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ABSTRACT

This paper presents a strategy for closed-loop control of a multi cylinder turbo charged Homogeneous Charge Compression Ignition (HCCI) engine. A dual fuel port injection system allows control of combustion timing and load individually for each cylinder. The two fuels used are isooctane and n-heptane, which provides a wide range of autoignition properties.

Cylinder pressure sensors provide feedback and information regarding combustion. The angle of 50% heat release is calculated in real time for each cycle and used for timing feedback.

Inlet air preheating is used at low loads to maintain a high combustion efficiency.

INTRODUCTION

HCCI is a hybrid of the well-known Spark Ignition, SI, and Compression Ignition, CI, engine concepts. As in an SI-engine, a homogenous fuel-air mixture is created in the inlet system. During the compression stroke the temperature of the mixture increases and reaches the point of auto ignition; i.e. the mixture burns without the help of any ignition system, just as in a CI engine. The first studies of this phenomenon in engines were performed on 2-stroke engines [1-6]. The primary purpose of using HCCI combustion in 2-stroke engines is to reduce the HC emissions at part load operation. Later studies on 4-stroke engines have shown that it is possible to achieve high efficiencies and low NO_X emissions by using a high compression ratio and lean mixtures [7-21]. In the 4-stroke case, a number of experiments have been performed where the HCCI combustion in itself is studied.

Since the homogeneous mixture auto ignites, combustion will start more or less simultaneously throughout the whole cylinder. To limit the rate of combustion under these conditions, the mixture must be highly diluted. In this study a highly diluted mixture is achieved by the use of excess air. Using a mixture that is too rich will cause a very rapid combustion and knock-related problems will occur. On the other hand, a too lean mixture will lead to an incomplete combustion or cause misfiring.

The proportion of air to fuel affects the timing of the start of combustion, which is also strongly influenced by the inlet temperature, fuel properties and compression ratio. In this study the timing of the combustion has been moderated by changing the proportions between the two fuels used; the easily self ignited n-heptane and the more self ignition resistant isooctane. Load has been controlled by varying the total amount of fuel, and pre heating has been used at low loads to improve combustion efficiency.

The aim of the study has been to outline, and test, a strategy for closed loop control of an HCCI engine. A robust control system is of course essential for any commercial engine, but is also very useful during research since HCCI is sometimes hard to control by hand, e.g. along with turbo chargers or EGR which can cause rapid changes in conditions.

EXPERIMENTAL APPARATUS

THE ENGINE

The engine used is a modified Scania DSC12, 12 liter, six-cylinder, turbo diesel engine, mainly used for truck applications. The original system for diesel injection has been removed and replaced by a low-pressure, sequential, system for port injection of gasoline. The engine has four-valve cylinder heads and consequently two inlet ports per cylinder. The injection system can thereby supply two fuels to each cylinder, one in each port. In this way the amount of each fuel can be individually adjusted for each cylinder from the controlling computer. The fuels used are isooctane and n-heptane.

In a previous study [22] the turbo charger used was dimensioned for the diesel cycle in a truck application. The turbine in this turbo charger was too big for an HCCI application and no boost was achieved. In this setup the turbine is smaller though still too large.

The inlet manifold has been extended to supply space for the injectors. In this way the injectors are placed just outside the inlet ports of the cylinder heads. Since fuel injection starts at Gas Exchange Top Dead Center (GETDC) and the fuel spray is directed along the ports, a very fast response to changes in fuel amounts is achieved.

A custom-made injection controller manages the injectors with one PIC processor per injector (PIC is a family of low cost one-chip computers). The PIC processors control the injectors via fiber optic cables and the device supplies a trigger pulse for each PIC processor at the GETDC of the associated cylinder. The PIC processors can be programmed, in real time, to turn injection on or off, to wait a certain angle between the trigger pulse and start of injection, and to keep injectors on for a certain amount of time. When injection is turned on, the PIC processor does nothing until it receives the GETDC trigger pulse, at which the processor starts to count degree pulses from the crankshaft encoder (5 pulses per crank angle degree). At the programmed angle the injection is turned on and stays on for the predetermined period of time. After injection is finished, the processor checks the parallel port before it returns to waiting for the trigger pulse. A result of this is that communication to a certain processor can be initiated only at the moment when injection is finished. If the controlling computer is too slow in sending the new command, the injector will not fire the next cycle. This phenomenon can be seen in Figure 4 and Figure 9 in the “Results” section below.

