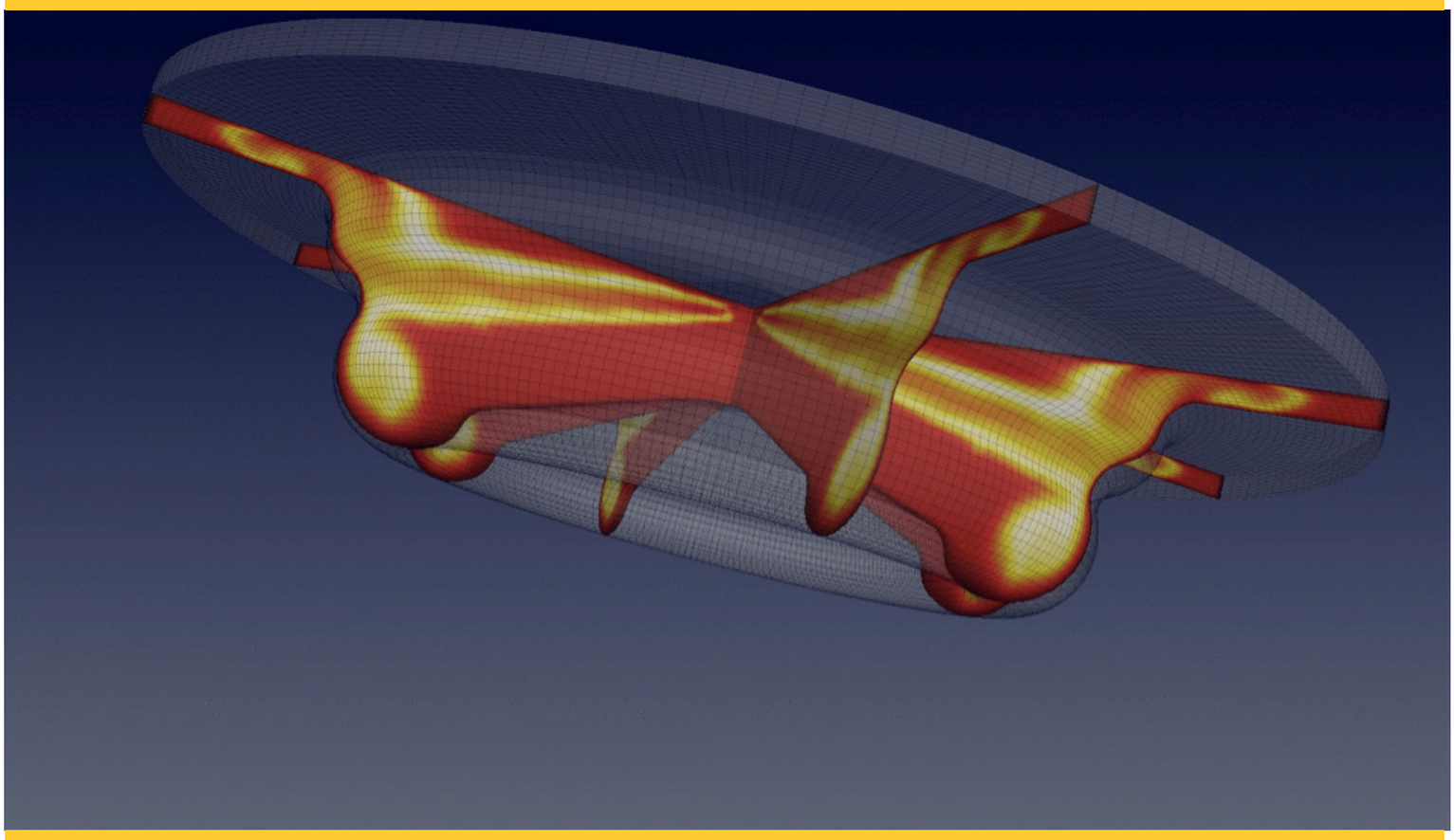
KCFP

Kompetenscentrum Förbränningsprocesser
Centre of Competence Combustion Processes

Annual Report 2011



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KCFP

Centre of Competence Combustion Processes

The Centre of Competence Combustion Processes, KCFP, started July 1, 1995.

The main goal of this centre is to better understand the combustion process in internal combustion engines.Of particular interest are the combustion processes with low enough temperature to suppress formation of NOx and particulates, PM, often called Low Temperature Combustion, LTC or Homogeneous Charge Compression Ignition, HCCI.

The Centre of Competence Combustion Processes has a budget of 22.25 MSEK per year. This is roughly one third each from the Swedish Energy Agency, STEM, Lund University and the Industry.

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1. The Partially Premixed Combustion Project

Partially Premixed Combustion, PPC, is a combustion process between Homogeneous Charge Compression Ignition, HCCI and the classical diffusion controlled diesel combustion. With PPC it is possible to moderate the charge stratification and thus control the burn rate better than with HCCI. In comparison to classical diesel combustion the NOx and particulates can be suppressed with orders of magnitude. KCFP has five different but linked subprojects on PPC.

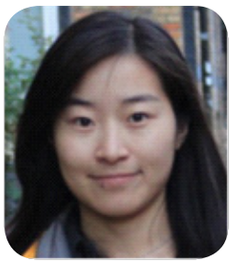
In the Heavy Duty PPC subproject, the first Ph.D. student, Vittorio Manente, has been running a Scania D13 truck size diesel engine with PPC using diesel fuel as well as gasoline in the 69-99 octane range. With gasoline it was possible to run from idle up to 30 bar IMEP even though only 26 bar was reported. Vittorio presented his thesis in September 2010 and was replaced by Peter Andersson. Peter has been focusing on ethanol as PPC fuel and initial tests has shown great potential with much less soot. In The Light Duty PPC project, PPC-LD, the Ph.D. student Patrick Borgqvist has been working with SI and HCCI reference tests in the low load regime. He has also designed a combustion chamber that should enable PPC combustion with the active valve train from Cargine. In the PPC-Fuels subproject the student, Hadeel Solaka, has been working with fuel and combustion interactions with the same fuels previously used in the PPC-HD project. Larger than expected differences between Heavy Duty and Light Duty engine geometries were found.

The PPC optical diagnostics have been in a build-up phase with of two kind of activities 2010. A new Scania D13 optical engine design has been developed and machined but due to delays in the lab, it has not been up and running during 2010 but will do so 2011. Due to the delay, the optical diagnostics activities 2010 have been performed in stationary test rigs instead. The PPC modelling subproject is based on the HCCI and SACI models developed in the previous phase of KCFP.

The PPC model with LES started with using gaseous jet modelling and will eventually move to include also sprays. The fundamental DNS simulations of hydrogen spray and combustion can give additional insight on the PPC type of combustion. The former controls project has due to budget constraints been reduced in the current phase of KCFP, thus less results can also be expected. Even so, the PPC control project has done mode switch SI-HCCI-SI using model based control in the PPC-LD engine having the fully flexible valve system. The developed control strategies can also be useful for PPC.

1.1 PPC - Heavy Duty

Introduction



Mengqin Shen
PhD Student

Partially premixed combustion is a promising way to achieve simultaneous high efficiency and low soot and NOx emissions in a diesel engine. Running with high ratio of EGR and lean combustion, there is a potential to get adequate premixing of fuel and air to avoid soot formation and at the same time maintain a low combustion temperature to avoid NOx formation. With proper fuel injection strategy, the combustion is controlled so that excessive heat-release rates are avoided, efficiency can be improved at high load.

Previous results showed that with 50% EGR and lambda 1.5, PPC could meet US10/Euro6 emission levels in a Scania D13 engine at some loads. Reducing the EGR ratio, higher efficiency can be achieved but more NOx will be generated. When running the engine near stoichiometric ratio, NOx, HC and CO emissions are no more an issue since a three way catalyst can be used to reduce them. It is, however, necessary to pay attention to the efficiency and soot penalties which can be substantial. During the past year, initial test was carried out to understand the effect of EGR and lambda on PPC. Diesel was selected as the first fuel to investigate the fuels influence on EGR and lambda effects.

Updated Engine System

During 2011, some time was spent to update the whole system and get it up and running, new components were added and some worn parts were replaced. A new EGR system was applied in May 2011. A new better-controlled back pressure valve and an EGR pump are used now to get the desired amount of EGR. The schematics of the updated system are shown in figure 1.

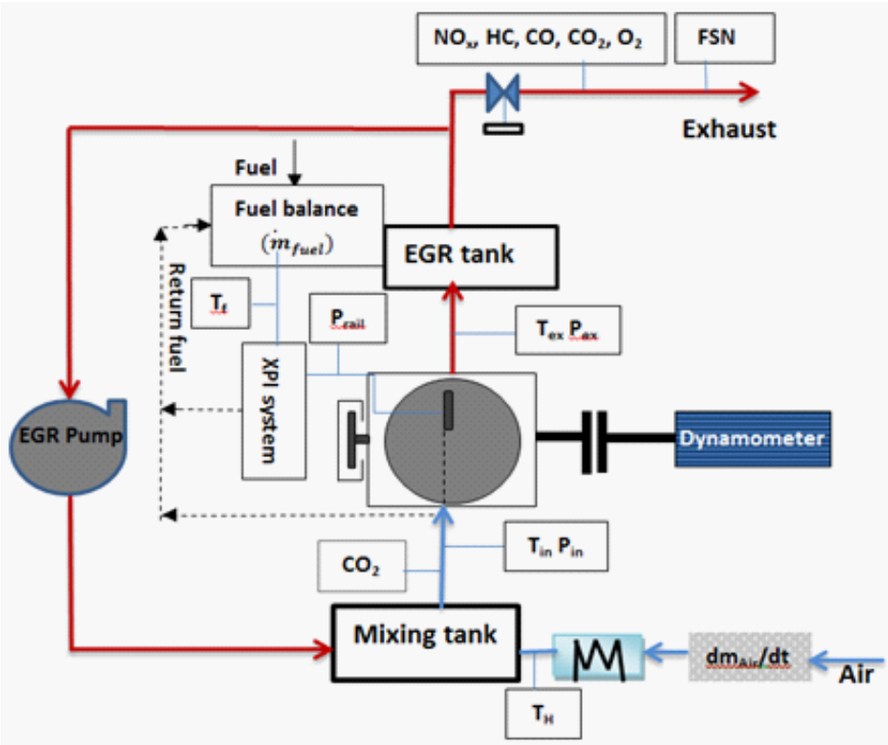


Figure 1 updated engine system

By adjusting the speed of the EGR pump, the EGR ratio can be controlled independent of the pressure ratio between exhaust pressure and inlet pressure. All the sensors were calibrated to assure the accuracy of measured values.

Diesel PPC Results

Diesel fuel was first selected to investigate the effect of EGR and lambda on soot and NOx formation. Figure 2 shows the effect of lambda (EGR=40%) on NOx and soot as well as the effect of EGR when lambda near stoichiometric ratio. With equal amount of fuel injected [FMEP=24bar] into the cylinder, as it shows in figure 3.(a), more soot was emitted as lambda decreased with EGR = 40% while NOx emission was high both at lean combustion and lambda near 1. When lambda was near stoichiometric ratio, figure 3.(b),increased EGR also causes less soot emission. In this case, EGR didn't have much effect on soot emission, compared to that with lambda=1.5 in figure 3. Thermal efficiency got much lower due to higher heat losses.

When operating the engine near stoichiometric ratio, both soot and NOx emissions are high, which indicates that diesel is not a good fuel for this type of PPC combustion.

