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Negative valve overlap peak pressure based in-cycle control for HCCI combustion using direct water injection

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Abstract

Homogeneous Charge Compression Ignition (HCCI), is a low temperature combustion method, which can significantly reduce nitrogen oxides (NO_x) emissions compared to current lean-burn spark ignition engines. The lack of direct ignition control leads to high cyclic variation with HCCI combustion. A fully variable electromagnetic valve train is used to provide the required thermal energy for HCCI through internal exhaust gas recirculation (EGR) using negative valve overlap (NVO). This leads to an increase in the cyclic coupling as residual gas and unburnt fuel is transferred between cycles through EGR. To improve combustion stability an experimentally validated feed-forward water injection controller is presented. Utilizing the low latency and rapid calculation rate of a Field Programmable Gate Array (FPGA) a real-time calculation of the gas exchange process and water injection controller is implemented on a prototyping engine controller. The developed and experimentally tested controller relates the upcoming combustion phasing to the peak NVO pressure. This control strategy aims to prevent the early rapid combustion following combustion during the NVO period by using direct water injection to cool the cylinder charge to retard combustion phasing back to the desired operating condition. The cooling provided by the direct water injection can also counter the additional thermal energy from any residual fuel that burnt during the NVO period. This control strategy aims to de-couple subsequent cycles by preventing early combustion cycles. The controller was experimentally tested and showed significant improvement in reducing the overshoot of indicated mean effective pressure (IMEP). The experimental testing also showed slight improvement in the combustion stability as shown by a reduction in the standard deviation of IMEP and reduced pressure rise rates.

1 Introduction

Homogenous Charge Compression Ignition (HCCI) is characterized by lean low-temperature combustion (LTC) with temperatures below the NO_x formation temperature. This gives HCCI the potential to significantly reduce NO_x emissions by up to 99% [1]. Therefore, exhaust aftertreatment systems can be reduced or simplified leading to reduced backpressure and further benefits [2]. Rapid global combustion in combination with reduced wall heat losses from LTC provides HCCI with fuel efficency benifits comparable with current stratified lean-burn combustion. The potential of HCCI combustion has been proven in numerous research projects[3, 5].

However, HCCI combustion use in production applications has been limited due to high combustion variability and a narrow operating range. HCCI combustion is enabled by compression induced autoignition of the premixed charge, which is highly dependent on the cylinder state after compression. For some operating conditions, the high variablity of HCCI results from the the lack of a direct ignition control method like spark timing in spark ignition (SI) engines or injection timing in traditional compression ignition (CI) engines. This has caused a strong research focus on control strategies for HCCI combustion timing. When using exhaust gas recircultation (EGR) as a means to provide the thermal energy required to achieve autoignition, a strong coupling between cycles can exist [6]. Consequently, the tendency for unstable combustion sequences increases, which proves to be problematic especially during engine transients and near the

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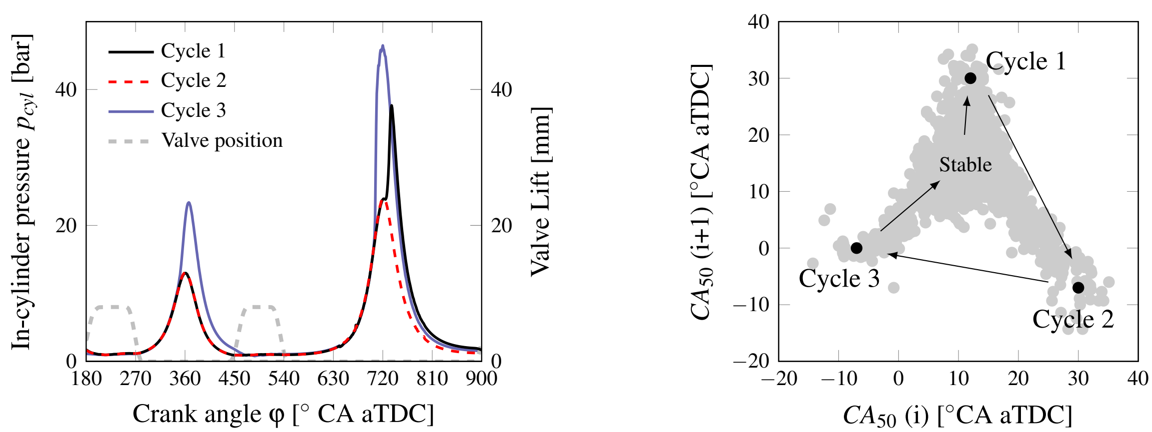
misfire limit. Distinct cyclic variations, characterized by a spontaneous shift from stable to unstable operation, has been investigated in [7–9]. To improve the stability of HCCI combustion these cycles must be decoupled [10].

