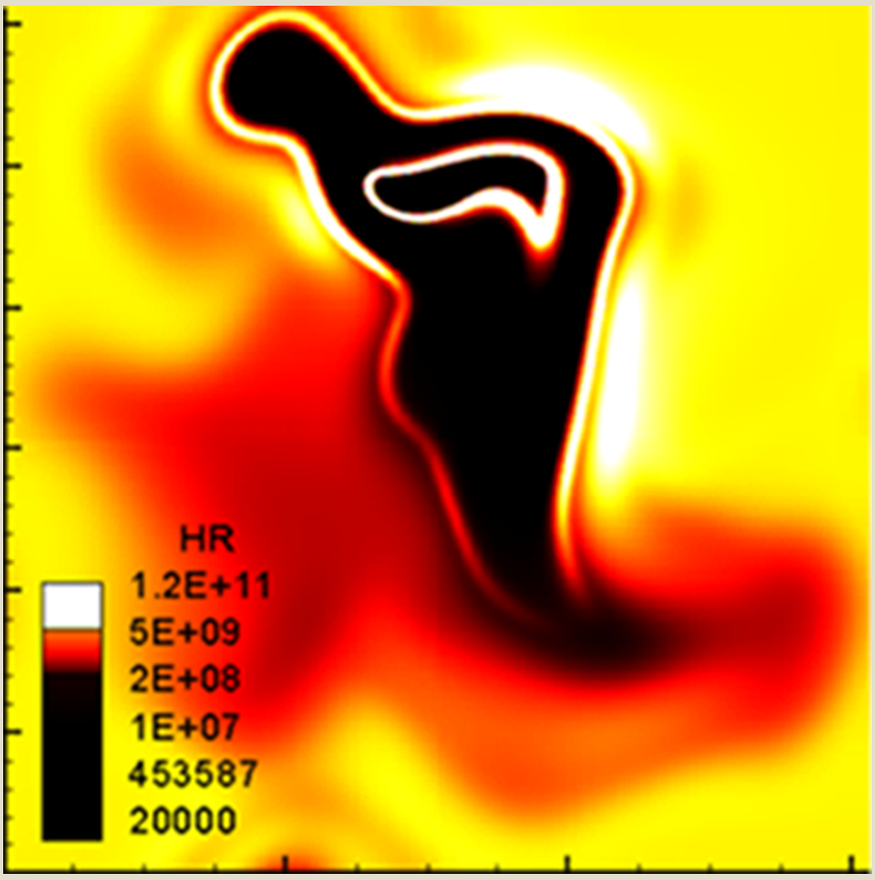
KCFP Annual Report 2015

CENTRE OF COMPETENCE COMBUSTION PROCESSES
FACULTY OF ENGINEERING, LTH | LUND UNIVERSITY



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 25.4 MSEK per year. This is roughly one third each from the Swedish Energy Agency, STEM, Lund University and the Industry.

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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.

PPC - Heavy Duty

Introduction

Mengqin Shen
PhD Student

Partially premixed combustion (PPC) is a promising way to achieve simultaneous high efficiency and low engine-out emissions in a heavy-duty engine. By utilizing high amount of exhaust gas recirculation (EGR), earlier injection than conventional diesel combustion is possible and combustion temperature can be reduced dramatically. The enhanced premixed fuel and air prior to ignition reduces the propensity for soot formation by avoiding locally fuel rich zones. Compared to diesel fuel, previous results in this project show that gasoline fuels actually have a more promising potential as engine-out soot emission is much lower while engine efficiency maintains the same or even higher. This emission difference between fuels raises the need for an improved understanding of the soot formation during combustion.

Experimental Setups

The research engine, in Figure 1 on the left, is based on a Scania D13 heavy duty diesel engine modified to run on only one of the six cylinders. A high pressure, cooled EGR system was utilized. The fast sampling valve tip was mounted directly into the cylinder head, replacing one of the exhaust ports, and the gas samples were taken between two injection jets in the cylinder. The sampled in-cylinder gas was diluted in three steps marked D1, D2 and D3 in Figure 1 on the right.

Results

EGR effect on diesel fuel

The fuel used in this experiment was Swedish diesel MK1. With absence of EGR, the highest peak soot mass concentration is formed (Figure 2.a). However, nearly all of the soot is oxidized with the in-cylinder gas during the ongoing combustion and expansion. EGR tends to reduce both soot formation rate and soot oxidation rate due to lower flame temperature and less oxidizer. Despite lower peak soot mass, a much stronger decrease in soot oxidation leads to increased soot emissions when adding EGR.

The number concentration of soot aggregates shows similar trends as the soot mass concentration during the combustion cycle (Figure 2.b). However the highest number concentration peak is found at low EGR in contrast to the soot mass trends (no EGR). The size measurements (Figure 2.c), total soot mass and particle number show similar trends during combustion. With higher EGR, the particle mean diameter decreases and particle number increases during the expansion stroke. Therefore, increased soot emissions with EGR are attributed to higher particle number despite of smaller particle size.

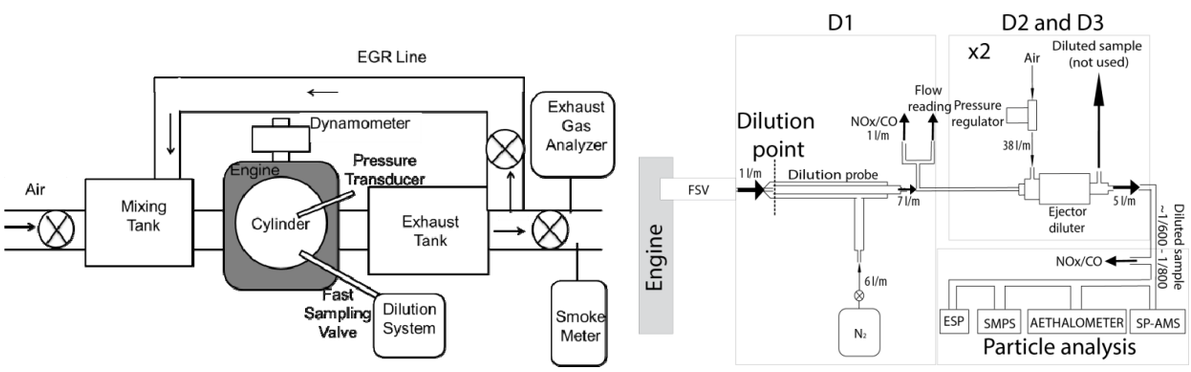


Figure 1. Single cylinder engine (left) and the dilution system used in the study (right)

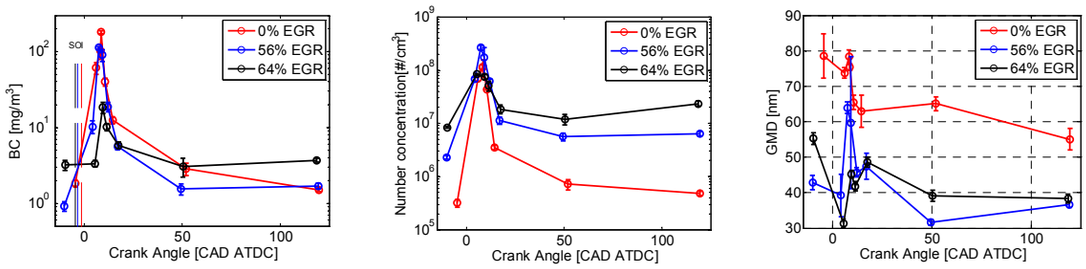


Figure 2 Black carbon(BC) mass, particle number concentration and geometric mean particle mobility diameter (GMD) variations during combustion at three EGR levels

Gasoline PPC vs. Diesel PPC

Figure 3 presents the soot particle-size distributions for gasoline and diesel PPC combustion, for different crank angle timing during combustion. Generally, before fuel injection (-16CAD case for gasoline and -10.2CAD case for diesel), the particle number is low for both fuels. After injection has occurred and combustion starts, particle number increases rapidly and the size distribution shifts to the right side, i.e. larger diameter. For gasoline, the major difference compared to diesel soot is the asymmetry apparent in the shapes of the gasoline particle distribution. During combustion, gasoline generates more particles but much smaller. The main reason for low soot emission with gasoline PPC can be smaller particles.

Conclusion

EGR tends to reduce peak soot mass formed during combustion for diesel fuel. Despite decreased particle size, increased particle number leads to higher soot emissions in the end. More particles are found during gasoline PPC combustion, but much smaller size contributes to a much lower soot emission compared to diesel PPC.

Future Work

For 2016, the focus of PPC work will be more on property study of exhaust soot for gasolines, diesel and ethanol, under various oxygen concentrations and different engine parameters, to have more fundamental understanding of the benefits of gasoline and ethanol as PPC fuels. Piston geometry study combined with different fuels, injection strategies will largely be carried out.

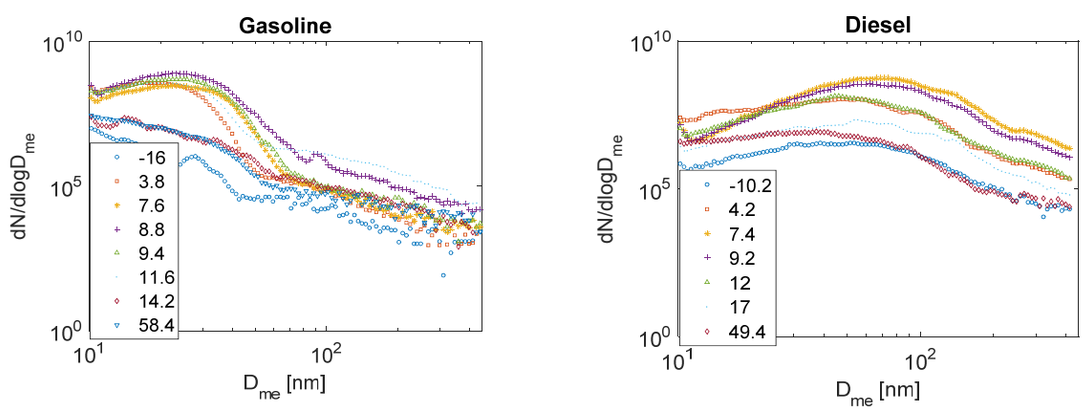


Figure 3. Number weighted mobility diameter distributions of gasoline particulate matter and diesel particulate matter at different crank angles during combustion.

PPC - HD Optics

Project scope

The project aim of PPC HD Optics is to gain a better understanding of the PPC type combustion for a heavy duty (HD) engine. The heavy duty engine to be used in the project is a Scania D13 engine. The project started in the beginning of 2014.

Sara Lönn
PhD Student

Activities during 2015

The mechanical work on the engine was finished in the spring of 2015. In parallel with this, the work with electronics, progressed and was finished late spring. Testing of the injection system, compressor and the burner for the production of EGR commenced while operating the engine with a metal piston. The piston was then changed to a pancake optical piston.

Joakim Rosell
PhD Student

During the summer of 2015 injection profiles were investigated by capturing the Mie scattering with a high speed camera to be used in CFD. This work was done in co-operation with Joakim Rosell and Alexios Matamis from combustion physics, and described further down.

Alexios Matamis
PhD Student

After this investigation the optical piston was changed to an open piston bowl in the early fall. However, as soon as the piston was motored, the quartz piston broke down in multiple pieces. This lead to a lot of collateral damage, the piston extension, the inlet and exhaust valves, and one side window in the optical liner all needed to be replaced. Furthermore, the cylinder liner for the optical setup had to be machined due to deposits of metal and scratches from the piston extension. The diameter had to be increased for one of the plates, since new scratches was found when it was reassembled and motored. Due to the previous piston failure, the distance between the piston and the cylinder head was investigated at different engine speeds to make sure that the piston does not hits inlet or exhaust valves. This was tested for both the open bowl metal piston and a quartz piston. A high pressure valve was installed to function as a leak flow for the common rail to compensate for the use of the function "skip fire", where for example the fuel injected is skipped for a number of cycles. The PI regulator values have been adjusted to provide better control of the common rail pressure in combination with the high pressure leak valve. Chemiluminescence images have also been captured using n-heptane as fuel.

#	P_{in} [bar]	**Dur** [µs]	**SOI** [CAD]
1	970-1100	900	-45
2	900-1000	900	-60
3	1050-1100	700	-30
4	950-1050	700	-45
5	1050-1100	700	-60
6	750	900	-30

Table 1. For all cases the inlet pressure was kept at 1.25 bar and the inlet temperature around 42°C.

Experimental Setup

A 3 W CW laser @445 nm was used as light source and directed in to the combustion chamber from the liner window. From beneath a Photron Fastcam SA-X2 high speed CMOS camera detected the Mie scattering of the fuel spray. A total of 50 frames at 25 000 fps were recorded with 768x688 pixels in resolution for six different injection settings, presented in Table 1.

Results

The collected HSV image sequences was thoroughly post-processed. Pixel distance representation was calculated and each spray of total eight was analyzed to support the CFD modeler. The post-processing process started to select the region of interest (ROI) for each (eight) spray plume. For every ROI i.e. spray, a threshold at 3 % of the maximum value was applied eliminating uncertain signal. The image was then rotated and translated in order to get the spray plume horizonta. These steps were done for every image recorded. In order to analyze the displacement of the spray, the tip of the spray had to be determined. We did this by first calculate the center of mass (CM) of every spray image, see upper right image of Figure 2. Each spray image was then filtered with a second order Butterworth filter. To find the tip of the spray at every ROI images two related methods were adopted. At the lower left image of Figure 2 the intensity of the filtered ROI image is represented, and the spray tip is considered to be where the normalized intensity has dropped to 10 %. For a more accurate position of the spray tip the gradient of the normalized filtered intensity was calculated. With some constraints, the spray tip position was calculated from the gradient plot seen in the lower right of Figure 2. Figure 3 shows an example of the spray tip displacement calculated with the described post-processing procedure.

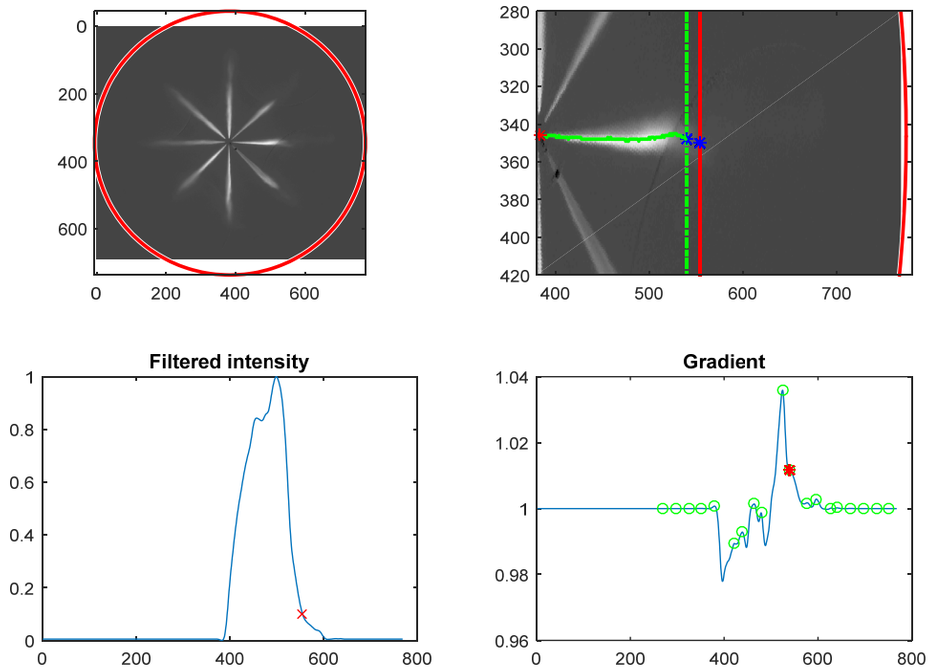


Figure 2. Example of spray tip determination. X-axis represents number of pixels from the nozzle. In upper right figure the red solid line represents spray tip position calculated from filtered intensity plot, green from gradient plot.

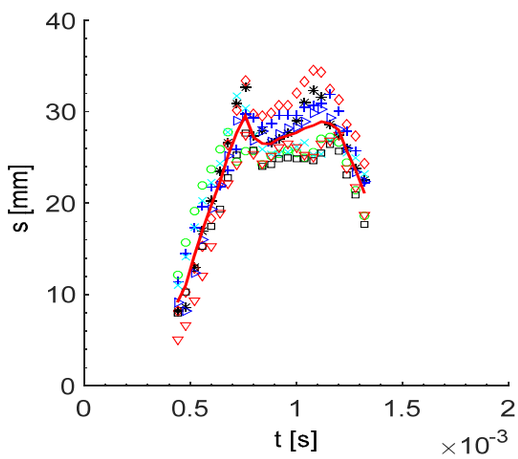


Figure 3 Spray tip displacement in mm as a function of time in seconds of case 6 in Table 1. The eight different symbols represent one spray each and the solid red line is the mean of the eight sprays.

Future Work

During the beginning of 2016 the combustion will be studied when the fuel is injected at different SOI while keeping the CA50 constant, in order to see how the combustion is affected by where the fuel is injected in relation to the piston.

HSV of PPC in a HD Engine

Introduction

Marcus Lundgren
PhD Student

Joakim Rosell
PhD Student

Optical diagnostics where conducted in a rebuilt heavy duty, non-swirling, single cylinder engine with optical access. High Speed Video (HSV) imaging of natural luminosity (NL) from the combustion chamber was recorded while investigating the general combustion process of PPC and how different injection strategies affect the process in terms of combustion stratification. In order to determine whether selective imagining of OH* or CH* from the broadband NL was possible or not, **spectral measurements** were applied. The spectrum recorded indicated a potential ratio between OH* and broadband NL hence further optical investigations were adapted. With a 550 nm long pass filter mounted on a stereoscope **Intensified High Speed Video recording** would enforce distinction between OH* imaging and broadband NL imaging. As no distinction was detected, the investigation confirms the deliberation that has to be considered when making conclusion from passive diagnostics techniques.

The experiments started in August 2016 and have resulted in a SAE paper. A presentation is expected at the "SAE 2016 World Congress and Exhibition".

Background

The Partially premixed combustion (PPC) concept combines both high thermal efficiency and low exhaust emissions of NOx and SOOT. PPC is classified as a combustion process between Homogeneous charge compression ignition (HCCI) and conventional diesel combustion (CDC). However due to the lack of combustion controllability for HCCI the focus has shifted to PPC. Even though the concept of PPC has been around for some time and substantial theoretical and experimental work has been done, optical studies of PPC in heavy duty engines are few. Aspects of mixing preparation are far from fully understood when it comes to stratification, turbulence, and combustion mixing. More knowledge is also needed to find a favorable geometry for the combustion chamber and inlet duct if they are to achieve the preferred mixing conditions for PPC. The aim of this project is, with HSV imaging, to achieve extended understanding of the general combustion phenomena of PPC and the transition from HCCI in an optical heavy duty engine, focusing on start of combustion and sensitivity to temperature variations.

Engine

Experiments were conducted in an optically modified 6-cylinder Volvo MD13 base engine. Specifications can be found in Table 1. An injection sweep was performed between the two extremes of -28 to -13 with an increment of 5 crank angle degrees (CAD). The purpose being to visualize the transition from a, in piston bowl, premixed combustion to a partially premixed combustion. For comparison a HCCI case was performed to compare a homogenous to a partially premixed combustion sequence. The effect of a double injection with the fuel mass split equally between the injections was also performed for comparison with the single injections cases of - 13 and - 28. All operating points were run for 81 cycles. Which means that each combustion sequence in skip fire contains 27 cycles.

The investigation was run in a moderate load of 7.5 bar IMEPg at 1200 rpm. The experimental operating conditions of start of injection (SOI) and duration (DURR) can be found in Table 2. The following parameters were set for the different operating points: Rail pressure: 1500 bar. Inlet pressure: 1.7 bar. Inlet temperature: 50 oC. 16 % of inlet oxygen and the fuel flow was measured to be ~77 mg/stroke. To save the optical components the engine was operated in skip fire mode, three combustions and ten motored.

Table 1

Displaced volume	2123 cc
Stroke	153 mm
Bore	131 mm
Connecting Rod	255 mm
Compression ratio	16:1
Number of Valves	4
Swirl	None
Fuel system	Delphi F2 common rail external pump
Orifices	7
Umbrella angle	150 deg

Table 2

SOI [CAD aTDC]	**DURR** [µs]
-13	930
-18	930
-23	930
-28	930
-28, -13	540, 540
-230	930

Optical Diagnostics

High Speed Video – Imaging of natural luminosity (NL) from the combustion chamber was recorded with a Photron Fastcam SA-X2 high speed CMOS camera with a resolution of 640x640 and a frame rate set at 27 000 fps. As the engine was running in 1200 rpm this equals 15 frames per 4 CAD. In order to protect the framing equipment from engine failure a 5 mm thick acrylic glass was placed between the engine and the high speed camera, see Figure 1.

Spectrometer Setup – The camera and its protective acrylic glass was replaced by an Acton 2300i spectrometer where the grating with 300 grooves/mm and a slit width at 400 µm was selected. An UV-achromatic Bernhard Halle lens, f = 100 mm, f# = 2, imaged the NL onto the entrance slit of the spectrometer. The position of the slit relative the imaged cylinder can be described as a thin chord through the imaged cylinder, not far from the cylinder midpoint. A PIMAX III ICCD camera with 1024x1024 pixels resolution imaged the spectra.

Intensified High Speed Video – In order to distinguish OH-chemiluminescence from the broadband NL of the combustion events a LaVision stereoscope was mounted on the UV-achromatic Bernhard Halle lens. The UV-achromatic lens with the stereoscope was then mounted on an image intensifier (Lambert HS IRO 2 stage) in combination with the high speed camera. To ensure that one of the simultaneously collected images did not contain any OH-chemiluminescence a long pass filter (OG 550) covered one channel of the stereoscope. The other channel was covered with a 315/15 single band bandpass filter.

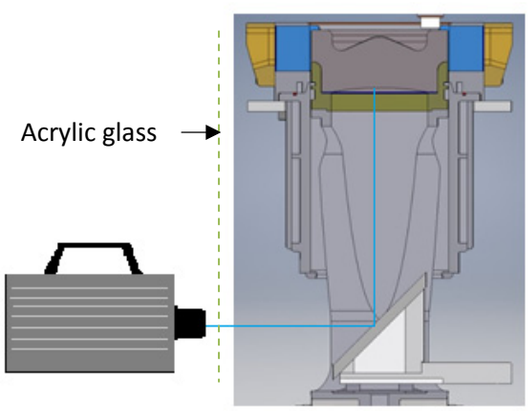


Figure 1. Bowditch design.