(a) Fuel MEP=24bar, EGR=40%

(b) Fuel MEP=24bar, lambda=1.06

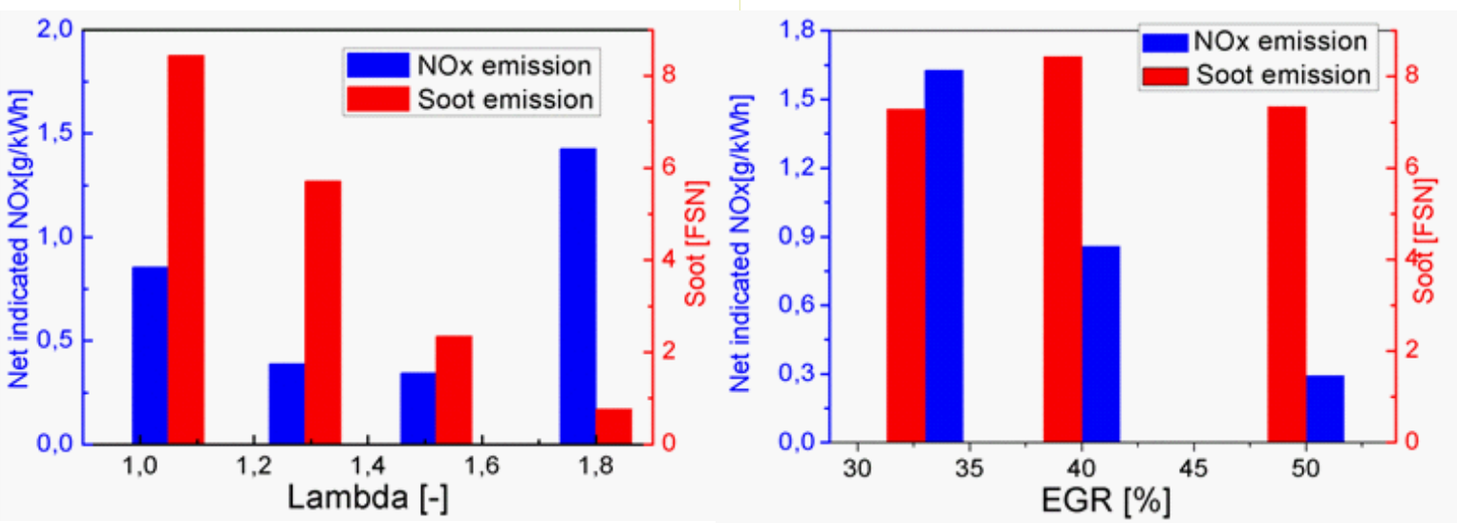


Figure 2 NOx and soot emissions as a function of lambda and EGR

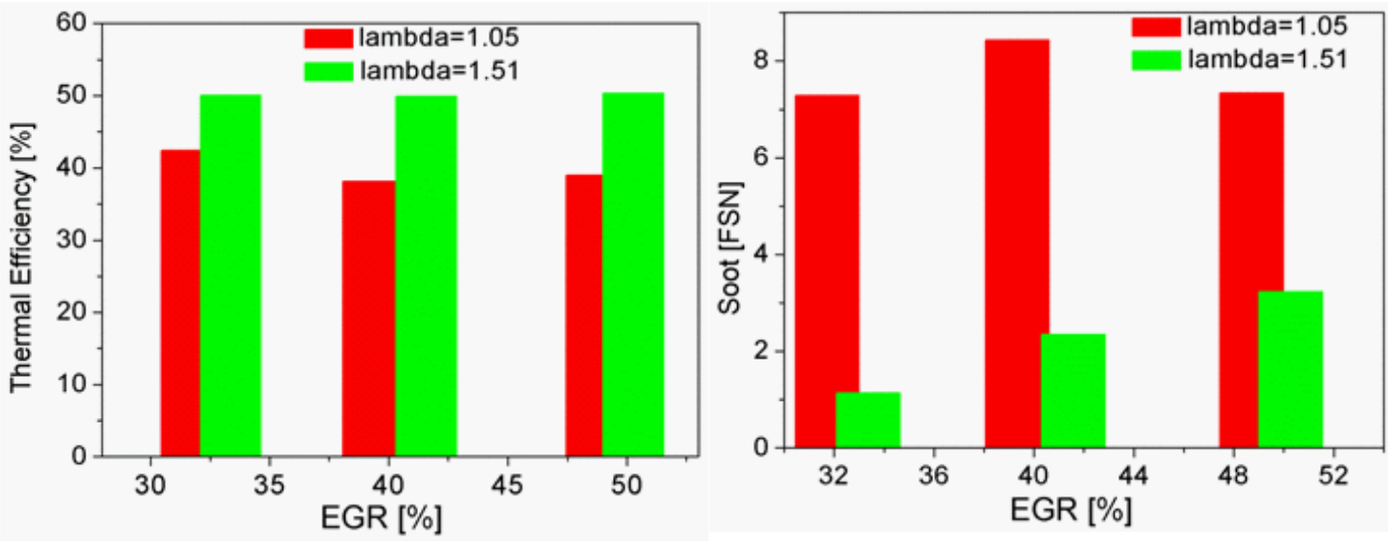


Figure 3 Thermal efficiency and soot emission as a function of EGR(Fuel MEP=24bar)

Future work

For 2012, the current campaign with 95% ethanol and two kinds of gasoline is to be finished. Further work is to modify the engine with two new combustion chambers, combined with both standard and new injectors to investigate how this influences the efficiency and what is the best combustion chamber for PPC.

1.2 PPC - Light Duty

Introduction

Gasoline partially premixed combustion is a promising way to achieve simultaneous high efficiency and low soot and NOx emissions in a diesel engine. Running on gasoline, there is a potential to get adequate premixing of fuel and air to avoid soot formation.

Patrick Borgqvist
PhD Student

With proper fuel injection strategy, the combustion is controlled so that excessive heat-release rates are avoided, which is one of the problems with HCCI combustion at higher loads.

The problem with gasoline partially premixed combustion is the limited operating region running with higher octane number fuels. See Figure 1. The purpose of this project is to increase the attainable operating region and increase the understanding of the combustion process. The approach is to use a variable valve timing system and different fuel injection strategies. Laser diagnostics will be applied to get increased knowledge and detailed information of the combustion.

The low load limitations of different fuels with varying octane numbers and the effect of negative valve overlap have been investigated. The experiments were performed at a low engine speed, 800 rpm, and the goal is to reach idle operating condition (approximately 1 bar IMEPgross).

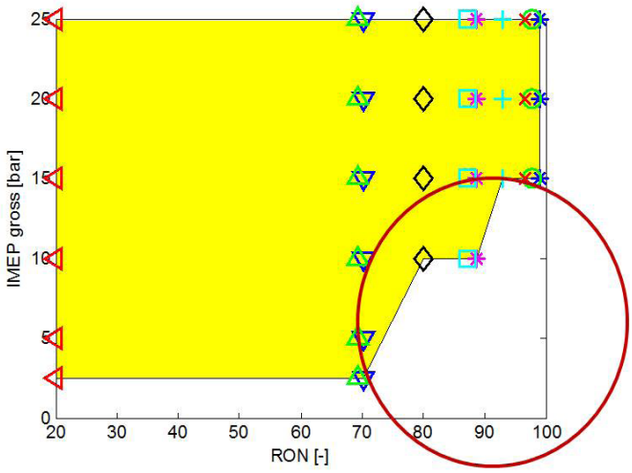


Figure 1. Attainable gasoline PPC operating region plotted against fuel octane number. Data was collected by Vittorio Manente on a heavy-duty engine [1].

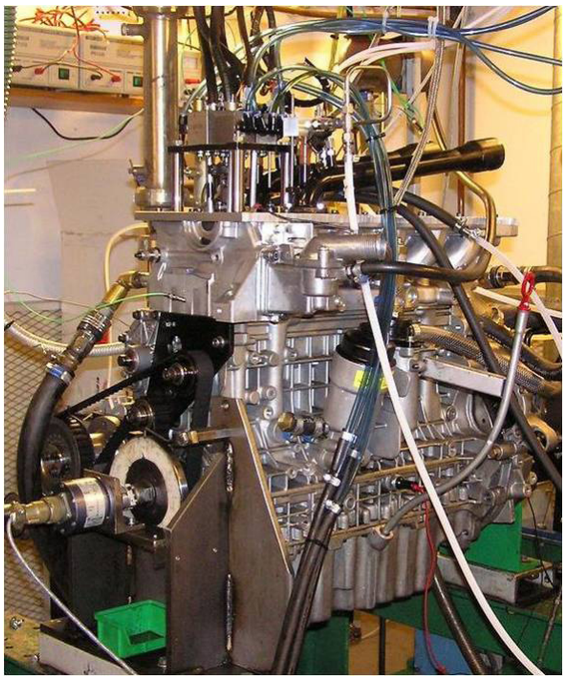


Figure 2. Research engine

Experimental Setup

The research engine, Figure 2, is based on a Volvo D5 light duty diesel engine. It is run on only one of the five cylinders and is equipped with a fully flexible pneumatic valve train system supplied by Cargine Engineering. The valve open speed is fast compared to conventional systems. In order to have valve clearance around top dead center with a standard Volvo D5 piston, the valves are operated with a minimum negative valve overlap of 60 CAD. The engine is run without boosted air. Intake gas (EGR and air) temperature is controlled with a heater in the intake manifold. The Intake gas temperature is constant 40° C, and intake pressure is the same as the ambient pressure, approximately 1 bar.

The engine control system was developed by Patrick Borgqvist and was made with LabVIEW 2009 software. The control system is run from a separate target PC, NI PXI-8110, which is a dedicated real-time system. The target PC is equipped with an R series multifunctional data acquisition (DAQ) card with re-programmable FPGA hardware, NI PXI-7853R, and an M-series data acquisition card, NI PXI-6251. The user interface is run on a separate Windows based host PC. The host and target PCs communicates over TCP/IP.

The fuels used in this experiment are diesel and two different gasoline fuels with octane numbers RON 69 and 87. The gasoline fuels were supplied by Chevron and the diesel fuel is Swedish diesel MK1.

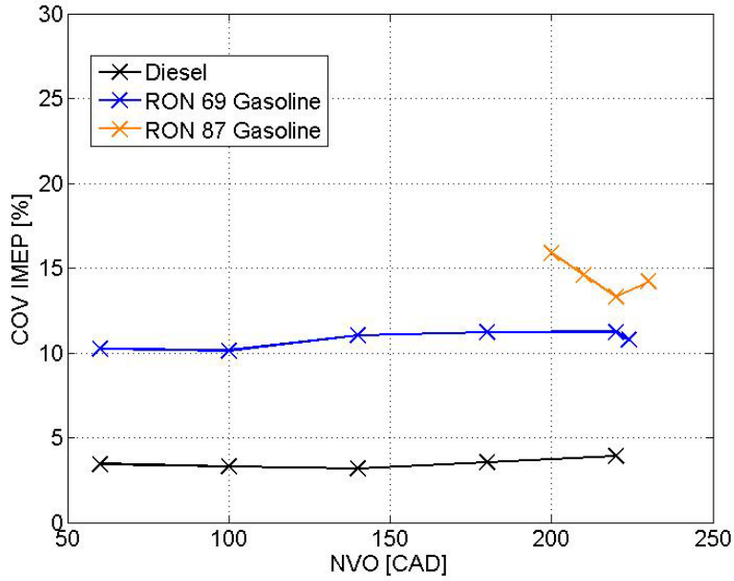


Figure 3. COV of IMEP for the different fuels plotted against NVO

Results

The 69 RON gasoline fuel can be run at as low load as 1 bar IMEPnet without a significant fraction of trapped hot residual gases. An increased fraction of NVO does not improve significantly on combustion stability in the cases with diesel and 69 RON gasoline, as the COV of IMEP is relatively unaffected, as seen in Figure 3. The 87 RON gasoline fuel can only be run with a significant fraction of trapped hot residual gases. The NVO setting that has to be used is as high as 180 CAD NVO and the minimum attainable load was approximately 1.75 bar IMEPnet. NVO has a significant effect on combustion stability in this case, as seen in Figure 3.

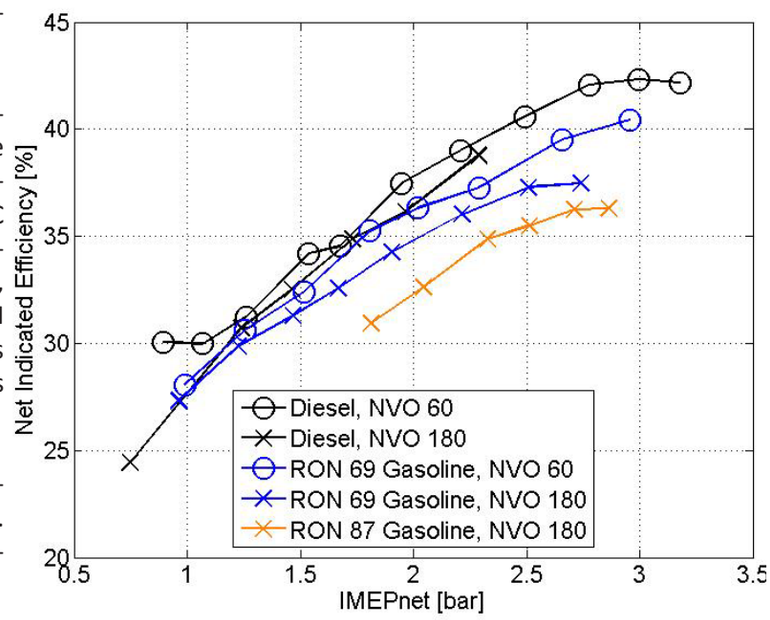


Figure 4. Net indicated efficiency plotted against engine load

The thermodynamic efficiency is higher for the gasoline cases compared to diesel but the combustion efficiency is significantly lower which result in a lower gross indicated efficiency for gasoline compared to diesel at low load operating conditions. NVO can be used to increase the combustion efficiency but the high penalty of reduced gas-exchange efficiency results in a lower net indicated efficiency, Figure 4. The main purpose of NVO is to increase the attainable operating region with high octane number fuels and as low NVO as possible should be used to get as high efficiency as possible.

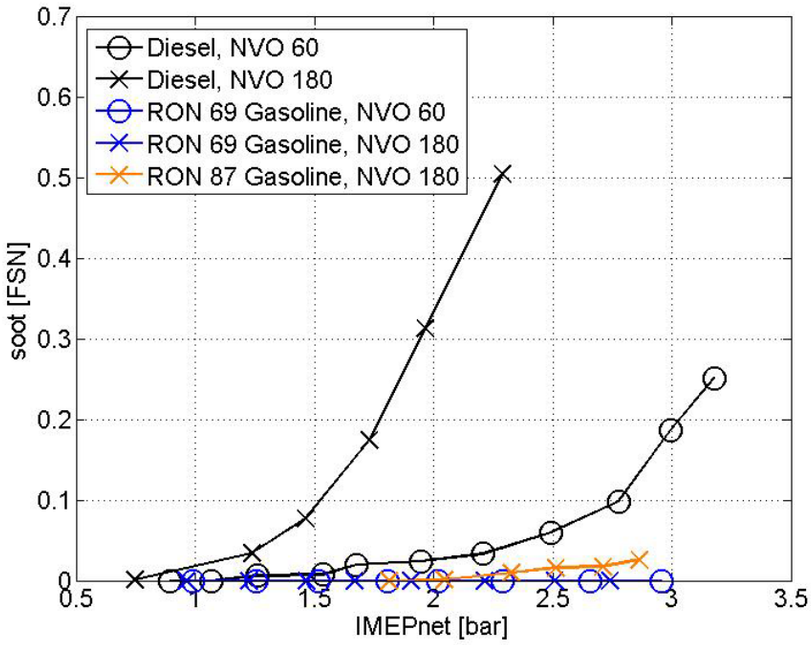


Figure 5. Soot emissions of diesel compared to gasoline with two different settings of NVO, 60 CAD and 180 CAD

In terms of emissions, HC and CO emissions are high for the gasoline cases compared to the diesel cases. NVO has a positive effect on combustion efficiency which results in lower HC and CO emissions. The soot emissions of the gasoline cases are low compared to the diesel case, Figure 5.

Future Work

The single cylinder metal engine experiments on low load operation with negative valve overlap will be finalized. The effect of fuel injection strategy and glow plug will be investigated. The engine will be rebuilt to optical engine configuration. As a first measurement campaign the suggestion is to run high speed fuel LIF experiments.

