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HCCI Operation of a Multi-Cylinder Engine

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Abstract

A six-cylinder truck-size engine has been converted for Homogeneous Charge Compression Ignition (HCCI) operation. This work demonstrates that it is possible to run a multi-cylinder engine under HCCI operation. The ultra-low NO_X characteristic of HCCI is also demonstrated. The sensitivity of combustion timing with respect to operating parameters is investigated, and a closed-loop control system is designed which utilizes the mixing ratio of two different fuels to control the timing of combustion. The performance of the control system is evaluated experimentally.

Introduction

The HCCI engine can be thought of as a hybrid between the two well-known Otto, or Spark-Ignition (SI), engine and the Diesel, or Compression Ignition (CI), engine. In the HCCI concept, a lean mixture of fuel and air is inducted into the cylinder, and is subsequently compressed until the point of auto ignition, which should ideally occur near the Top Dead Center (TDC).

Many studies of HCCI engine combustion [1-14] have been conducted, but most of them have been performed on single-cylinder research engines, which do not provide sufficient torque for dynamometer testing. Stockinger et al. [15] did however demonstrate brake thermal efficiency of 35% on a 4-cylinder 1.6 liter engine at 5 bar Brake Mean Effective Pressure (BMEP). This study demonstrates a brake thermal efficiency exceeding 40% at 5 bar BMEP and a specific NO_X level of approximately 10 mg/kWh.

One of the most difficult problems associated with practical implementation of HCCI is the fact that it lacks a direct control of combustion timing. SI and CI engines each have direct control of combustion timing through spark timing and fuel injection timing respectively, whereas for HCCI engines, combustion timing is a complicated function of equivalence ratio, fuel octane number, residual gas fraction, Exhaust Gas Recirculation (EGR), intake temperature, compression ratio, valve timing, and intake pressure. At some operating points the HCCI engine is even unstable, in the sense that with constant operating parameters the combustion event drifts towards progressively earlier combustion causing hotter combustion chamber walls, which in turn makes combustion occur even earlier. Hence, some kind of closed-loop control of combustion timing is necessary for stable HCCI operation at arbitrary operating points.

This study devises a control strategy for closed-loop control of combustion timing on an HCCI engine using two fuels with different octane ratings. The control input is the mixing ratio of the fuels, which will determine the effective octane rating of the fuel mixture, which has a strong influence on the combustion timing. Pressure sensors mounted in all six cylinders provide cylinder-individual feedback. Simple heat release analysis is performed based on the pressure measurements, and the crank angle of 50% heat release (CA50) is determined. CA50 is the quantitative feedback used for the combustion timing control. The feasibility of closed-loop combustion-timing control of the HCCI process is demonstrated, and the importance of eliminating time delays in the system is noted.

Experimental Apparatus

The test engine 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.

The turbo charger, which is designed for diesel operation, produces very little boost at the low exhaust temperatures prevalent in HCCI combustion. Future studies, with a significantly smaller exhaust turbine, will explore the possibilities of high load HCCI operation. This will of course be at the expense of high exhaust back pressure, resulting in pumping losses.

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 TDC and the fuel spray is directed along the ports, a very fast response to changes in fuel amounts is achieved.

Custom-made injector drivers control injection timing and duration. The controlling computer sends fuel injection commands in terms of timing and duration over a parallel port to the injector drivers, which execute the commands.

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 can be controlled continuously between 0 and 37 kW. A charge-air cooler is also present for cooling the intake air at higher loads.