Results

Image Signal analyzing

The analysis was performed by finding the first occurring combustion in the NL image sequence. When pinpointed the angular and radial coordinates was collected. Figure 3 below shows the location of inlet and exhaust valves interpreted with intensity as a function of theta. Which simplifies the interpretations of the coordinate plot discussed in the results section.

Start of Combustion

Figure 4 displays the planar coordinates of the start of combustion. Each marker represents first second and third combustion for each injection. The horizontal line in the figure represent the piston bowl rim and the vertical lines are the fuel spray targets.

Observing all injection timings at once it is noticeable that neither the first, second nor third combustion show any trend in location as the bulk temperature increases. However, the radial band width where the SOC takes places shifts as the injection is advanced.

Both the earlier discussed -13 case and the -18 case show similar results in SOC with a narrow band width. Most of the markers can also be seen to avoid the fuel spray trajectory lines. Comparing with the combustion sequence in Figure 2, it's noticeable that the combustion for the -18 case spreads over a wider band, as the fuel has longer time to spread in the combustion chamber.

Early-injection cases of -23 and -28 show more premixed fuel and air mixture as the combustion propagate longer into injector tip region. The liquid fuel penetrates further before vaporizing as the ambient temperature is cooler and the density is lower. After the fuel hits the cylinder wall the fuel vapor continuous its trajectory back to the injector tip region. Seen in Figure 4 the radial band width has increased, which indicates a wider stochastic spread of locally fuel rich zones. The increase in band width of start of combustion between the -18 to -23 are very significant. One hypothesis can be that it is a geometrical effect in combination of longer ignition delay. I.e. the earlier injection hits the piston wall higher and with the upwards moving piston the fuel travels down in the bowl and inward in the bowl center. For the late injection cases the injections hits in the piston bowl, hence the fuel trajectory will be upwards the bowl wall, casing the fuel vapor to tumble in a tighter radial band.

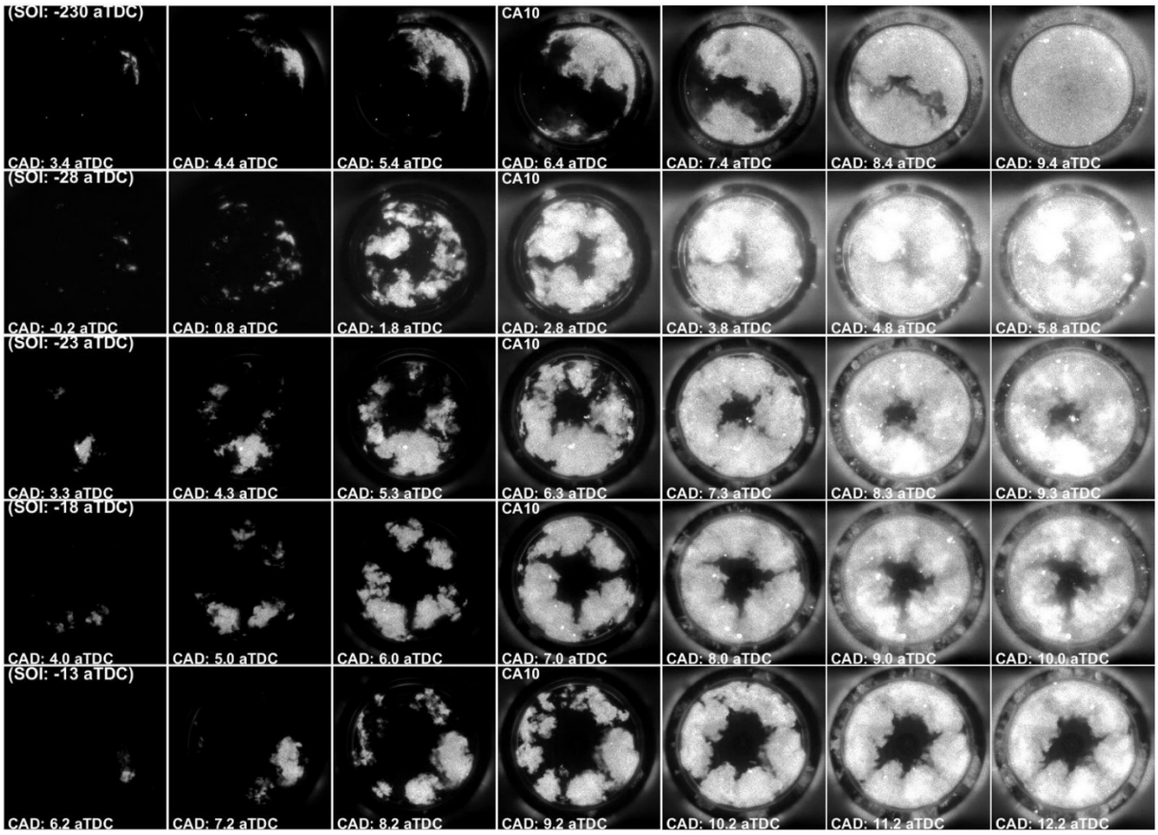


Figure 2. Natural luminosity images over the injection sweep. Each image sequence is synced with the ten percent heat released marker (CA10) in the middle image. In top left corner of each sequence is the start of injection displayed.

Start of combustion for the HCCI case show trends of igniting at the periphery of the piston bowl, Figure 4. Indicating that the local temperature of the piston and squish region ignites the combustion. For 2nd and 3rd combustion cases the start of combustion tends to leave the bowl rim as the global temperature increases with the residual gases from previous combustion event. Part of the explanation can be that hot gases from the squish volume are escaping into the piston bowl and igniting the accumulated air and fuel mixture inside the piston bowl. In the NL image sequence is visible how the combustion starts at the bowl rim, and proceeds rapidly around the piston bowl rim in both directions, simultaneously as the bulk volume ignites. Since this is only line-of-sight can no conclusions can be drawn if the kernel starts to appear close to the cylinder head or in the piston bowl wall, image distortion of the optical piston also complicates observations.

Spectra around the start of combustion (SOC) and later were collected to determine whether the chemiluminescence from OH* and/or CH* would stand out enough from the broadband NL. Five of the collected spectra are shown at inclining CAD aTDC from 7 to 15 in Figure 5. They are the average over 20 collected spectra per CAD and normalized by the highest and lowest value of all five spectra. The efficiency of the spectrometer has been compensated for and spectra below 7 CAD were no NL was detected is not

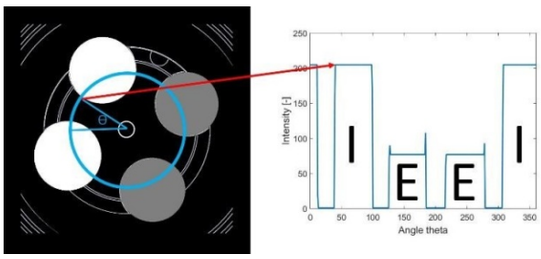


Figure 3. Image distortion of the piston bowl shape, as well as the investigated band of 73.5 mm.

shown. Chemiluminescence from excited OH (OH*) has its strongest peak around 310 nm whereas chemiluminescence from CH* has its strongest peak around 431 nm. OH* is present for all investigated CAD but the short-lived CH* only seems to be present at 7 and 9 CAD aTDC. The broadband background in the spectra is most likely dominant of chemiluminescence from CO2* superimposing all other peaks [18], [19]. Due to the too low ratio between CH* and background in Figure 5, the conclusion is that CH* imaging with a bandpass filter centered at 431 nm will not be reliable. OH* though seems to have a potential of being imaged. However, in order to make sure that there will be no interferences from NL of mentioned radicals and soot great care has to be taken to avoid misinterpreting the data. This concern is also greater at higher soot tempera-

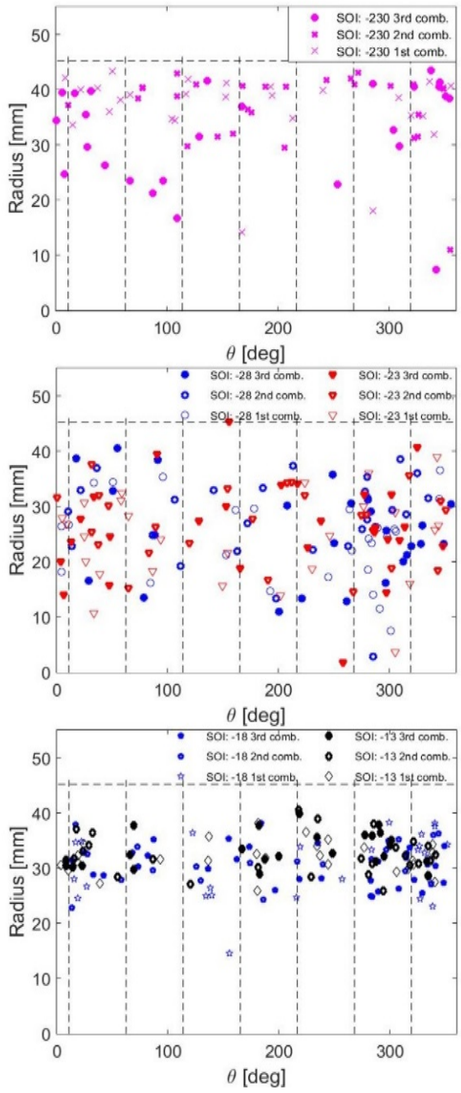


Figure 4. Angular and radial location of start of combustion. Each marker represents the coordinates for each start of combustion location. The horizontal line represents the piston bowl rim. Each vertical line represents the seven-hole-injector spray target.

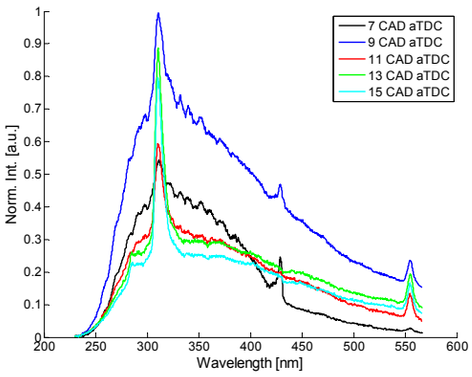


Figure 5. Emission spectra collected at five different CAD.

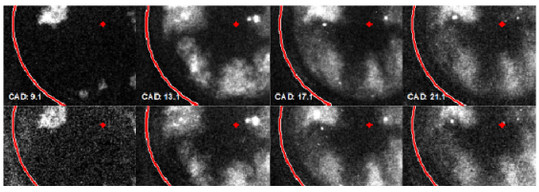


Figure 6. Simultaneously line of sight imaging of the combustion event at indicated CAD aTDC. Upper row: OH-filter. Lower row: OG 550 + OD 0.1 glass filters.

tures since the radiation in the UV-region will be stronger with increasing temperature.

Figure 6 shows images recorded with the stereoscope and no characteristic difference can be detected from looking at 310 nm where OH* should be present, or at the NL above 550 nm with no OH*. This confirms that for the investigated case there is no need trying to distinguish OH* chemiluminescence from the rest of the NL since NL evidently gets transmitted through the narrowband interference filter superimposing the OH*.

Looking at the ratio between OH* and broadband NL in Figure 5 it might be tempting to attach an OH bandpass filter on the camera and expect to be imaging OH. As these results show, that would not be effective. This result confirms that caution always has to be considered when drawing conclusions from data obtained by passive diagnostics techniques were the signal is obtained from the integrated line of site i.e. from the whole combustor chamber volume.

Future Work

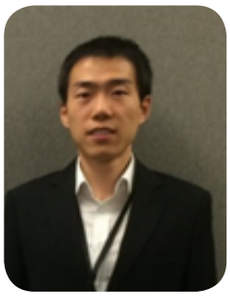
Laser based, in situ, optical diagnostic techniques.

Investigate the transition from HCCI to PPC with a flat piston to exclude geometrical interference.

PPC - Light Duty

Introduction

Slavey Tanov
PhD Student



Zhenkan Wang
PhD Student

Partially Premixed Combustion (PPC) is used to find alternative combustion processes, which are able to reduce both NOx and PM without affecting fuel efficiency in particular operating conditions. It can be considered as an intermediate process between the fully mixed HCCI concept and conventional spray and diffusion controlled compression ignition combustion. The benefit of PPC compared to traditional HCCI combustion is that the fuel injection timing can be used as control parameter for the combustion timing. The air-fuel mixture should not be fully mixed as combustion rate will be high. On the other hand should fuel be mixed enough to prevent overly rich mixtures that form soot and stoichiometric zones giving NOx. PPC is normally generated by applying direct fuel injection rather late in the cycle.

High-speed PIV measurement in a light-duty PPC engine

The investigation of in-cylinder flow structure in a light-duty PPC engine by high-speed particle image velocimetry (PIV) contains two parts. This is the first time high speed PIV has been implemented in a PPC engine with the re-entrant piston bowl shape.

1. The objective of the first part is to investigate the influence of different injection strategies on in-cylinder flow field and fuel-air interaction and mixing with this kind of combustion chamber design. The measurements were performed with fuel injection but without combustion. This was achieved by means of ultra-high (exhaust gas recirculation) EGR levels.

2. In the second part, the effects of injection strategies on in-cylinder flow structures in the PPC combustion mode were investigated with their associated rate of heat release.

Experimental setup of high-speed PIV measurement

A commercial double cavity Nd:YLF high speed laser, Litron LDV 300, was used for the PIV measurement. It was synchronized with the optical engine and a high-speed camera (SpeedSense 900) which was placed at another side of the optical liner. The time interval between the double frames was 20 µs. TiO2 was used as the PIV seeding and its particle size is about 3 µm. The seeding was taken into the cylinder with intake air. The EGR was supplied by the exhaust gas from an external burner burning conventional (MKI) diesel at stoichiometric conditions to produce CO2. More detailed description of the PPC engine and experimental setup could be found in previous annual reports and in the references. The different injection strategies for PIV measurement includes single, double and triple injections with varied start of injection timings (SOI) and injection durations.

Results

Since high-speed PIV was implemented, cycle-resolved data was evaluated. Cycle-resolved results mean it is possible to resolve the turbulence in one cycle. In such case, the impact from cycle to cycle variations could be prevented during the calculation. Both flow field and turbulent kinetic energy (TKE) were evaluated and presented. The detailed methods of calculation can be found in the references.

Figure 1 describes the averaged TKE of motored case and single injection cases from -20 CAD bTDC to 20 CAD aTDC. For case S16, SOI of the injection occurred at 16 CAD bTDC. The spray plume entered directly into the piston bowl region and impinged on the wall of piston bowl as it was observed by Mie-scattering recorded with high speed PIV camera. The spray itself carries a lot of flow momentum and TKE during the injection event. It also introduces a great deal of turbulence via air entrainment, turbulent diffusion and evaporation of volatile fuel. The amount of TKE caused by the injection was dominant during the compression stroke. The bulk flow structure generated by the injection created a clockwise vertical vortex in the piston bowl as shown in Figure 2 as point a in Figure 1.

A count-clockwise vortex occurred together with a clockwise vortex in case S16 while case S24 only had one clockwise vortex near TDC as shown in Figure 3. They represent point c and d in Figure 1. The high TKE occurred at the interface between the counter-clockwise vortex and the clockwise vortex in case S16, i.e. in the shear layer between them. After the TDC position, the TKE of case S16 decreased quickly due to the turbulence dissipation and reverse-squish flow.

The cycle-resolved averaged TKE of case D53-17, case D52-14 and case D52-20, which represented different SOI timings of second injection for double injection, are illustrated in Figure 4. The TKE level near TDC depends on the second injection and its fuel amount, which means the fuel injected inside the piston bowl mainly control the turbulence in the range of 20 CADs before and after TDC. With later SOI timing, the turbulence imparted by the fuel injection could be retained and TKE level became high as shown in Figure4. It was found that in case S24, the spray was hitting on the lip of the piston bowl rim by Mie-scattering images. The turbulence arose by the injection event was lost.

In figure 5, the cycle-resolved averaged TKE of case D53-17, T63-30-17 and T63-30-12 are compared. The later SOI timing of the latest injection, the higher TKE level could be achieved. It can be concluded that if too little amount of fuel is injected into the piston bowl that will only influence the turbulence before TDC and such influence will disappear after TDC.

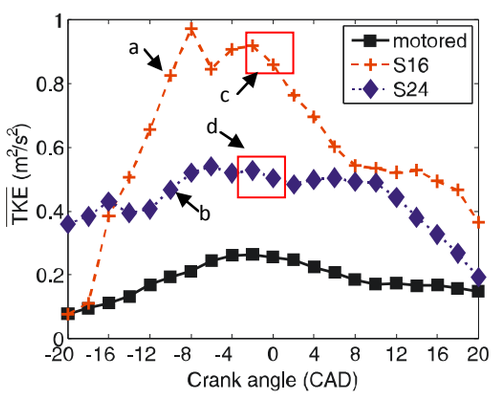


Figure 1. Cycle-resolved averaged TKE of motored case, S16 case and S24 case.

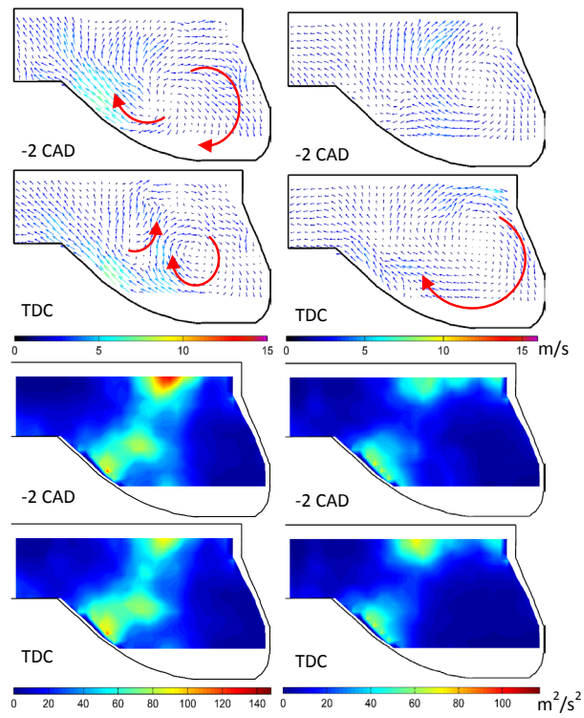


Figure 3. Ensemble averaged flow field and cycle-resolved averaged TKE for case S16 (left column, point c in Figure 1) and case S24 (right column, point d in Figure 1).

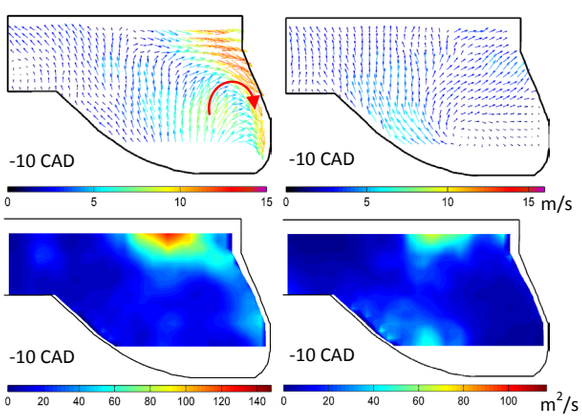


Figure 2. Ensemble averaged flow field and cycle-resolved averaged TKE for case S16 (left column, point a in Figure 1) and case S24 (right column, point b in Figure 1).

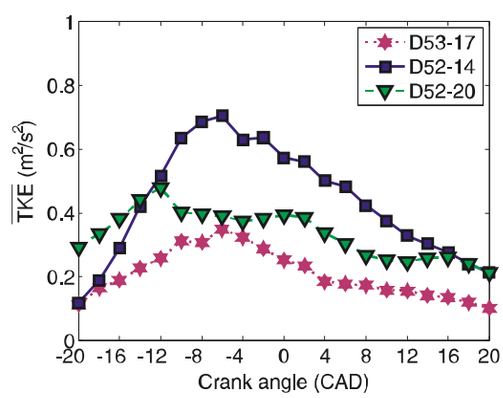


Figure 4. Cycle-resolved averaged TKE of case D53-17, case D52-14 and case D52-20.

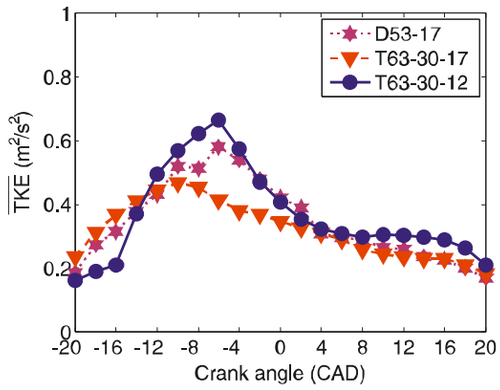


Figure 5. Cycle-resolved averaged TKE of case D53-17, case T63-30-17 and case T63-30-12.

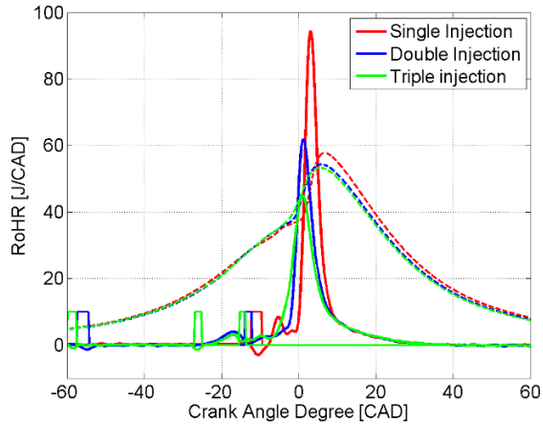


Figure 6. Heat release, pressure trace and injection signal for all injection strategies.

Rate of heat release

The rate of heat release was acquired during PIV measurement via a pressure sensor mounted on the engine cylinder head and presented in figure 6 for each injection strategy. All three cases give PPC-like combustion with long ignition delay, long combustion tail and a high peak heat release rate. The last injection at 16.5 CAD bTDC is used to trigger the high reactive mixture and control the combustion phasing.