References

[1] An Advanced Internal Combustion Engine Concept for Low Emissions and High Efficiency from Idle to Max Load Using Gasoline Partially Premixed Combustion, V. Manente, B. Johansson, P. Tunestal, C. Zander, W. Cannella, SAE 2010-01-2198

1.3 PPC - Optics

Wall Temperature Measurements

Johannes Lindén
PhD Student

As outlined in the previous report there is an interest in measuring the in-cylinder wall temperature. This is especially the case when running combustion systems with high pressure rise rates such as HCCI and PPC. There is a possibility that the induced pressure oscillations can rupture the thermal boundary layer and thus cause a severe increase in heat transfer to the relatively cold wall. Such phenomena could be investigated using thermographic thermometry. The surface of interest is coated with a thin layer of a suitable thermo sensitive material often referred to as a thermographic phosphor. Then the coating is illuminated with laser radiation and shortly thereafter, when the excited phosphor relaxes, light is emitted. This resulting radiation from most thermographic phosphors has a duration and spectral characteristics that is dependent on the temperature. Measurements of the signal decay generally results in a temperature reading with higher precision compared to corresponding spectral measurements. Decay times are measured using photo multiplier tubes (PMT) which limits this approach to point temperature measurements. Spectral investigations are often based on taking the ratio between two recordings performed at two different wavelengths. This can be performed using high sensitivity cameras which enable two-dimensional measurements. These techniques have been applied in engines before both within KCFP and at other research centers. Thermographic phosphors have proven to be a robust technique capable of delivering qualitative and quantitative data also when applied in harsh environments such as engines. However, what has been lacking is detailed knowledge about the precision and accuracy of the measurement technique. The two main approaches (decay time and spectral) are both in need of thorough studies in order to reach a level where absolute temperatures can be presented with reliable error bars. The latter is highly important if small temperature variations on an in-cylinder wall should be possible to detect in a trustful manner. An extensive effort has been focused on refining the two-dimensional temperature measurement technique in order to bring it to the desired level.

This study includes detailed investigations on: reproducibility, spectral ratio dependence of phosphorescence intensity, laser intensity, detector gain, detector integration time and phosphor layer thickness. All these parameters must be taken into account if real quantitative data with known measurement errors should be possible. It has been revealed that commonly used ICCD cameras suffer from non-linearity when illuminated with to high irradiance at a much lower count level than expected.

Figure 1 illustrates this problem and indicates that one is restricted to count levels below 30 000 counts (in this particularly case) in order to avoid results that could be mistaken for as false temperature readouts. This is quite remarkable since the maximum counts allowed are 65 000.

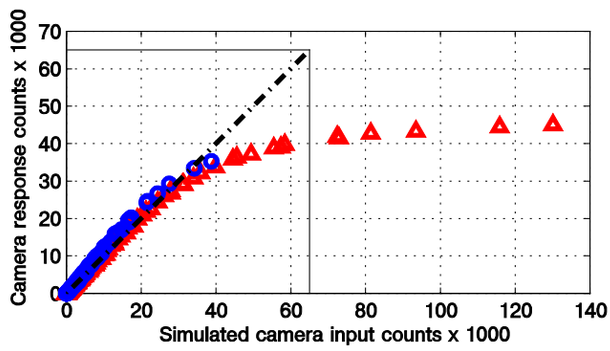


Figure 1 Camera response function for the two different filter wavelengths (400 nm - blue circles; 456 nm red triangles). The straight dashed line indicates the ideal linear response curve, limit to 65 000 counts by the A/D converter's bit depth.

Once sure that the cameras are used in a suitable manner, ensuring stability, reproducibility and linearity, it is possible to expect that the spectral intensity ratio should depend on the surface temperature only. This means that even if the original measurements performed at two different wavelengths should show intensity variations due to laser intensity variations or variations in phosphor concentration, the resulting ratio between these two images should only reflect temperature, se example in Figure 2.

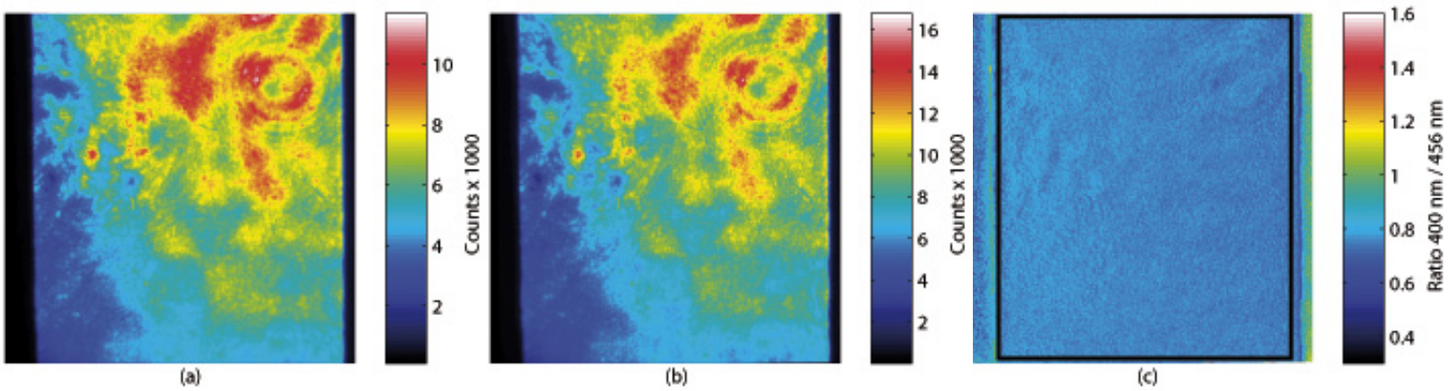


Figure 2 Examples of measurement images at 331 K (59°C). A sample of the thermographic phosphor BAM coated on a surface was placed in a water bath and illuminated with laser. Images if phosphorescence at 400 nm (a) and at 456 nm (b) are divided and results in a ratio image (c). It can clearly be seen how variations in the raw data, caused by laser intensity variations, are canceled out when the ratio is calculated.

The phosphor sample used to achieve the images in figure 2 is submerged in a liquid bath and images are achieved at several temperatures between 0 and 200 degrees C. This is done in order to investigate the best precision achievable. .

For each temperature, the mean and standard deviation of the ratio within the mark area in figure 2 c is evaluated. This results in a graph illustrating the relation between ratio and temperature, see figure 3

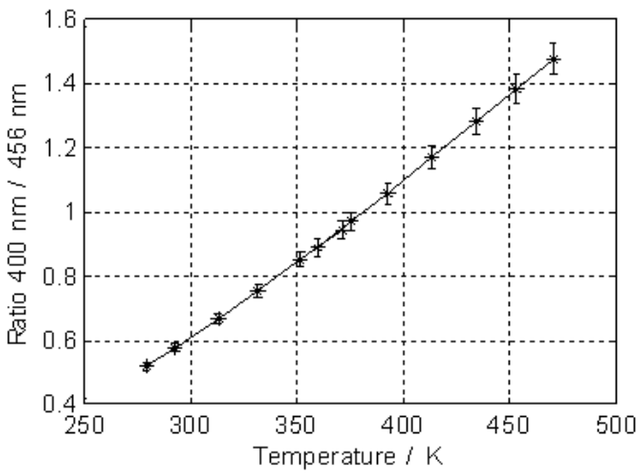


Figure 3 Relation between phosphorescence ratio and temperature. The error bars indicates the standard deviation of the evaluated ratio, which together with the slope of the line (0.005 K-1) enables conversion of the error bars into precision in Kelvin.

As can bee seen, the slope is quite linear (0.005 K-1), and it is clear that the error bars are increasing with temperature, indicating that the precision is decreasing with higher temperature. At room temperature the size of the error bar is ±0.015, giving a precision of ±3 K, or 1 %.

Figure 4 shows the degradation in precision with temperature, from 280 to 471 K.

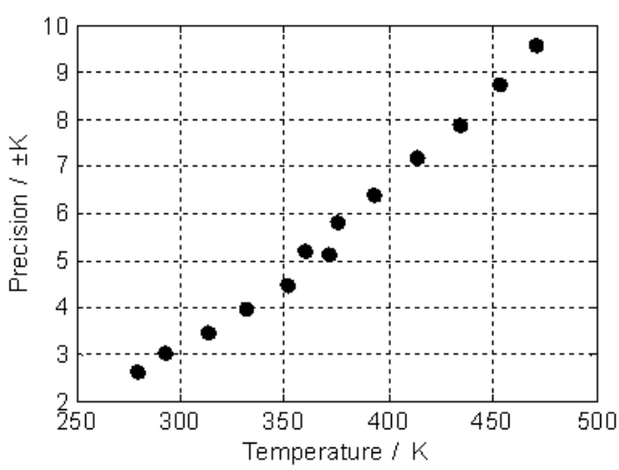


Figure 4 Evaluated precision versus temperature of data achieved using the second configuration setup.

As can be seen, there is a discontinuity between the measurements at 360 and 370 K. This is due to a drift over time in one of the ICCD cameras used, becoming noticeable because of a break in the measurements for 3 hours. During this break the sensitivity of one of the cameras drifted, causing this degradation in precision. This instability in the cameras is hardly noticeable in other applications, and indicates how sensitive the two-color ratio method is to small interferences.

From figure 4 it can be seen that the precision at the lowest temperature (280 K) is ±2.6 K, or 0.9 %, and at the highest temperature (471 K) is ±9.6 K, or 2 %. In order to get an understanding of the meaning of these results when applied on a realistic application, one has to consider the precision in relation to the spatial resolution, since these two quantities involved a trade-off in this case: the precision might get better as the spatial resolution decreases. The area of the phosphorescence images achieved in these experiments are 2.5x2.5 cm which in the image corresponding to about 1000x1000 pixels. This gives an maximum resolution of 40 pixels/mm, or 25µm/pixel. This however is the maximum resolution, and doesn't necessarily say anything about how easily details in the image are distinguished. This dependence on a various numbers of conditions such as optics, focus, motions and obstructing objects in front of the camera. This apparent resolution, henceforth only resolution, has to be estimated using a resolution test target. Doing so one gets a value of the resolution in lines per mm. Therefore, in addition to the measurements, an image of a resolution test target was achieved, in order to measure the spatial resolution.

In order to investigate the relation between spatial resolution and precision, the individual images from each camera was software smoothed with Gaussian filtering using MATLAB, prior to division. In that way the standard deviation within the resulting ratio image decreases, and hence the precision of the ratio is increased, but with the expense that the spatial resolution decreases. By repeating the software filtering on the image of the resolution test target, an estimation of the spatial resolution was achieved. By repeating this procedure with different strengths of the software Gaussian filtering, both to the image of the resolution test target and to the resulting ratio images, a relation between precision and spatial resolution is achieved.

Figure 5 illustrates how these precisions are improved when the phosphorescence images are subjected to Gaussian software filtering.

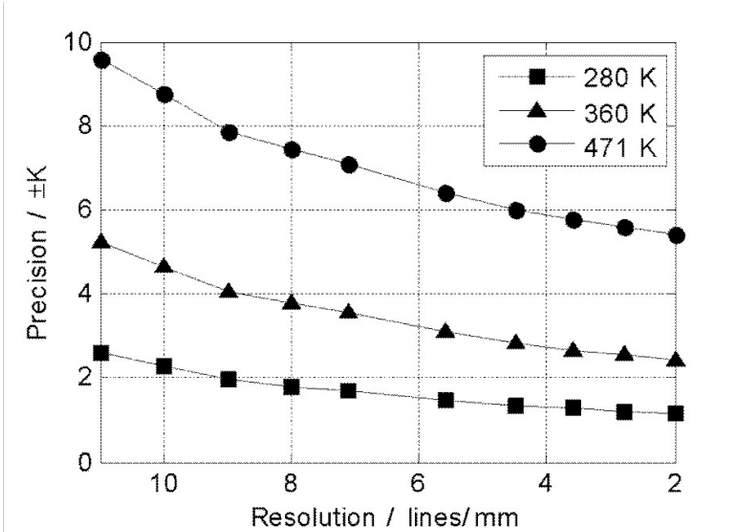


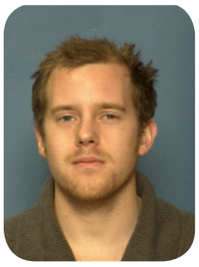
Figure 5 The increase in precision with decreasing resolution at three different temperatures measurements with the second configuration setup, i.e. with two cameras.

For the lowest temperature, the precision is increased to ±1.2 K, or 0.4 %, as the resolution is decreased to 2 lines/mm. For the highest temperature, the precision is increased to ±5.4 K, or 1.2%. Accordingly, in the case of the second configuration setup, the precision vary from 0.4 % to 2 %, depending on absolute temperature and resolution.

During this work it has been noticeable how sensitive this two-color ratio method is two small interferences. There was a non-linearity discovered in the ICCD-cameras used, and the cameras still suffer from an internal drift in sensitivity over time, that causes minor problems (as the discontinuity in figure 4). Despite these problems, a precision of less than 2 % for 2D temperature probing is truly outstanding for a technique with such high temporal resolution.

1.4 PPC Modeling

Introduction



Rickard Solsjö
PhD Student

Mehdi Jangi
PhD Student

The aim of the KCFP modeling project is to perform detailed numerical simulations of Partially Premixed Combustion (PPC) and diesel combustion, to improve the understanding of the underlying physics involved in PPC engines that employ advanced injection strategies. Large eddy simulation (LES) is utilized as a tool to simulate the combustion process in experimental rigs such as heavy-duty diesel engines, operating both in diesel and PPC mode. In LES the energetic large-scale structures of the flow are resolved; and as such the important turbulent structures of fuel and air mixing are captured, and the development of shear layer instabilities and vortex shedding are predicted.