Figure 1a shows the experimentally measured cyclic pressure signals of three consecutive cycles. Cycle 1 is a good representation of a standard cycle with a normal combustion phasing of 12° CA aTDC. It is then followed by cycle 2 which can be considered a incomplete combustion with a very late combustion phasing. Then, due to the incomplete combustion, residual fuel is transferred to the next cycle through internal EGR. As the combustion phasing is very late, the in-cylinder temperature increases which increases the temperature of the exhaust gas transferred to cycle 3. There is also the possibility that during the negative valve overlap (NVO) recompression a portion of the residual fuel ignites (as seen in cycle 3) and leads to a further temperature increase of the residual exhaust gas. The result is an increase in the temperature of the fresh air charge and the temperature after compression. This leads to an early combustion phasing with a high pressure rise rate (PRR). An early combustion phasing is not desired as the high PRR leads to increased combustion noise and possible engine damage [11][12]. Overall, high cyclic variation of combustion also tends to reduce thermal efficiency and increase raw engine out emissions [13].

The combustion phasing, CA_{50}, gives the angle of 50% heat released in the combustion process. Variation in CA_{50} is representative of the stability of the the combustion process. A return map is used to show the relationship between the combustion phasing of the current cycle, $CA_{50}(i)$, and of the following one, $CA_{50}(i+1)$. In stable operation, two consecutive cycles are not correlated, the return map shows random scatter around the combustion phasing mean. The spread of the data points represents the stochastic variation from cycle to cycle [6]. However, when a distinct pattern or branching can be seen on the return map as is the case in Figure 1b, a direct coupling between cycles exists. To effectively stabilize combustion the spread of the data points and distinct 'V' should be reduced.



(a) Cyclic variation in the in-cylinder pressure p_{cyl} trace

(b) Corresponding combustion phasing CA_{50} return map

Figure 1: Distinct HCCI cyclic variation

Curently, research using water injection in conventional SI combustion has focused on its potential as a method to reduce fuel consumption [14], reduce NO_X emissions and to enhance performance due to knock reduction [15]. Similar to SI combustion, water injection in HCCI combustion provided a significant increase in achievable engine loads [16, 17] and a decrease in NO_X emissions [18, 19]. However, these tests are in steady state operation, while this work will focus on using water injection to improve HCCI combustion stability by attempting to decouple subsequent cycles. A developed and experimentally validated in-cycle feed-forward control algorithm utilizing direct water injection will be shown in the next sections.

2 Experimental setup

In this work, a single cylinder research engine (SCRE) outfitted with a fully variable electro-magnetic valve train (EMVT) is used to test developed water injection controller. The flexibility of the valve timing allows for engine operation with various valve strategies, however, for this work only combustion chamber exhaust gas re-circulation through NVO will be used. This valve strategy is used to retain some of the exhaust gas in the cylinder which provides the thermal energy for the autoignition process to begin. The piston used in the engine has enlarged valve pockets to provide free running operation at with any valve timing, this however limits the in-cylinder mixing compared to bowl shaped pistons.

