HCCI Combustion Stability Improvement Using a Rapid Ignition System

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Summary

When compared to traditional spark ignition engines, homogeneous charge compression ignition (HCCI) has the potential to significantly reduce NO_x raw emissions. Since HCCI does not provide direct control of the combustion timing it can suffer from high cyclic variability. With Spark Assisted Compression Ignition (SACI), it has been demonstrated that misfires can be reduced but with increased NO_x emissions. To merge the advantages of the HCCI and the SACI combustion processes, a Field Programmable Gate Array based calclation of the gas exchange process is combined with a rapid spark system. A feedforward controller based on the cylinder pressure or heat release is developed to determine if a spark is required to assist the HCCI combustion process. This paper presents the development and experimental validation of the controller on a single cylinder research engine. With the new system, the combustion stability can be significantly improved, shown by a reduced standard deviation of the indicated mean effective pressure and a reduction of cyclic variation.

1 Introduction

Homogenous charge compression ignition (HCCI), is a part-load combustion method, which is characterized by lean low-temperature combustion (LTC) with temperatures below the oxides of Nitrogen (NO_x) formation temperature of 2000 K. As a result, HCCI has the potential to significantly reduce NO_x emissions, while maintaining a high fuel efficiency, comparable with current stratified lean-burn combustion, with NO_x emissions reduced by up to 99 % [1], [2]. Therefore, expensive exhaust aftertreatment systems can be reduced or simplified [3]. Improved efficiency is also possible when using HCCI combustion due to the rapid global and spatial combustion in combination with reduced wall heat losses from the LTC. The potential of HCCI combustion has been proven in numerous research projects [3]–[5].

Near the misfire limit, 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 [5],[6]. Consequently, the tendency for unstable combustion sequences increases, which proves to be problematic especially during engine transients and near the misfire limit. Distinct cyclic variations, characterized by a spontaneous shift from stable to unstable operation, has been investigated in [7], [8]. Various control interactions have been explored to reduce this cyclic coupling and extend the HCCI operating range but with varying success.

One of the main challenges with HCCI is the increased unburnt hydrocarbon (uHC) and carbon monoxide (CO) emissions that are a result of both cool combustion temperatures, rapid combustion and complete or partial misfires where little or no fuel is burnt. This limitation of HCCI can be reduced by the addition of a supporting spark, however, this supporting spark increases the local combustion temperature causing an increase in NOx emissions. The combination of spark ignition leading to HCCI autoignition in the end-gas is referred to as spark assisted compression ignition (SACI) which provides the ability both extend the load range of HCCI. However, the use of SACI leads to reduced combustion stability compared to HCCI[9],[10].

In this work, a rapid ignition system (RIS) will be controlled using a real-time gas exchange model running on a field programmable gate array (FPGA) to provide a spark only on cycles with late combustion phasing. The model is able to calculate the cylinder state within 0.1 crank angle degrees (° CA) and using the RIS a spark can be introduced within 5-10 s to help prevent misfire cycles while limiting the number of cycles which require spark.

2 Experimental setup

A single cylinder research engine (SCRE) outfitted with a fully variable electro-magnetic valve train (EMVT) is used to validate the online FPGA gas exchange model. The EMVT is controlled using the FPGA board which is part of the engine control unit (ECU). The flexibility of the valve timing allows for engine operation with combustion chamber exhaust gas re-circulation through negative valve overlap (NVO). The fuel injector is a piezoelectric outward-opening hollow cone injector that is controlled using the FPGA. The fuel used for all testing in this work is conventional European Research Octane Number (RON) 96 gasoline containing 10 % ethanol. The engine speed is controled to 1500 1/min for all tests.

An FEV Combustion Analysis System (CAS) is used to record cylinder pressure at a 0.1° CA resolution for offline analysis. A microprocessor and FPGA board are contained in the dSPACE MicroAutoBox II prototyping ECU which is used for the real-time gas exchange calculation and engine control. Both injectors along with the EMVT set points are directly controlled by the FPGA board allowing for rapid control intervention. Full engine specifications and a detailed description of the ECU can be found in [11],[14].

The rapid ignition system (RIS) is designed for providing the ability to have a spark begin immediately after the ignition trigger signal is received. To achieve such a fast response, a high-frequency current system is utilized. Traditional ignition systems utilize stored magnetic energy which is charged to the coil to provide a spark between the gap of the plug. However, these conventional ignition systems require significant charging time (typically 3-10 ms) when compared to the desired control action speed as further discussed in the following sections. In the RIS, the high frequency current supplied to the primary coil from the power supply unit generates an alternating magnetic field, and the field yields alternating high voltage on the secondary coil immediately. Thus the RIS does not require charging time, allowing for the spark to begin within approximately 5-10 s, which is significantly shorter than that of conventional systems, and fits to the proposed control strategy.

3 Control strategy and motivation

To improve HCCI combustion stability, large cyclic variations need to be reduced. Figure 1 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 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. An early combustion phasing is not desired as the high pressure rise rate leads to increased combustion noise and possible engine damage [12]. Overall, high cyclic variation of combustion also tends to reduce thermal efficiency and increase exhaust emissions [13].