The remaining unresolved phenomena in LES include (a) sub-grid scale (SGS) transport process; (b) spray/gas interaction; (c) turbulence/chemistry interactions. A PPC model has been developed to take into account these phenomena. The performance of the PPC model has been evaluated using several experimental facilities, e.g. Sandia Spray A rig and the Scania D13/D12 engines. The PPC model has been used to simulate the effect of swirl on PPC and the jet-jet interaction in a multiple jets diesel engine.

Evaluation of LES PPC Models

In LES, the spatial resolution is on the Taylor micro-scale, which is generally larger than most spray droplets. To simulate the spray breakup process, Lagrangian Particle Tracking (LPT), is used. The primary breakup process has been neglected in the model since the liquid core is rather small in modern injection systems with high injection pressure and small nozzle orifice. The effect of SGS turbulence on the mixing and transport process is modeled using a one-equation Smagorinsky model. The chemistry/turbulence interaction is modeled using time-scale model, which essentially requires the direct integration of the chemical kinetic mechanisms into flow transport simulations. We developed a high efficiency chemistry/turbulence coupling method, the so-called Chemistry Coordinate Mapping (CCM), which typically speeds up the simulations by a factor of 30. The models are implemented in OpenFoam, an open source CFD code. The code has a dynamic mesh motion handling the piston, and the intake- and exhaust-valve motions.

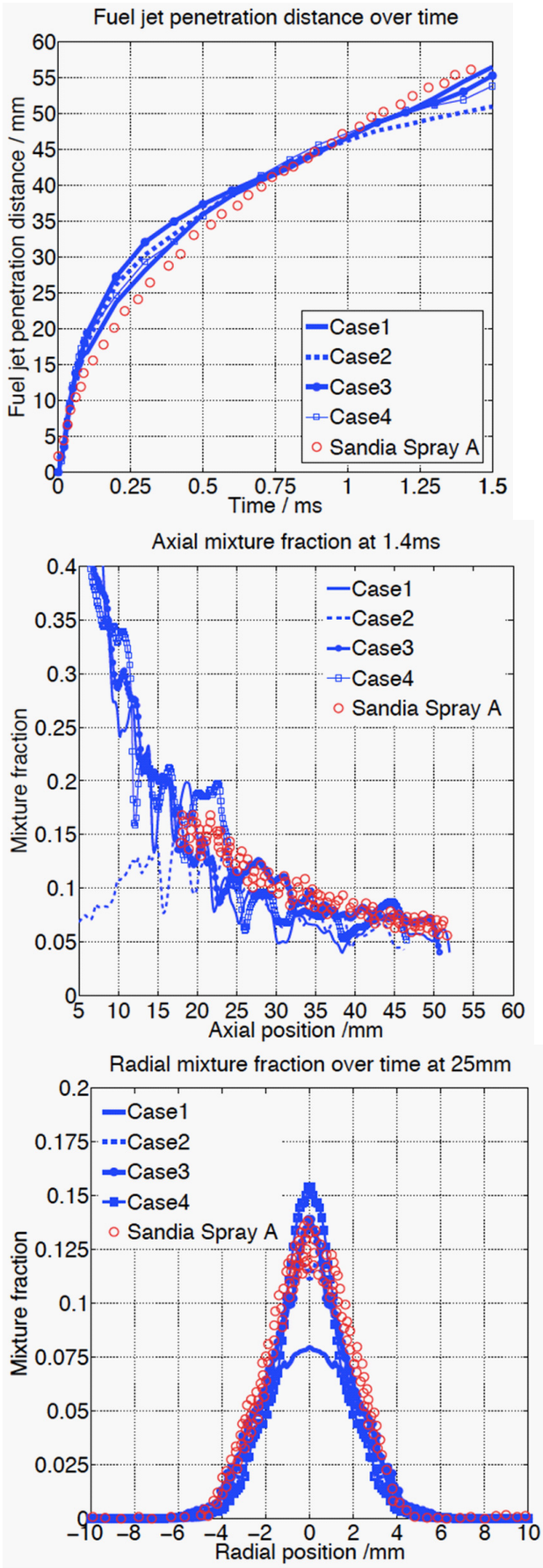


Figure 1. Vapor fuel penetration distance (upper), axial vapor fuel distribution (middle) and radical vapor fuel distribution (bottom), compared to the experimental data performed at Sandia (red circles)

To evaluate the performance of the model, simulations were performed in the Sandia Spray A rig, a constant-volume chamber without combustion. This is to isolate the effects of the flow transport and spray process from the chemical reactions in PPC engines. The simulations were performed at high-pressure injection conditions, approximately 1500 bar. Figure 1 shows the temporal evolution of vapor penetration and the spatial distribution of the fuel vapor using different grid resolutions, and number of spray parcels (case 1-4). The results demonstrated the effect grid resolution, the level of detailed description of the liquid phase of the fuel and additional liquid distributions on the simulation results. The results showed that to truly capture the fuel and air mixing and turbulence, the close to nozzle resolution is important. It was found that a large number of tracking particles of the liquid fuel is necessary for the fuel and air mixing and not to over predict the liquid penetration distance even at high spatial resolution of flow field. The vapor fuel distributions and penetration distance were compared to the baseline experimental case performed at Sandia, spray A, with satisfactory agreement. Details of the results are given in Solsjö et al. [1]. Further evaluation of the PPC model in the presence of combustion in Spray A rig and the D13 engine is ongoing.

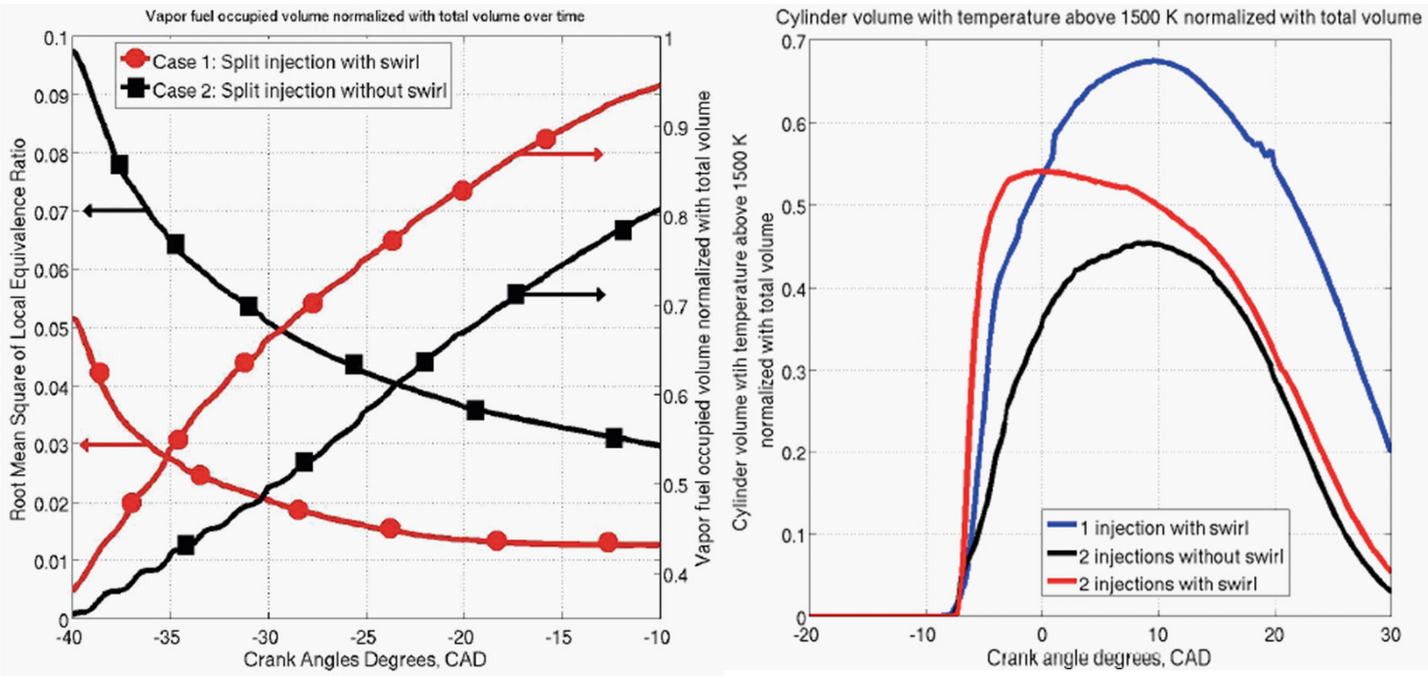


Figure 2. Vapor fuel occupied volume normalized with total volume (left), cylinder volume temperature above 1500K normalized with total volume (right)

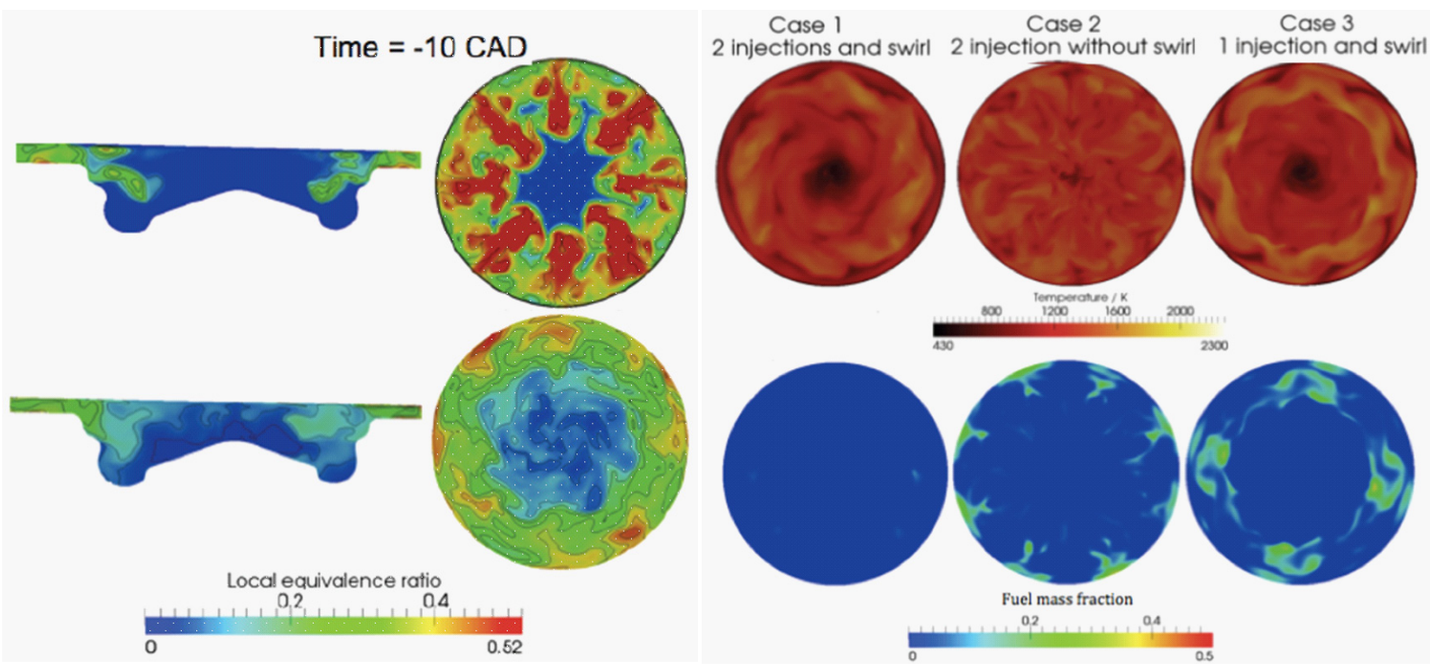


Figure 3. Snapshots of the equivalence ratio at second injection (left) and snapshots of temperature and fuel mass fraction at 30 CAD ATDC (right)

Swirl Effects on Ethanol PPC in a PPC Engine

Using the LES PPC model we simulated the combustion process in a PPC engine modified based on the Scandia D13 engine. The injection strategy is similar to the experimental tests in Scandia D12, with a pilot injection at -60 CAD BTDC and a main injection at -10 CAD BTDC. Three cases were considered; two simulations with split injection, with and without swirl, and one single injection case with swirl and injection at -10 CAD BTDC. Figures 2 and 3 show the fuel distributions at different piston positions. For the split injection case with swirl present in the engine, 25% less fuel was injected. The results reveal the important characteristics of the phenomena occurring in the engine. With sufficient time between the pilot injection and the main injection, fuel distribution becomes more homogeneous in the presence of swirl compared with that of non-swirling case. It was found that with 25% less fuel, the pressure-rise-rate for the two split injection case with swirl is almost identical to that of non-swirling case; indicating a higher efficiency with swirl. Also, the combustion fronts were found to be more uniformly spreading in the cylinder when swirl was present, while in the non-swirling case the combustion front is limited to the neighborhood of the spray fuel jet. One-step chemistry and simplified swirl flow were used in the simulations; it was expected that the simulated combustion phasing might not be exactly comparable to the experiments. The trend that the results reveal can be correct; however, this will be confirmed in the in future engine experiments. The results of this work are to be presented in Solsjö et al. [2].

[1] R. Solsjö, X.S. Bai, “Injection of fuel at high pressure conditions: LES study”, SAE Paper 2011-24-0041, 2011

[2] R. Solsjö, M. Jangi, M. Tunér, X.S. Bai, “Large eddy simulation of partially premixed combustion in an internal combustion engine”, SAE Paper 2012-01-0139, 2012

1.5 PPC - Fuels

Introduction

Hadeel Solaka
PhD Student

The diesel engine has a high efficiency hence it is an interesting and important source of energy for efficient transports. Diesel engine fuel efficiency is limited by emission Aftertreatment due to emissions legislation. However, two important pollutants from diesel engines nitrogen oxide (NOx) and particulates matter (PM) are still challenging.

By introducing a new concept at Lund University called partially premixed combustion (PPC), which is a promising way to achieve high efficiency and low NOx and soot emissions in a diesel engine. One recent focus of the PPC Fuel project has been to identify the limited operating region with partially premixed combustion by running four fuels in the gasoline boiling range together with Swedish diesel fuel (MK1) as a reference fuel in a light-duty diesel engine.