Single Injection: Under the operating conditions considered here, the fuel injector was energized at 15 CAD bTDC for a period of 600 μs (3 CAD). The first column in Figure 1 illustrates the development of the flow-field structure inside the bulk shape. Without fuel injection (at –18 CAD), the flow structures in the bowl are primarily generated by piston motion. After fuel is injected at 600 bar, the flow pattern changes due to the associated displacement of the small seeding particles. The fuel jet gradually penetrates into the ascending piston bowl. Interactions of the jets with the bowl wall result in the formation of a strong, clockwise-rotating vertical structure within the bowl with velocities around 8–9 m/s. This vortex is set up during late compression time, and is clearly visible during the first part of the main heat release. At 6 CAD aTDC, the vortex is still present, but the velocity magnitude is much smaller compared to the TDC position. In addition, the centre of the vortex shifts upwards in the bowl while the piston descends. Another notable observation at this CAD is a small counter-clockwise vortex that forms close to the cylinder centreline (near the piston "pip") with velocities around 3 m/s. This is very close to what other authors found in a DI diesel engine with a similar re-entrant bowl-shape piston. Beyond 18 CAD, the fluid close to the piston surface follows the descending movement of the piston. At 20 CAD and beyond, the data from the bowl region are no longer available because the region moves out of the measurement area.

In addition to the ensemble average velocity field, the time sequence of the spatial turbulence distribution can be seen in the second column of Figure 7. At 12 CAD bTDC, when fuel is already injected and redirected at the piston bowl, high turbulence intensity (~120 m2/s2) can be obtained. This is mainly because the fuel jet itself carries a large amount of flow momentum and kinetic energy, which is transported to the in-cylinder fluid by entrainment, turbulent diffusion, and evaporation. Further at TDC when the main heat release starts, high velocities inside the bulk produce higher vorticity and turbulence. Also during this CAD period, high turbulent kinetic energy (~80 m2/s2) is generated at the top of the piston, which can be caused by the squish turbulence. After 6 CAD, a rapid decrease in the turbulence intensity can be seen. The large-scale flow structure has broken down and the turbulence starts to dissipate with the descending motion of the piston.

Double injection: The temporal and spatial development of the flow within the combustion chamber as the piston approaches TDC is clearly demonstrated in Figure 2. In this case, fuel was injected at 60/16.5 CAD bTDC and with 600 bar rail pressure. At –12 CAD, shortly after the second injection, a small-scaled vortex forms at the outward location of the piston bowl. The second injection leads in less spray momentum available in the bowl, which leads to a weaker clockwise toroidal vortex. As the piston continues to move toward TDC, the small vortex is compressed and breaks down. After sufficient mixing of the fuel/air mixture, a small amount of heat release has occurred close before TDC. At this CAD and beyond, the vertical plane flow structure is mainly dominated by combustion-induced flow. When combustion occurs, the gas expansion due to heat release acts to displays the seeding, which further lead to an enhanced flow vortex, created low in the bowl. After the main heat release at 14 CAD, double vortex flow, as seen in the single-injection case, can be observed but in a weaker form. A small counter-clockwise toroidal vortex is formed close to the piston pip. As mentioned in the single-injection case, beyond 20 CAD the bowl region is

out of the examination window and no data of the bulk flow was obtained. The character of the turbulence flow calculated is also illustrated in the second column of Figure 2. We note that the colour bar scaling of the turbulence intensity is much smaller compared to the single-injection case. With a lower duration time of the second fuel injection, the turbulence mixing is consequently reduced. At –12 CAD, high TKE levels (~20 m2/s2) at the outward bowl and near the piston lip can be observed. Towards the end of the compression stroke (TDC), the charge is compressed into the bowl and together with the combustion flow the charge is directed towards to the piston pip, creating high turbulence levels in this region. It can also be noted that high turbulence intensity is generated in the upper part of the piston area due to the process of compressing swirling charge into the bowl.

Triple Injection: The triple-injection case shows some similarities to double injection, and differs greatly from single injection. Figure 9 illustrates the triple-injection strategy in which fuel was injected at 62.5/22.5/17 CAD bTDC. For the last fuel injection only a small amount of the fuel penetrates briefly into the piston and is used to trigger the combustion. Without the third injection, the premixed fuel/air does not combust. At –16 CAD, the spray momentum of the last injection sets up a clockwise rotating small vortex at the outward position of the piston. When the compression strokes continue, no vortex is visible at – 6 CAD. However, at TDC and beyond, velocity vectors in the planar flow field get smaller. These small vectors indicate the location of combustion, because of the absence of small particles that were pushed towards the centreline by the flow of the combustion. Then at 8 CAD (long after the main heat release), the structure of the double vortex, as in single and double injection, is recognizable again. Beyond 10 CAD, the double toroidal vortex observed earlier appears to dissipate, and after 18 CAD the flow structure of the cylinder is predominantly in the direction of the downwards moving piston.

In Figure 9, the second column shows the evolution of the TKE intensity distribution at the centre plane. Once the small clockwise toroidal vortex at the piston lip is generated, a high turbulence level (~30 m2/s2) can also be observed at this position. After 8 CA, the vortex breaks down; and as the compression stroke proceeds, the squish flow penetrates a smaller distance towards the cylinder centreline. The experimentally measured TKE level in the upper portion of the cylinder however, appears considerably stronger than in the double-injection case. From TDC until 6 CAD aTDC, high TKE intensities are estimated at the inward portion of the bowl and also at the upper region of the piston, due to more squish and swirl interaction. As the pistons begin to move downwards, the turbulence levels starts to decrease as well. We note that at 8 CAD and 12 CAD, late-cycle turbulence is perceptible. This continued turbulence production is most likely due to combustion-induced expansion, which causes velocity fluctuation within the piston bowl.

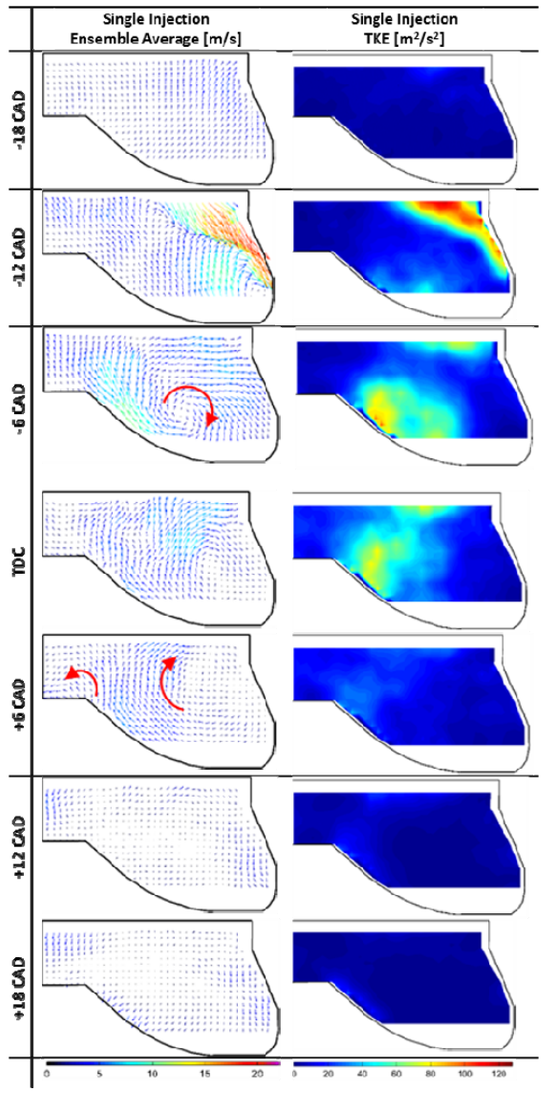


Figure 7. Mean flow field and TKE field at different crank angle positions for the single-injection operating condition.

For an overview the differences in flow-field structure for motored conditions and single- and multiple-injection strategies, turbulence through the cycle has been calculated for the fired and non-fired case by the moving window method over the sampled crank angle degree period. From Figure 10, it is clear that the motored condition has the lowest value for the turbulence.

In general both the fired and non-fired have the same turbulence behaviour from start of injection until TDC, respectively. Short after TDC, when heat release occurs the turbulences of both cases are diverging. For fired case the flow turbulence starts to increase additionally due gas expansion from the resulting premixed combustion and for non-fired case it decreases progressively. Further the results presented in this figure highlight an increase in turbulence (3 m/s) for triple injection before 15 CAD bTDC for both, fired and non-fired cases. This increase in turbulence is mainly dominated by the carried fluid momentum from the

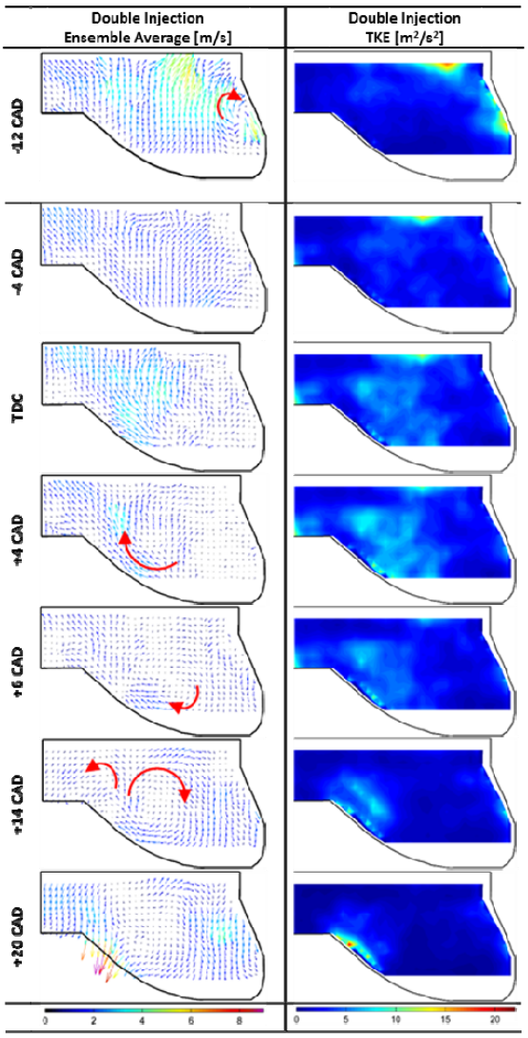


Figure 8. Mean flow field and TKE field at different crank angle positions for the double-injection operating condition.

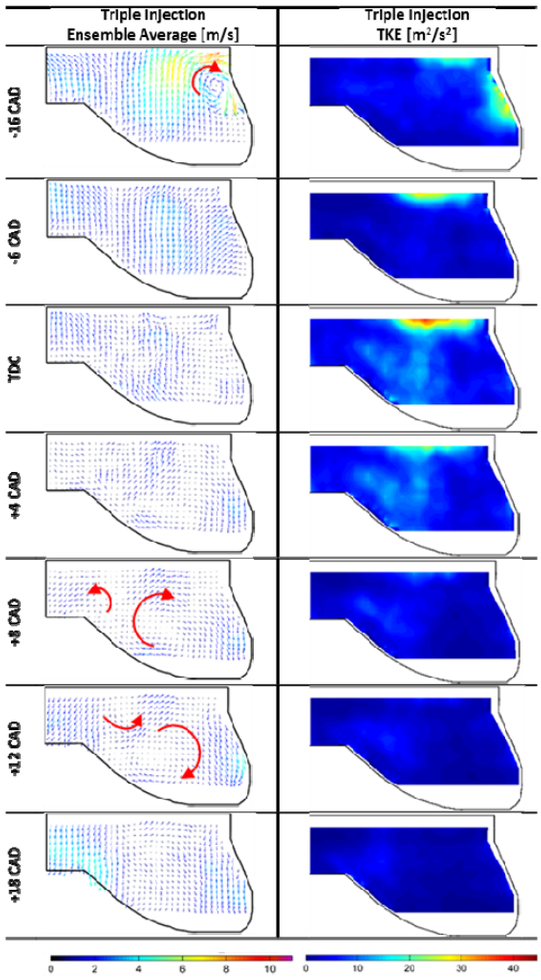


Figure 9. Mean flow field and TKE field at different crank angle positions for the triple-injection operating condition.

second and third fuel injection event, namely at –29.5 and 62.5. Since the time between the first and second injection for the double-injection case is very large, this leads to smaller velocities and turbulence. If we compare single injection (injection duration ~2.9 CAD) with multiple injections (injection duration ~1.9 CAD for double injection and ~1.7 CAD for triple injection), the magnitude of the turbulence for single injection slightly decreases until –5 CAD. Since the timing for single injection was –15 CAD, and the total fuel mass gradually penetrates into the piston bowl, high turbulence and velocities are generated due to the spray momentum. It is evident that the single injection has the highest turbulence value and the motored case has the lowest, as expected. Around –2 CAD, a second peak can be observed for the single injection; this can be a positive side effect of the turbulent squish flow (the flow in the squish volume is compressed and pushed into the piston bowl), which enhances the turbulence again to higher values after a slight decrease. Around the TDC position, the measured turbulence shows the same behaviour for all injection strategies. It can be seen that the turbulence increases when the piston approaches TDC, and the flow from the squish volume moves inward towards the piston centreline and increases turbulence. The time between 4 CAD bTDC and 8 CAD aTDC approximately corresponds to the time duration of the heat release for double and triple injection. Once the air/fuel mixture is combusted, a continued flow turbulence is generated. Miles et al. explained that this turbulence is most likely due to high shear regions generated as the combustion-induced gas expansion causes increased velocities. In general, when the piston moves downwards, the generated turbulence breaks down and dissipates for all injection strategies.

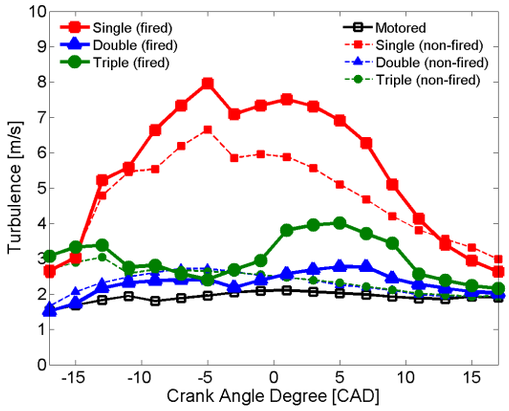


Figure 10. Temporal magnitude turbulence vs CAD for all injection strategies.

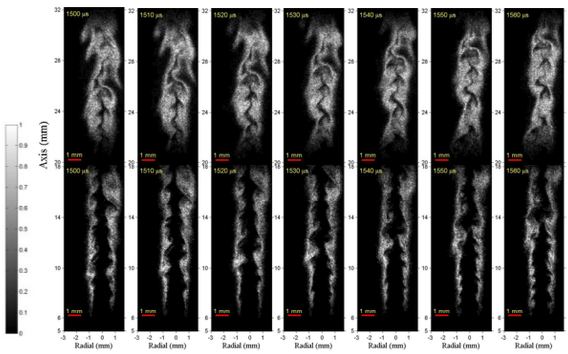


Figure 11. Result of CH_2O with 66 m/s flow speed and Φ=1 at 10 µs temporal resolution.

Table 1. Specification of burst-mode laser

Name	Quasimodo	
Pulse width	10 ns (adjustable)	
Duration of pulse sequences	maximum 10 ms	
Spectral Bandwidth	< 1 GHz	
Beam diameter	8 mm	
Frequency	10 kHz	100 kHz
Pulse energy	at 10 kHz	at 100 kHz
@ 1064 nm	1.4 J	160 mJ
@ 512 nm	600 mJ	53 mJ
@ 355 nm	300 mJ	37 mJ
@ 266 nm	100 mJ	10 mJ
Pulse numbers	100	1000

Application development of a 100 kHz burst-mode laser

Table 1 depicts the specifications of this burst-mode laser. The repetition rate of the laser is adjustable. The higher repetition rate, the lower pulse energy it can produce. The pulse energy is comparable with a common 10 Hz Nd:YAG laser at 10 kHz repetition rate. Within in a burst, the pulse train contains 1000 pulses (10 ms burst duration) at 100 kHz. Due to the advantage of high temporal resolution (10 µs) with the high speed burst-mode laser, its application could be used to obtain the transient combustion behaviour in the ICE, such as the ignition of HCCI or PPC. It can also achieve the cycle-resolved information for the investigation of the development of turbulent combustion in the PPC engine.

The measurement of CH_2O in a LUPJ burner at high turbulent flame conditions were performed and developed. The results were shown in figure 7. Along the horizontal direction, the time interval is 10 µs. That demonstrates the potential capability of utilizing it for the investigation of combustion inside optical engines.

Future Work

In 2016, PLIF measurement of fuel distribution will be performed to better understand the mixing process of PPC.

Reference

Zhenkan Wang, Slavey Tanov, Hua Wang, Mattias Richter, Bengt Johansson, and Marcus Alden. High-Speed Particle Image Velocimetry Measurement of Partially Premixed Combustion (PPC) in a Light Duty Engine for Different Injection Strategies. No. 2015-24-2454. SAE Technical Paper, 2015.

Tanov Slavey, Zhenkan Wang, Hua Wang, Mattias Richter, and Bengt Johansson. Effects of Injection Strategies on Fluid Flow and Turbulence in Partially Premixed Combustion (PPC) in a Light Duty Engine. No. 2015-24-2455. SAE Technical Paper, 2015.

PPC - Light Duty

Experiments

Introduction

Nikolaos Dimitrakopoulos
PhD Student

The diesel engine has been the main means of vehicle and ship propulsion for many decades due to two main reasons, high reliability as well as high fuel efficiency. Despite these advantages, the biggest drawback of diesel combustion is the exhaust emissions. These come in the form of nitrogen oxides (NOx) and particulate matter (PM). As the emission legislation becomes more stringent, controlling them becomes more challenging and the cost of the aftertreatment system can be as high as the cost of the engine itself.

Background

One of the first alternative combustion processes that promised diesel-like fuel efficiency but without the high exhaust emissions was homogeneous charge compression ignition (HCCI). The main drawback of this process was the limited load range as well as the challenging combustion control. Partially premixed combustion (PPC) concept which was introduced by Lund University is an alternative way to reach the same, or even higher fuel efficiency while keeping the exhaust emissions to a minimum. Running the engine with gasoline-like fuels gives an adequate time for fuel and air mixing, which reduces the PM formation and using excess air or exhaust gas recirculation (EGR) reduces the combustion temperatures to a level that NOx formation is at a minimum.

Experimental Setup

During the previous years, most of the fundamental research on PPC was carried with optical and metal single cylinder engines and the results were promising for the feasibility of the PPC concept. For this new PPC project it was decided that a commercial multi cylinder metal engine will be used, while keeping the engine as close as possible to the original configuration. That means that the turbocharging and intake system that is installed on the engine from the manufacturer will not receive major modifications (Figure 1).

The research will be based on a Volvo 2litre diesel VED-D4 engine, which complies with the Euro 6 emission requirements. The main modifications to that engine is a long route EGR system and a watercooled intercooler in order to control better the intake temperature. The engine control system is based on Labview 2014 (Figure 2) and is controlled by a dedicated real time NI PXI system that has also data acquisition capabilities (Figure 3).