Figure 2 shows the return map for the combustion phasing, CA_{50}. 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 so the return map would show random scatter around the combustion phasing mean. The spread of the data points represents the stochastic variation from cycle to cycle [5]. However, when a distinct pattern or branching can be seen on the return map as is the case in Figure 2, a direct coupling between cycles exists. To effectively stabilize combustion the spread of the data points and distinct 'V' should be reduced.

In previous work the cyclic variation of HCCI has been reduced by preventing the early combustion following a misfire (cycle 3 in Figure 1) by using direct water injection to cool the trapped exhaust gas to retard combustion phasing back to the desired value[14], [15]. However, that leaves cycle 2 unaffected and the engine experiences a misfire leading to higher emissions and lower efficiency. The use of water injection adds the requirement for a second injection system and the requirement for a high quality water source for implementation in production vehicles [16]. To overcome these limitations the use of a spark interaction during cycle 2 in Figure 1 is presented in this work. This supporting spark will be used only when necessary and helps preserve the NO_x benefits of HCCI combustion by keeping the combustion temperature low.

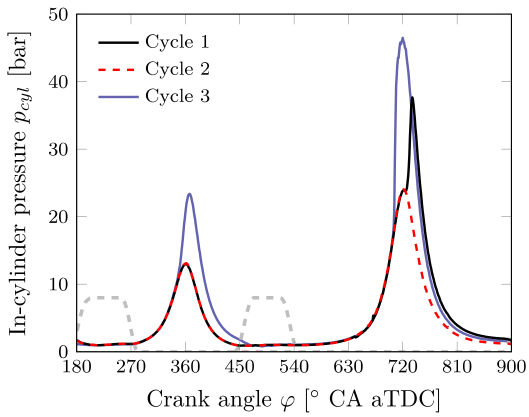

Fig. 1: Cyclic variation in the in-cylinder pressure at n = 1500 1/min, IMEP = 4.0 bar

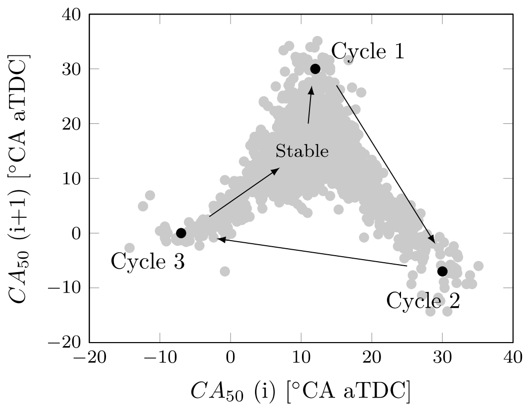

Fig. 2: Return map of combustion phasing, CA_50 at n = 1500 1/min, IMEP = 4.0 bar

4 Pressure based control strategy

The high cyclic variability of HCCI combustion can be seen when examining the range of in-cylinder pressure of 200 cycles at a specified crank angle as seen in Figure 3. In this figure, the motoring pressure (cylinder pressure without combustion) can be seen where all the cycles overlap when late combustion occurs. This difference in measured cylinder pressure compared to the baseline motoring pressure provides a method to determine if auto-ignition has started.

The large variation in cylinder pressure presented in Figure 3 shows that at a given crank angle there is a large variation in cylinder pressure. By selecting a specific CA and then examining the correlation of the in-cylinder pressure with the combustion phasing of that cycle results in Figure 4. When considering the correlation between cylinder pressure and the combustion phasing of that cycle at an engine angle before TDC (-10 or -5° CA aTDC) is it very difficult to observe anything other than cycles with very early combustion phasings. This is not useful for spark control as it is nessary to distinguish late cycles from early ones. However, when considering the cylinder pressure after TDC a clear correlatation between combustion phasing and cylinder pressure can be seen. This has limited applications as in order for the spark to have the maximum impact on the combustion we want it as early as possible. At 5° CA aTDC a clear distinction between cycles with a combustion phasing later than 10° CA aTDC and earlier than 10° CA aTDC can be made. These are the cycles that are considered as having a late combustion phasing and should be supported with a spark.

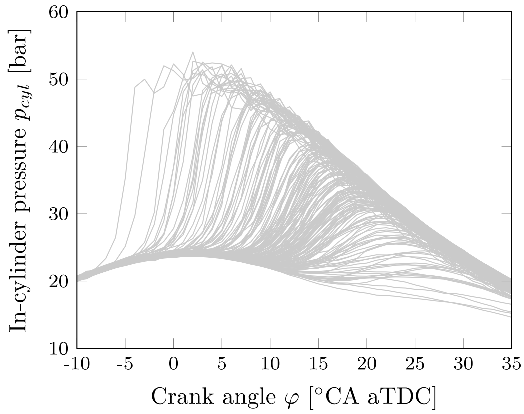

Fig. 3: Cyclic variability in the cylinder pressure during HCCI combustion at n = 1500 1/min, IMEP = 4.0 bar