Figure 1 shows a schematic of the engine, and

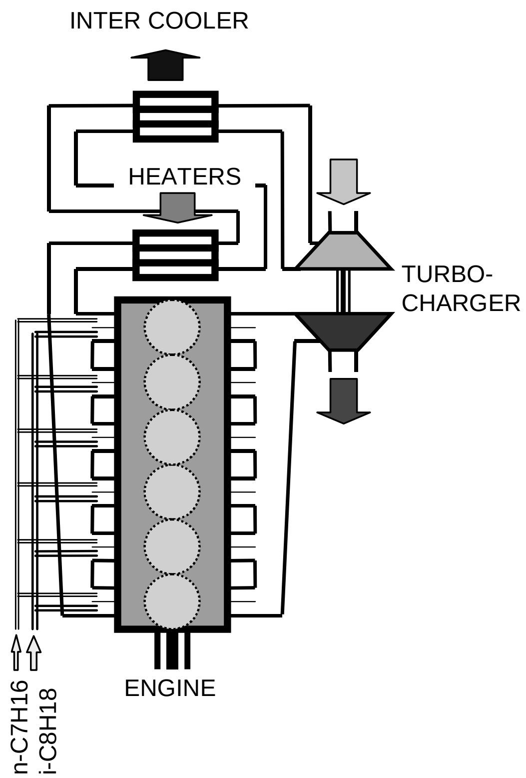

Figure 1: Schematic of the engine converted for closed-loop HCCI operation.

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

Displacement volume	11 705 cm^3
Compression Ratio	18:1
Bore	127 mm
Stroke	154 mm
Connecting Rod	255 mm
Exhaust Valve Open	82° BBDC
Exhaust Valve Close	38° ATDC
Intake Valve Open	39° BTDC
Intake Valve Close	63° ABDC

Performance and Emissions

The performance in terms of brake thermal efficiency and specific NO_X of the HCCI engine is presented in Figure 2 and Figure 3. A wide range of engine speeds and loads are included in the experiments, and for each operating point the combustion timing is optimized with respect to efficiency by adjusting the octane number of the fuel mixture. Most of the experiments are performed without inlet air pre-heating, but for 1000 and 1500 RPM slight pre-heating of the inlet air is included also for comparison.

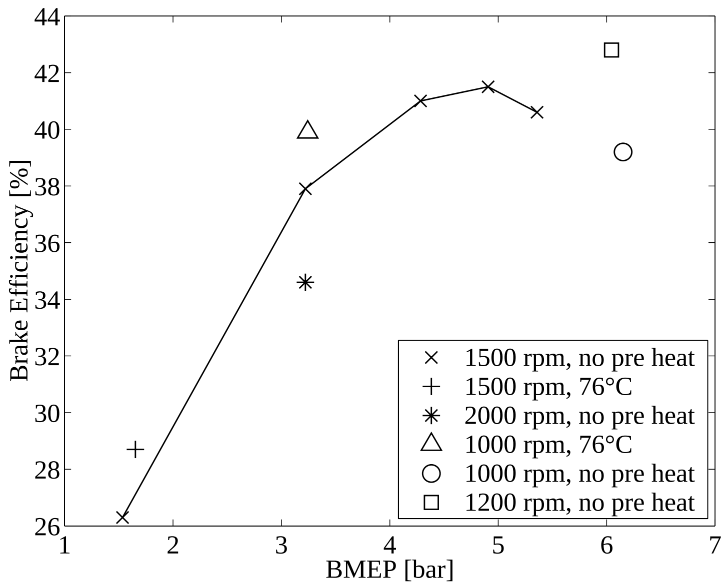

Figure 2: Steady-state brake thermal efficiency versus load (BMEP).

Using the mid-range engine speed of 1500 RPM as bench mark, Figure 2 shows that the efficiency peaks at approximately 5 bar BMEP. The drop in efficiency for lower loads is due to poor combustion efficiency caused by very low combustion temperature. At the lowest loads, the equivalence ratio is only 0.17. The efficiency drop for higher loads is due to increased heat losses to the combustion chamber walls, caused by the increase in peak temperature.

Using again the bench mark of 1500 RPM, Figure 3 shows that the specific NO_X is essentially zero up to 5 bar BMEP, where a rapid increase with load is observed. This also, has to do with the increase in combustion temperature with load. Below 2100 K the rate of NO_X production in an engine is essentially zero, but increases rapidly with temperature above this threshold [16].