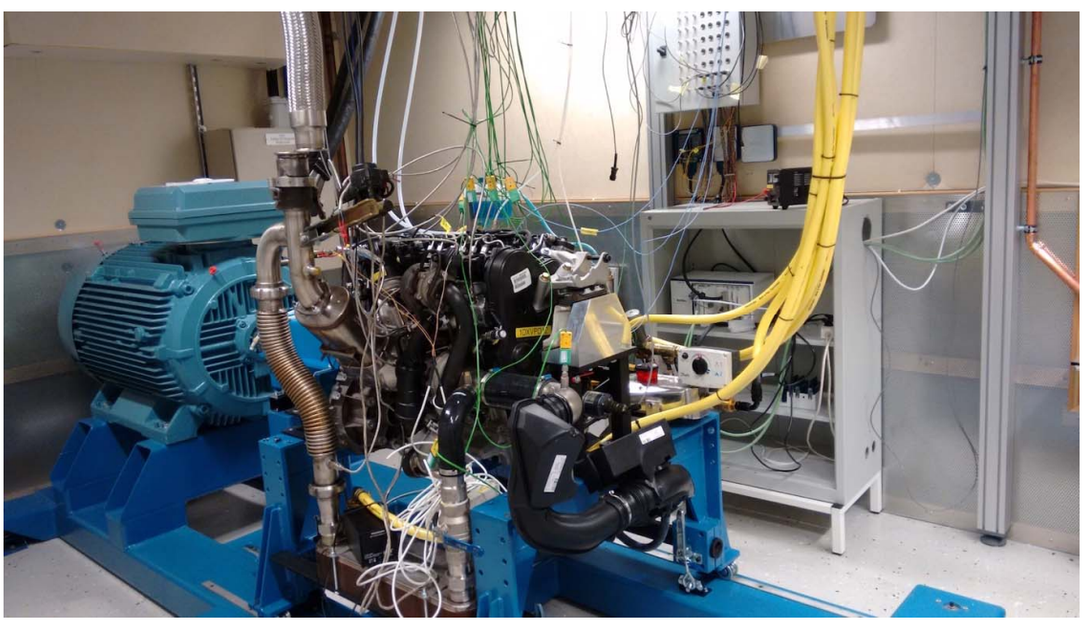


Figure 1. Volvo VED engine modified for PPC operation

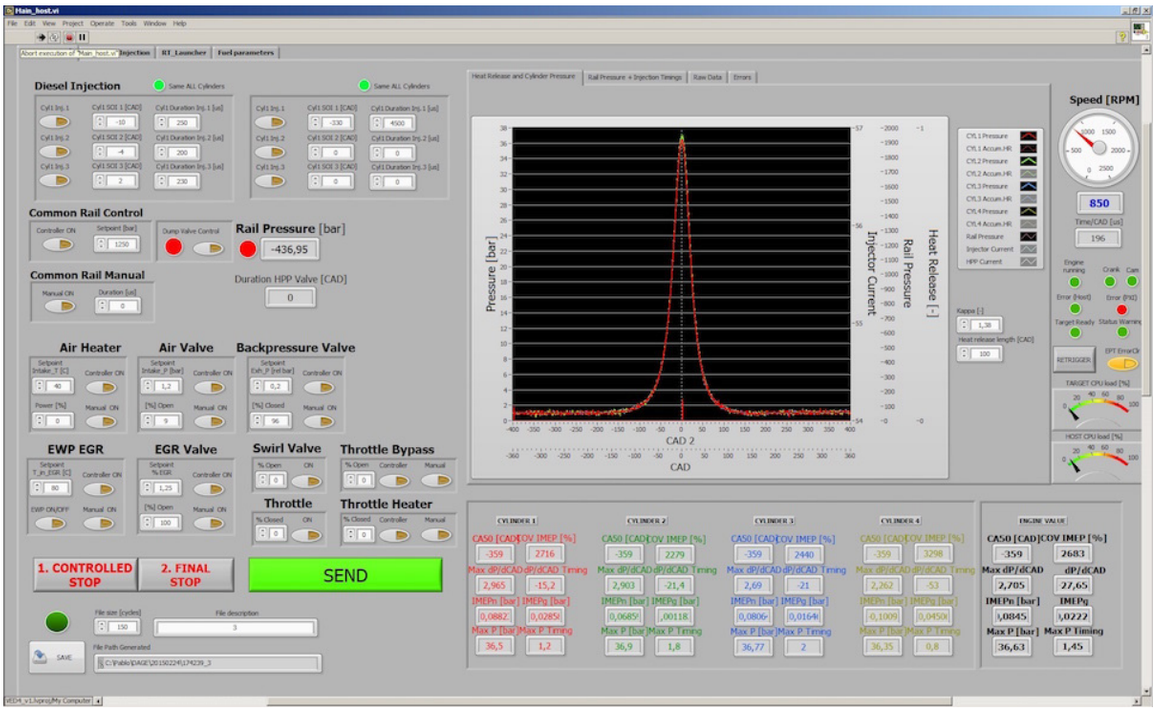


Figure 2. Labview engine control software

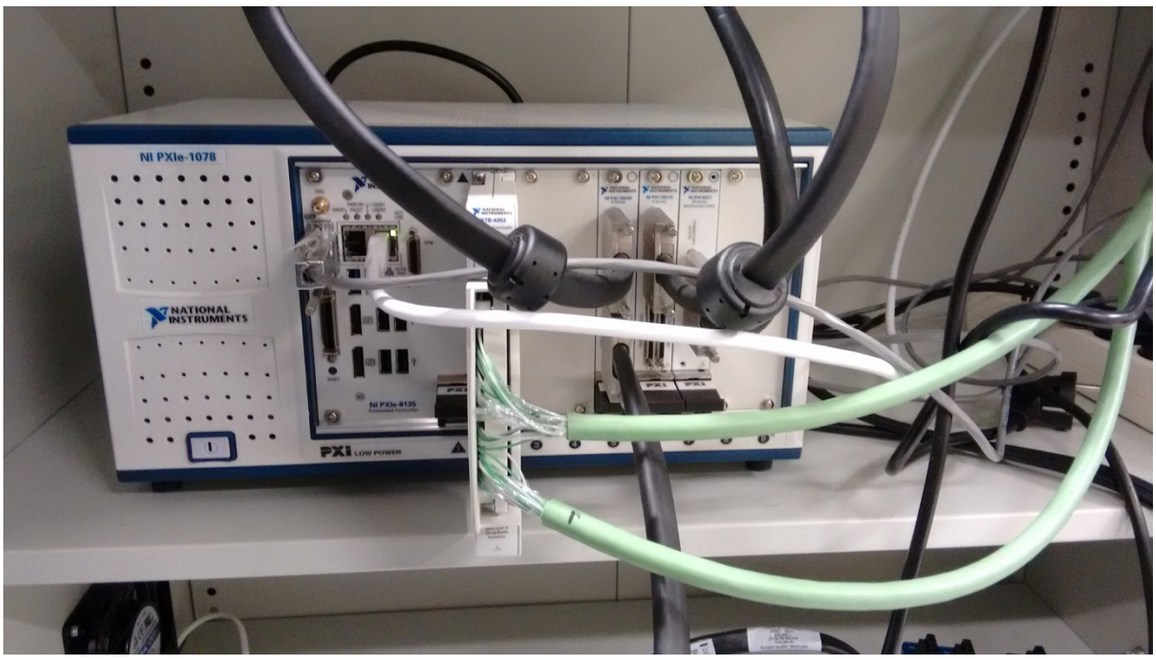


Figure 3. NI PXI real time engine control system

Future Work

Previous research has shown that low RON gasoline has the best behavior under PPC operation. As a result, after the complete installation and calibration of the engine, a load sweep will be performed with RON 70 gasoline, followed by advanced injection strategies in order to extend the low load operation limit as low as idle operation.

PPC - Modeling

Introduction

Christian Ibron
PhD Student

The aim of the CFD modeling project is to perform detailed numerical simulations of PPC combustion to improve the understanding of the mixing process and the underlying combustion physics. Reynolds averaged Navier-Stokes (RANS) simulation, Large Eddy Simulation (LES) and Direct Numerical Simulation (DNS) are used as tools to simulate the fuel injection, mixing, turbulent flow, and the combustion process and to provide with detailed time dependent, three-dimensional, fields of velocity, composition, and temperature of high temporal and spatial resolution. LES and DNS are needed to capture the important energetic large-scale and turbulent structures, such as vortex formation and shear layer instabilities, which control, e.g., fuel and air mixing and fuel/air entrainment. RANS is an approach widely used in industry application. It is important to identify the advantages and limitations of the different approaches.

In the past year the CFD modeling project has been focusing on the following two works; (a) RANS study of in-cylinder flow dynamics in the optical VOLVO D5 engines, which is in close collaboration with the PIV measurement projects; and (b) DNS of PPC combustion of n-heptane/iso-octane mixture under PPC engine relevant conditions. The latter study provides model-free numerical data that can reveal insight to the physics important in the PPC process.

CFD study of in-cylinder flow dynamics

In the PPC optical engine experiments subprojects, effort has been made to measure the incylinder flow field in light-duty PPC engines using high-speed PIV. The CFD project has thus been focusing on the simulation of the incylinder flow in the experiment engine. The optical engine is based on the Volvo D5 engine modified with a quartz piston and a quartz cylinder liner to allow for optical access. A blended fuel, PRF 70, consisting of 30% n-heptane and 70% iso-octane in volume percentage was used in the experiment to achieve long ignition delay for PPC. Motored run, single injection, double injection, triple injection, with and without combustion cases are studied in the experiments. In the CFD simulations RANS approach is currently employed to study the various experimental cases to (a) validate the RANS CFD approach for simulation of incylinder flows in light-duty PPC engines, and (b) provide more detailed flow field data (three-dimensional) supplementary to the experimental data for a comprehensive understanding of the flow structures, production and decay of the turbulence, and the mixing field in different piston positions.

Figure 1 shows the mean velocity vector fields at three different piston positions from the RANS and PIV measurements, under the motored run condition. The engine speed is 800 rpm, and the compression ratio is 11.3. The intake valve geometry is not considered in the simulations; the initial velocity field at BDC is assumed to be a solid-body rotational flow with a swirl ratio of 2.6, estimated based on the engine experiments. At the motored run condition due to the absence of fuel injection the flow field in the later compression stage and the expansion stage is highly depending on the piston motion. At -20 CAD the piston motion leads to the formation of two flow streams upward, one in the piston bowl region and one in the squish room, which counter flow towards each other and form a

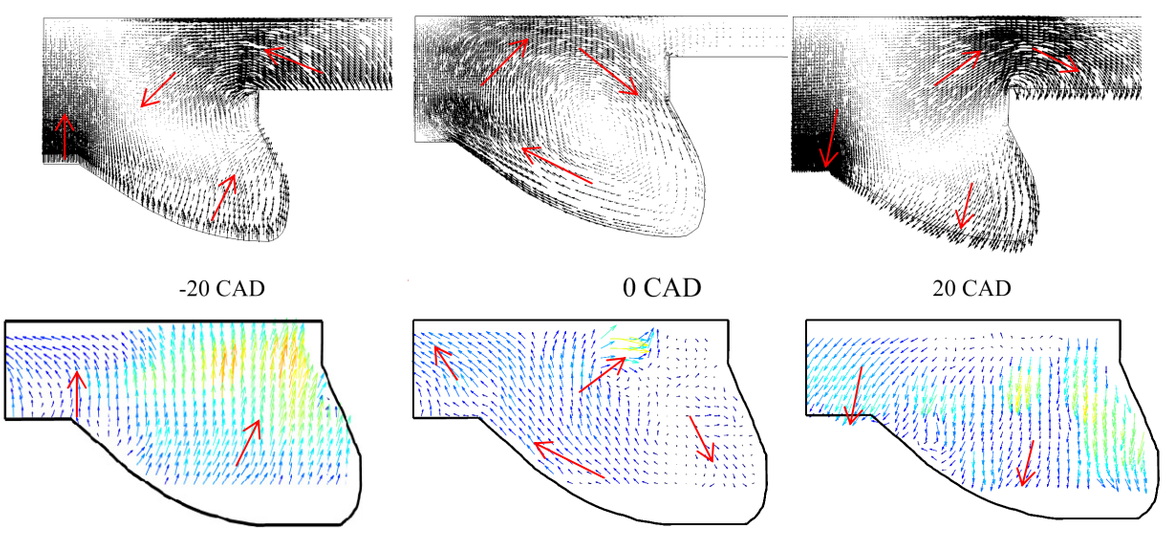


Figure 1. Mean velocity vector field at different piston positions under motored run condition. Upper row: RANS results, lower row: PIV results.

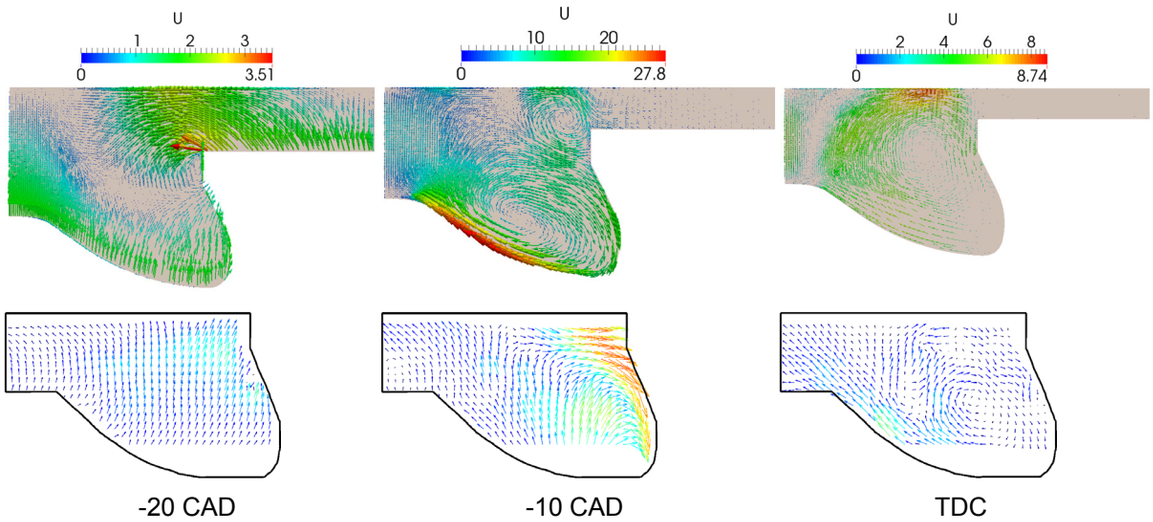


Figure 2. Mean velocity vector field at different piston positions under a single injection condition. The injection is at -16 CAD. Upper row: RANS results, lower row: PIaV results.

low speed region in the middle of the bowl. At TDC the piston speed is zero and no such counterflow streams can be supported, which results in a clockwise rotating large-scale recirculation zone. At 20 CAD in the earlier expansion stroke the piston is moving downward, forming a two flowing streams departing from the center of the cylinder. The swirl flow in the third dimension fills up the void left by the piston motion. The RANS results are comparable to the PIV at the TDC and 20 CAD. At the -20 CAD the PIV results did not reveal a significant low speed region. The discrepancy between the RANS results and the PIV results could be due to (a) the RANS model, (b) the neglecting of the intake valve and intake manifolds in the CFD simulations. In the ongoing study we are working on the implementation of the real intake manifolds and intake valve motion. Simulations using a more advanced model, i.e. the LES approach, will be considered when the entire engine geometry is properly modeled. It is expected that this study will clarify a few uncertainties in the current CFD simulations, e.g., the uncertainty due to the intake geometry and initial condition, as well as turbulence modeling.

Figure 2 shows the single injection case (without the onset of combustion) with the start of injection at -16 CAD and injection duration of 600 microseconds. In the experiments, high amount of EGR is used to quench the ignition of the injected fuel. The flow field at -20 CAD is fairly similar to that of the motored run case; at -10 CAD, which is after the end of injection, the RANS results show two large scale recirculation zones, one in the bowl region and one in the squish room. In the PIV the bowl region recirculation zone is evident but the squish room is not in the shown experimental window. In between the two-recirculation regions a high-speed flow stream is evident, which is due to the liquid fuel jet. In the PIV results the fuel jet induced flow stream is stronger than that predicted by the RANS model. At TDC the recirculation zone in the squish room is compressed away, while the recirculation zone in the bowl is maintained during the compression process. The difference between the RANS results and the PIV results can be due to the turbulence model and the intake geometry, which was not considered in the RANS study shown in Figure 2. A future investigation is needed to scrutinize the uncertainties caused by such simplifications.

DNS study of n-heptane/isooctane mixture under PPC engine relevant conditions

In parallel to the RANS studies of the incylinder flows, DNS studies of the combustion process under PPC engine relevant conditions are carried out. The results are presented in Refs. [1,2]. The fuel is PRF70, consisting of 30% n-heptane and 70% iso-octane in volume percentage, which was used in the light duty PPC experiment to achieve long ignition delay for PPC. To resolve the fine details of the reaction layers it is necessary to have a mesh size around one micrometer, which essentially limits the computation to only small domains. In the current study we have used a uniform grid of 5123 cells, with a resolution of 1.22 micrometers mesh size. This spatial resolution was found to resolve the fine reaction layers with at least 10-15 mesh cells. The domain size is therefore a cube with each side of 614 micrometers. Figure 3 shows the initial distribution of an iso-surface of iso-octane mass fraction (YC8H18=0.0082) with the distribution of temperature shown on the iso-surface. The small-scale pockets are from the first injection (early injection). The relatively large-scale fuel-rich pocket in the center of the domain is from the second injection. The DNS case is therefore a simplified case with two injections. The engine speed is 1100 rpm, and the compression ratio is 14. The initial field is set at 10 CAD before TDC, where the initial pressure is 40 bar. The local temperature ranges from 750 K to 1200 K in the domain, and 50% EGR was used in

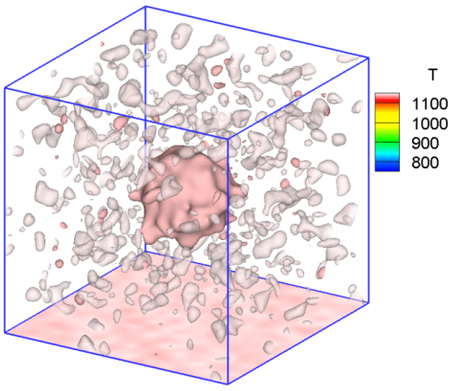


Figure 3. Iso-surfaces of iso-octane mass fraction (Y_{C8H18}=0.0082) colored with temperature in the initial field.

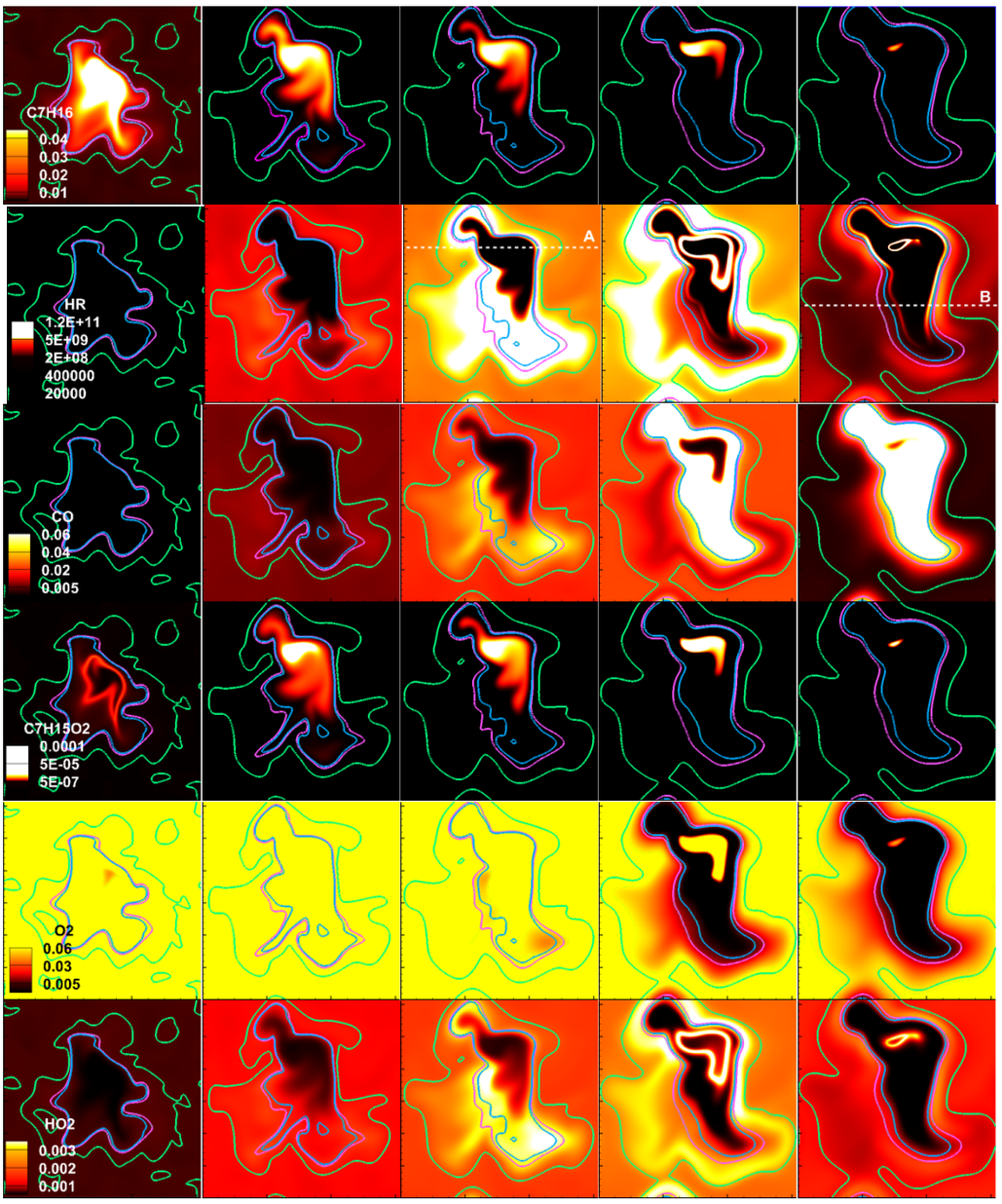


Figure 4. Distributions of heat release rate (HR) and mass fractions of n-heptane, CO, O_2, HO_2, and C_{7H15O2} in the middle plane of the domain at various instances of time, from left to right, 0.0317 ms, 0.135 ms, 0.171 ms, 0.2 ms, and 0.237 ms. The iso-lines correspond to the local equivalence ratio (Φ) of 1.2 (the innermost blue line), 1 (the middle purple line), and 0.5 (the outermost green line).

the simulation. More detailed information of the DNS case is given in Ref. [1].

To look into the details of the ignition process in the 3D PPC case, a series of 2D distributions in the center plane of the domain are plotted in Figure 4. The iso-lines of the local equivalence ratio (Φ) are also shown in the figure. Based on the equivalence ratio the domain can be divided into three regions: the lean nearly homogeneous charge (LHC) region where the fuel is from the first injection (Φ< 0.5), the fuel-rich charge (FRC) in the center of the domain where the fuel is mainly from the second injection (Φ> 1.2), and in between the two regions the charge with Φ around unity (0.5 <Φ< 1.2) is referred here as the stoichiometric charge (SC) region.

As seen in Figure 4, at 0.0317ms the ignition process is at its earlier stage. In the FRC region the fuel-rich pocket is distorted by turbulence, while HR is not significant and no CO is formed. $C_7H_{15}O_2$ is formed along the periphery of the fuel-rich pocket, indicating low temperature ignition reactions taking place in the region. In the LHC region the concentration of HO_2 is high, indicating the high temperature ignition reactions in the LHC.

At 0.135 ms, a significant heat release is seen in the LHC region due to the initial high temperature. Furthermore, ignition is shown to take place also in the SC and FRC regions. Fuel is consumed quickly in the LHC, SC and part of the FRC region forming intermediate species such as CO. In the central FRC region a high-level $C_7H_{15}O_2$ is observed indicating the onset of low temperature ignition in the entire FRC region. At 0.171 ms, the highest heat release rate is seen in the SC region and part of the FRC region, where a high-level CO is also found. In the LHC region HR is relatively low owing to the low F (< 0.5).

As combustion proceeds to 0.2 ms, the fuel is seen only in a small region in the central FRC region where $C_7H_{15}O_2$ is still observable. In the FRC region outside the fuel-rich pocket CO is at its highest concentration. The peak temperature is in the SC region due to the close-to-stoichiometric condition. A thin layer of high HR is seen to embrace the fuel pocket. This thin reaction zone is rather similar to a rich premixed flame front, as indicated by the O_2 distribution in the fuel-rich pocket. CO in the LHC region is oxidized further to CO_2 in a HCCI mode.

At 0.237 ms, CO oxidation is almost completed in the LHC region. The fuel, the oxygen, $C_7H_{15}O_2$ and HO_2 in the FRC region are consumed almost completely (except in a small region in the center of the FRC region). The entire FRC region is filled with CO. Thereafter, CO oxidation becomes the dominant process, via the elementary reaction OH + CO = CO_2 + H. Since in the FRC region oxygen is consumed completely in the process of the CO formation, the consumption of CO in the FRC region is controlled by the mixing of oxygen in the LHC with the CO in the FRC region. This process is the well-known diffusion controlled combustion.

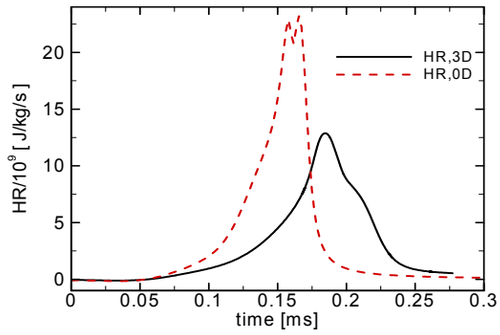


Figure 5. Temporal evolution of the heat release rate in the 0D HCCI case and the 3D PPC case.