The intake system has been fitted with three electrical heaters, with a total capacity of 37 kW, between the inter cooler and the intake manifold. Heating is only used at low loads to keep emissions of HC and CO down, i.e. to keep combustion efficiency up. Heating power is adjusted by individually turning two of the heaters on or off and by adjusting the power of the third heater between 0 and 10 kW.

Figure 1 shows a schematic drawing of the engine system.

Apart from these changes the engine is in its original configuration, with both pistons and cylinder heads unchanged.

The properties of the engine are summarized in Table 1.

Table 1. Geometric properties of the engine. Valve timings refer to 0.15 mm lift plus lash.

Swept Volume	11 705 cm^3
Compression Ratio	18:1
Bore	127 mm
Stroke	154 mm
Connection Rod	255 mm
Exhaust Valve Open	82° BBDC
Exhaust Valve Close	38° ATDC
Inlet Valve Open	39° BTDC
Inlet Valve Close	63° ABDC

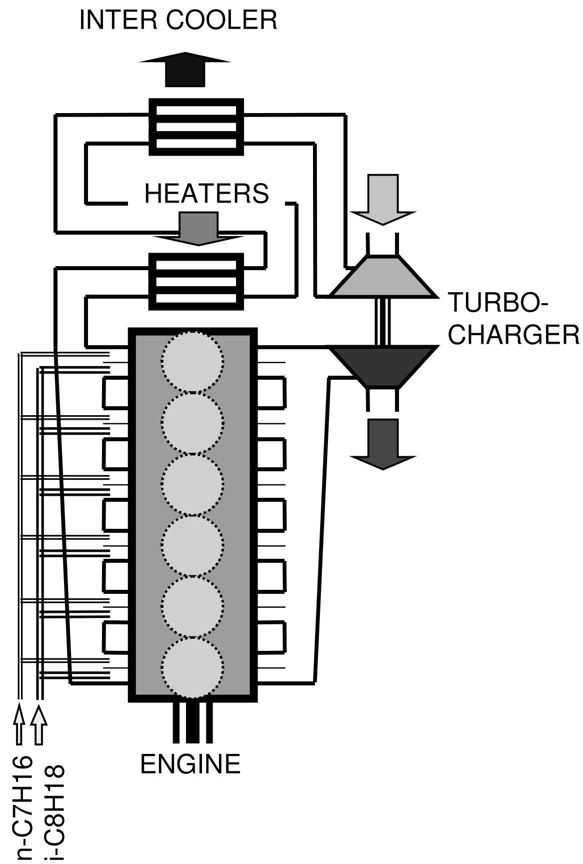

Figure 1 Engine system.

SENSORS AND MEASURING SYSTEMS

Each cylinder is equipped with a cylinder pressure sensor to give the control system a means of monitoring the combustion. The sensors are piezoelectric transducers connected to external charge amplifiers.

Each exhaust port has a thermocouple, measuring the exhaust stagnation temperature of each cylinder individually. Thermocouples are also used for monitoring inlet air temperature, before and after the turbo compressor, after the inter-cooler and after the heaters. Strain gauge absolute pressure sensors are used for measuring inlet air pressure and exhaust pressure before and after the turbine.

The signals not of primary interest for controlling the engine are monitored by a low frequency sampling logger.

Cylinder pressures, inlet-air temperature before and after the heaters and the inlet pressure are all monitored by the controlling computer. Cylinder pressures are sampled at a rate of 1800 samples per cycle for each cylinder. Inlet conditions are sampled once per cycle.

A high-resolution encoder (1800 pulses per revolution) connected to the crankshaft provides the crank angle base for cylinder pressure measurements. A multiplexed 16 bit A/D-converter, clocked by the crankshaft encoder, with simultaneous sample and hold circuits and an on-board First In First Out buffer (FIFO) of 65536 samples is used for sampling cylinder pressure. The device sampling cylinder pressure communicates with the controlling computer using the parallel port.

Inlet air pressure signals and inlet air temperature signals before and after the heaters are sampled by a 16 bit multiplexed A/D-converter on a multifunction card connected to the PCI bus of the controlling computer. The same card is used to control the heaters by two digital channels and a 12 bit analog output.

The controlling computer is a 600 MHz Pentium II with 128 MB RAM. The computer does all computation and no DSP capability is available on any of the data sampling devices. The operating system used is Windows 98 and the application has been developed using Delphi 5.

ENGINE STARTUP

The engine is started by letting the dynamometer motor the engine at a desired speed. Heaters are turned on and tuned for the appropriate inlet air temperature. Finally, fuel injection is turned on and the octane number is lowered until proper combustion is obtained.