Experiments and Inlet Conditions

The obtainable load region for stable partially premixed combustion (PPC) was examined at loads between 2 and 8 bar IMEPg in a HSDI diesel engine operated on one cylinder. The engine was operated at 1500 rpm with 50% heat release completion (CA50) at 6 crank angle degree after top dead center (TDC). A single injection strategy was used, wherein the start of injection and injection duration were adjusted to maintain the desired load and CA50, as the injection pressure was kept constant at 1000 bar. During the experiments the inlet mixture temperature was 335 K and the desired λ=1.5 and EGR ratio 53±1%.

Results

The higher RON fuels had a higher low-load limit. The low-load limit for higher RON fuels was 7 bar, for low RON fuels it was 5, bar and for MK1 it was 3 bar IMEPg. With both EGR ratio and λ constant, the inlet pressure was a function of load (see Figure 1). However, in order to maintain a stable combustion below the low-load limit, an extended boosting, with a corresponding increase in λ, was used (see Figure 1 right). Figure 1 to the left shows the applied extended boosting at lower loads. This strategy also gives information about the relevance of the target λ =1.5, since different fuels have different low-load limits the emission performance can be compared between both the fuels and in relation to λ. During the experiment the EGR ratio was kept at 53±1%, hence the air-fuel ratio was increased below the low-load limit due to instable combustion or misfire.

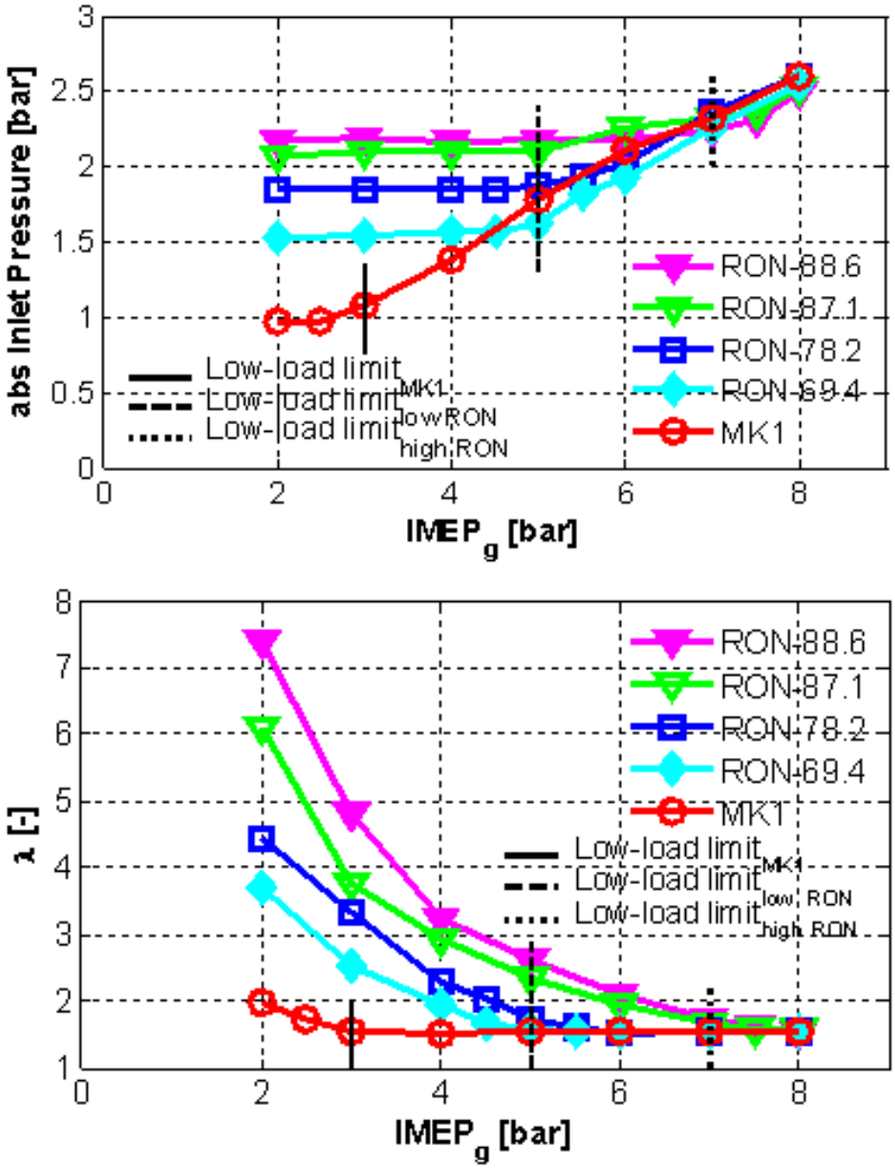


Figure 1. Absolute inlet pressure (upper) and lambda (bottom) as a function of load for different fuel

Figure 2 shows the smoke emissions (upper) and the emission index of NOx, gram NOx per kilogram fuel (bottom) as a function of the load. Between 2 and 4 bar IMEPg, all fuel had smoke levels below the detection limit 0.01 FSN. The 69.4 RON gasoline fuel had detectable smoke at 8 bar IMEPg, while MK1 produced significant smoke levels between 5 and 8 bar IMEPg (see Figure 2 upper). The rate of soot oxidation depends mostly on turbulence, temperature and available oxygen. Since similar combustion phasing, boosting and inlet temperature is used at the highest load for all fuels the soot formation can be expected to depend on fuel characteristics and mixture formation. The difference in ignition delay is, however, small between the low RON fuels and MK1 at the highest loads. It is thereby indicated that the fuel composition had a far stronger influence on the smoke level than any characteristics of the combustion process.

For a given diesel engine, there are three major factors that influence NOx formation including air-fuel ratio, EGR and combustion phasing. Also the NOx level depended both on fuel properties and λ. For the gasoline fuels, the NOx level was close to 0.2 g/kg fuel over the low-load limits. At lower loads, below the low-load limits, the NOx level increased rapidly as the load decreased mostly due to an increased combustion temperature and λ. At the lowest load the level was between 1.2 and 2 g/kg fuel (see Figure 2 bottom).

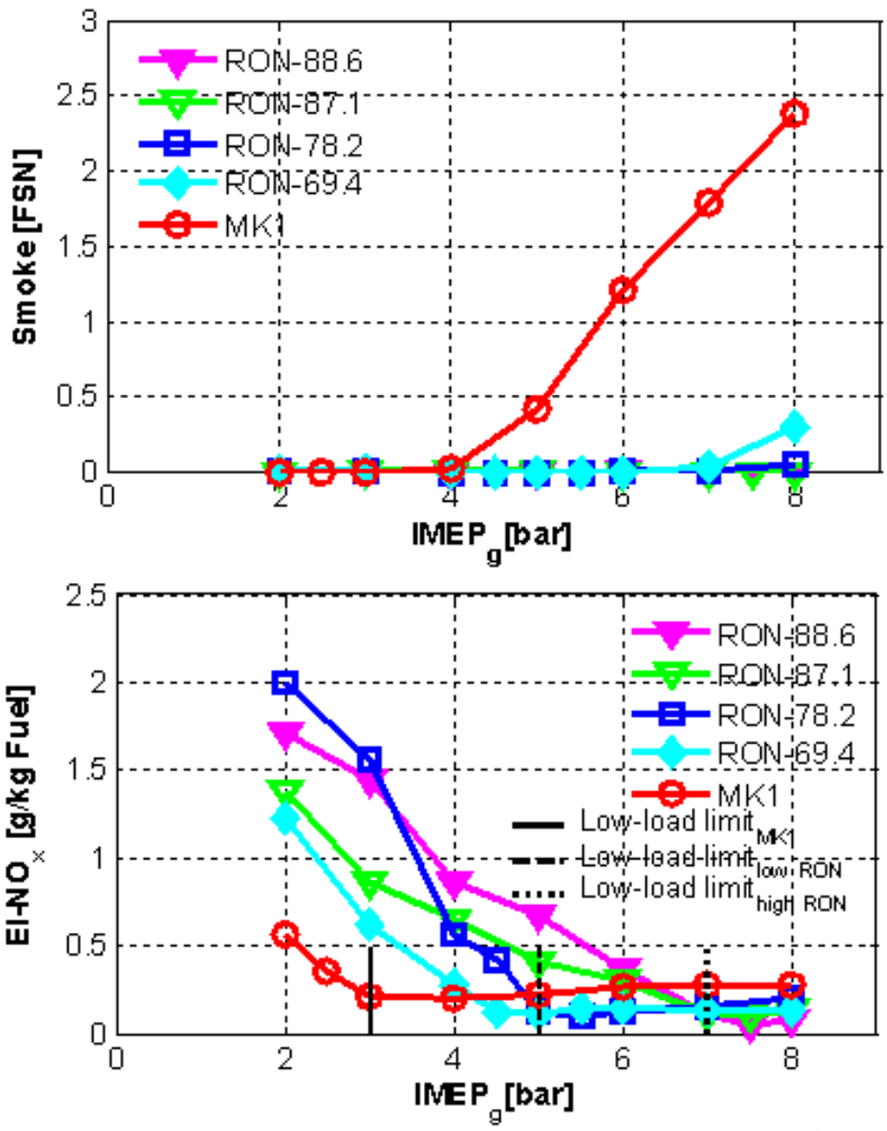


Figure 2. Smoke (upper) and emission index NOx (bottom) as a function of load for different fuel

Conclusions and Summary

- The low-load limit with the stable combustion at λ=1.5 was between 5 and 7 bar IMEPg for the gasoline fuels, and a higher limit for the higher RON values. Diesel had the lowest low-load limit at 3 bar IMEPg.
- By increasing λ with kept EGR ratio, with extended boosting, all fuels could be operated down to 2 bar IMEPg. The higher oxygen fraction in combination with the higher pressure at the top dead center favored the combustion stability.
- The fraction of premixed combustion was not distinctly different between the MK1 and the gasoline fuels. However, MK1 showed the longest combustion duration independently on load. This was an indication of a slower reaction rate during the premixed combustion for MK1.
- The duration of the low temperature reactions depended on the ignition delay, where a prolonged ignition delay gave an increased duration of LTR.
- The fraction of the LTR decreased with increased load from around 5% at 2 bar IMEPg to around 3% at 8 bar IMEPg. MK1 had a similar fraction as the gasoline fuels at the low load, but above 5 bar IMEPg the LTR was not presented.
- Smoke was higher for the MK1 than for the gasoline fuels
- HC and CO emissions were lower for MK1 than for gasoline fuels below the low-load limits, while above the low-load limits gasoline fuel showed lower emissions than MK1.
- The indicated efficiency was not distinctly different between the fuels at higher loads. However, below the low-load limits MK1 and the lowest RON fuel showed higher values.

Future Work

The coming research is focused on finding a combination of reference fuel (PRF) with toluene and ethanol. Using a DoE model to find the proper blend of the fuels, that has similar ignition quality and study their effect on LTR phase and emissions.

1.6 PPC - Control

Introduction

For the purpose of fuel efficiency and low emissions, the focus of this project is model-based control of PPC. Physics-based models are favored due to the improved portability of the resulting controllers as well as the added insight into the physical process. During the year, a physics-based single-zone model of PPC combustion has been developed and implemented in the JModelica.org platform. The overarching goal of the modeling efforts is to be able to use it explicitly for control design and optimization and the model complexity and implementation were chosen with this in mind. The single-zone approach reduces the complexity of the resulting model compared to multi-zone models and the chosen platform allows optimization problems to be fomulated based on the model equations. An initial use of this possibility is automatic calibration of the model parameters, which has been achieved.

Anders Widd
PhD Student

Patrick Borgqvist
PhD Student

Ongoing and future work involves optimal control of injection timings and valve strategies to achieve a desirable heat release profile.

Modeling

The model is a single-zone model, meaning that it does not consider spatial variations within the cylinder. The model includes heat losses to the cylinder wall as well as vaporization losses.

The in-cylinder pressure is computed using the first law of thermodynamics. The total change in thermal energy inside the cylinder is composed of the heat release from the combustion of fuel, the vaporization losses, and the heat losses. The vaporization losses were computed using the heat of vaporization for the fuels used while the heat losses were computed assuming only convection between the cylinder wall and the gas and used the Hohenberg expression for the convection coefficient.

The injected fuel was modeled as either vaporized, prepared, or burned fuel with first order differential equations governing the transitions.

The vaporization and subsequent mixing were modeled as proceeding at a constant rate while the burn rate was set to the Arrhenius rate. A quadratic dependence on burn duration was included, which has been used successfully in previous models.

Implementation

The model was implemented in the Modelica language, an object-oriented, equation-based modeling language aimed at modeling of complex physical systems. To enable optimization based on the model, the open source framework JModelica.org was used. JModelica.org extends Modelica with support for dynamic optimization and solves the resulting optimization problem using a collocation method. The derivatives were calculated using a CasADi (an open-source tool for computer algebra implementing automatic differentiation) and the numerical optimization was done using IPOPT (an interior point solver for large scale nonlinear optimization). Since discrete events in the model (such as start of combustion) can be troublsome in the optimization, smooth approximations were used. Simulating the resulting model after compilation took approximately 0.1 seconds on a desktop computer.

Calibration

The model contains several parameters that need to be adjusted to the specific engine and fuel used. Calibration of physics-based single-zone models is often time consuming. Most reports on calibration of engine models are sequential so that a few parameters at a time are determined. The approach presented here allows for a simultaneous automatic calibration of the unknown parameters. Particular care was taken to proper numerical scaling of the model variables and derivatives. The calibration was then formulated as an optimization problem aiming to minimize the quadratic error between the measured pressure and that of the model during the closed part of the cycle.