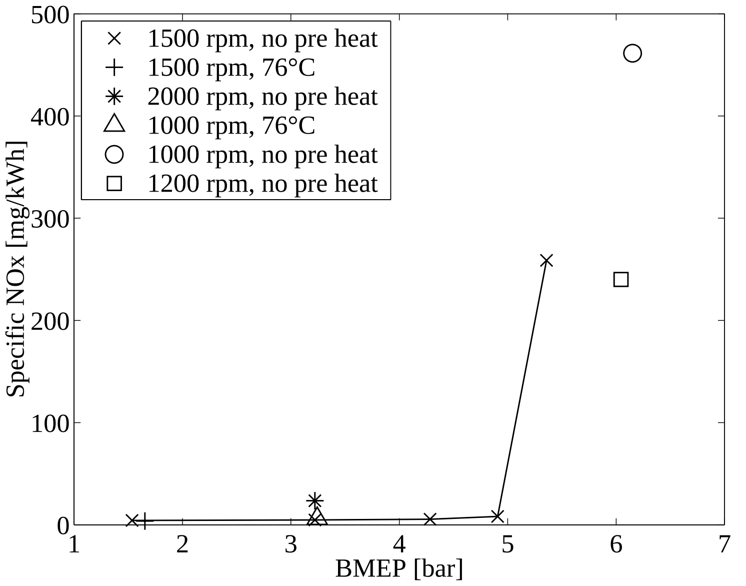

Figure 3: Specific NO_X versus load (BMEP).

The load limit of approximately 6 bar BMEP is imposed by the increasing rate of combustion at higher equivalence ratios. This rate increase causes peak cylinder pressures above the rated values for the engine. The peak temperature also increases, which results in higher NO_X emissions and heat losses to the combustion chamber walls.

Control System Performance

The control system consists of six independent PID controllers for cylinder-individual combustion timing control, six independent PID controllers for cylinder-individual load control, and one PID controller for inlet air temperature control.

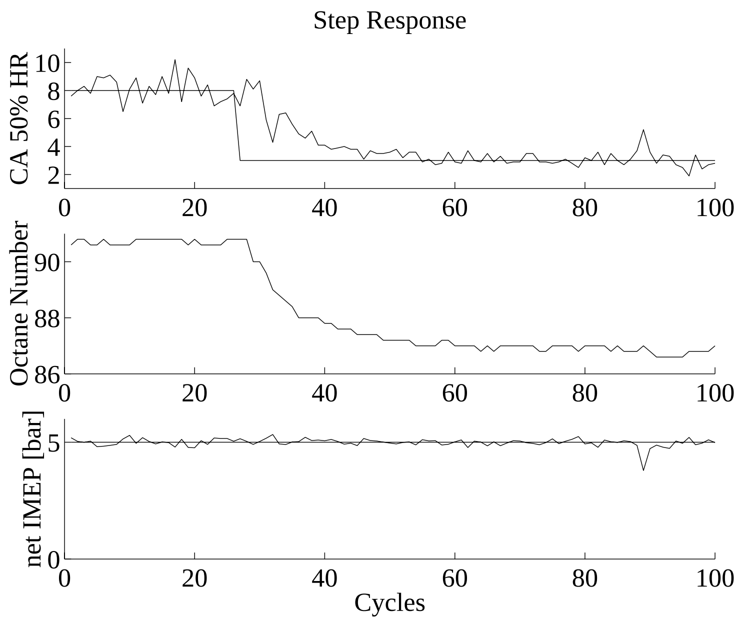

Figure 4: Control system response to a negative step in the CA50 set point. Simultaneous control of combustion timing (CA50) and load (IMEP). Experiment conducted at 1500 RPM, 5.0 bar IMEP.

Figure 4 shows the control system response to a change in the CA50 set point from 8° to 3° after TDC. A time delay of 4-5 cycles is observed. This delay has to do with cylinder pressure acquisition and communication of fuel commands to the fuel injectors. The delay limits the bandwidth of the closed-loop system, but it is seen that the correct steady-state CA50 is reached in approximately 30 cycles.