This starting procedure is not suitable for any kind of real-life applications, but is used for convenience in these experiments.

STRATEGY

The intention was to demonstrate a functioning control system for a dual fuel HCCI engine rather than propose a concept for a commercial engine. For this reason the “less practical but more scientific” fuels of isooctane and n-heptane were chosen. It was also decided on not to put any limits on mixing ratios between the two fuels.

Because of the nature of HCCI, the control system cannot directly control the crank angle of ignition. In stead, ignition timing has to be controlled by secondary parameters and the desired torque from the engine has to be achieved in a way that allows timing control of heat release to work. In this system the amount of isooctane and the amount of n-heptane can be directly controlled for each cylinder and the inlet air heating power is also directly controlled, however shared, by all cylinders.

The problem is solved by, for each cylinder, having one PID controller for timing control and one for IMEP control, plus having one PID controller for inlet air temperature control. The controller for timing has to be gain scheduled, since the angle sensitivity for a change in mixing rate between the fuels varies a lot depending on the situation.

CONTROL SYSTEM

OVERVIEW

The controlling computer interfaces the engine system by a fast sampling A/D-converter for the cylinder pressure traces, a multifunction card for slower sampling of inlet conditions and control of heat power and a device actuating the injectors.

The user interface allows manual control of injection as well as closed loop control. The closed loop control can be turned on individually for each cylinder for timing and subsequently for net Indicated Mean Effective Pressure (IMEP). The user is continuously updated with pressure curves and values for mixing rates of fuels, total amounts of fuel, the angle of 50% heat release (CA50) and the net IMEP. Inlet conditions and engine speed are also displayed.

The application is multi threaded to allow maximum performance. Reading from A/D-buffers, calculation of the engine status and control is made in one thread, called the “calculation thread” for convenience. Communication to injectors, screen updating and disc saving are done in other threads.

DATA FLOW

During operation, the two A/D-converters sample data and transfer it to a RAM-buffer. The calculation thread empties these buffers and checks how many complete cycles are available for all cylinders. The work is then done only on the latest cycle available for all cylinders. As it turns out though, the computer normally makes the calculations faster than data is generated, and consequently all cycles are normally analyzed.

When at least one complete cycle is available for all cylinders, the engine status is calculated for that particular cycle. The engine status contains data such as engine speed, inlet temperature, inlet pressure, amount of fuel injected, CA50, and IMEP. This data is used for the control algorithm.

The control algorithm receives the engine status data as input. The control algorithm starts by checking for alarms. Alarms are set in the engine status and can be caused by an excessively high maximum pressure or pressure increase or an unexpectedly low heat release. If alarms are found, an offset to the set point in timing or IMEP will be made to avoid damage of the engine. If very high alarm levels are reached, injection will be turned off!

After going through the alarms, the closed loop control is performed for timing and net IMEP for each cylinder and also for inlet temperature as specified by the user. When this is done, injection and heat is adjusted according to the controller outputs.

After calling the control function the engine status is sent to the screen updating thread. Subsequently the disc saving thread receives updated data, and saves the engine status and binary cylinder pressure data as instructed by the user.

Finally the used data in the temporary buffer for binary pressures is removed before the A/D-buffers are emptied again.

Figure 2 below illustrates the most important parts of the data flow.

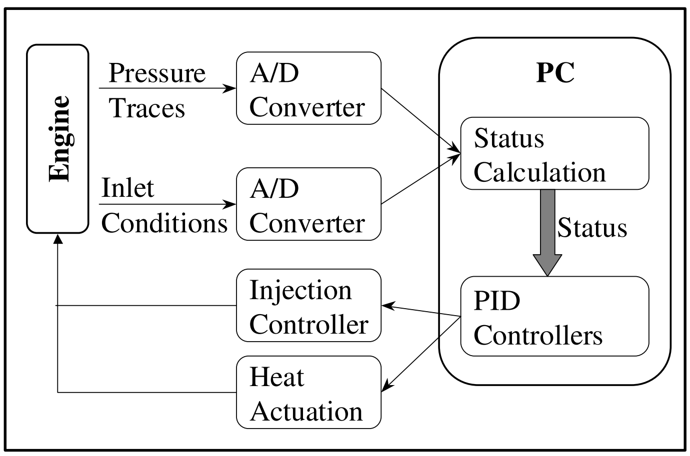

Figure 2: Schematic view of the data flow.