Fuel is injected into the SCRE through an piezoelectric outward-opening hollow cone injector. Conventional European Research Octane Number (RON) 96 gasoline containing 10% ethanol is used for all tests in this work and the fuel pressure maintained at 100 bar. For the injection of distilled water directly into the combustion chamber an identical injector is used. An injection pressure of 50 bar is used throughout this work at it ensures evaporation of the water while still allowing small water injection amounts. The engine geometry and testing conditions are listed in Table 1.

Table 1: Single cylinder research engine parameters

Parameter	Value
Displacement volume	0.499 L
Stroke	90 mm
Bore	84 mm
Compression ratio	12:1
Intake / exhaust pressure	1013 mbar
Oil and coolant temperature	90 °C
Engine speed	1500 rpm
Fuel rail pressure	100 bar
Intake temperature	50 °C

In-cylinder pressure is measured using a Kistler 6041 piezoelectric pressure transducer. The intake and exhaust manifold pressures are measured using Kistler 4045-A5 piezoresistive pressure transducers. The position of the valves are measured using FEV Europe GmbH (FEV) conductive lift measurement sensors. The angular position of the crank is measured using a 0.1° CA resolution optical encoder. All of these signal are simultaneously input to both the Combustion Analysis System (CAS) and prototyping engine control unit (ECU).

For offline analysis an FEV CAS is used to record the cylinder and manifold pressures and valve position at a 0.1° CA resolution. The ECU contains a microprocessor and FPGA board which are used for the real-time gas exchange calculation and engine control. Both injectors and the EMVT system are directly controlled by the FPGA board allowing for rapid control intervention. The control algorithms that are developed are run on the FPGA board along with existing calculations with a sample time of 12.5 ns on the FPGA used. These additional calculations include a thermodynamic zero point correction for cylinder pressure referencing. Along with a cylinder pressure indication system and angle calculation module as presented by Pfluger et al [20]. The processor in the ECU has a sample time which is set to 0.5 ms. Details of the MicroAutoBox II (MABX) prototyping ECU can be found in [10, 21].

3 Controller design

The gas exchange model previously developed in [21] is used to provide a calculation of cylinder pressure. This is used instead of measured pressure to reduce measurement noise and provide disturbance rejection. The model is capable of calculating the cylinder state in 0.1 ° CA when using the FPGA for the calculation.

To decouple consecutive cycles as seen in Figure 1, the increased pressure during NVO is used to represent the combustion of residual fuel burning during the NVO period. This burning fuel increases the temperature of the residual gas transferred to the upcoming cycle leading to an advanced combustion phasing and increased pressures. Direct water injection immediately after intake valve closing (580° CA aTDC) will then be used to help reduce the increased cylinder temperature caused by the fuel burnt during the NVO period.

The relationship between peak in-cylinder pressure during NVO and the combustion phasing of the subsequent cycle is shown in Figure 2. This correlation shows a distinct relationship between cycles with peak NVO pressure above 14 bar and early combustion phasing. This is the expected trend and can be seen as cycle 3 in Figure 1a. There is a large variation of the combustion phasing when the cylinder pressure is between 12 and 14 bar, therefore only the early cycles provide a prediction of an early combustion phasing. These cycles are separated from the main point cloud as shown by the black points. These selected cycles are fitted with a linear regression, shown in red, to produce a prediction of the upcoming CA_{50} based on the NVO pressure.

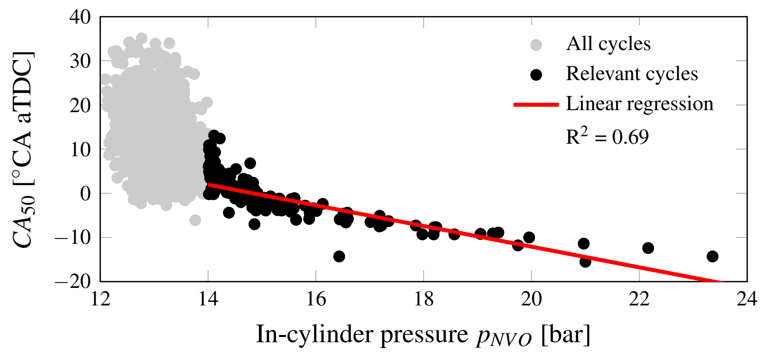


Figure 2: Linear regression of combustion phasing CA_{50} based on max cylinder pressure during NVO recompession, P_{NVO}.