Experimental Setup

The model was calibrated against data obtained from a Volvo D5 lighty duty engine operated on a single cylinder. The engine was equipped with a fully flexible pneumatic valve train system and direct injection of fuel. The engine was run naturally aspirated without cooled exhaust gas recirculation. The engine had a displacement volume of 4.8 liters and was run on diesel fuel. The experiments were performed at an engine speed of 1500 rpm and the start of injection was varied between 16.7 and 12.7 degrees before top dead center. Data sets of 1000 cycles were collected and the average pressure and heat release traces were used in the validation. The engine and engine control system were described in more detail in the previous report.

Calibration Results

Figure 1 shows the measured and model pressure. The data was recorded with start of injection 15.6 degrees before top dead center. The pressure decrease in response to fuel evaporation and heat losses before combustion is slightly exaggerated, but the overall agreement compares well with models of similar complexity.

Figure 2 shows the total change in thermal energy from the model and calculated from the measured pressure. The dip due to vaporization and the overall heat release are captured fairly well.

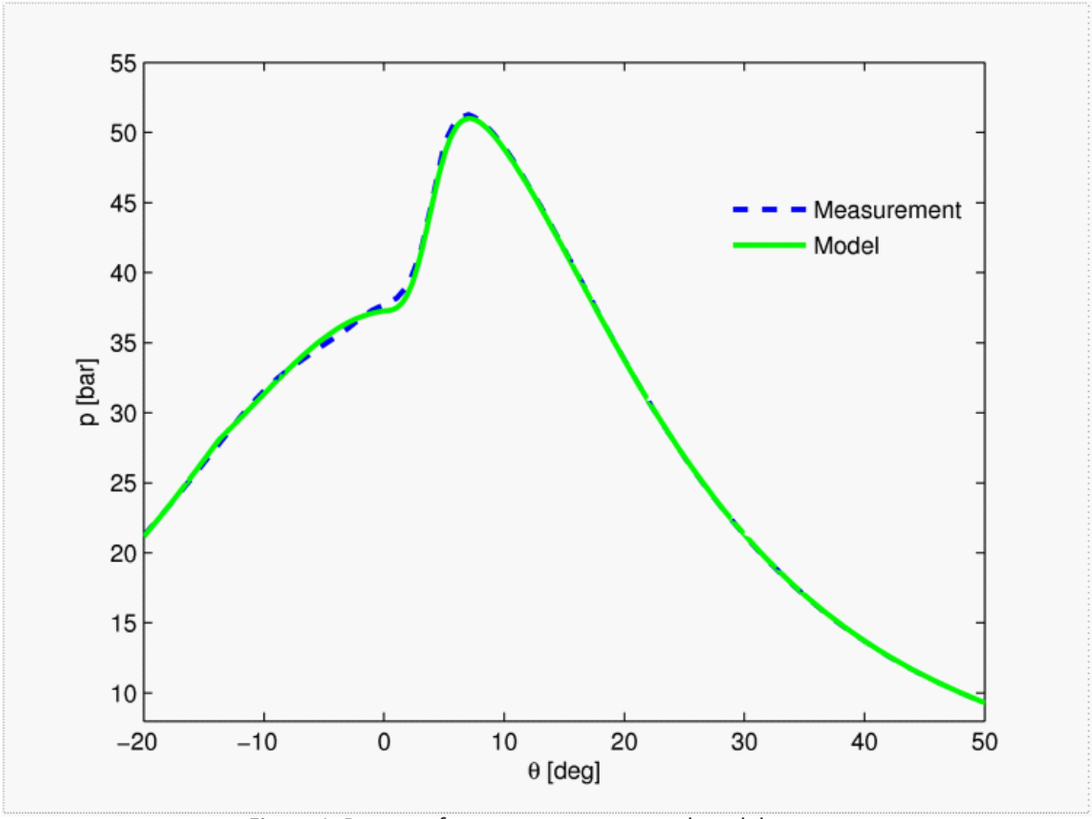


Figure 1. Pressure from measurements and model output

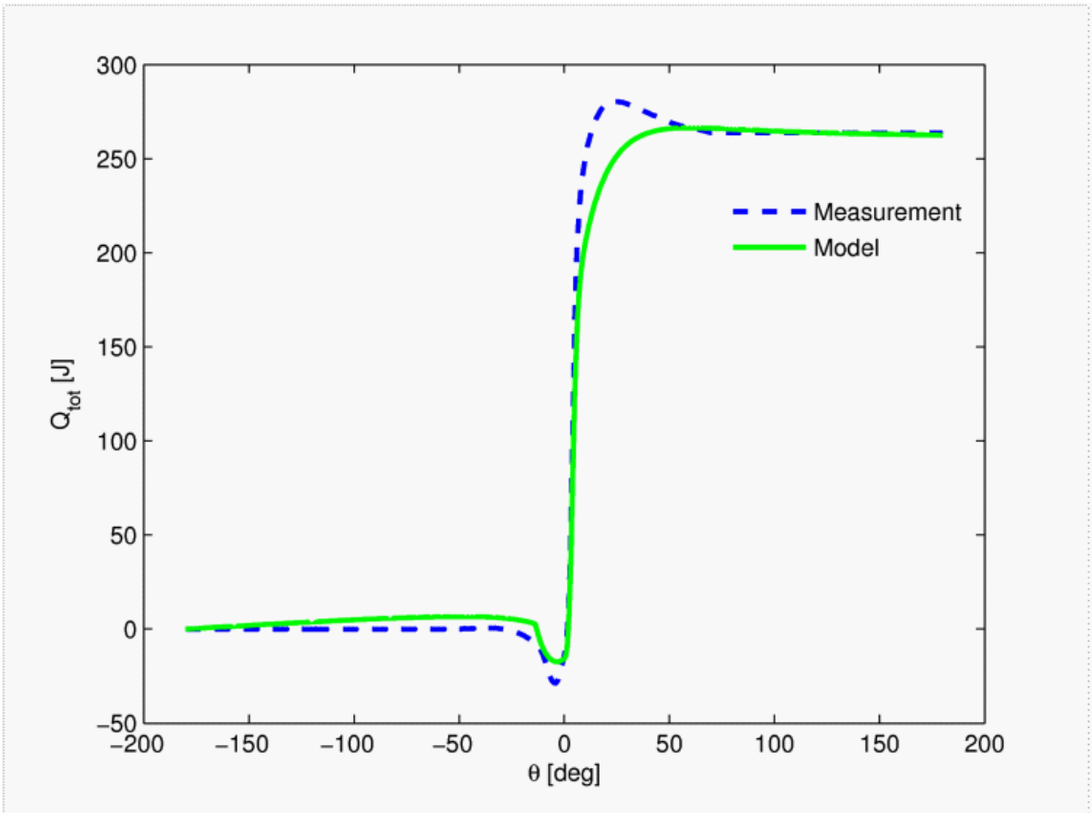


Figure 2. Heat release from measurement and model output. The heat release was computed excluding effects of heat transfer and vaporization

Figure 3 shows the pressure traces for four different fuel injection timings (15.7, 13.7, 16.7, and 12.7 degress before top dead center) and the corresponding model results. The model parameters were not altered for the different cases. The agreement is fairly good in all cases with slightly worse results for the two earlier injection timings.

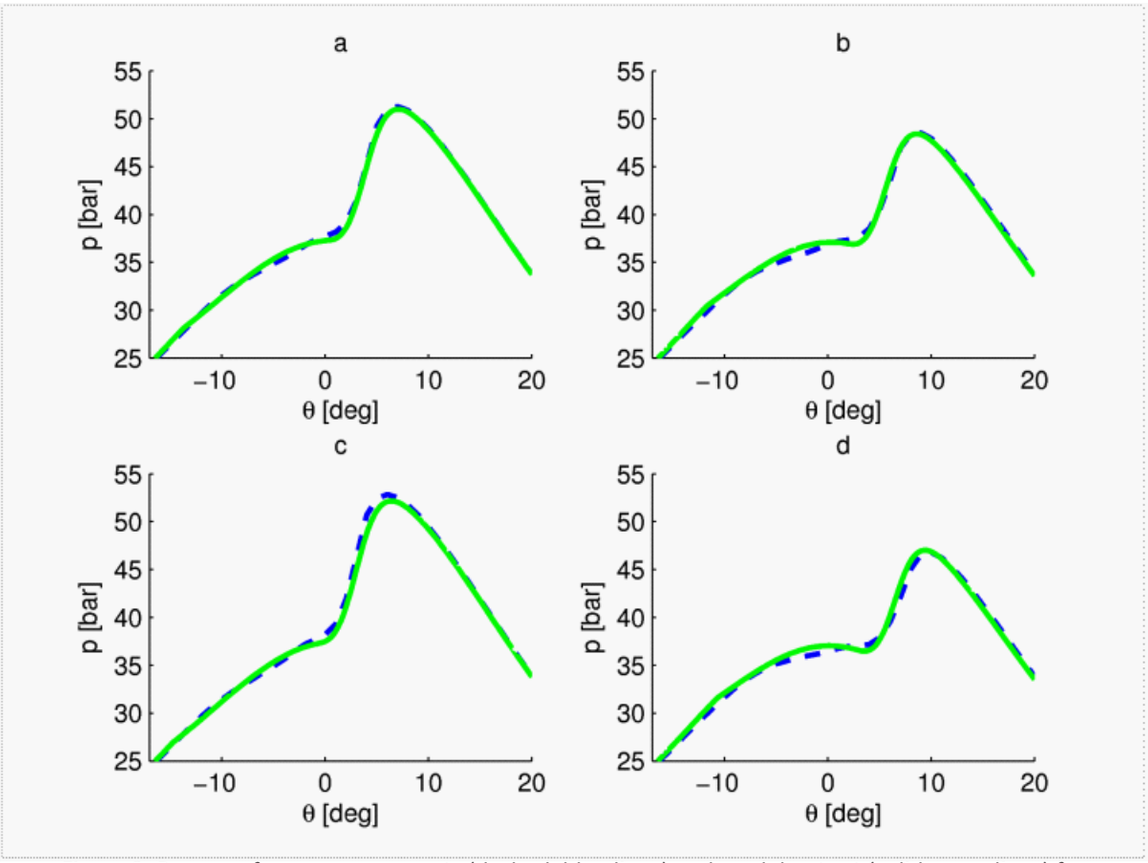


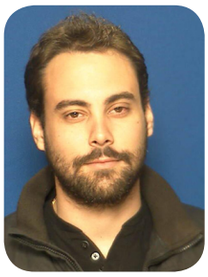
Figure 3: Pressure from measurements (dashed, blue lines) and model output (solid green lines) for start of injection at 15.7, 13.7, 16.7, and 12.7 degrees before top dead center.

2. The Generic Diesel Project

Due to its relatively high efficiency, the diesel engine is an important power source for road transport. Its importance can be expected to increase as the demands on fuel efficiency increase. However, its exhaust emissions of nitrogen oxides and particulates remain challenging. The Generic Diesel (GenDies) project currently focuses on various in-cylinder mechanisms that explain trends in soot emissions.

Clément Chartier
PhD Student

After the fuel jets impinge on the walls of the diesel combustion chamber they form wall jets. The air entrainment into these wall jets is believed to influence the soot oxidation rate. During the autumn 2011, this mixing process was investigated using a novel laser diagnostic called SLIPI. The results are currently under analysis, but some preliminary results are shown below.

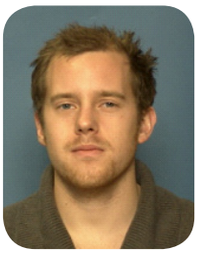


Yann Gallo
PhD Student

Hot gases from neighboring jets have been found to shorten the lift-off length on burning fuel jets, which is a controlling factor for the soot formation in the jet. Two papers were written on this topic. One is an experimental study that demonstrates the effect, the other is a computational study (LES) showing that the phenomenon is due to hot products. Both are described below.

Johan Sjöholm
PhD Student

A new Ph.D. student, Yann Gallo, was hired to the project. He is currently planning measurements of late-cycle soot oxidation rates using the laser extinction technique.



Rickard Solsjö
PhD Student

Finally, we present high-speed laser imaging of spray penetration during late post injections in a light-duty diesel engine. This experiment was conducted in collaboration with Guillaume Lequien of the Cecost Transient Sprays project.

Rikard Wellander
PhD Student

2.1 Quantitative In-Cylinder Fuel Measurements in a Heavy-Duty Diesel Engine using Structured Laser Illumination Planar Imaging (SLIPI).

Quantitative laser induced fluorescence (LIF) is at best difficult in a optical engine. Problems with pressure and temperature dependent LIF signals can sometimes be handled with careful calibration but problems associated with scattered light usually remain. Scattered laser light will cause a volume illumination creating erroneous signals. Scattered signal light generally decreases the signal as photons are lost between the laser beam and the detector. However, these photons can be detected at altered angles and may thus blur the image.

By using a technique called Structured Laser Illumination Planar Imaging (SLIPI) it is possible to remove the effects from scattered light. SLIPI uses laser sheets that are spatially modulated with a sinusoidal pattern, $E_{laser} = E_o \sin(\omega x + \phi)$
The detected signal (I1), given some scattering, can then be expressed as $I_1 = I_c + I_s \sin(\omega x + \phi)$ where I_s has maintained the modulation and I_c has lost the original modulation through e.g. scattering.

Combining three measurements with varying phase, (ϕ, = 0, ± 120) one can calculate Ic according to $I_c = 1/3[I_1 + I_2 + I_3]$, thus recreating the signal one would get if the laser sheet were not modulated. More significant is that Is can also be retrieved according to $I_s = \frac{2}{\sqrt{3}}\sqrt{[(I_1 - I_2)^2 + (I_1 - I_3)^2 + (I_2 - I_3)^2]}$ SLIPI can thus remove all scattered light and show the original signal from the laser sheet.

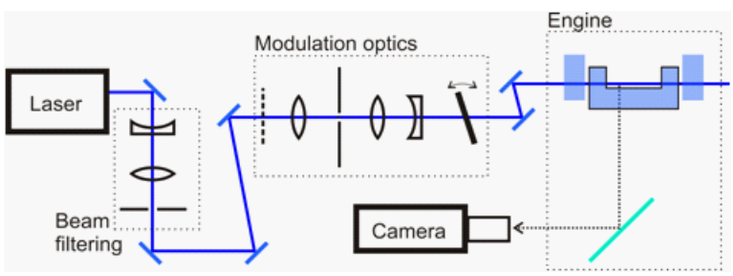


Figure 2.1: Sketch of the optical setup for SLIPI measurements. The modulation optics consists of: grating, focusing lens, Fourier filtering, sheet forming lenses and a rotating plate.