CALCULATION OF ENGINE STATUS

The engine status is the central variable containing almost all data necessary to describe the engine state. It contains both general information valid for the complete engine as well as cylinder specific data. Listed below are the important parts of the engine status:

- Engine speed
- Inlet temperature
- Inlet pressure
- Heat power
- Cycle index
- Time of cycle
- Pressure traces for each cylinder, converted to Pa.
- CA50 for each cylinder
- Net IMEP for each cylinder
- Amount of fuel for each cylinder
- Fuel mixing rate for each cylinder
- Peak pressure for each cylinder
- Peak pressure-rate (Pa per crank angle) for each cylinder
- Net heat release for each cylinder
- Cylinder individual alarms for too high peak pressure, too high pressure-rate and too low heat release.

Some of these values are measured directly, others have to be calculated from measured data. Most important is the cylinder pressure data. Because of the nature of piezoelectric sensors, the offset of the cylinder pressure will not be fixed. Net IMEP computation does not suffer from this, but the computation of net heat release will be wrong if the pressure offset cannot be correctly determined.

To calculate the pressure offset the program assumes a known $\gamma = C_p/C_v$. The pressure trace is then fitted to $(p - p_o)V^{\gamma} = C$, using least squares nonlinear regression.

Net heat release is calculated for the total mass in the cylinder as the integral of

$$\frac{1}{\gamma - 1} VdP + \frac{\gamma}{\gamma - 1} PdV \ ,$$

starting at crank angle -30°, BTDC, and finishing 90° later. In the absence of combustion this is a decreasing function of crank angle. To find CA50 for a cycle, the lowest value of the heat release integral is found first. Then the highest value at an angle *after* the lowest value is found. The difference between these two values defines the net heat release. CA50 is the angle between these two angles where the heat release integral is halfway between the lowest and highest values.

Peak pressure is picked as the highest pressure reading without filtering, whereas the highest pressure rate is found after low pass filtering of the pressure data. The filter used is a 1:st order Butterworth filter.

Cycle index and time of cycle are used for the controllers to be able to work either time based or cycle based. The heat controller is time based and the controllers for timing and net IMEP are cycle based.

THE PID CONTROLLER

Heat control, timing control and IMEP control all use the same PID controller code, but with individual parameters of course. The implementation of the controller is discrete and it is executed once per cycle, if the computer is fast enough. For each time the controller is executed it receives the time as input. This way the same controller code can be used for either time-based control or cycle-based control.

The proportional part of the controller adds a contribution to the output proportional to the difference between the set point and the measured value. The derivative part adds contributions proportional to the derivatives of set point and measured value separately, with individual gains and low-pass filtering.

The integral part of the controller integrates the error and gives a contribution proportional to the integrated sum. If the controller saturates, the sum will be kept at a value giving 100% or 0% output regardless of the integral result; this is to prevent integrator windup.

A feed forward term, given as an input to the controller, is added to the output after low pass filtering.

Basically two reasons exist for low-pass filtering the derivative part and the feed forward term: One is that both the derivative part and the feed forward term can be sensitive to signal noise because of the high gain from measured value to controller output for these parts. Amplification of high frequency noise would feed through to the controller output and prevent smooth operation of the engine. The other reason for low pass filtering is the risk of saturating the controller. If the controller is saturated due to a large contribution from either the derivative part or the feed forward term, the integrator sum is reduced to prevent integrator windup. But if the term causing saturation disappears or decreases rapidly again, it takes a long time for the integrator sum to build back up again. The result is a short period with full controller output followed by a long period with low output and consequently an error that may grow. The low pass filtering used is a Butterworth filter of the 1:st order.

The parameters for the PID controller are summarized below:

- Proportional gain
- Integral gain
- Set point derivative gain
- Measured-value derivative gain
- Cut off frequency for low pass filtering of set-point (for derivative use)
- Cut off frequency for measured value (for derivative use)
- Cut off frequency for feed forward term
- Maximum controller output
- Minimum controller output

HEAT CONTROL

The set point for the heat controller is a temperature selected by the user. Naturally, the engine performance is indeed dependent on the inlet temperature. At low loads, a high inlet temperature is preferable to maintain a high combustion efficiency. At high loads on the other hand, a low inlet temperature is desirable to lower the rate of heat release and avoid premature ignition. In the present configuration however, the user selects inlet temperature and heat is not used actively to control combustion.

Since ignition timing is controlled by fuel mixing ratio rather than inlet temperature, a fast response to changes in inlet temperature set point is not very important. Keeping the inlet temperature constant during changes in conditions is important though! Changing speed for example, will in itself affect the timing of the combustion. This would be an even bigger disturbance for timing control if inlet temperature changed as well.