Then using the relationship between the peak pressure during NVO and a predicted combustion phasing of the upcoming cycle the amount of water injection can be predicted. This feedforward controller is then implemented on the FPGA and experimentally validated.

5. Results and discussions

The proposed control strategy is tested on the SCRE, where 1000 consecutive cycles are collected and the water injection controller is activated after the first 500 cycles. The impact of the controller on IMEP can be seen in Figure 3, where the reduction in IMEP overshoot can be seen by the decrease in cycles with an IMEP > 5 bar from 8 to 1. The water injection controller is able to effectively prevent the overshoot in IMEP caused by the early rapid combustion after an incomplete combustion by cooling the exhaust gas transferred between cycles. The cycles with below average IMEP remain as the use of water can not advance the combustion phasing and these cycles are not the target of the proposed control strategy.

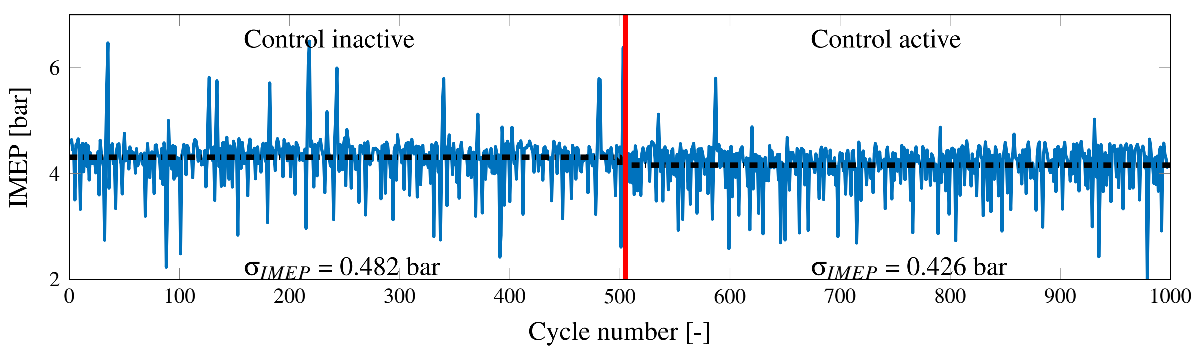


Figure 3: IMEP change due to in-cycle control. Controller is activated after 500 cycles.

Along with the improvement in IMEP shown a similar improvement could also be seen in PRR as shown in Figure 4. Here the average PRR is reduced by 6.3% but more importantly the peak PRR over 500 cycles is reduced from 7.97 to 7.02 bar/°CA. The reduction in both average and peak PRR is desirable as operating with reduced pressure rise rates leads to reduced engine noise and wear.

The main goal of the presented controller was to improve HCCI combustion stability by de-coupling subsequent cycles as presented in Figure 1. The effect of the water injection controller on the combustion phasing return map can be seen in Figure 5. Here the blue colored points are directly influenced by the water injection controller in cycle $i+1$. The combustion phasing in the current cycle is not affected but the combustion phasing in the following cycle is retarded by