Figure 2.1 shows a sketch of the setup used in order to implement SLIPI in an optical engine rig. These measurements represent the first SLIPI measurements in an optical engine and the first gas phase LIF measurements using SLIPI. The engine was a Scania D12 engine, modified for single cylinder operation with a Bowditch piston extension granting optical access. The horizontal laser sheet entered the piston bowl from the side through a quarts window in the liner and through the quarts piston top that was glued to the piston extension.

A quarts plate, fixed on a rotation stage, was used to shift the phase of the modulation. The setup was found to be very sensitive to vibrations and an alternative method of calculating Is had to be developed. This new method can cope with phase differences that differ from 120 degrees without sacrificing signal linearity.

Figure 2.2 illustrates the results from the measurements. The first image (top left) shows the measured image with modulation of the laser light giving horizontal lines across the image. The laser enters the image from the left. The part of the image showing the quarts piston has been blocked due to excessive quartz fluorescence. The second image shows the reconstructed LIF image (IC).

The scattered laser light illuminating the entire combustion chamber and the liquid fuel close to the centrally mounted direct injector (marked with a white point to the right in the images) are clearly visible. The bottom images show the final SLIPI image (left) and a comparison LIF image (right) recorded without modulation of the laser sheet. A background image (without fuel injection) has been subtracted from the LIF image in order to make a fair comparison with the SLIPI image.

Note the difference in signal strength close to the bowl wall. The regular LIF image fails to capture the wall jets accurately due to scattered laser light and quarts fluorescence interfering with the fuel LIF signal.

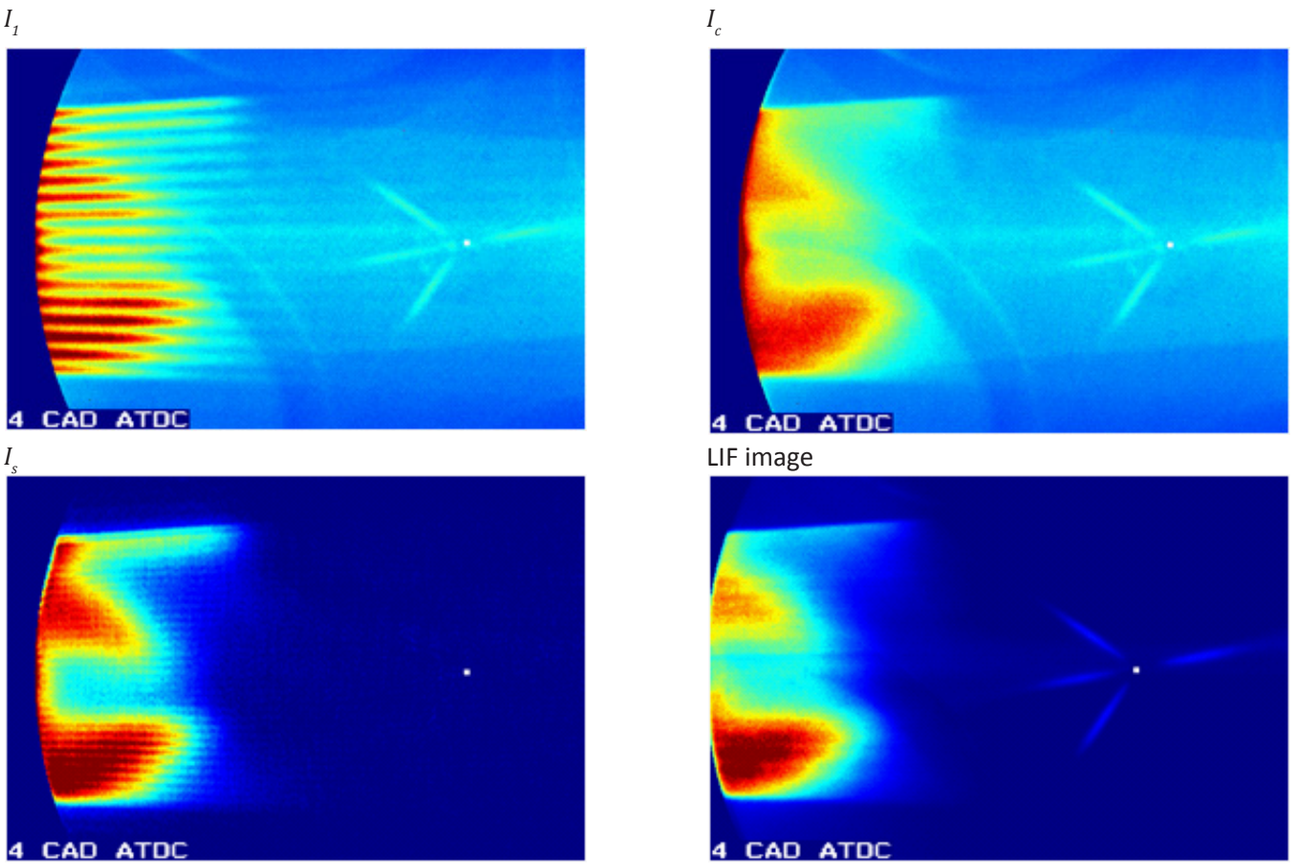


Figure 2.2: Raw image (I1) with modulation (upper left), the reconstructed LIF image (Ic) (upper right), SLIPI image (Is) (lower left) and a regular LIF image captured without modulation (lower right) but with correct background subtraction. All images are recorded under the same engine conditions.

2.2 Influence of Jet-Jet Interactions on the Lift-Off Length in an Optical Heavy-Duty DI Diesel Engine

Introduction

Several investigations have reported that the lift-off length on diesel jets depends strongly on the ambient temperature. The spacing between adjacent jets is thereby expected to influence the lift-off length, as it affects the amount of hot, burned gases present between the jets. It is known that the soot formation rate in quasi-steady jets is influenced by the amount of air entrained in the jet.

Numerous studies of the lift-off length in diesel combustion have been performed in constant-volume combustion vessels. This type of device is able to accurately reproduce ambient conditions of most automotive diesel operating conditions in terms of temperature, density or degree of dilution.

However, the larger volume greatly reduces the effect of combustion on ambient pressure and temperature. Furthermore, the large distance to the chamber walls removes the wall interaction effect. Finally, since single hole injectors are usually employed, there are no jet-jet interactions.

The current study investigates the influence of inter-jet spacing on the lift-off length for quasi-steady jets, to gain insight into the effects of the burned gases recirculated upstream of the lift-off region. An empirical expression by Pickett et al., based on combustion-vessel experiments, is used for lift-off length prediction and compared to the experimental lift-off length results.

Method

Experiments were performed using both symmetrical and asymmetrical 4-hole injector nozzles in order to gain insight into the effects of jet-jet interactions on the lift-off length in realistic engine-conditions. OH-chemiluminescence images were recorded during the quasi-steady stage of the combustion, using an interference filter centered at 310 nm (FWHM 10 nm) in front of a 105 mm UV-Nikkor lens along with a Hamamatsu image intensifier. The lift-off length is defined here as the distance between the nozzle and the first appearance of OH radicals in the jet, i.e. the distance to the high temperature mixing-controlled reaction zone.

Results

Figure 2.3 shows a scatterplot of measured lift-off lengths versus predicted ones for the current study. If the empirical expression mentioned in the introduction were applicable to the optical engine data, all points would collapse onto a line with slope 1, which they clearly do not. The large variations in the lift-off length data illustrate how sensitive the stabilization process is to disturbances affecting the entrained gas temperature. These phenomena are not completely reproduced in a combustion vessel and can explain the differences observed.

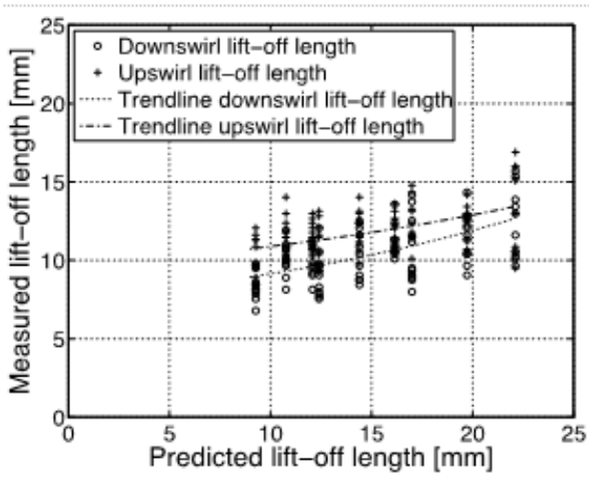


Figure 2.3: Comparison between the measured and predicted lift-off length for all conditions investigated in the factorial study.

The data also showed asymmetrical lift-off lengths on the jets. Figure 2.4 shows a boxplot to compare the distribution and central tendency of the data. The distributions of the upswirl and downswirl lift-off lengths have similar degrees of dispersion and skewness. However, the median value for downswirl measurements is approximately 15% lower than for the upswirl dataset. A plausible explanation for this trend is that the swirl field displaces hot gases around the jet. As mentioned in the experimental hypothesis, Pickett et al. suggested that reservoirs of hot gases surrounding the jet stabilize the lift-off length by a process relying on autoignition. In that picture, the asymmetric lift-off would be explained by more hot gases being present on the downswirl side, facilitating autoignition. This is in line with the observation that burned gases affects the lift-off length.

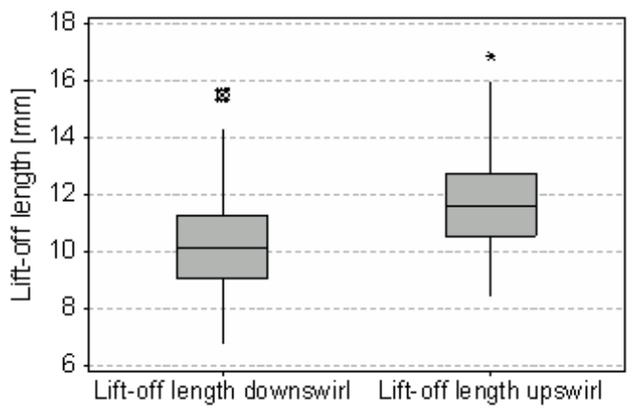


Figure 2.4: Boxplot representation of the lift-off length distribution on the downswirl and upswirl side of the jet.

2.3 LES of Jet-Jet Interaction Effects on Multi-Hole Injectors

LES of jet-jet interaction in the presence of swirl is carried out using the LES PPC model based on detailed chemistry. As optical investigations usually are restricted to single-species visualization, LES can provide information about intermediate species, such as species appearing in low-temperature reactions. Furthermore, LES results can be used to examine the temperature distribution adjacent to the lift-off positions of diesel flames. In accordance with the above GenDies experiment, LES was performed with two different nozzle configurations; a symmetrical setup with 90° inter-jet angle and an asymmetrical setup with 45° and 135° inter-jet angle. The injection pressure was 2500 bar. The simulations successfully captured the influence of swirl, which causes an asymmetrical lift-off behavior. As shown in Figure 2.5, the downswirl lift-off length was estimated to be 13-18.5% shorter. This should be compared to an average of 15% shorter lift-off lengths observed in the experiments.

The lift-off length in the simulations was under-predicted compared to the experimental observations, but the general trends agreed. It was shown that hot products from adjacent jets were transported in the azimuthal direction, creating high temperature regions around the lift-off positions. For the jet with an inter-jet angle of 45°, the adjacent ambient gas temperature was on the order of 300-350 K higher than for the 90° and 135° inter-jet angles. This likely explains the experimentally observed difference in lift-off length. The results indicated that local temperature distribution would be significantly different if the inter-jet angles were larger than 90°. The LES results showed subsequent development of the initial low-temperature reaction process, the transition to high-temperature auto-ignition, and finally the stabilization of the lifted diesel flames.

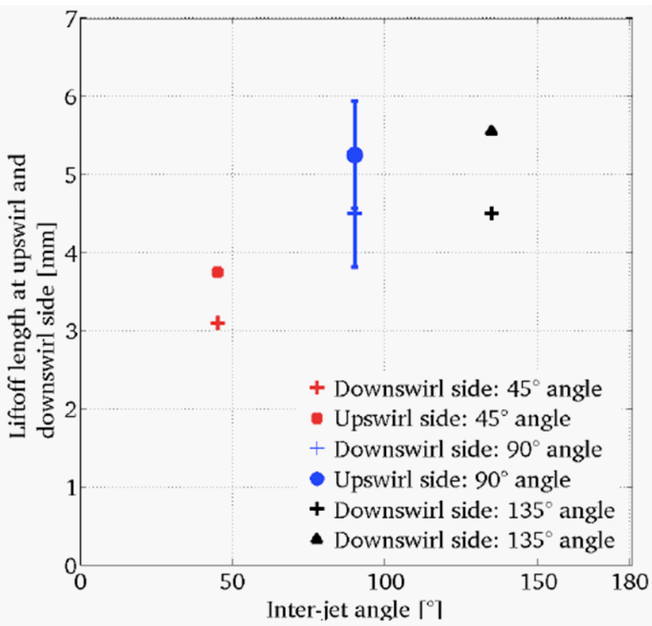


Figure 2.5: Lift-off length at upswirl and downswirl side of the jets.

2.4 Soot Measurements During the Late Cycle

Previous studies of soot processes in the GenDies project have focused on the early phases of the cycle. The late cycle-oxidation is believed to be a major factor determining the trends in engine-out particulate emissions. To study the oxidation rate a quasi-quantitative technique is needed. The laser extinction method measures how much laser intensity is lost over the beam path through the cylinder and the extinction is proportional to the volume fraction of soot in this path. To probe a volume representative of as much as possible of the cylinder volume, a vertical beam path is employed. The laser beam is introduced through the cylinder head and extracted vertically through the optical piston. To achieve this, a specially designed cylinder head has been manufactured with a hole for the optical fiber. Yann Gallo currently develops the setup and the experiments will begin during the spring 2012.