Figure 5 shows a comparison between the volume-averaged total heat release rate (HR) from the 3D DNS (PPC case) and the corresponding results from the 0D run (the HCCI case). It is seen that auto-ignition in the HCCI case occurs earlier, and HR in the HCCI case is significantly higher than the PPC case. There are significant differences between the HCCI and PPC processes. In the HCCI case the combustion process is of a sequential manner: the major heat release is after the fuel being consumed, whereas in the PPC case there is an overlap between the consumption of the fuel and the consumption of the intermediate species, which results in a slower heat release process. The HR profile in the HCCI case shows two local peaks. The first is at 0.157ms, which coincides with the peak of CO profile; thus, the first HR peak is owing to the rapid CO oxidation reaction (CO + OH = CO_2 +H). The second HR peak is at 0.165ms, which coincides with the peak of HO_2 profile, thus the second HR peak is due to the exothermic reaction of HO_2 formation (H+ O_2 + M = HO_2 + M).

Future Work

In the coming future work CFD analysis of the incylinder flow, mixing and combustion process will be carried out on the Volvo D5 optical engine geometry under operation conditions studied in the experimental subprojects. The full intake manifolds and valve motion will be considered to characterize their effect on the incylinder flow and combustion process. LES will be used to study the cases to quantify the impact of turbulence models on the incylinder flow, fuel/air mixing and the combustion process.

References

[1] F. Zhang, R. Yu, X.S. Bai, *Direct numerical simulation of PRF70/air partially premixed combustion under IC engine conditions*, Proc. Combust. Inst., 35:2975-2982, 2015.
[2] F. Zhang, R. Yu, X.S. Bai, *Effect of split fuel injection on heat release and pollutant emissions in partially premixed combustion of PRF70/air/EGR mixtures,* Applied Energy, 149:283-296, 2015.

PPC - Control

Introduction

This project focuses on Partially Premixed Combustion (PPC) multi-cylinder engine control. The purpose is to develop advanced control oriented model and control algorithm to optimize the engine efficiency and maintain low engine emission at the same time. The effort were taken both on gas exchange system control and combustion process control. In the last year, one part of the research work is to investigate the PPC multi-cylinder engine efficiency; the other part is to use advanced control method to control the combustion process.

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Optimal settings of lambda and EGR for high efficiency PPC engine

The experiments covered lambda (1.4-1.9) and EGR (40%-60%) and the NIE results are shown in the Figure 1. The highest NIE was located at low lambda and EGR range from 50% to 55%.

From the NIE results, two trends were observed. One trend is that as lambda decreased, the NIE increased, the other one is that the NIE increased with EGR and reached the peak at EGR from 50% to 55%, then drop with further EGR increase.

The main contributors of the NIE trend is Thermodynamic Efficiency (TE) trend. The TE was plotted in Figure 2. Peak TE located at region of EGR between 50% to 55% and lambda above 1.5. The ratio of highest TE and lowest TE is 18% which is significantly larger than the CE ratio of 1%. Since the GIE was calculated by CE times TE. This means that the trend in GIE was more influenced by the thermodynamic process.

The thermodynamic efficiency is the result of 100% efficiency minus the exhaust-loss-energy percentage and the heat-transfer-energy percentage in the total fuel energy. The trends of both energy percentage can be used to understand the trend of thermodynamic efficiency. Both energy percentage are illustrated in Figure 4.

The peak of exhaust-energy and heat-transfer-energy located in the opposite region (Right side in the upper figure and left side in the lower figure). Low lambda and high EGR leads to a high waste energy in exhaust and low heat transfer, while high lambda and low EGR leads to a low waste energy in exhaust and high heat transfer. High EGR and low lambda dilutes the combustion and reduces the in-cylinder temperature and results in low heat-transfer. The same region is characterized by the long combustion duration, which results in more waste energy in the exhaust.

The high EGR region has too much waste energy in exhaust and the low EGR region has too much heat-transfer energy, the region in the middle represents the best trade-off of both energy losses and has the highest TE.

The variation of the combustion efficiency is quite limited and thus the trends in GIE are mainly determined by the thermodynamic efficiency.

As in Figure 3, GIE and NIE increased with load. GIE reached the peak of 50.9% at 11.2 bar IMEPn (Point A) and NIE reached the peak of 49.7 % at 13.5 bar IMEPn (Point B) and then decreased with the load. The brake efficiency in-

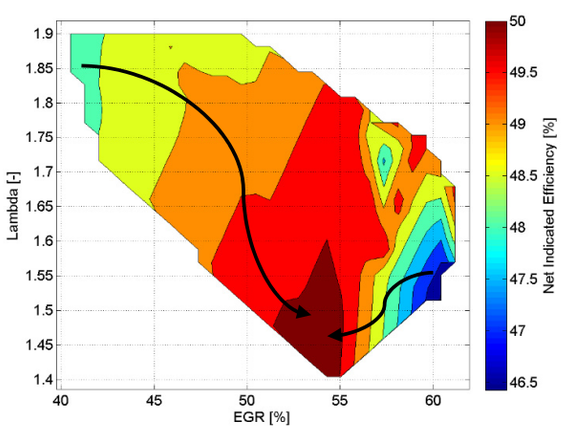


Figure 1. Net indicated efficiency.

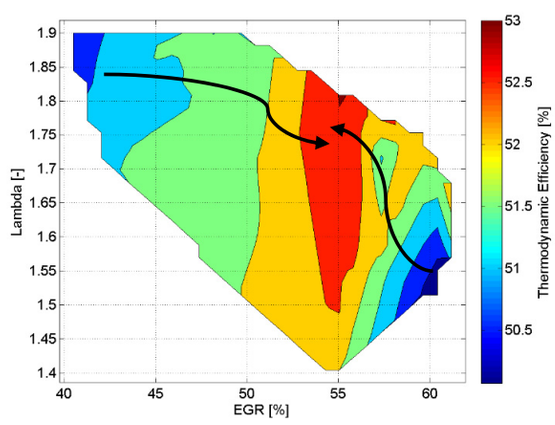


Figure 2. Thermodynamic efficiency. Peak TE located at region of EGR between 50% to 55% and lambda above 1.5. The ratio of highest TE and lowest TE is 18% which is significantly larger than the CE ratio of 1%. This means that the trend in GIE was more influenced by the thermodynamic process.

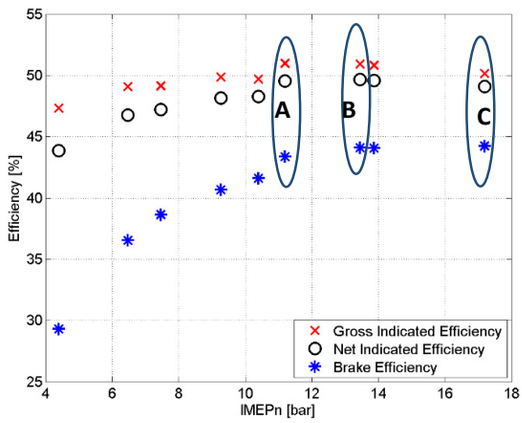


Figure 3. GIE, NIE and brake efficiency were plotted against IMEPn.

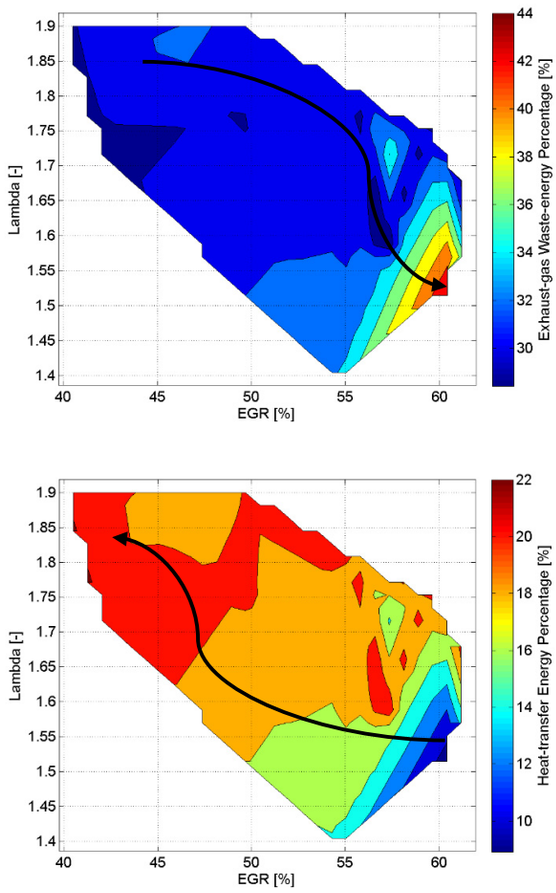


Figure 4. Exhaust-energy percentage in total fuel energy in upper figure and heat-transfer-energy percentage in total fuel energy in lower figure.

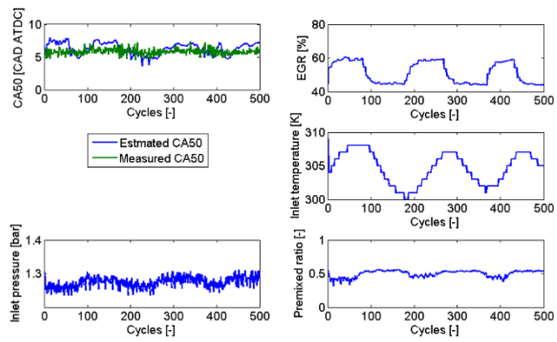


Figure 5. Model-based combustion timing control

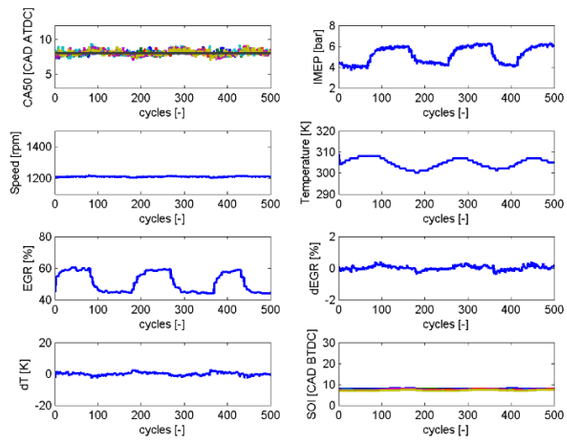


Figure 6. Model-based combustion timing control

creased with load and a maximum of 44.2% was observed at 17.2 bar IMEPn (Point C).

More details, results and discussion can be found in:
An experimental investigation of a multi-cylinder engine with gasoline-like fuel towards a high engine efficiency
Yin, L., Ingesson, G., Johansson, R., and Tunestål, P. (2016). Submitted to the SAE world congress, April 13, 2016, Detroit, USA

Model-based combustion - Timing control

Partially Premixed combustion (PPC) is a promising combustion concept to achieve high engine efficiency. The combustion timing of PPC is affected by both inlet-charge condition and the fuel injection, a simple map-based feed-forward control method is not sufficient for controlling the combustion during transient operation. This part of work investigates a model-based control method to solve this control problem. A non-linear model was developed to capture the main trend of inlet-gas condition and injection. The model was validated in the experiments.

A Model Predictive Controller (MPC) was designed to control injection timing and the gas-exchange system. The controller was verified in a multi-cylinder heavy-duty PPC engine.

More details about the controller can be found in:
Model based Partially Premixed Combustion (PPC) timing control Yin, L., Ingesson, G., Johansson, R., and Tunestål, P. (2016). Submitted to the IFAC Symposium on Advances in Automotive Control, June 20-23, 2016, Kolmården, Sweden

Future work

A combustion timing reference tracking problem was formed and investigated using MPC in this report, the next step will form a optimization problem which optimize the net indicated efficiency with a proper heat release by adjust the inlet condition and injection (Octane number might also be a control output), under the emission constraints.

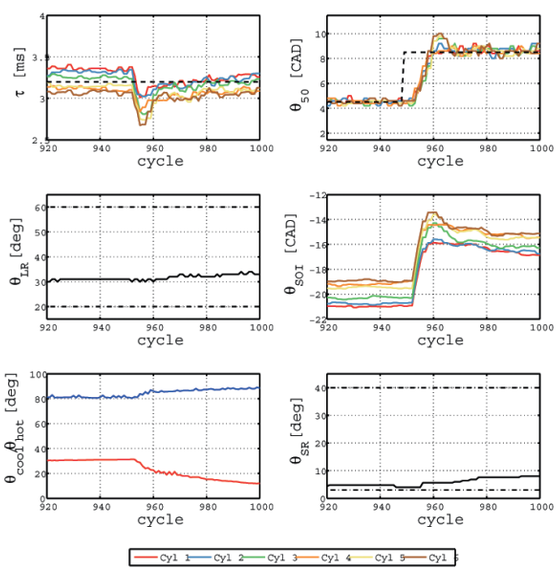


Figure 7. In this figure, a model predictive control experiment is conducted on a six-cylinder engine where the ignition delay (upper right) is regulated while the combustion timings (upper left) is tracking a set-point step change. In order to delay the combustion timings, the fuel injection timings (middle right) have to be delayed during the combustion stroke. Since the in-cylinder air is hotter later during the compression stroke, control action has to be taken by the gas-exchange system to keep the ignition delay constant. The in-cylinder charge has to be diluted with more recirculated exhaust gas and the inducted air has to be cooled, this is done by opening the exhaust gas recirculation valves (lower right, middle left), and opening (closing) the cooled (non cooled) air-path valve prior to the intake manifold.

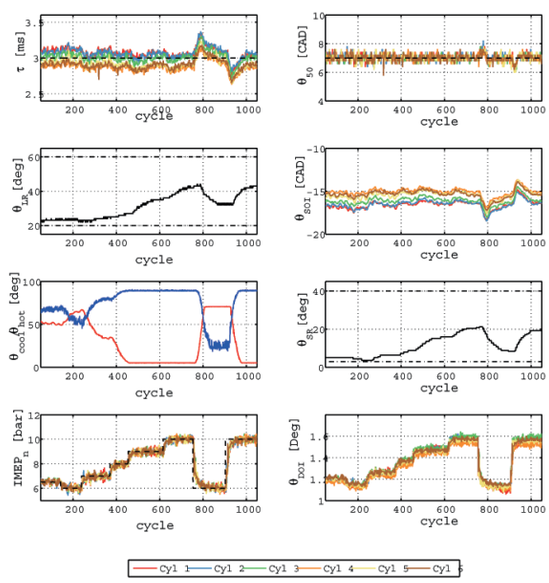


Figure 8. System inputs and outputs during 1000 cycles for which steps in the IMEPn set point are made. In addition to the signals displayed in the previous figures, IMEPn is plotted together with its set point in the lower left figure, the injection durations are presented in the lower right figure.

Two-stage turbocharge experiments to optimize the multi-cylinder engine efficiency further (depends on the procedure).

Model-Based Control of Combustion Timing and Ignition Delay

In low temperature combustion concepts such as partially premixed combustion, the ignition delay, τ, should be long enough in order to ensure sufficient fuel and air mixing before the start of combustion. It is also necessary that the combustion timing θ_{50} is sufficiently well phased for high thermodynamic efficiency. Since the ignition delay and combustion timing are intimately coupled, the decoupling of these two quantities gives rise to an interesting multiple input, multiple output control problem where the control of the air system and the fuel injection system have to be combined. In a multi-cylinder engine this problem becomes underdetermined or uncontrollable with more outputs than inputs. During the first half of 2015, a model-based cycle-to-cycle closed-loop controller of the ignition delay and the combustion phasing was presented and experimentally evaluated.

The controller design was based on the principle of model predictive control (MPC) which is a suitable design for multiple input/output systems with actuator constraints. Ignition delay and combustion phasing were extracted from cooled in-cylinder pressure sensors and controlled by manipulating injection timings, gas mixture temperature and exhaust-gas recirculation (EGR) ratio using a dual EGR-path system and a fast thermal-management (FTM) system. The physics-based ignition-delay model used by the controller was presented in the previous KCFP annual report.

The controller was successful in tracking τ and θ_{50} set points both during stationary conditions and load and speed variations. This can be seen in Figures 7 and 8. It was found that the MPC tuning procedure was a trade-off between speed and sensitivity to model-errors and cycle-to-cycle variation. If the allowed control action was to be increased it was more likely that the system outputs would overshoot during set-point changes, something that is probably caused by insufficient model prediction performance. In this work, the controller was tuned to give slow but reliable performance. The gas-system model used in this work was limited since only the static relation between valve positions and the inlet manifold gas state in a small operating range was included in the model. Incorporating a more detailed physics based gas-system model into this controller framework is considered to be future work.

The MPC framework yielded a simple way of prioritizing system output behavior, it also took system interaction effects into account. Input constraints and the cost of using EGR was also incorporated in the controller. Comparable controller performance could probably be obtained by using decentralized controllers for instance by letting θ_{50} be controlled by the injection timings locally and then let the mean τ be controlled by the gas-system valve positions. This would demand less on-line computations, however, the framework presented here is more general.

The MPC framework could of course also be extended to cover high-level performance measures such as emission levels and efficiency instead of set-point tracking, this would however require an extended engine model with more states. A sufficient ignition delay was here considered to be a marker for a low temperature combustion mode with favorable emission properties. In future work this controller design will be evaluated with emission measurements to conclude that this hypothesis holds or if supplementary control actions need to be taken in order to fulfill emission constraints.

More about the controller can be found in:
Simultaneous Control of Combustion Timing and Ignition Delay in Multi-Cylinder Partially Premixed Combustion. Ingesson, Gabriel; Yin, Lianhao; Johansson, Rolf and Tunestål, Per (2015) In SAE International Journal of Engines 8(5).

Double Fuel-Injection Control

It was previously discovered in KCFP research that the long ignition delays in single injection PPC give rise to very high pressure-rise rates due to violent HCCI-like combustion rates. High pressure-rise rates is an indicator for high audible-noise levels and could also cause mechanical engine damage, therefore, the pressure-rise rate has to be kept below certain levels in order to ensure silent and safe operation.

A remedy to the problem of high pressure-rise rates in PPC is to introduce a pilot fuel injection, e.g. by having an early pilot injection with less than half of the fuel and a main injection containing the majority of the fuel amount. The pilot injection sets a lean mixture environment that decreases the ignition delay of the main injection which decreases the combustion rates. This technique is also used in conventional diesel engines both to improve low-load performance and to decrease emissions and engine noise levels.

With the increased amount of fuel-injection events, the amount of calibration work for optimized engine performance for different loads and speeds grows exponentially. Therefore, it is of course very appealing to find fuel-injection controllers that automatically find fuel-injection timings and fuel distribution among the multiple injections. During the latter half of 2015 experimental PPC work in order to find engine output characteristics w.r.t. double-injection parameters was carried out. The experimental results were then used for design of a model predictive controller that was able to track combustion phasing while guaranteeing an upper bound on the mean maximum pressure-rise rate.

The findings in the experimental results can be summarized accordingly:

- The ratio of the pilot fuel injection duration to the total fuel injection duration can be used to control the pressure rise rate (see Figure 9), the controllability decreases with separation between the pilot and main fuel injection timing.
- The net indicated efficiency increases slightly with pilot fuel injection duration for a given load and combustion timing, the opposite holds with very early pilot injection timings, this is probably linked to the observed increased HC-emission levels.
- The pilot ratio is a trade-off between NOx and soot emission levels.
- The combustion timing is controlled by the main fuel-injection timing.

A model predictive controller was then designed based on the previously mentioned findings while a Kalman filter was used to attenuate noise levels on the computed pressure-rise rate and combustion timing. Experimental evaluation of the controller indicated that it was successful in maintaining an upper bound for the pressure rise rate using the pilot ratio while keeping the combustion phasing at a predefined value, both in steady state and during load and speed transients, see Figure 10.

More about the controller can be found in:
A Double-Injection Control Strategy For Partially Premixed Combustion. Ingesson, G., Yin, L., Johansson, R., and Tunestål, P. (2016). submitted to the IFAC Symposium on Advances in Automotive Control, June 20-23, 2016, Kolmården, Sweden

On December 2nd, Gabriel also defended his Licenciate Thesis. *Model-Based Control of Gasoline Partially Premixed Combustion*. Ingesson, Gabriel (2015)

Which can be found at:
https://lup.lub.lu.se/search/publication/8301285

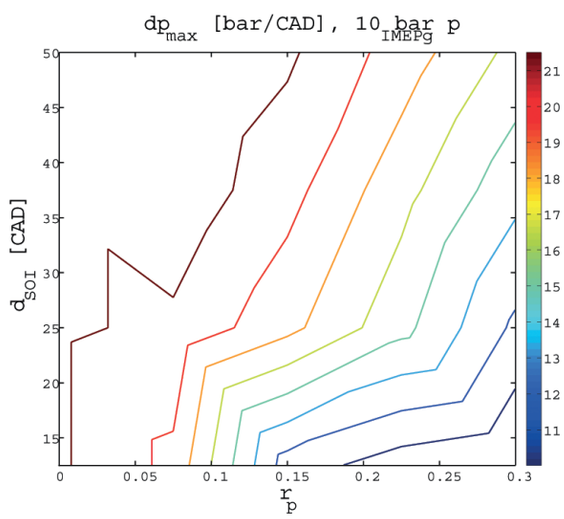


Figure 9. Maximum pressure-rise rate, dpmax, contour plot versus the distance between main and pilot injection *d*SOI and the pilot ratio *rp* at 10 bar IMEPg. It is clear that *rp* can be used to control *dp*max since dpmax decreases with *rp*. The *dp*max controllability is also shown to be higher for small *d*SOI.

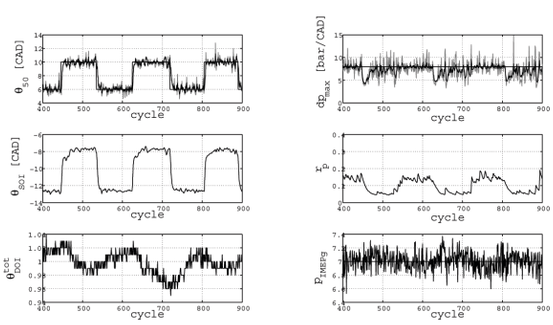


Figure 10. Input and output data during a sequence of combustion-timing set-point changes. In the upper diagrams the filtered combustion timing and pressure-rise rate is presented in black together with the corresponding raw signals which is presented in grey. As combustion timing is advanced, the pilot ratio (rp) is forced to increase in order to fulfil the specified Pressure-rise rate constraint of 8 bar/CAD, when the combustion timing is retarded, the pilot ratio is decreased due to its absolute-value cost in order to maximize the ignition delay. During the experiment, load was controlled with the total fuel injection duration which can be seen in the lower part of the figure.

The Generic Diesel Project

The diesel engine is a favored power source for road transport due to its high efficiency, but its emissions of nitric oxides and particulates remain a challenge. The Generic Diesel (GenDies) project focuses on studies of the in-cylinder processes controlling the soot emissions.