Heat control is time based rather than cycle based, though for practical reasons, the controller is run once per cycle since inlet conditions are updated on a cycle basis.

The heat controller uses feed forward to improve controller performance during transients. The calculated feed forward value is heating power, based on engine speed, inlet pressure and desired temperature increase over the heaters. No low pass filtering of the feed forward term is used, because of the low-pass character of the heating system itself.

The PID controller for heating control makes use of proportional, integral and set point derivative gains. The derivative part for measured value is not used.

IMEP CONTROL

Load is controlled by adjusting the total amount of fuel injected to each cylinder every cycle. The feedback is net IMEP, which is computed from cylinder pressure. It is desirable to be able to change load as fast as possible. Unfortunately a change in load, or IMEP, has a large influence on the timing of combustion! For this reason the response from the IMEP controller has to be limited by the performance of the timing controller.

In this first approach, the problem of IMEP affecting combustion timing was solved by making the IMEP controller slower than the timing controller. Only the integral part is used for IMEP control. An integrated load/timing controller is probably a better, although more complicated, approach.

The IMEP controller is implemented for cycle based control.

TIMING CONTROL

Timing control turned out to be the most challenging part of this work. Timing of combustion is controlled by the mixing ratio between the two fuels n-heptane and isooctane. In this paper, for ease of notation, the mixing ratio of the fuels is referred to as the octane number, even though this is not quite correct.

An increase in octane number will always delay the combustion and in this way the octane number is a powerful tool for timing control. However, the sensitivity of combustion timing to a change in octane number is found to vary between roughly 0.15 and 15 degrees per octane number unit. The sensitivity is measured as the change in CA50 divided by the change in octane number:

$$S \equiv \frac{d(CA50)}{dO}$$

It was found that the sensitivity is strongly dependent on octane number, but also on speed, inlet temperature, fuel amount and the timing of combustion. Inlet pressure probably also affects the sensitivity, but due to leakage problems with the inlet system at the time, this couldn't be included.

Isolating the different dependencies is important, but also very difficult – it is impossible to change parameters individually. The approach chosen was to assume that the sensitivity could be described as a product of functions, where each is a function of only one variable.

$$S = f_n \cdot f_{Ti} \cdot f_{mf} \cdot f_O \cdot f_{\theta 50}$$

It could be argued that the timing, i.e. CA50, is actually a function of the other parameters and consequently should not be included. However, the experimental data shows low repeatability for combustion timing, and thus the sensitivity function improves by including timing as well.

Inlet temperature is measured just in front of the inlet manifold. Heat transfer between the inlet air and the manifold, ports and cylinder walls will affect the actual temperature trace during compression. So will evaporation of fuel. Variations in these phenomena are not monitored and these are probably reasons for the apparently low repeatability of combustion timing.

The experimental data for sensitivity includes 345 operating points, and the relative standard deviation of the prediction error, using the sensitivity function at these points, is just below 3%. Figure 3 shows a plot of the fit between measured and estimated sensitivities.

The timing controller is gain scheduled in order to be able to handle the variation in sensitivity. The gain parameters are divided by the estimated sensitivity before entering the control algorithm.

Timing control uses all parts of the PID controller except for the feed forward, which is not used. Timing control is cycle based.

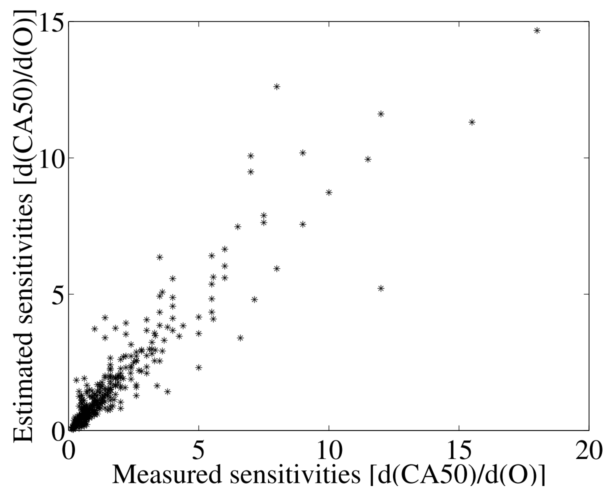

Figure 3: The estimated sensitivities plotted versus measured sensitivities. The correlation coefficient is 0.93.

RESULTS

This section evaluates the control system by subjecting it to various excitations, some more and some less realistic.