In Figure 5, the control system response to a ramp increase in engine load is shown. Here, simultaneous control of combustion timing and load is active on all six cylinders. It is seen that the combustion timing can be kept within a few crank angle degrees from the set point of 5° after TDC throughout the load change. The slope of the load ramp is limited by the closed-loop bandwidth of the combustion timing control loops.

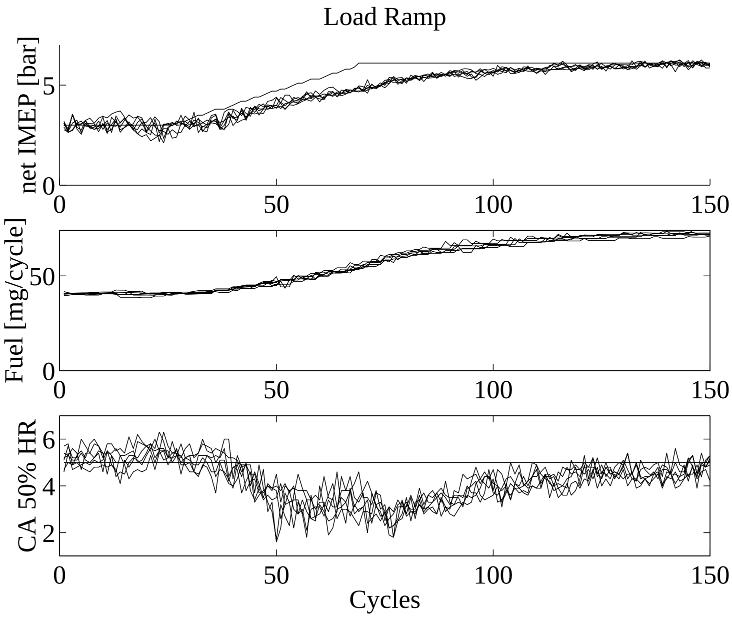

Figure 5: Control system response to a load ramp. Simultaneous control of combustion timing (CA50) and load (IMEP) on all six cylinders. Experiment conducted at 1500 RPM.

Conclusions and Future Work

This study demonstrates the possibility of operating a heavy-duty multi-cylinder engine under HCCI with high brake thermal efficiency (>40%) and low NO_X emissions (~10 mg/kWh). The need for closed-loop control of the combustion process is indicated by the experiments, and a strategy is devised and implemented. The choice of dual-fuel as control input is chosen because of its relative ease of implementation compared to other possible choices. Also, it provides a very strong control authority on the combustion timing. More practical solutions, which will be investigated in future work, include variable compression ratio, variable valve timing, and controlled EGR.

Acquisition of cylinder pressure data and communication of fuel commands to injectors has to be optimized in terms of time between measurement and control action. In the present configuration, buffering of cylinder pressure measurements and synchronization with fuel injector controllers causes a delay of 4-5 cycles. This delay severely limits the bandwidth of the closed-loop system.

The engine loads demonstrated in this study are fairly low (<6 bar BMEP). More recent unpublished studies indicate that this can be extended to as much as 17 bar BMEP with a better-suited turbo charger and a different choice of high-octane fuel.

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Sydkraft, the Swedish National Energy Administration (STEM), and the Center of Competence of Combustion Processes have sponsored this research. The engine has been borrowed from Scania AB. The technicians at the Combustion Engines Division of the Lund Institute of Technology have contributed by doing the major part of the physical setup of the engine.

Abbreviations

HCCI	Homogeneous Charge Compression Ignition
SI	Spark Ignition
CI	Compression Ignition
TDC	Top Dead Center
BMEP	Brake Mean Effective Pressure
CA50	Crank Angle of 50% heat release
IMEP	Indicated Mean Effective Pressure
EGR	Exhaust Gas Recirculation

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