cylinder Heavy-duty Natural gas Engine” ASME, ICES2010-35012

[34] M. Kobayashi, Y. Aoyagi, T. Adachi, T. Murayama, A. Noda, Y. Goto, H. Suzuki “ Effect of High Boost and High EGR on the Super Clean Diesel Engine” COMODIA 2008, OS-A1

[35] T. Glade, L, Ljung “Control Theory — Multivariable and Nonlinear Methods” Book, English edition, 2000

[36] W. C. Strahle “An Introduction to Combustion”, Combustion Science and Technology Book Series, Vol. 1, Gordon and Breach Publishers, Amsterdam, 1993

[37] N. Apostolescu and R. Chiriac “Optimum Control of S.I. Engine with a Lambda Capability” SAE Paper 960603

[38] U. Kiencke, L. Nielsen “Automotive Control Systems, For Engine, Driveline, and Vehicle” end edition, 2004

[39] K. Collier, R. L. Hoekstra, N. Mulligan, C. Jones and D. Hahn “Untreated Exhaust Emissions of a Hydrogen Enriched CNG Production Engine Conversion”, SAE Paper 960858

[40] S. Munshi, C. Nedelcu, J. Harris, T. Edwards, J. R. Williams, F. Lynch, M. R. Frailey, G. Dixon, S. Wayne, and R. Nine “Hydrogen-Blended Natural Gas Operation of a Heavy-Duty Turbocharged Lean-Burn Spark Ignition Engine”, SAE Paper 2004-01-2956

[41] G. Pede, E. Rossi, M. Chiesa, and F. Ortenzi “Test of Blends of Hydrogen and Natural Gas in a Light-Duty Vehicle”, SAE Paper 2007-01-2045

[42] I. Saanum, M. Bysveen, P. Tunestål, S. Trajkovic “HCCI Combustion of Natural Gas and Hydrogen Enriched Natural Gas Combustion Control by Early Direct Injection of Diesel Oil and RME” SAE Paper 2008-01-1657

[43] P. Tunestål, M. Christensen, P. Einewall, T. Andersson, B. Johansson “Hydrogen Addition for Improved Lean Burn Capability of Slow and Fast Burning Natural Gas Combustion Chamber” SAE Paper 2002-01-2686

[44] M. Kaiadi, P. Tunestål and B. Johansson “Using Hythane as a Fuel in a 6-Cylinder Stoichiometric Natural gas Engine” SAE Paper 2009-01-1950

[45] M. Kaiadi, P. Tunestål and B. Johansson “How Hythane with 25% Hydrogen can affect the Combustion in a 6-Cylinder Natural gas Engine” SAE Paper 2009-01-1950

[46] Yusaf, T., Al-Atabi, M., and Buttsworth, D., "Engine Performance and Exhaust Gas Emissions Characteristics of (CNG/Diesel) Dual-Fuel Engine," SAE Technical Paper 2001-01-1808, 2001, doi:10.4271/2001-01-1808.

[47] Shen, J., Qin, J., and Yao, M., "Turbocharged diesel/CNG Dual-fuel Engines with Intercooler: Combustion, Emissions and Performance," SAE Technical Paper 2003-01-3082, 2003, doi:10.4271/2003-01-3082.

[48] Ahmad - I, N., Babu, M., and Ramesh, A., "Experimental Investigations of Different Parameters Affecting the Performance of a CNG - Diesel Dual Fuel Engine," SAE Technical Paper 2005-01-3767, 2005, doi:10.4271/2005-01-3767.

[49] Attard, W., Kohn, J., and Parsons, P., "Ignition Energy Development for a Spark Initiated Combustion System Capable of High Load, High Efficiency and Near Zero NOx Emissions,"*SAE Int. J. Engines* 3(2):481-496, 2010, doi:10.4271/2010-32-0088.

[50] Attard, W. and Blaxill, H., "A Single Fuel Pre-Chamber Jet Ignition Powertrain Achieving High Load, High Efficiency and Near Zero NOx Emissions," *SAE Int. J. Engines* 5(3):734-746, 2012, doi:10.4271/2011-01-2023.

[51] Geiger, J., Pischinger, S., Böwing, R., Koß, H. et al., "Ignition Systems for Highly Diluted Mixtures in SI-Engines," SAE Technical Paper 1999-01-0799, 1999, doi:10.4271/1999-01-0799.

[52] Shah, A., Tunestal, P., and Johansson, B., "Investigation of Performance and Emission Characteristics of a Heavy Duty Natural Gas Engine Operated with Pre-Chamber Spark Plug and Dilution with Excess Air and EGR," *SAE Int. J. Engines* 5(4):1790-1801, 2012, doi:10.4271/2012-01-1980.

[53] Pre-chamber spark plug from Multitorch GmbH http://www.multitorch.de

[54] Shah, A., Tunestal, P., and Johansson, B., "Applicability of Ionization Current Sensing Technique with Plasma Jet Ignition Using Pre-Chamber Spark Plug in a Heavy Duty Natural Gas Engine," SAE Technical Paper 2012-01-1632, 2012, doi:10.4271/2012-01-1632.

[55] Gussak, L., Karpov, V., and Tikhonov, Y., "The Application of Lag-Process in Prechamber Engines," SAE Technical Paper 790692, 1979, doi:10.4271/790692.

[56] Gussak, L., Turkish, M., and Siegla, D., "High Chemical Activity of Incomplete Combustion Products and a Method of Prechamber Torch

Ignition for Avalanche Activation of Combustion in Internal Combustion Engines," SAE Technical Paper 750890, 1975, doi:10.4271/750890.

[57] Semenoff, N., ”Chemical Kinetics and Chain Reactions”, Oxford University Press, 1935