TIMING CONTROL STEP RESPONSE

The control system's response to step changes of the CA50 set point is shown in Figure 4 and Figure 5. Both the positive and the negative steps are conducted at an operating point of 5.0 bar net IMEP, an engine speed of 1500 RPM with closed loop control and no heating of the inlet air. These experiments are conducted on one cylinder for simplicity. The other cylinders are motored.

A step change of the CA50 set point is not a very realistic scenario, but serves to illustrate some characteristics of the control system.

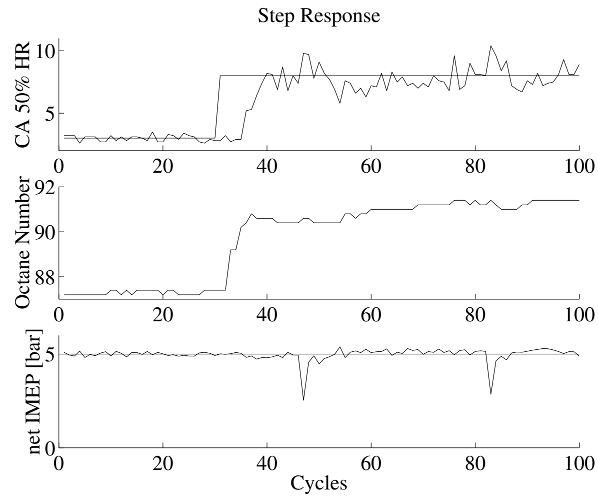

Figure 4: Positive step change of the CA50 set point.

In Figure 4 the CA50 set point is instantaneously changed from 3° to 8° ATDC. The delay of about 4 cycles between the set point change and the reaction in the measured CA50 is a result of buffering of cylinder pressure data and the fact that, in most cases, it takes two cycles to change values for both fuel injectors of one cylinder.

The IMEP graph shows two dips during the late CA50 part. One of the injectors has not fired due to a delay in the communication between the injector and the controlling computer. The problem is described above in the section “Experimental apparatus”.

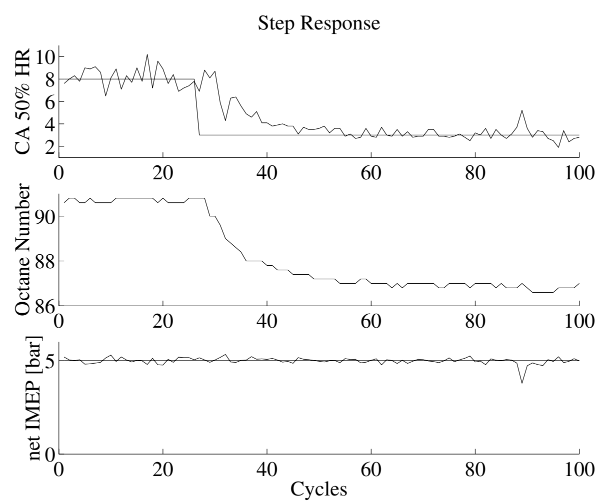

Figure 5: Negative step change of the CA50 set point.

Figure 5 shows the corresponding negative set point change from 8° to 3° ATDC. The same time delay between set point and measured value is seen in this experiment.

Comparing Figure 4 and Figure 5 shows a slower response for the negative step change. The reason for this is the time delay mentioned above in combination with a higher sensitivity at higher octane number and CA50. Higher sensitivity makes the controller more conservative in changing the output. Thus a slower response.

IMEP SET POINT RAMP CHANGE

The control system response to ramp changes of the IMEP set point is shown in Figure 6 and Figure 7. These experiments are conducted on the entire engine, with all timing and IMEP control loops active. Both the negative and the positive ramp changes are conducted at 1500 RPM, with no inlet air heating and a constant CA50 set point of 5 degrees ATDC.

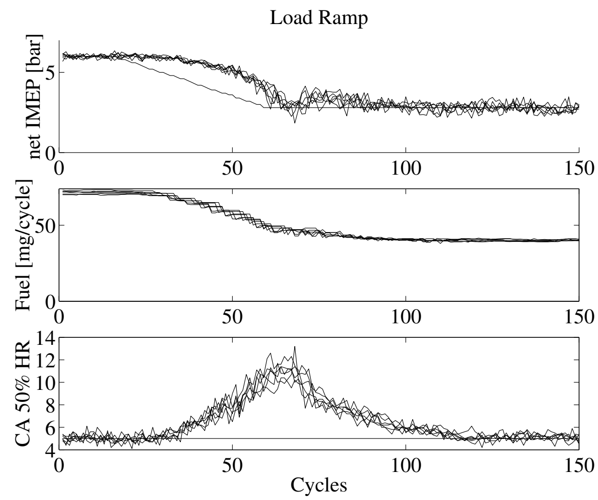

Figure 6: Negative ramp on the load set point.