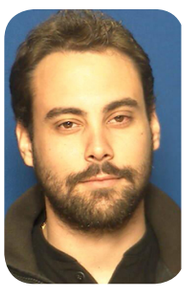
Soot emissions are the net result of formation and oxidation of soot in the cylinder. Since the oxidation is efficient in modern engines, only a small fraction of the soot is emitted. Several GenDies studies have identified in-cylinder soot oxidation as the dominant factor explaining the trends in soot emissions.

A laser extinction measurement setup was developed to identify the factors that have the greatest influence on the soot oxidation rate. The technique is relatively simple but provides semi-quantitative soot concentrations with high temporal resolution, making it possible to measure the soot oxidation rate in situ. It has previously been used to study the influence of EGR on the soot oxidation rate and has now been used to study the effects of several other variables. These results will provide a baseline for future experiments with more advanced laser diagnostics.

One of the limiting factors for the soot oxidation rate is the availability of oxidizers. An experiment was performed where soot was imaged simultaneously with the important oxidizer OH, to investigate how variations in the in-cylinder flow affected the area of the interface between these two species. The results are summarized below.

A new light-duty optical diesel engine has been designed, built and installed in the GenDies laboratory. It will primarily be used for a new activity focusing on multiple injection strategies.

In-cylinder soot oxidation measured using laser extinction



Yann Gallo
PhD Student

After the previous extinction campaign, a slightly improved setup was used to study how a wider set of parameters affect the late-cycle oxidation of soot. The improvements mainly involve a more powerful laser source and a narrower filter in order to improve the signal-to-noise ratio and to reduce the risk of total extinction of the laser at highly sooting conditions. The laser beam is introduced vertically into the cylinder as shown in Figure 1, allowing measurement throughout the whole expansion stroke.

Zheming Li
PhD Student

Six parameters were included in the experiment. The first, the in-cylinder gas density, was chosen since it should affect the frequency with which a soot aggregate encounters oxidizing molecules during the expansion stroke. The in-cylinder temperature was included as it probably controls the kinetics of the oxidation process. The remaining parameters were chosen due to their assumed effect on the in-cylinder flow and mixing rate during the late cycle; the injection pressure, the engine speed and swirl ratio should have direct effects on both, and the nozzle hole diameter is assumed to affect the fluid motion through its effect on the spray momentum.

Figure 2 shows an example of how the soot concentration evolves in the cylinder during the cycle. The soot concentration is directly related to Kext L and is clearly affected by a change in injection pressure. At higher injection pressures the amount of soot formed is reduced, probably due to increased air entrainment into the burning jets, but also due to reduced residence times in the soot-forming regions of the jets. The soot oxidation rate can be determined from the decay rate of the curve after the end of injection, which also seems to be affected by the injection pressure. A detailed analysis of the complete dataset will be carried out during the spring of 2016.

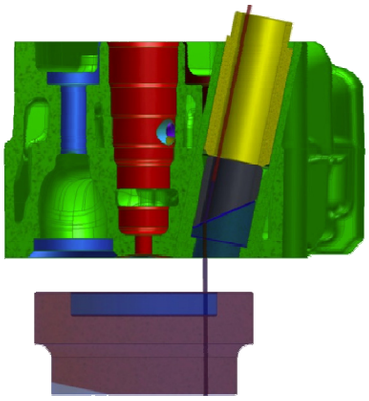


Figure 1. Cross section of the cylinder head and optical piston. The red line represents the laser beam, which is introduced through a slanted window in the cylinder head and guided vertically through the cylinder.

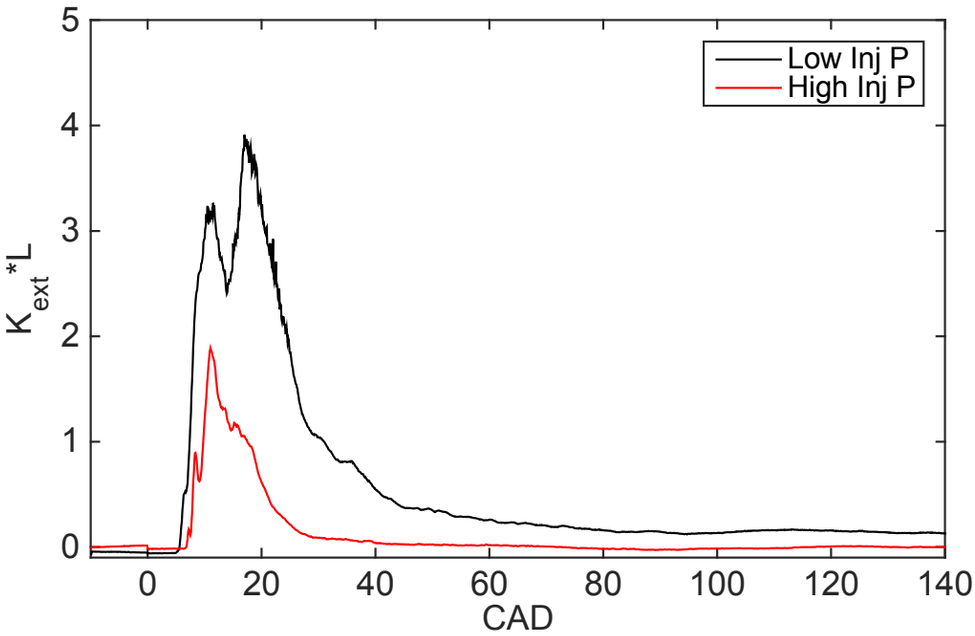


Figure 2. Extinction signal from TDC to exhaust valve opening (140 CAD ATDC). The injection pressure affects both the amount of soot formed and the late-cycle oxidation rate.

Soot oxidation studied by simultaneous OH-LIF and LII imaging

OH has been identified as the dominant oxidizer of soot under diesel conditions. PLIF imaging of OH and LII imaging of soot was carried out simultaneously in an optical diesel engine. The images were acquired after the end of injection in the recirculation zone between two adjacent diesel jets. As seen in Figure 3, the signal is first dominated by soot (red) and later by the soot-oxidizer OH (green). This means that the oxidation zone is moving through the laser sheet, and the analysis should be made around 10-12 CAD, when both soot and oxidizer are present in the imaged region.

Ted Lind
PhD Student

Zheming Li
PhD Student

Scalars such as the proximity between OH and soot, the total amount of OH and soot and also the wrinkledness of the soot and OH regions were extracted from the images and compared with trends in engine-out soot emissions. The OH and soot were considered in proximity of each other if they were separated by less than 10 pixels, corresponding to ~0.8 mm. The wrinkledness was computed according to

$$\boldsymbol{Wrinkledness} = \frac{\boldsymbol{Perimeter\ of\ the\ cloud}}{\sqrt{\boldsymbol{Area\ of\ the\ cloud}}}$$

It was found that the total amount of OH in the images is negatively correlated with the soot emissions, as is the spatial proximity between the OH and soot regions. This indicates that OH is an important soot oxidizer and that it needs to be located close to the soot to perform this function.

The total amount of soot in the images shows no apparent correlation with the soot emissions, indicating that the amount of soot formed is a poor predictor of the emission trends. There was also a lack of correlation between the wrinkledness and the soot emissions. The lack of a correlation between the wrinkledness of the soot cloud and the emissions of soot could be interpreted as a lack of importance of O2 as a soot oxidizer under these conditions. On the other hand, these 2D measurements of 3D structures may not capture important aspects of the flame topology. Whether the shape of the clouds affects the soot oxidation rate or not requires further study.

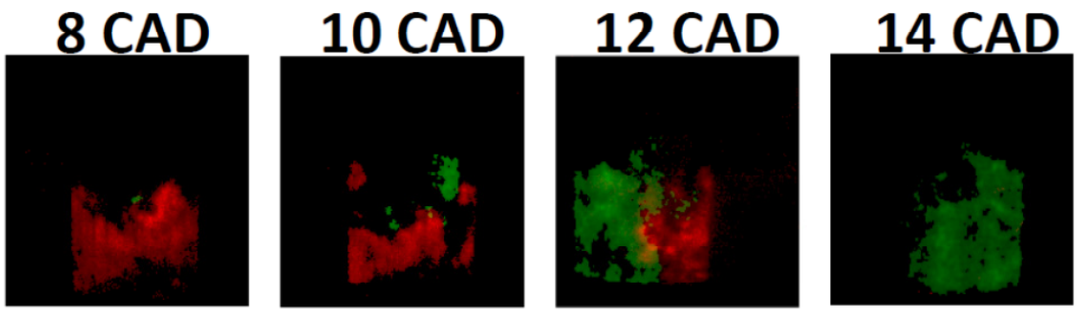


Figure 3. The OH-LIF signal (green) shows the location of the oxidizer in relation to the soot (red). Between 10 and 12 CAD ATDC both soot and oxidizer are present in the imaged region.

Mechanisms of pilot and post injection strategies in a LD, optical diesel engine

Michael Denny
PhD Student

This new activity in the GenDies project aims at understanding the details of how multiple injection strategies affect the combustion process. Pilot injections are implemented primarily to control combustion noise and NOx emissions. The heat released by the combustion of the pilot injection increases cylinder pressure and temperature, which reduces the amount of pre-mixed combustion of the subsequent main injection by reducing the ignition delay. The tradeoff is generally increased soot emissions. When compared to a single injection strategy phased to minimize NOx emissions, there is often a fuel efficiency benefit when adding a pilot injection. In doing so, the main injection event and therefore 50% burn point, can be phased earlier in the cycle.

Due to current customer expectations on perceived noise and tightening emissions regulations, pilot injections have become commonplace, and their comparison to a single injection strategy becomes less relevant at this stage of maturity of diesel engines, even considering the improvements in fuel injection equipment. Instead, the theme of this project is to compare multiple injection strategies to each other.

Fuel injection equipment technology has advanced over the years by increasing operating pressures, shortening dwell times between injections, and reducing the minimum injected mass per injection event. This has led to a change in injection strategies to achieve the desired engine-out emissions and noise levels. While these results can be achieved through a DoE of calibration variables, the mechanisms governing these new trends in injection strategies are still unclear. Discovering and explaining them are the foci of this project.

A new light duty optical engine based on the latest production engine geometry from Volvo Cars is currently under installation, replacing the older I5D Volvo Cars engine. The first experiment will study 3 different multiple pilot strategies. In addition to common engine diagnostics such as in-cylinder pressure sensing, high-speed video of the injection and combustion event will be recorded. The recorded events will be used to determine the combustion flame type and distribution both spatially and temporally, via the use of several optical filters. The results from this first experiment will help guide the project's direction in the future.

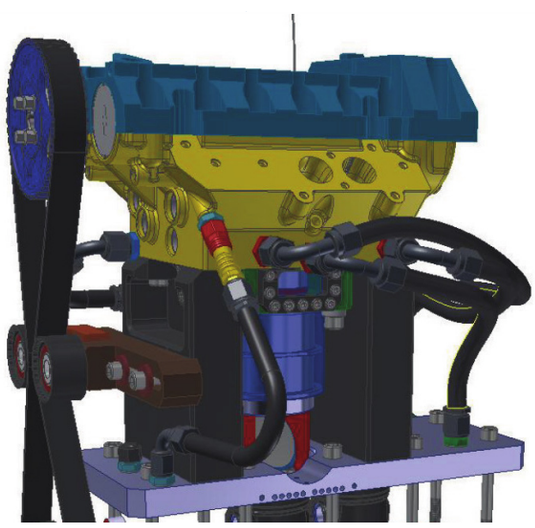


Figure 4. 3D model of the optical topworks of the new GenDies optical light-duty diesel engine.

The Gas Engine Project

Introduction

Ashish Shah
PhD Student

The Gas Engine Project 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. 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 in heavy duty natural gas engines.

Two most feasible alternative ignition systems were identified, namely diesel pilot injection and pre-chamber ignition system but it was soon realized that the former has already received considerable attention and has been commercialized 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 ignition device and hence gaining deeper insight into this mechanism is the objective of the current phase of the gas engine project.

Background

Earlier in this phase, experiments were performed using multi and single cylinder engine setups to compare the performance of conventional and pre-chamber spark plugs. It was observed that simply replacing a spark plug with a pre-chamber spark plug (i.e. with no additional pre-chamber fueling), results in improved combustion stability but without any considerable extension in lean or dilution limit with EGR. For some operating regimes, the NOx emissions were higher with pre-chamber spark plugs. Better combustion stability with pre-chamber spark plug is expected, since charge motion inside the pre-chamber is not significantly affected by cyclic variation of main chamber charge motion and also because the ignition resulting from pre-chamber system is spatially distributed as opposed to a single point spark. The inability of an unfueled pre-chamber spark plug in considerably extending lean and dilution limit with EGR is due to over-leaning of charge inside the pre-chamber due to the lack of proper scavenging at the beginning of every cycle.

To address these issues and based on a literature study of pre-chamber ignition systems, further studies were conducted with fueled pre-chambers. A pre-chamber assembly capable of fuel injection, spark ignition and pressure measurement was designed and manufactured for this purpose, details of which can be found in previous annual report. A particular operating strategy, called Avalanche Activated Combustion (originally 'Lavinia Aktyvatsia Gorenia' in Russian), commonly referred to as LAG-ignition process, was chosen for further studies as it had documented benefits but its application to heavy duty natural gas engines is less explored. Experiments were performed at the load of approximately 10 bar gross IMEP and at pre-chamber geometrical settings as recommended for the LAG-process and it was found that fuel rich combustion in the pre-chamber (excess air ratio, $\lambda < 0.5$), considerably extends the lean limit of operation with the main chamber excess air ratio of up to $\lambda = 2.9$. Peak indicated efficiency of 47.6% was recorded for main chamber λ of 2.4 at pre-chamber λ of 0.2.

Following these findings, experimental studies on the effect of pre-chamber volume and nozzle diameter were conducted wherein both the parameters were varied around the recommended settings used for previous studies. It was observed that the pre-chamber volume fraction of 2.4% is an optimum tradeoff between ignition characteristics and NOx emissions, whereas the setting of nozzle diameter depends of the relative importance of lean limit of combustion and effectiveness of ignition at a reduced dilution level.

Recent Progress

Following the above findings, a CFD study was conducted to further understand the fluid dynamic aspects of mixing between the pre-chamber jets and the main chamber charge, which was is an important factor controlling ignition in the main chamber. The simulations were transient 3D RANS type for the time duration between start of jet ejection from the pre-chamber and 10% heat release in the main chamber (main chamber ignition). Experimentally measured pre- and main chamber pressure was used as transient boundary conditions. Figure 1 shows the computational domain.

One of the main results obtained from this simulation study was the jet penetration profile for various pre-chamber volume and nozzle diameter cases. Fig. 2 shows the average jet penetration velocity for all the cases and it can be seen that the penetration velocity increases with increase in pre-chamber volume and at a given pre-chamber volume, a smaller nozzle diameter causes higher velocity jets. This can be directly correlated to the experimental observed flame development angle behavior for these pre-chamber settings. Fig. 3 shows the turbulence intensity in the main chamber for different pre-chamber volume cases, but at equal time duration after the start of ejection. It can be seen that a jet from a larger pre-chamber engulfs greater fraction of main chamber charge, which burns close to stoichiometric conditions when the main chamber ignites. This explains the experimentally observed trend of increasing NOx emission with increase in pre-chamber volume. Further data analysis related to simulation activity can be found in the publication – SAE Technical Paper 2015-01-1890.

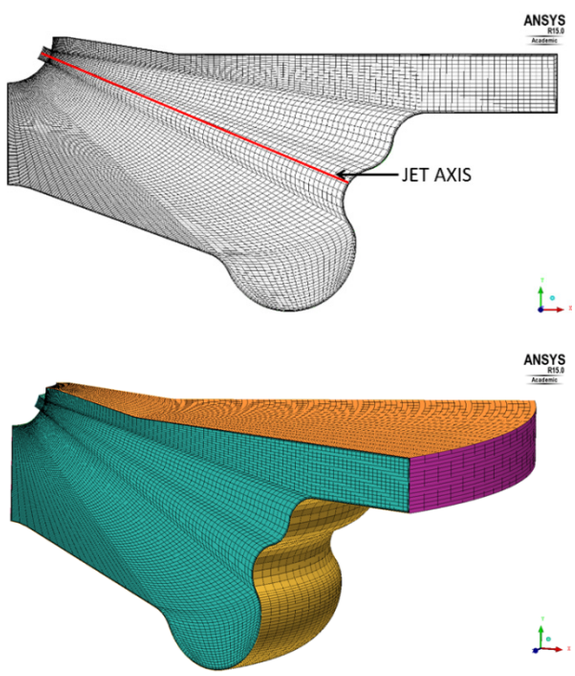


Figure 1. The computational domain for CFD simulations

Following studies conducted so far, in which the benefits of fuel rich combustion in the pre-chamber were highlighted and also the optimal settings for pre-chamber volume and nozzle diameter were found, the dependency of these optimal settings on engine size was unclear, since all the experiments with fuel-rich pre-chamber combustion strategy were conducted on a truck size engine. It was therefore of interest to understand the scaling requirements of pre-chamber ignition. Further experiments were therefore conducted on a medium speed large bore (Bore = 200 mm) engine from Wärtsilä, to evaluate if the optimality of pre-chamber volume previously found is due to its absolute volume or its volume relative to the engine's displacement volume.

The most important experimental result is presented in Figure 4, which is the comparison of normalized (for in-cylinder turbulence) flame development angle (in absolute time scale) for experiments in the truck size engine (Scania) and large bore marine engine (Wärtsilä), expressed on absolute and relative pre-chamber volume basis. From the first figure, where the data is expressed on absolute

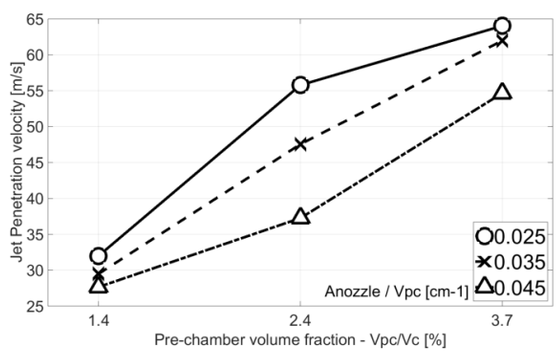


Figure 2. Effect of pre-chamber volume and nozzle diameter on the jet penetration velocity

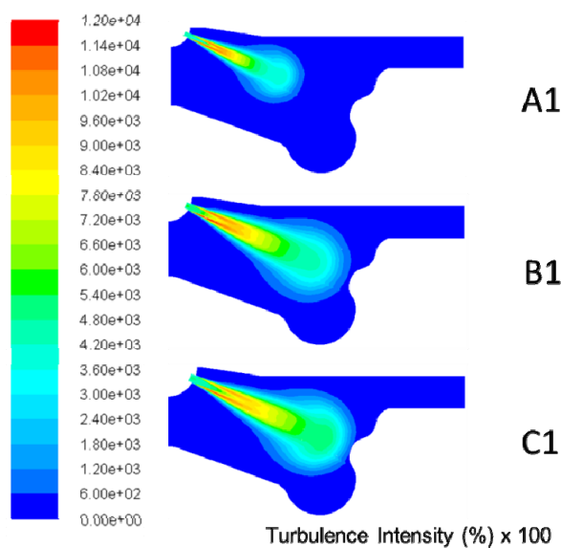


Figure 3. The effect of pre-chamber volume on the volume of jet plume

volume basis, it can be seen that a pre-chamber of a given volume behaves very differently, with respect to main chamber ignition characteristics, in two different engines. In other words, the optimal pre-chamber volume is strongly affected the displacement volume of the engine. The same data, when replotted on relative pre-chamber volume basis, shows a trend of reducing flame development angle duration with increase in the pre-chamber volume. It can therefore be concluded that optimal volume of the pre-chamber is 2.4% of the compressions volume of the engine, and hence it scales with engine size. Further results from these experiments can be found in the student (Ashish Shah)'s doctoral thesis.

Future work

The student (Ashish Shah) successfully defended his doctoral thesis on 16th December, 2015 and hence this phase of the Gas Engine project, which focused on pre-chamber ignition system, came to an end. Overall, it was shown that pre-chamber ignition system operating with fuel rich pre-chamber combustion strategy is a simple and very effective alternative ignition strategy for heavy duty gas engines. Recommendations for optimal settings for geometrical and operating parameters of the pre-chamber ignition system were also made. The student's suggestions for future research on pre-chamber ignition system can be found in his doctoral thesis, and information on the next phase of the Gas Engine project can be obtained from KCFP.

Ashish Shah's doctoral thesis is available at
http://lup.lub.lu.se/record/8228129

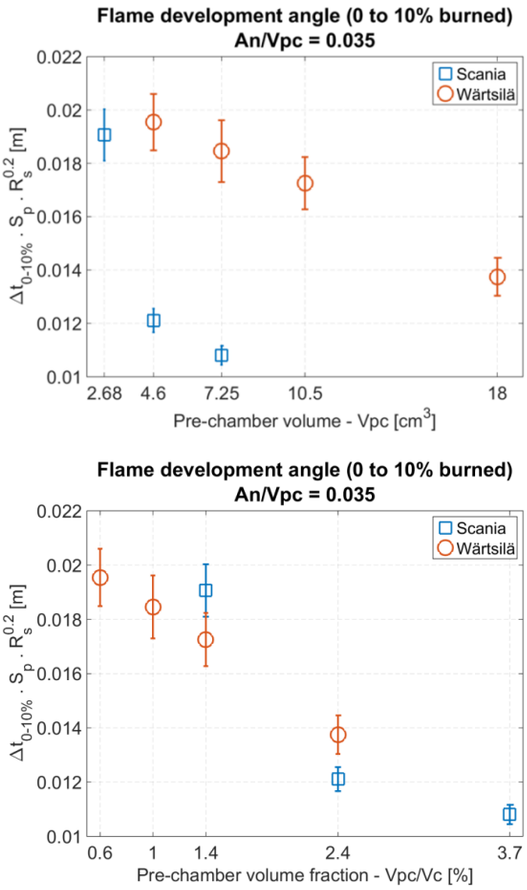


Figure 4. Comparison of flame development angle duration in the Scania and the Wärtsilä engine, expressed on absolute (above) and relative (below) pre-chamber volume basis

KCFP PhD Students who Graduated in 2015

Guillaume Lequien graduated within the ELOF project and defended his PhD thesis, "Investigations of Diesel Sprays in Optical Engines", at the Department of Energy Sciences on May 29, 2015.