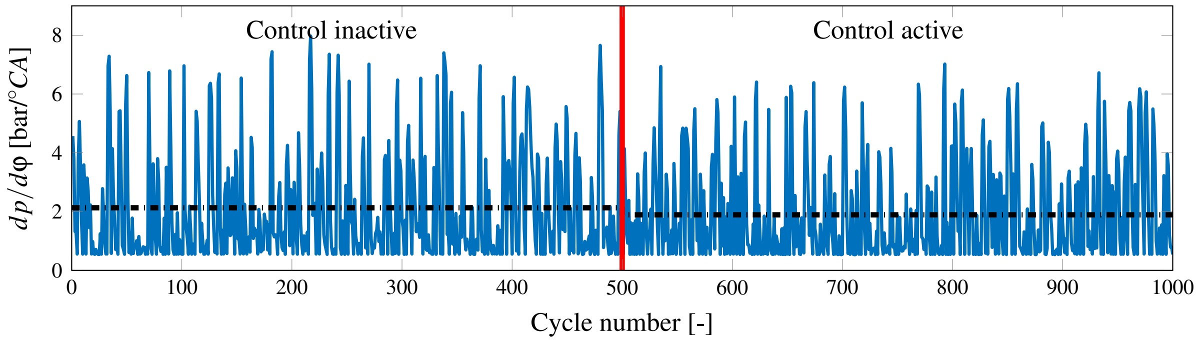


Figure 4: Peak pressure rise rate (PRR) change due to in-cycle control. Controller is activated after 500 cycles.

water injection seen as a upward vertical shift in the data points in the lower right leg of the figure. The water injection directly reduces the right leg of the return map, while the reduction in the left leg is due to the prevention of an early combustion phasing following a cycle that also has an early combustion. Using Figure 1b to clarify further, the controller would prevent cycle 2 which would also prevent cycle 3 by shifting the combustion phasing back into the stable central area. As the controller only injected water in 77 of the 500 cycles the remaining cycles which were not directly affected by the controller (dark gray data points) also show a reduction in their spread in the far ends of the return map. Overall, reducing the spread of the CA_{50} return map indicates that with the controllers activated an improvement in combustion stability is achieved.

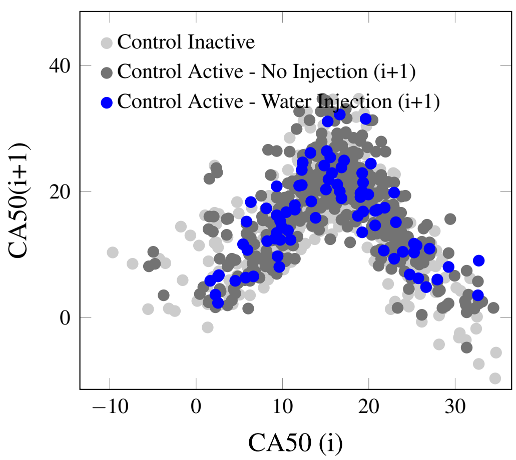


Figure 5: CA50 return map improvement

Figure 5 also shows that the controller interacted in cycles which were not the target of this control strategy. Cycles where an average combustion phasing is followed by another average cycle (located in the central area of the return map) also triggered the controller for water injection. This effect can be attributed to the relatively large cyclic variability in combustion phasing associated with NVO pressure between 14 and 15 bar as shown in Figure 2. These cycles can have either an early combustion phasing or normal combustion phasing and still have the same NVO pressure.

There are still some very early combustion phasing cycles (dark gray in the lower right of the return map), these cycles could not be detected by the controller as there was no combustion in the NVO recompression, however, there was a late combustion in the previous cycle that lead to hotter exhaust gasses being trapped and transferred to the subsequent cycle. Another possibility is that combustion is late in the NVO period leading to a pressure rise that was countered by the increasing cylinder volume. Taking the peak pressure during the NVO recompression is a limitation of the method as information is needed to completely capture the in-cylinder state.

4 Conclusions

The proposed controller showed a improvement in the combustion stability of HCCI as the controller was able to decouple some cycles and reduce the IMEP overshoot. However, the improvement in the standard deviation of IMEP and average pressure rise rate was modest. The controller was also unable to accurately predict the upcoming combustion phasing for all cycles and injected water in cycles where it was not required. This control strategy shows that in-cycle control is possible, however, only considering the in-cylinder pressure does not provide enough information about the in-cylinder state to accurately predict an early combustion phasing every cycle.

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