The timing control loop has some problems with the negative load ramp. These problems are associated with the fact that there is a time delay in the timing control loop. The problem is further accentuated by the increase in sensitivity with late timing. This is also the reason for not requesting a step change in load.

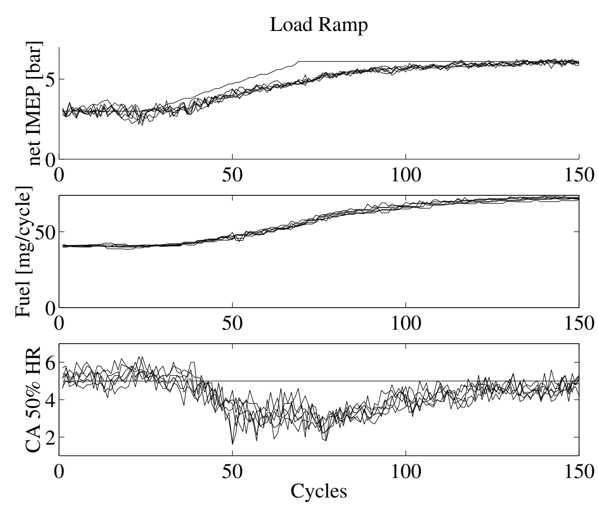

Figure 7: Positive ramp on the load set point.

The timing control is significantly tighter for the positive load change. This is because there is no problem with combustion stability for early timing.

SPEED RAMP CHANGE

Figure 8 to Figure 11 show the control system response to ramp changes of the engine speed. These experiments are conducted on all cylinders with all timing and IMEP control loops active.

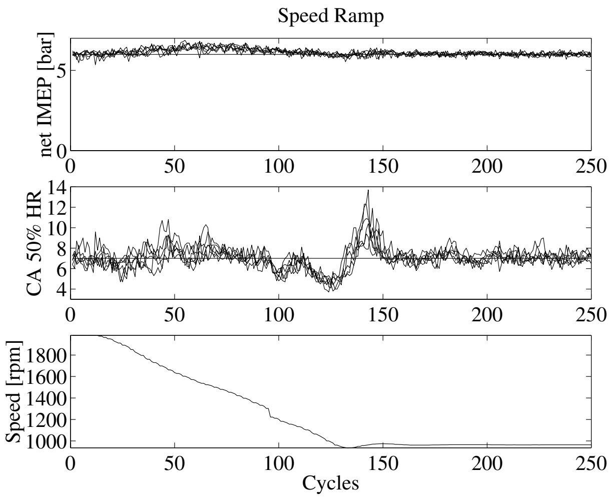

Figure 8: Timing and load response to a negative speed ramp.

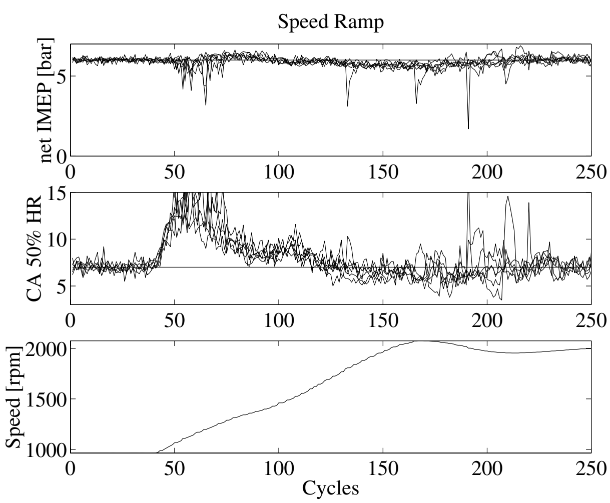

Figure 9: Timing and load response to a positive speed ramp.

Figure 8 and Figure 9 show the IMEP and timing control response to negative and positive speed ramps with no inlet air heating. The most difficult speed ramp is the one where the speed increases, since higher speed requires a faster combustion and, again, there are some problems associated with combustion instability due to the time delay in the timing control loop. Figure 9 also shows some misfiring problems, around cycle 200, caused by communication problems between the computer and the injection controller. Another problem is the change in intake pressure as the speed increases as seen in Figure 10. The sensitivity map used for gain scheduling of the controllers is mainly based on operating points where the intake pressure is close to atmospheric. The estimated sensitivity may therefor not be very accurate at boosted operating points.