Ashish Shah graduated within the Gas Engine project and defended his PhD thesis, "Improving the Efficiency of Gas Engines using Pre-Chamber Ignition", at the Department of Energy Sciences on December 16, 2015.

20 years with KCFP

By Bengt Johansson,
Ph.D. student, assistant, associate and full professor within KCFP and director of the center 2003-2015

My time at Lund University and hence within KCFP has come to an end. Hence it can be the right time to look back and reflect on the past. The KCFP was formed July 1 1995 and was the first competence center which included research on combustion in internal combustion engines in Sweden. It was followed by the CERC at Chalmers only half a year later and then CICERO/CCGEx at KTH in 2006. The more combustion oriented CeCOST was started a few years after KCFP in Lund as well. I have had the privilege to work within KCFP as a Ph.D. student (kind of), research assistant, assistant professor, associate professor and finally full professor. I have since the start been a project leader for a number of projects within KCFP on various aspects of combustion in engines and since 2003 also been the director of the center. All this means that I have become an integral part of KCFP and perhaps KCFP has been an integral part of me as well. So what has been done and what became of it all? I will try to give my personal reflections on the 20 years and tell some of the interesting stories I remember.

KCFP has been conducted in seven different phases with a project plan and budget for each phase. Hence it can be argued that KCFP has not been one long activity but rather seven research grants stacked after each other. Members of the center have joined and dropped out during the years depending on the research topics investigated and the quality of the research. The seven (or actually 8) phases are listed below.

Phase 1: 1995-1997 (1/7 1995- 30/6 1997, two years)
Phase 2: 1997-2000 (1/7 1997- 30/6 2000, three years)
Phase 3: 2000-2003 (1/7 2000- 30/6 2003, three years)
Phase 4: 2003-2005 (1/7 2003 – 30/6 2005, two years)
Phase 4.5: 2005 (1/7 2005- 31/12 2005, half a year)
Phase 5: 2006-2009 (1/1 2006-31/12 2009, four years)
Phase 6: 2010-2013 (1/1 2010-31/12 2013, four years)
Phase 7: 2014-2017 (1/1 2014- 31/12 2017, four years)

I will try to remember some of the highlights within the phases. This is not a complete list of all activities nor a final report of KCFP, just my personal reflections. I will name a few of the Ph.D. students I worked with but not all. Many of them worked mostly outside KCFP and some I did not supervise that much.

Phase 1: 1995-1997 (1/7 1995- 30/6 1997, two years), 18 MSEK total budget

At the start of KCFP in July 1995 my predecessor as head of the combustion engine group, prof. Gunnar Lundholm, was appointed the director of the center. At the time KCFP should have a number of research topics, not only engines but also gas turbines, furnaces and fundamental combustion. I became responsible for the two projects concerning combustion engines that could be fitted into the budget. It was on two-stroke engine and Otto-Atkinson engine operation.

The two stroke engine project was very interesting. It was formed with Husqvarna and Berema (Atlas Copco) two Swedish companies making handheld two-stroke engines. The major concern was to understand the in-cylinder flow and through that understand why a less expensive design on the transfer channels from crankcase to cylinder gave around 30% more hydrocarbons. Since I was about to graduate on velocity measurements in the cylinder of a truck engine, I had very fresh experience on setting up an LDV system and we started to build an optical Husqvarna engine. The Ph.D. student, Martin Ekenberg was very enthusiastic and we soon got an engine running. It took some failed experiments with sinking a chain saw engine block intro concrete to make it robust and we broke a number of optical engines; good thing they were rather small and hence cheap. Eventually we managed to map the entire flow field in the cylinder with single point LDV and also some imaging of the flow with mie-scatting from above. To get information on the exhaust emissions the fuel concentration was mapped with laser induced fluorescence just outside the exhaust port. This was done together with the Combustion Physics division and a student, Öivind Andersson, whom I eventually recruited to the combustion engine group. He is now full professor here. The two-stroke project was very interesting and I could implement the techniques learnt during my own Ph.D. 1989-1995. The knowledge was very fresh as I actually became project leader and supervisor half a year before I graduated myself. The frustrating part was that the industry members had a hard time translating the knowledge gained directly to a better product. The project was thus only for the first phase of KCFP, two years, but it resulted in a number of papers, one of which was selected for an SAE special publication on two-stroke engines. Ironically, the paper selected is the one with least citations of all the ones generated. Martin Ekenberg continued to a project with Saab on direct injected gasoline and built the first optical engine in Lund with a piston extender in that project. He graduated in 2002 and is since a patent attorney. I think his logic selecting that trade was that working is no fun anyway and then he could just as well make money doing it.

The Otto-Atkinson project was also started at the start of the KCFP in July 1995 with Fredrik Söderberg as Ph.D. student. The basic idea here was to investigate the fluid flow in a four stroke SI engine for Volvo Cars and study the effects of valve events. The idea was to run early or late

inlet valve timing and detect how that would affect the bulk flow, the turbulence generated and how that in turn affects the combustion rate. The engine was an almost standard Volvo 850 engine but with a special single cylinder head that had windows in the sides of the pentroof combustion chamber. The 10 mm diameter windows enabled optical access with LDV along the centerline through the combustion chamber. The idea was to use special design camshafts to get the valve lifts needed and those camshafts were to be delivered as in-kind contribution from Volvo Cars. The problem was the delivery time of the special camshafts, they never came. We had to do something while waiting so it was decided to run the standard camshafts but with late phasing of the inlet valve. This gave both late inlet valve closing as intended but also late inlet valve opening and hence a significant amount of negative valve overlap, something later used in HCCI to control combustion. On top of that we deactivated one of the inlet valves to generate swirl. With the velocity measured along the line from window to window we could easily see the transition from tumble type flow to swirl type and measured the turbulence and how homogeneous it was. With very late phasing of the inlet valve and using only one valve we could even operate with faster combustion at lambda=1.8 than the original setting gave with lambda=1. This was in fact a result on the same level as what we got with special built camshafts made by ourselves once we got tired of waiting for Volvo to deliver them. I also got an idea of throttle-less operation of an SI engine by rearranging the inlet ports within this project. The idea was to put one inlet and one exhaust valve on each camshaft. This means that the effective duration of both inlet and exhaust valves are extended if the two camshafts are phased different. Hence the engine load can be controlled with late inlet valve closing but without negative valve overlap. A cylinder head according to the principle was designed and built by Volvo and we tested it with respect to both engine performance and in-cylinder flow measurements, this time with the PIV-technique. It was discussed to file a patent of the x-head as it was called but it never happened.

The fluid flow results within the Otto-Atkinson project was analyzed with the wavelet technique. This was the first time this technique was used for LDV data. What could be extracted was the frequency content in the signal as a function of time. I really did not understand wavelets myself but took help by Jan Holst at Mathematical Statistics and a few of his students. With this analysis it was possible to understand the breakdown of large scale (slow) eddies to fine scale (fast) turbulence and the difference between swirl and tumble in that aspect. It can be noted that this project was not always behind schedule. In fact, during the fall of 1996 we finished a measurement campaign six weeks before the deadline. This has not happened since, in any project. Fredrik Söderberg graduated in 2002 and is now working as a patent attorney as well.

Phase 2: 1997-2000 (1/7 1997- 30/6 2000, three years)

When it was time to start phase 2 of the KCFP in 1997 it was not possible to get the interest and hence funding from Husqvarna and Berema to continue the two-stroke activities. On the other hand the budget increased from 3x2 MSEK the first year and 3x4 MSEK the second year in phase 1 to 3x6 MSEK per year in phase 2. The 3x means that the energy agency, Lund University and industry all put in equal amounts, in this case 6 MSEK/year. With larger budget it was possible to start two new projects on engines and I was the project leader of both also in phase 2.

The first was called "gas engine". It was a project that continued activities started in 1993 during my Ph.D. period. My thesis was on cycle to cycle variations due to variations in the early flame development in SI engines. In order to study them an optical engine was built. But even if it was not part of my thesis I was curious how much the fluid flow would change with a change in the combustion chamber shape. Hence I initiated a measurement campaign with a range of different combustion chambers. Since it was a modified diesel engine the combustion chamber was located within a bowl in the piston. Hence by simply machining different bowls in piston blanks the flow could be altered a lot. Again the LDV technique was used, this time through a small window replacing the spark plug near the center of the combustion chamber. The flow was changed to the extent that the combustion duration was reduced to half in some cases. The links between fluid flow, turbulence, combustion and emission formation were published in two SAE papers. The first focused on the link between fluid flow and combustion and the second was on combustion and emission formation. I did most of the work with LDV and in-cylinder pressure analysis but Krister Olsson did the emissions part. He has been working within the combustion engine group since the 1970:s and was my teacher in IC engines before I joined as a Ph.D. student at the restart of the group in 1989. Before that he was pretty much the group. He was a very reluctant engine researcher though and during the emission testing he often repeated that running an engine in the lab was as fun as watching wet paint dry on a wall. Well, he did get a Licentiate Degree from the the results and I even managed to get him to Detroit for an SAE paper presentation. At the party after his defense I of course gave him a can of white paint.

Back to the gas engine project. This was a continuation of the earlier work with combustion chambers, now with a set of six new combustion chambers. The work was first done in the single cylinder Volvo TD100 which was used during my Ph.D. but then moved to a completely new multi-cylinder engine in the basement. Patrik Einewall started as a Ph.D. student on the project 1997 and graduated in 2003 after a number of experimental

campaigns in both the single and full engine. We quickly learned that it is not simpler to operate six cylinders than one. All of a sudden you do not get one answer to your question, you get six and they are all different. Balancing cylinders became much of his task. The gas engine project took a break after Patrik got is Ph.D. in 2003 and returned in 2006 when it was time for the reformulation of the KCFP to engine related combustion studies only. This time the student, Mehrzad Kaiadi, was supervised by Per Tunestål and the project was more focused on engine controls. Only two combustion chambers were tested and my participation in the project was more limited. Ashish Shah continued after Mehrzad and was my last graduate in December 2015. He worked mainly on prechamber combustion.

The second new project 1997 was called ATAC-engine. The term stands for Active Thermo-Atmosphere Combustion and was introduced by Onishi in 1979. The combustion concept is perhaps better know as HCCI (Homogeneous Charge Compression Ignition) today. The HCCI all started with a question during my thesis defense in 1995. The question was the following: "You have now spent five years studying the cycle to cycle variations in SI engines in some detail. But how should you handle the problem and get rid of them?" My answer was just a silent "I don't know, haven't thought about that". This was of course very annoying and I spent the spring 1996 thinking about that question. How can we take away the cycle to cycle variations in gasoline engines? At the SAE meeting in San Antonio September 1996 I got the answer. I was there to present my paper on the summary of my thesis, but there I also did what I normally do at SAE meetings, listen to as many other paper presentations as possible. One that caught my attention was a literature study on HCCI performed by a Canadian research group (SAE paper 962063). The title was HCCI using methanol, gasoline and diesel fuel. There was the answer! When switching from SI to HCCI combustion the cycle to cycle variations are gone. And no one cared! There were less than ten persons in the room and no one understood the greatness of what was presented. I had to rush home and get started on HCCI before the rest of the world understood.

Directly when coming home from the SAE meeting I started preparing for the first HCCI experiments in Lund and also started thinking about how to get the recourses to do research on it. The first was rather simple, I asked our excellent technician, Bertil Andersson, to remake the TD100 used for my Ph.D. for HCCI. From the review paper I learned that you need high compression ratio and high inlet temperature. We had some piston blanks from the gas engine work and since we thought that combustion chamber shape was not important, Bertil just cut it with the aim of 21:1 compression ratio. Well, it turned out to be 19:1 in reality but high enough anyway. For inlet temperature we took the simplest path possible. A hot air gun was installed aiming at the inlet of the engine. A hot air gun is normally used to blow hot air on a surface that needs to be cleaned from old paint or something similar or perhaps for melting glue. It normally has three levels; I, II and III depending on the amount of hot air needed. For the first HCCI experiments that translated to 40°C inlet temperature (O), 60°C (I), 80°C (II) and 100°C (III). But we needed also 120°C for the experiments with natural gas. The solution to that problem was rather simple though; two hot air guns. Since I was lacking any type of funding for HCCI I had to rely on free labor, called M.Sc. students in the University world. Magnus Christensen volunteered and hence became the first researcher doing HCCI in Lund. He did his master thesis work during the fall 1996 and the spring 1997. He then continued as a Ph.D. student and when done in 2002 he had published some 26 papers on HCCI, still a record. He is now working at Volvo Trucks running single cylinder experiments, not only HCCI though.

OK, so I had an engine, a student for free for some time but would need real funding so I could enroll Magnus as a Ph.D. student and get some real research done. I thus asked the board of the KCFP if I could talk about HCCI and ask for funding at the board meeting in the fall 1996. This was actually before we had any results from the lab. Since I was presenting the results from the two-stroke and the Otto-Atkinson projects I took the liberty to presents my ideas on HCCI during the meeting as well. The board then asked questions like:

- Who else is doing HCCI in the world? I had to answer: not many.
- Why is HCCI not already in production if it is that good? A: Don't know
- Why study a combustion concept that is so sensitive to external conditions? A: Huge benefits.
- The extremely low NOx sounds unreasonable. Is the literature information really correct, most likely not? A: Don't know, we have not done the experiments ourselves.

I could not really answer any questions and it all ended with me not getting any priority at all in the discussions on what to spend money on in the next phase of KCFP, 1997-2000.

So no one really believed in the HCCI literature. I needed real engine data from the lab and once I had that it was time for the next board meeting. I then timed my presentation of results of ongoing projects such that I was done just before the coffee break. Before the break, I asked if I could present some results of HCCI fresh from the engine lab, if nothing else as entertainment to the coffee. The board said a reluctant yes to that. I then presented the results that later became the first HCCI paper from Lund. The NOx was 1000 times less than for the same engine run in SI mode and the indicated efficiency was above 50% in some of the operating points. Those operating points also happened to be where gasoline

cars operated most of the time. With the results at hand the board realized that I was on to something and said they would consider HCCI provided that I would get the required industry sponsoring. After visits at Volvo, Scania and Volvo Cars they agreed to put HCCI on the priority list for KCFP. I also talked to the Wärtsilä office in Trollhättan and convinced them to join the center for phase 2. Well, it also took a second trip to Wärtsilä, now to Vasa, Finland. It was a very surreal experience there; I was called in to an executive meeting room with a very large table in the middle. On one side of the table were some 6-8 middle aged men in formal suits. I was asked to sit alone on the other side. The chair of the meeting introduced them all and then asked me only two questions. Who are you and what do you want? I then introduced myself as a rather fresh researcher with a fresh Ph.D. in a topic not related to the research I wanted to talk about and I was coming from a university that had very limited prior history in engine research. What I wanted was to present a new and novel combustion concept that would reduce NOx and soot and improve efficiency a lot. By that time when I opened my deck of OH transparencies I saw that they all started to look at their watches and lost interest. Thankfully this was before the introduction of smartphones so I did not loose them totally. When I was done with the presentation, I just expected a polite "thank you, don't call us, we call you" but that was not what happened. The chairman just said "how much" and I gave him the number for Wärtsilä joining KCFP. Then the discussion was ended with a short "OK".

With a new member of KCFP and the current Volvo, Volvo Cars and Scania interested in HCCI I got funding for a project in phase 2, 1997-2000. It was not as simple as just starting a project though. The KCFP at the time had a number of activities. In order for the combustion engine group to get a student on HCCI the combustion physics group needed 1.25. One student should focus on chemical kinetics and the 0.25 student on optical diagnostics.

Our first paper on HCCI was published at the SAE fall meeting 1997 and presented there by Magnus Christensen. The engine was operated on isooctane, ethanol and natural gas. Those three fuels were not in the literature study of the year before and hence we generated a "world's first" paper. The interest in the paper was very large.

In all essence the SAE paper was the content of Magnus Christensen's master thesis. By the time of the master presentation we had generated a second and a third paper on HCCI as well and even upgraded the test cell with a real air heater. Magnus' master thesis can perhaps be worthy of a side story. We realized by the summer 1997 that Magnus needed to get his master degree in order to be accepted to the Ph.D. program. Hence a thesis needed to be produced but it should not interfere too much with the important work in the lab with generating experimental data on HCCI. Thus I told Magnus to just take the first SAE paper and make it into thesis format with single column and large figures. So he did and then it was soon time for the opposition. What neither Magnus nor I were ready for was that the opponent was a Lund student that did his master thesis in a German company. This student spent half a year to study the parasitic losses of different auxiliary systems like belt drives to generators, fuel pumps etc. A totally uninteresting topic in my view but he had put his heart into the report. A masterpiece of more than a hundred pages with perfectly scanned images, a complete and long reference list and a lengthy section on the methods used when cataloging the auxiliaries and their losses. When this student got hold of Magnus' thesis he totally lost it. Magnus' thesis did not have a methods section at all. HCCI was not explained in detail, at least not so the student could understand it, the reference list was short and on top of all the page numbers were not consistent in the report. He kept on for more than half an hour directing his anger to Magnus but since he is not a person you excite that easily (not sure it is even possible) the opponent refocused his anger to me as a supervisor. He then kept on for the better part of an hour stating that he felt ashamed for getting a degree from a university that could pass such substandard work and that possibly I should be banned from any further supervision. I decided to keep my mouth shut completely, thinking that if I would start talking I perhaps would in fact be thrown out from the University based on what I would say. As I did not respond the opponent again refocused, now to Gunnar Lundholm, asking what kind of substandard research group he was running. Again he did not get that much reply, if Magnus was hard to get upset, Gunnar was even more calm. How did it all end? Magnus got his master degree, continued to Licentiate and Ph.D. and has generated papers with more citations than entire research groups doing research for decades. I have not heard of the opponent or any of his work since the defense.

With Magnus employed as Ph.D. student we could focus on some engine experiments. We ran the engine with EGR instead of air (paper 2), added water instead of EGR (paper 3) and also boosted the inlet pressure and thus increased the load range of HCCI from the modest 4-5 bar IMEP to 14 bar (paper 4). This latter study was presented at the SAE world congress 1998. At that time the interest of HCCI had grown substantially. The paper was presented in room W2-69 at Cobo center, Detroit. The room had a capacity of around 80 persons sitting. All those seats were taken and people were standing along all the three walls not used by the screen, in triple rows. Even outside the room people were standing in a cone formation looking through the open door! My guess is that more than double the number of allowed persons were in the room. Magnus presented the results and during the Q&A got the question on the maximum pressure during the cycle. We reported 250 bar in the paper and it was questioned

if the engine could actually handle that pressure level. Magnus just replied "The engine is a Volvo TD100". And by that he thought further discussions were unnecessary. The current chairman of KCFP, Sören Udd, was sitting in the audience. He later told me that he got a comment from his boss at Volvo also sitting in the room; "I always told you that the TD100 engine was over specified, now we know." What we did not know at the time was that the engine was perhaps not as strong as we thought. A later check showed that the connecting rod was a bit shorter at the end of that measurement campaign. Hence the effective compression ratio was much less in the end. That also eventually explained the quite large variation in results from our first paper on HCCI 1997 and the one with boosted conditions 1998. We were careless enough to repeat the same experiments in both papers but presented up to 50°C higher inlet temperature needed in the 1998 paper for the same case. We did not think too much about that at the time but some of the industry sponsors did. I had to spend a lot of time trying to convince Cummins and Caterpillar that Lund could do good measurements and at the same time blame 50°C on day to day variations in the lab.

With the start of HCCI (or ATAC as it was called at the time) within KCFP we also started the chemical kinetic modelling. As a coincidence Fabian Mauss was hired as professor at that time within the division of combustion physics. He soon got a student, Per Amneus, that started to use Magnus' engine data as a base for chemistry simulations. That collaboration quickly generated results and the first simulations were added to the boosted paper 1998. Many more simulations resulted with much of Magnus' database used for validation and calibration of the models. His data was used by Lawrence Livermore National Labs and University of Wisconsin-Madison to mention a few.

The industry response during the first meetings within the ATAC project were quite interesting. During the first meeting in 1997 there were endless questions and very large interest. We overshot the time for each presentation and the entire meeting. The industry members wanted to know exactly how Magnus ran the engine, how to start the combustion and how to tune the combustion phasing. Then when it was time for the second meeting we prepared a bit more for questions like that but all the industry members were sitting quiet and no one asked anything. We were wondering what happened, was HCCI dead already? Well, during the following dinner it was clear that all industry members had started their own experiments on HCCI and did not want the other participants to know that they did or how they were running their engines. So no questions can either mean that the topic is dead or that it is very hot.

During 1998 we got a first extension to the work within KCFP with a separate project with Scania. It was a project focusing on understanding HCCI and using optical diagnostics for doing so. Anders Hultqvist was hired as an industrial Ph.D. student. The idea was that he should build his own optical Scania engine with piston extender and large optical access. He did so but that took some time. Hence to have something to do while the engine was being built he joined Magnus in operation of the TD100. This engine could have optical access but limited to four windows on the side of a spacer between engine block and cylinder head. Anders and Magnus worked together with Mattias Richter from Combustion Physics who was fortunate(?) to get the 0.25 position with KCFP for optical diagnostics on HCCI. At first, passive measurements were performed, but later also laser induced fluorescence and other techniques. The first results from these activities were released in 1999. I still think these are some of the strongest papers on HCCI released from Lund and when I heard that we got the Horning award for best SAE paper 1999 I was fully convinced that it was for the optical paper. It turned out that that was not at all the case. Magnus, Anders and I got it for a paper on operating conditions needed for HCCI with different fuels. There could be many reasons why we got the award for that paper. One is that the list of authors was short, only three persons. Another can be that the paper was rather short and quite easy to understand. A third could be that it perhaps explained something about HCCI. A fourth reason could be that it was our time since the Lund HCCI papers attracted much attention also before this paper. Or it could be a combination of all the above, I do not know. The paper was rather simple. Take an engine with variable compression ratio and adjustable inlet temperature. Run it with a range of fuels from n-heptane (octane number 0) to isooctane (100) or mixtures of diesel and gasoline and then note the combinations of compression ratio and inlet temperature that must be used to get the correct combustion phasing with HCCI. Simple but still my most cited paper. Anders got his Ph.D. in 2002 and is now Professor and head of the combustion engine group at KTH. Mattias stayed at Combustion Physics after graduation and will soon become full professor as well.