2.5 Visualization of Late Post-Injections using a kHz Laser System

Late post injections are commonly used in modern diesel engines in order to enable high enough temperatures in the exhaust stream to enable regeneration of the Diesel Particulate Filter. At these late injection timings, in-cylinder densities are low compared to Top Dead Center. The liquid spray penetration length in these conditions is longer than for burning jets at higher ambient densities, thus increasing the probability for the fuel to impinge on the cylinder wall. This risk is especially significant in small-bore engines. When wall impingement occurs, liquid fuel may pass by the piston rings and mix with the crank case oil, modifying oil viscosity and causing further engine damages. The problem is aggravated when biodiesel is blended into the ordinary diesel fuel, due to the lower volatility of these fuels and therefore longer liquid penetration length.

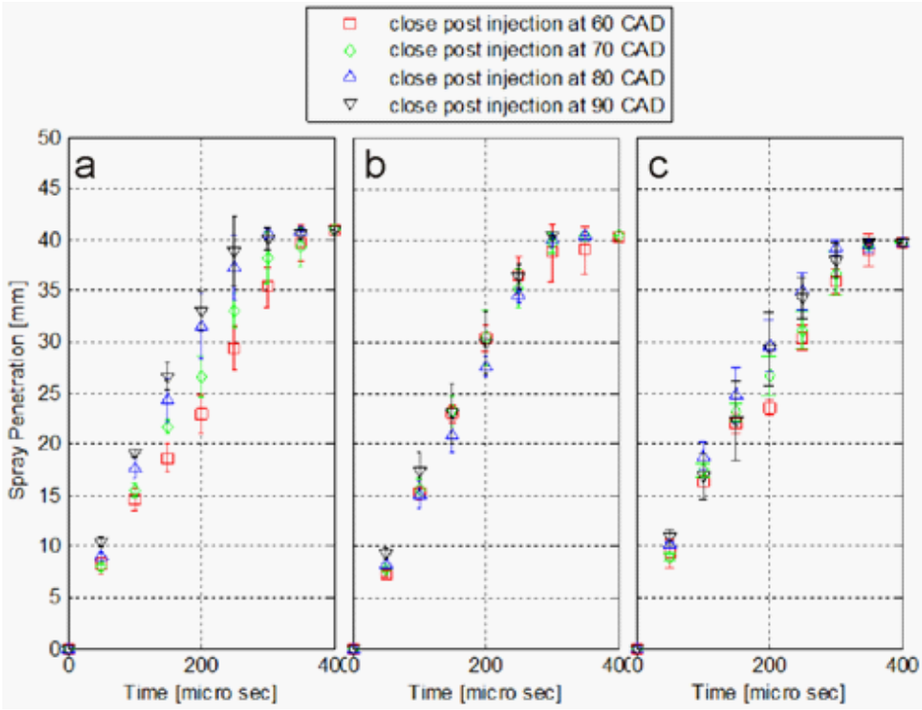


Figure 2.6: Late post spray penetration length as a function of time after start of injection. (a) VSD10, (b) MK1 and (c) VSD10 10 % RME

A study of the liquid penetration length is carried out on an optically accessible light duty diesel engine of Bowditch design. Different fuel injection strategies commonly used for the particulate filter regeneration are investigated. The study aims at finding out how the spray penetration length is affected by different injection strategies. An additional focus is put on the effects of blending biodiesel into ordinary diesel fuel. The study is motivated by the current trends to increase the portion of bio-derived fuels in trade fuels, especially in Europe, where rapeseed methyl ester (RME) is the most common biofuel blended with trade diesel. The Swedish standard diesel fuel MK1 and a reference fuel free of RME (VSD10) are tested. Then a Biodiesel fuel containing 10% of RME is tested in order to evaluate the impact of this blend on the liquid spray penetration length.

Laser pulses, short enough to freeze the motion of the spray1, are formed into laser sheets oriented in the vertical direction and directed along the spray to illuminate the droplets in a cross section of the spray. Light scattered from the droplets is imaged through the quartz liner onto a high speed CMOS camera located perpendicular to the spray and laser light propagation. The spray penetration at the time of the laser pulse is extracted from the recorded image. The laser light is emitted from a solid state high speed Nd:YAG laser operated at 20 kHz. Thus the spray evolution can be resolved with a time resolution of 50 µs. In order to reproduce combustion and in-cylinder conditions equivalent to all-metal engines, a metal piston with a realistic bowl shape is mounted on the test engine and the in-cylinder conditions are matched to data obtained from a multi cylinder engine running at relevant operating points.

The data are still under evaluation. In Figure 2.6 the spray penetration is shown as a function of time after start of injection for one of the tested spray durations and late poste timings.

3. The Gas Engine Project

Introduction

Ashish Shah
PhD Student

The Gas Engine Project at Lund University aims to explore and understand the combustion phenomenon in engine operating on gaseous fuels and develop technologies as an alternative to present day diesel operated heavy duty engines which are facing severe challenges like stringent emissions norms, high technology cost and unsustainable fuel supply.

Over the past few years, Natural Gas has emerged as the most promising gaseous fuel due to its benefits in terms of emission reduction with cheap after treatment devices and increasing fueling station network worldwide.

Project 5th Phase – New PhD Student

Previous phases of this project focused on understanding the basic combustion phenomenon, effect of combustion chamber geometry, cylinder to cylinder and cycle to cycle variations, different fueling strategies and use of control systems to reduce emissions and achieve higher operating efficiencies. Majority of this work was done using conventional spark plug ignition system and after having optimized all other operating parameters like boost pressure and EGR rate, it has been seen that the major limitation in extending lean limit of operation is the capability of ignition system to reliably ignite a leaner mixture. To address the issue, the fifth phase of this project, which commenced in June 2011, will focus on alternate ignition techniques for a gas engine which can sustain a leaner operation (diluted either with excess air or EGR). Mehrzad Kaiadi defended his PhD thesis in early 2011 and a new PhD student, Ashish Shah, started working on the 5th phase from June 2011.

Experimental Work

After a brief literature survey, three main alternate ignition techniques, namely pre-chamber spark plugs, pre-combustion chamber and diesel pilot injection, were identified. Different researchers have named these techniques in different ways but finally the techniques boil down to these three basic categories. Out of these, pre-chamber spark plugs were investigated first since they required no major modification to the existing engine setup.

The experimental engine is 6 cylinder 9 liter turbo-charged (VGT) engine which is equipped with multi-point gas injection. All the experiments till now in the 5th phase have been done without any after treatment device. Pre-chamber spark plugs were procured from MULTITORCH GmbH and are as shown in the figure 1 below. Initially a basic experiment was conducted to briefly examine the performance and emission characteristic when operating with pre-chamber spark plugs. It was observed that ignition capability was significantly enhanced and flame development angle and combustion duration reduced by about 40 % of those with conventional spark plugs at certain operating point.

Figure 1: View of MULTITORCH Pre-chamber spark plug

Towards the end of 2011, detailed experiments have been done to access the capability of pre-chamber spark plugs to achieve higher operating loads and extend the dilution limit. It has been observed that pre-chamber spark plugs enhance the ignition substantially but cause pre-ignition due to very hot pre-chamber surface at loads exceeding 10-12 bar IMEPg. This causes a minimum amount of dilution required, either with excess air or EGR to operate above 10 bar IMEPg which reduces the available operating window. This can be seen in figure 2 and 3 below. Ignition being more powerful, the pre-chamber spark plugs have been found to sustain leaner operation, extending lean limit with excess air by 0.8 Lambda units and with EGR by about 5% units. There are three main limiting factors which limit higher operating load, namely exhaust temperature (700 deg. C), pre-igniton and turbocharger over-speeding at higher boost. The dominant factor when operating with pre-chamber spark plugs is pre-ignition and that with conventional spark plugs is exhaust temperature and turbocharger over-speeding at loads exceeding 20 bar. With such limitation, the maximum load (IMEPg) achievable with pre-chamber spark plug is 20 bar with excess air and 19 bar with EGR dilution. The same with conventional spark plugs was found to be 22.5 and 21 bar with excess air and EGR dilution respectively.

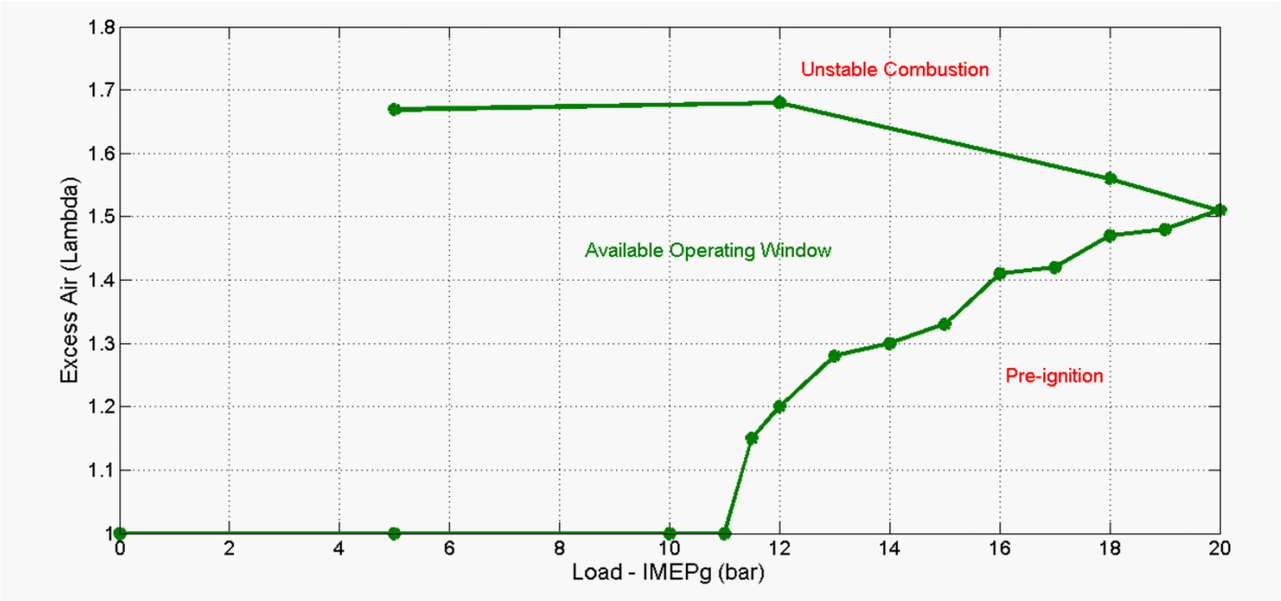
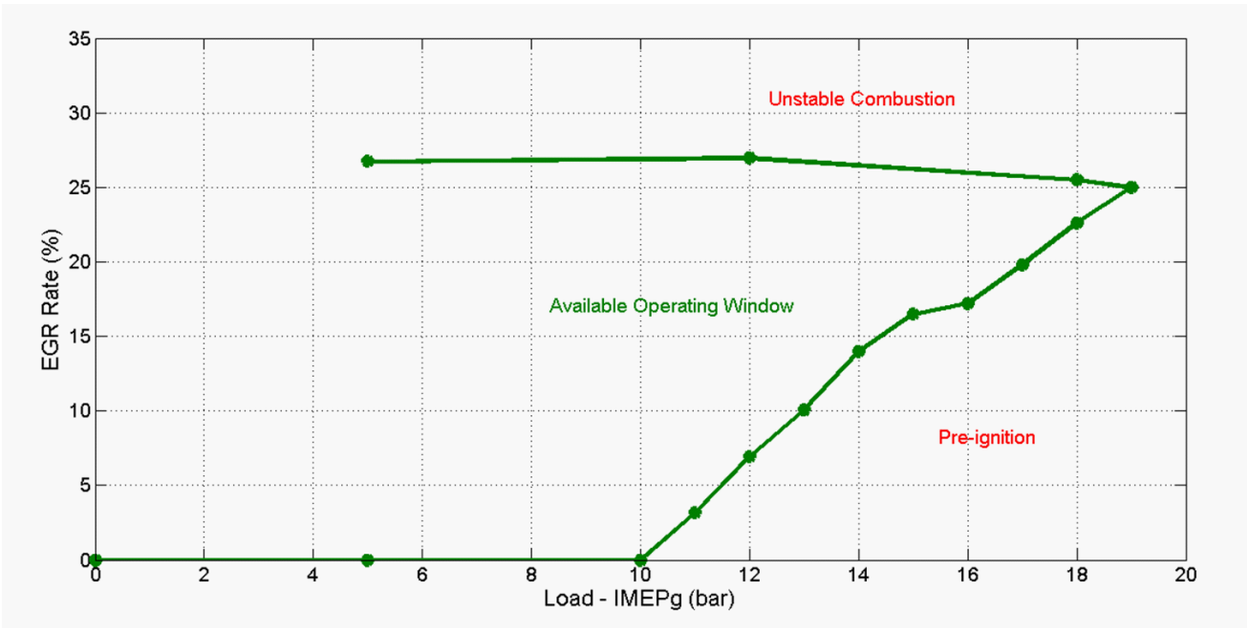


Figure 2 (above): Minimum dilution with Excess Air (Lambda); Figure 3 (below): Minimum dilution with EGR



Summary and Future Work

Though pre-chamber spark plugs can enhance ignition resulting in shorter ignition duration and flame development angle, they are limited by pre-ignition at load exceeding 10 bar IMEPg. Further experiments are planned with altered pre-chamber geometry, different hole size, different pre-chamber material and with combustion chamber with lower turbulence. It is also planned to study the effect of pre-chamber indexing with respect to combustion chamber geometry. After completing experiments with pre-chamber, a diesel pilot injection system will be procured and adapted to the engine, experiments with which are planned towards the end of 2012.