Since IMEP is mostly dependent on the total amount of fuel injected per cycle, and depends very little on engine speed and fuel ratio, the speed change causes very little change in IMEP, which is to be expected.

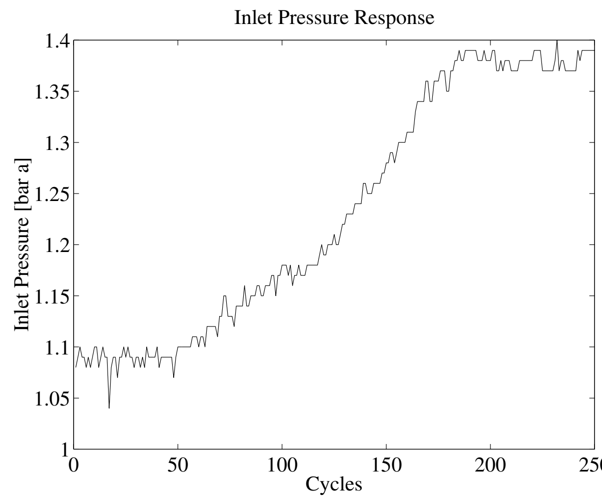

Figure 10: Intake pressure response to a positive speed ramp.

Figure 11 shows the heat control loop response to a negative speed ramp at 2.0 bar net IMEP. The temperature is kept within a few degrees of the 80°C set point as the heat power requirement decreases with the speed.

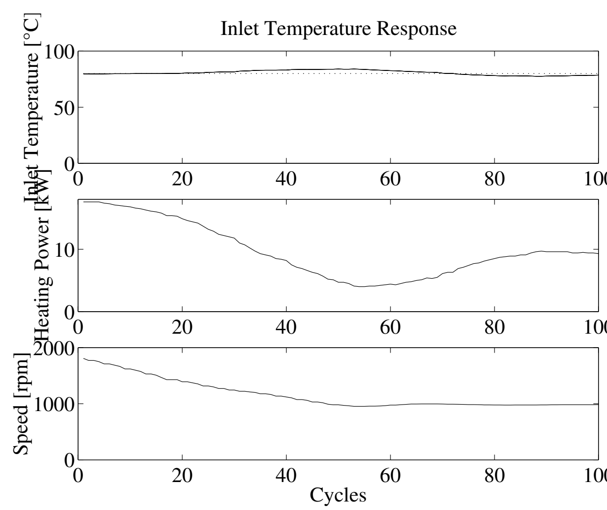

Figure 11: Inlet temperature response to a negative speed ramp.

CONCLUSIONS

The work has shown that it is possible to control an HCCI engine by the use of two fuels. The approach has proven to work for a wide range of operating points: speed between 700 and 2000 rpm and IMEP up to 6.5 bar.

Timing control is very much dependent on an algorithm for estimation of sensitivity that always gives values of the right order of magnitude. The optimization of the timing controller is limited by the quality of the sensitivity map. To avoid instability, the controller gains have to be conservative if sensitivity estimates cannot be trusted.

The bandwidth of the load control loop is limited by the performance of the timing control. With the present controller configuration it is possible to double or half the load in approximately 50 cycles. This could be considerably improved with a better timing controller.

An important property of the control system is the delay between sensor measurements and control computation and between control computation and actuation. The present system has a delay on the input side of about 3 cycles, which is definitely a limiting factor when trying to achieve a fast response.

Calculation of heat release is done on a crank angle window of -30° to 60° ATDC. IMEP calculation uses the complete cycle, but could be modified to work on the same window. This would give 300 crank angle degrees for data acquisition, control computation, and actuation before next cycle starts. This would give no delay and allow for a significant improvement in controller response. To achieve this, data acquisition and fuel injection hardware would have to be improved. The required speed of the controlling computer would not change.

ACKNOWLEDGMENTS

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

CAD	Crank Angle Degree
CA50	Crank Angle of 50% Heat Release
EGR	Exhaust Gas Recirculation
TDC	Top Dead Center
ATDC	After Top Dead Center
GETDC	Gas Exchange Top Dead Center
Net IMEP	Net Indicated Mean Effective Pressure
γ	C_P/C_V
n	Engine Speed
T_i	Inlet Air Temperature
m_f	Injected Fuel Mass
O	Octane Number
θ50, CA50	Angle of 50% Heat Release
S	Combustion timing sensitivity, d(θ50)/d(O)
PIC processor	A low cost one chip computer
FIFO	First In First Out Buffer