A second extension to the HCCI work was formed with a project together with Caterpillar 1998 as well. We were approached by Rey Agama from Caterpillar at the SAE fall meeting in San Francisco 1998. Caterpillar was very interested in HCCI and wanted to work with us on the concept as we were the world leaders on the topic at the time. Ray took us (Magnus, Anders and me) to the best fish restaurant in SF to discuss what to do and the terms of the contract. Magnus looked at the menu and looked troubled. He could find 150 different kinds of fish but where was the steak? Rey understood the problem after some translation and asked the waiter if he could recommend us a good steakhouse since there had been a mistake in the selection of restaurant. The waiter replied that on the second last page of the menu there were indeed four types of steaks in the strange

event that you would go to the best fish restaurant in SF, and possibly the world, and would like to eat steak. As it turned out the steak was really good, not only Magnus had steak but all of us did. After this bonding dinner the links between Caterpillar and Lund were forged and we started a project 1999 before Caterpillar joined KCFP at the start of phase 3 in July 2000. Rey visited Lund a number of times during phase 3 and also hosted Jan-Ola Olsson when he spent two three-month periods at Caterpillar during this time. Jan-Ola was a Ph.D. student working with HCCI within KCFP.

Jan-Ola lived with Rey in his house during the stay in Peoria and that generated another interesting episode. As a Swede Jan-Ola thought a suitable way of transport is walking. Hence he walked from Rey's house to Peoria city center and back since the distance would not justify driving a car. This is not normal behavior in the Midwest; there you drive always and hence there are no sidewalks. That did not stop Jan-Ola from walking but the police did. Jan-Ola was picked up by the police when walking one evening and questioned about his strange behavior. Could he not afford a car or why else would he walk? After some discussions the police drove him to Rey's house and after verifying with Rey that indeed Jan-Ola was a Swede with only a strange definition of suitable means of transport he was released. This incident happened during the first three-month period. Jan-Ola learned from the mistake the second round and decided not to provoke the Midwest culture. He thus got a bicycle and took to the roads. It did not help much though, he got picked up by the police anyway. Apparently using a bicycle in the Midwest is as strange as walking. After that Jan-Ola used a car.

I lost contact with Rey as he changed positions within Caterpillar after around 2005. The last contact was at the 2006 SAE fuels and lubricants meeting in Toronto. I was there giving a keynote talk on HCCI and Rey flew in just to introduce me. He then named me Mr. HCCI after the impact of the HCCI work in Lund on the world research in the area. As a consequence you can reach me at my private email mrhcci@gmail.com which I registered just after the Toronto event.

Jan-Ola was the first student within a new branch of the HCCI activities in Lund. Since HCCI is fundamentally unstable it requires close attention and control at all times. Uncontrolled HCCI will ruin an engine in fractions of a second at high loads. I was able to get an HCCI control project funded by Sydkraft, the Swedish power utility company in 1999. It was the Sydkraft member of the board in KCFP, Lars Sjunnesson, that thought HCCI could be a concept of interest also for stationary power production. Hence I got funding for a first multi-cylinder engine. We rebuilt a Scania 12 liter six-cylinder diesel engine to be operated with HCCI using two fuels to control the combustion phasing. Instead of adjusting the engine conditions to the fuel we adjusted the fuel to the engine condition. As Jan-Ola was setting up the engine and the control system, Per Tunestål joined the group after graduating from Berkeley on closed loop combustion control. Per took control over the controls oriented Sydkraft project and it was all merged into KCFP when it was time to start phase 3 of KCFP. Jan-Ola and Per could run the Scania engine at 20.4 bar IMEP and 16 bar BMEP with a modified turbo based on a turbo compound system. This is still the highest IMEP reported with HCCI even though John Dec at Sandia is now close with 20.0 bar. With the first generation controls and some occasional "oops" we repeated the one-way variable compression ratio of the TD100 though. The engine got gradually shorter connecting rods and the crank no longer had exactly 120° between TDC:s. It is understandable that the high load record has not been broken. Jan-Ola graduated in 2004 and is now working at Volvo Cars. More control projects followed but that story is perhaps best told by Per since he was in charge.

Phase 3: 2000-2003 (1/7 2000- 30/6 2003, three years), 18 MSEK/year

When it was time to restart KCFP for phase 3 in 2000 the interest for HCCI was huge and Lund got a reputation to present very interesting results. Thus there was always a long line of persons from industry and academia that liked to talk after the papers were presented at SAE or similar events. Talking is nice but if the industry liked to know more I just said that full information comes with a membership in KCFP. Having a clear proposal and a fixed price tag on the information was very good and helped me a lot in the discussions. Many if not all engine companies were interested but those that finally signed the KCFP contract 2000 were Caterpillar, Cummins and Saab making it a total of 7 with Volvo, Volvo Cars, Scania and Wärtsilä already in KCFP. More followed during the phase. Toyota joined 2001 with Hino (Toyota Heavy Duty) in 2003 and Nissan in 2004 for the next phase. At that time the HCCI project had grown out of the KCFP costume as the Energy Agency and Lund University funding was fixed in 2000 and could not be increased. At one time the ATAC (HCCI) project had a larger budget than the total KCFP. But still the engine work could not have the full budget of KCFP.

During the phase a number of side projects on HCCI were started. With Saab we started the VCR-HCCI using the Saab variable compression ratio engine called V-eps. We got two students, one for control, Göran Haraldsson, and one for engine performance, Jari Hyvönen, who worked together. They eventually mastered the controls and could run the engine in a full transient test cycle in HCCI only, another world's first. Both Göran and Jari graduated in 2005. Jari is now working at Wärtsilä and Göran at Volvo. An interesting note is that Jari's last paper was a thermodynamic efficiency comparison of the v-eps, a conventional 4-cylinder GM engine and

Jan-Ola's Scania. Those results have been the backbone of the HCCI efficiency lecture I have presented possibly 100 times since 1996 but the paper is not that much cited anyway. Perhaps because it contained too much information, perhaps because it highlighted some fundamental drawbacks with HCCI. Jari and Göran were replaced by Hans Aulin and Tomas Johansson for another pair doing controls and engine work. They graduated in 2011 and 2010 and are now working for BorgWarner and Koeningsegg. During this phase the project was managed by GM and the Germans in that organization had little understanding of engine research. It was all cancelled when Saab went bankrupt in 2011.

Another side project was the national project called GIHR within the green car initiative. It was a large project with Ingemar Magnusson from Volvo as project leader. Lund, Chalmers and KTH were all involved with the largest part for the combustion engine group in Lund. The GIHR was a starting point for the transition from HCCI being a Lund thing to a Swedish specialty. It was a bit mixed emotions for me with GIHR. Of course it was nice to have additional students and thus being able to do more research but on the other hand the control of the HCCI work was not 100% in my hands anymore. Leif Hildingsson did much work on HCCI with direct injection in a Volvo Car optical engine. When GIHR ended he continued within KCFP and graduated 2006. After that he did a post doc at Shell and worked on initial gasoline PPC work there. He is now working for Rolls-Royce, UK. Henrik Nordgren worked with a HD Scania engine that later transformed to the PPC combustion mode, more on that later. Petter Strandh did closed loop control of a Volvo HD engine much like Jan-Ola did before but used variable valve timing for control and not only fuel blends. Petter is now working at Volvo.

So what did we do within KCFP during the phase? A very significant contribution was the application of laser diagnostics to the HCCI combustion process. We did laser induced fluorescence of fuel, formaldehyde and OH. In the beginning we used a conventional 10 Hz laser and hence could get one picture per cycle. This was good but not really sufficient as the combustion is very structured and the structure changes randomly from cycle to cycle. It is hence not possible to track the process by taking images from many cycles at one crank angle position at a time and generate an averaged time sequence that way. Luckily Marcus Aldén at Combustion Physics had developed a multi-YAG system. This can oversimplified be explained as eight lasers in a box and eight cameras in another box. This enabled a short movie of the process in an individual cycle. It was the first time anyone could see the transformation from homogeneous bulk oxidation to a structured process with HCCI.

What also happened during this phase was that I got promoted to Professor 2001 and graduated my first batch of Ph.D. students in 2002. It was a rather busy year with Martin, Fredrik, Magnus and Anders all graduating. There was also a fifth, Olof Erlandsson. He was the first simulation student I supervised. The project was fully funded by the Energy Agency but linked to Wärtsilä. Olof did a very good job using the experimental data generated by Magnus and Jan-Ola to predict what could be achieved with HCCI and a humid cycle or otherwise. He also generated two side projects by himself. One was on Swedish hot bulb engines from 1890 to 1964. He found out that the Lund University library is one of two national libraries that should store all documents generated in Sweden. Some of that happened to be the test results from "Statens Maskinprovning" that tested all engines sold on the Swedish market for tractors and stationary engines. Hence Olof ordered all the test protocols and compiled the engine efficiency as a function of load for hundreds of test points. It all resulted in an SAE paper 2002 and a test run of a hot bulb engine in the lab the same year. The second side project was to use all the data generated by Magnus Christensen in the TD100 to compile a model for HCCI. He generated a simulation program that could run in real time. With five or more inputs we get five outputs and a pressure trace, much like when running an engine in the lab. Since 2002 all new students on HCCI have started with using the simulation program before running a real engine in the lab. A virtual engine is much cheaper and faster to repair. Can't state that we did not blow up HCCI engines after 2002 though.

A much less positive event late during the phase was that my predecessor as head of division and my supervisor, Gunnar Lundholm, was diagnosed with cancer in the spring of 2003. He thus could not continue as director of KCFP when it was time to restart again in July 2003. I had to take over with very short time to prepare. When Gunnar passed away in January 2004 I also had to take over as head of the division of combustion engines.

Phase 4: 2003-2005 (1/7 2003 – 30/6 2005, two years)

With phase 4 I was all of a sudden director of KCFP. I got an almost fixed project list and budget to work with and it did not give much room for change. I did bring in Hino and Nissan as new members of KCFP and started a separate project with Toyota on optical diagnostics but had to spend more time on administration. The HCCI projects moved on and generated a number of new results. It was during this phase that the interest in HCCI world-wide peaked. The energy agency could only fund a two-year project instead of the three years before due to political movements within the Swedish combustion engine community.

Phase 4.5: 2005 (1/7 2005- 31/12 2005, half a year)

The phase 4.5 was a result of these political movements. The half a year phase would align KCFP with CERC at Chalmers and the new center to be at KTH; CICERO. The years 2004 and 2005 were a bit frustrating. Lund was clearly #1 in the world on the #1 hot topic within engine research but what was discussed at the board meetings of KCFP was how to improve the situation at KTH and how to strengthen Swedish academia as a whole. There were discussions on forming a large national competence center for combustion engines preferably run by KTH or if that was not possible Chalmers. That would have been the end of the activities in Lund. Discussions were sometimes intense and in a meeting in Skövde, Urban Johansson from Scania made a clear statement to me: "One more word from you and I will phone back to Scania and two minutes later we are out of KCFP for good ". I then thought about the tactic I used during Magnus Christensen's master thesis defense and decided to do just that, keep my mouth shut for the remainder of the meeting. But Urban was a bit confused as I did not leave the room, perhaps a more normal response in such a situation. Sometimes you do not need to talk to express your point though.

Phase 5: 2006-2009 (1/1 2006-31/12 2009, four years)

With phase 5 there was a complete change of operation of KCFP and the other engine research activities in Sweden. At Chalmers the CERC continued and at KTH CICERO was formed. It was decided that CICERO should focus on gas exchange since it was the strong point of SAAB that at the time had the engine R&D site in Södertälje and Scania was happy with that as long as more activities came to Stockholm. Strangely enough it was decided to give the directorship to Henrik Alfredsson in fluid mechanics and not to the professor in combustion engines. As a result, CICERO was focusing on fundamental fluid dynamics and only ¼ or less of the funding was for the combustion engine group.

For KCFP the restart 2006 meant three things. The center was refocused to combustion in engines only. Thus the budget for engine related work was tripled within KCFP. The other thing was that the funding was secured for a 4+4 year period. We finally had a foundation to build on. We lost Hino and Cummins in the transition with the half year 4.5 phase but gained Chevron as a new partner instead. We started new side projects with Volvo, Delphi and Shell as well. We also got a first EU funded project on mini-engines. It should be noted, however, that the budget for engine research in Lund did not triple just because the budget within KCFP did. The GenDies project that was started 2001 by Rolf Egnell within CeCOST was transferred to KCFP at the same time as the gas turbine work was moved from KCFP to CeCOST.

The third thing that happened at the start of phase 5 in 2006 was perhaps even bigger even though I did not do a very good job executing it. I killed the HCCI research.

When it was time to start planning for phase 5 we were ten years into the HCCI era and it was quite clear that industry started to doubt the direct applicability of HCCI. In academia Lund still had the highest possible status though. I had thus two options, focus on more and more academic HCCI work and squeeze out all that can be extracted from HCCI. I am sure that would have been the path resulting in the highest academic recognition another ten years or so. The other option, that I took, was to cancel the HCCI activities and move on to something a bit different. This was a move to more mixed combustion modes i.e. a step back from pure HCCI towards something more similar to SI or something more similar to diesel. The first was named Spark Assisted Compression Ignition, SACI, and the latter Partially Premixed Combustion, PPC. In the work plan for phase 5 of KCFP HCCI was no longer part of the plan. It is perhaps a bit ironic that what put Lund on the map for combustion engine research in the world was not part of the major expansion of the activities in 2006.

Changing something that drastically was not done in a day though. The students had a hard time getting the fuel injection systems up and running and did a lot of HCCI work with simple port fuel injection a number of years into the phase. In fact, when Andreas Vressner graduated in 2007 he had not done anything but HCCI. It was still a very good thesis and he is now working at Volvo.

We did start the SACI work earlier than 2006 with some work in the Saab v-eps engine by Jari and in a Volvo inline 6 by Håkan Persson. Håkan's project was officially called "unorthodox otto engines" before 2006 but was in reality SACI. Håkan continued from the multi-cylinder to an optical engine within KCFP and got his Ph.D. in 2008 before heading to Volvo Cars. As he graduated we felt that we knew as much as needed on SACI and did not push further in the field. It was a very unstable combustion mode and the benefits were not great.

The PPC work started in 2003 with a first paper on PPC using diesel fuel published in 2006. Henrik Nordgren worked on a Scania single cylinder in the mean time within the GIHR project. For some reason it was decided within GIHR that we at Lund should work with concepts with very early fuel injection giving a close to homogeneous charge in the cylinder. Late injection should be left to others (Chalmers). Henrik did so and also investigated the fluid flow in the cylinder in some detail with PIV. In the very end of his studies we decided to care less about these "guidelines" though and ran with late injection timing and hence what we would call PPC today. Henrik is now working at Scania.

In 2007 it was time for Vittorio Manente to change from his EU-project on mini engines running HCCI to PPC within KCFP. I remember that I did have an introductory talk with Vittorio when he changed project. In the mini engine project, he ran model airplane engines designed for SI combustion and 15 bar peak pressure in HCCI mode with 50 bar peak pressure. Once combustion got started and the residual gas slowed down combustion that was OK but the first cycle does not have any residuals. Hence it was quite common that the engine connecting rod broke during start-up. I think he ruined 17 connecting rods in one week as a record. This was not a problem with a model airplane engine since the connecting rods are cheap but when he moved on to the Scania truck engine, connecting rods are much bigger and more expensive. Hence I told him to take care and not to brake the Scania. So he did. Well he did break the Scania a few times as well but not as often as the mini engine at least.

Vittorio started running PPC with diesel fuel but gradually also with more high octane fuels like gasoline and ethanol. We soon realized that PPC is very well suited for gasoline like fuels at mid to high loads and Vittorio started to optimize the conditions for PPC. At the end he managed to reach 57% indicated efficiency at 8 bar IMEP with a standard combustion chamber and 55% in the load range 4-18 bar with a slightly reduced compression ratio. These were remarkable numbers, some 10% points higher than what could be achieved with pure HCCI and much better than conventional diesel combustion. He also showed that PPC could be operated up to 26 bar IMEP even though he broke the transmission when aiming for 30 bar. He also tested fuels. We got different kinds of gasoline from Chevron and tried to sort out what were good and bad components within the mixtures. Vittorio got his Ph.D. in 2010 and is now working at Aramco, Paris after periods at Volvo Cars and Volvo. He did not publish more papers than Magnus Christensen before graduating but it was close.

Vittorio's work triggered an interest in PPC much like the HCCI work did before but perhaps at a slightly more modest scale internationally. We started a project on running PPC with too high octane number at too low load. I called it the limbo project even though it had a different official name. Patrick Borgqvist did the limbo with a Volvo car single cylinder engine having a Cargine active valve train to trap hot residuals in one way or another. He finally succeeded wtih the task and could run PPC with commercial gasoline at idle. Patrick graduated in 2013 and is now working at SAAB.

Before moving on to Phase 6 of the KCFP it is worth mentioning the political struggles that resulted from the international evaluation of the competence centers in 2009. It was not a problem for KCFP, by then the citations from the early work on HCCI in Lund had full impact and hence we were OK. The problem was in Stockholm as the director there did not understand the importance of the evaluation. Hence it was recommended that CICERO would be discontinued. The discussions during the board meetings were quite interesting and entertaining for me. Perhaps it should be mentioned that as a compromise in 2006 when some industry members liked one national competence center and some three independent centers we got an organization with three centers but with our three board meetings taking place back to back at the same location. The directors of the three centers should then take part in also the other two board meetings. We also got an umbrella organization first called 3KC and the later SICEC. Hence four meetings in two days.

As a result of the failed evaluation of CICERO it was decided to shut it down. But the director of the center then got the assignment to immediately form a new center with very similar content like the bird Phoenix. The only requirement the Energy Agency had was that it should have a new name. When the new name CICERO2 was proposed the Energy Agency representative slammed his hand in the table so hard I even woke up. The new name became CCGEx and the director was replaced.

The Energy Agency representative, Bernt Gustafsson, also gave me as KCFP director a special assignment at the start of phase 6. I had to become the Swedish delegate in the International Energy Agency (IEA) implementing an agreement for combustion, and to be task leader for the new HCCI task. It turned out to mean two additional trips per year to a nice place. I thus had to spend a week per year at an IEA meeting and also host the event in Lund once. At first I though this was perhaps a bit of a waste of time and money but after a few years I started to realize the importance of meetings with high ranking people from other countries in a relaxed way once per year. It was actually quite useful time spent. We got a DoE project from the US after a few years and a week alone per year with the Energy Agency representative does not hurt.

Phase 6: 2010-2013 (1/1 2010-31/12 2013, four years)

At the start of phase 6 it was time to fully focus on PPC. Well at least half focus. The PPC activities got around ½ of the KCFP budget and Gendies got 1/3 with the remainder 1/6 to gas engine work. But with that funding it was time to focus on understanding PPC and the effects of fuels. We got some results done including those that were sufficient for Hadeel Solaka to graduate in 2014 but the phase was largely disrupted by the large laboratory refurbishment. The location and status of the combustion engine lab has been discussed since 2002 when our neighbor MAX-lab claimed our lab space. Endless meetings and discussions later it was finally decided to rebuild the lab in 2012, a decade later, perhaps

a normal time for a decision at Lund University. It was a rebuild of the lab with a cost of 54 MSEK. The project was kept under the budget but even so the cost was later states as 73 MSEK and then 83 MEK due to the fact that half was paid directly by the university (?!) We got six new test cells and also two existing cells refurbished. The plan was to do the refurbishment in new locations in a way to not disturb the current test cells and thus enable engine experiments during the refurbishment. Not much of that turned out to be true. The lab was unusable for most of 2013 and a bit of 2014. This also meant that when it was time to summarize the PPC work during phase 6 not that much had really happened. We could really reuse the work plan for phase 6 almost intact at the start of phase 7.

Phase 7: 2014-2017 (1/1 2014- 31/12 2017, four years)

It was now time to start the current phase of KCFP. With help from Scania we managed to get Cummins to rejoin KCFP for phase 7 and Wärtsilä decided to step up from minor to full paying member. We also got BorgWarner to join due to the interest in PPC and a well timed dinner with the later VP of research at BW at an SAE event in Detroit. The additional SME:s of Dantec, Loge and Swedish Biomimetics made a list of 14 participants within KCFP. We did loose Caterpillar though. As Cummins rejoined they thought it was best to leave. It could also be the effect of me giving a PPC talk at Caterpillar in which I stated that the RCCI results of Rolf Reitz at Wisconsin-Madison were questionable. As it turned out 100% of the persons in the room got their degree from the ERC at Wisconsin.

In this phase it was also time to start a Ph.D. student program on the next thing after PPC. During the efforts in understanding why the PPC results of Vittorio Manente were so good it became clear that the major thing was the reduction of heat losses to the walls. With high enough dilution the temperature after combustion is lower and hence the temperature difference from burned gas to wall will also be less. But running with high dilution is a challenge. If we should operate at lambda=3 instead of lambda=1 we need three times the air into the engine. This can be achieved with a turbo provided that the exhaust energy is sufficient to drive the turbine. The problem is that lean mixtures also give low exhaust temperature and hence less energy for the turbine to work with. This was the starting point to the Double Compression Expansion Engine, DCEE. What if we feed the engine with really high boost pressure from a piston compressor? It can then be considered as a split cycle with half the compression in the first cylinder and half in the second smaller one. And what if we then do the same with the expansion? According to simulations this concept has the potential to reach 65% indicated and 60.5% brake efficiency even though it is only 62%/56% published this far. The key is to use a combined compression ratio of 60:1 and a lean mixture. The DCEE resulted from some thinking from my side in combination with a quiz walk. I never thought I would have anything good to say about quiz walks. Generally, I think they are a total waste of time and with smartphones possibly a bit outdated. The quiz walk took place at an Energy Agency event in Örebro as a break between long sessions of project presentations. It was by the way there that I did the most slide dense presentation this far with 67 slides in 7 minutes. Some state that they understood what we did within KCFP after that presentation so apparently I failed with my purpose.

Anyway, why was the quiz walk essential? It was because Staffan Lundgren was as little interested in a quiz as I was so we started to talk DCEE instead and soon realized that we should work more on this, much more. At the Energy Agency quality check presentation for the resulting project proposal we got the question why we did not ask for more money. Then I knew this will be big.

I decided to keep the DCEE outside KCFP instead of integrating it into the center at the restart 2014. The reason is that it already generated a number of patent applications and will generate more in the future. Patents are always a big problem to handle within a center. I remember when we tried to patent the concept of closed loop combustion control of HCCI back in 1998. Or perhaps I have to tell that story some other time.

It is now the end of 2015 and it is time to end this not too short reflection of my years within KCFP. The combustion engine group has changed completely from the persons involved in the beginning to what it is today. The group is strong with very competent faculty and staff and I have no doubt that they will do great things also after I leave. After all, I hand picked them all. Best of luck and I am sure we will keep in contact also in the future.

PS. Per, thanks for the proof reading. You will at least have one less task to do in Lund after I leave. DS.

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