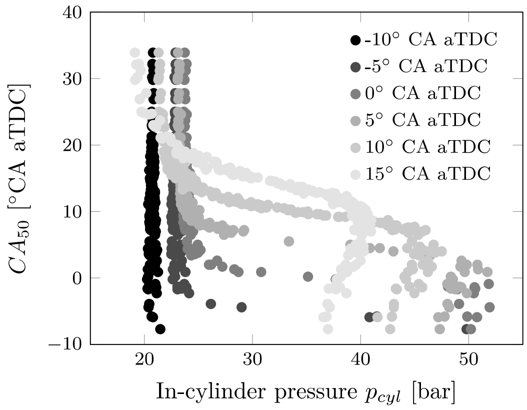

Fig. 4: Correlation of combustion phasing and cylinder pressure at a specified crank angle. n = 1500 1/min, IMEP = 4.0 bar

At 5° CA aTDC if the cylinder pressure has not exceeded 26 bar it is assumed that the combustion will be late or misfire and a spark is required. There will be some cycles when the spark will be activated even though the ignition timing is only slightly retarded at around 12° CA aTDC, however, it is important to keep the spark as early as possible to ensure it has enough time for the flame to propagate through the fuel/air mixture.

5 Heat release based control strategy

The increase in cylinder pressure is due to the start of the combustion process, which can also be shown using the accumulated heat release. The advantage of using heat release over just measured pressure is that a very large change in value is observed at the start of the combustion process. Therefore, the effect of signal noise can be minimized. A similar result to the pressure based correlation (Figure 4) is found when the heat release at a specified angle aTDC is correlated to the combustion phasing of that cycle.

As in the pressure correlation, 5° CA aTDC is selected as the timing when to check if combustion has begun. If not a spark is activated. In this case, a heat release threshold of 0 J is selected as a physical representation of the start of combustion. This threshold predicts the combustion phasing will be approximately 15° CA aTDC or later.

6 Ignition timing constraints

The above two control strategies are then implemented on the FPGA side of the engine controller as shown schematically in Figure 5. The proposed feedforward control strategy takes the cylinder pressure or cumulative heat release at an engine angle of 5° CA aTDC and compares it to the specified threshold (discussed in the previous section). If it is determined that the combustion has yet to begin the RIS is sent a signal for the spark to begin.

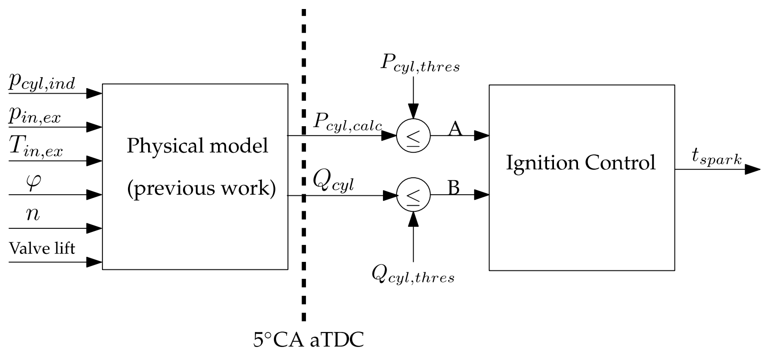

Fig. 5: Feedforward spark control structure

With HCCI combustion the cylinder state is very close to the auto-ignition point of the fuel and air mixture even in cases of late combustion or misfire. Therefore only a small amount of additional energy is required for the auto-ignition process to begin. However, after the piston reaches TDC the expanding cylinder volume lowers the cylinder pressure and increases the amount of additional energy required for combustion to begin. Therefore, it is desirable to have the spark occur as early as possible.

Using the rapid calculation rate of the FPGA allows for the calculation of the heat release (using the existing gas exchange model[11]) and the calculation of the controller output with a delay of only 301 FPGA samples (or 3.76 μs for the FPGA used). When combined with the high speed of the RIS ignition system the spark begins in 5-10 s (0.045-0.09 ° CA at 1500 1/min) after the engine reaches 5° CA aTDC. Therefore when using an FPGA based model and RIS system the spark can occur when the cylinder state is close to auto-ignition and provide the maximum time for the fuel to burn before the exhaust valve opens. Using the proposed control strategy with traditional ignition systems (which require around 3-10 ms for coil charging) would significantly reduce the amount of time for the fuel to burn.

When compared to the existing gas exchange model[11] the proposed controller consumes very little of the available FPGA resources. Using under 1 % of the available flip-flops and look-up tables.

7 Combustion stability

The proposed feedforward controller was tested using both the in-cylinder pressure and heat release as control input on the SCRE with the goal of reducing the cyclic variation shown in Figure 1. The performance of the controller at stabilizing the combustion phasing can be seen in Figure 6. Here 1000 consecutive cycles are presented with the controller being activated after 500 cycles. The variation in combustion phasing after cycle 500 is significantly reduced after the controller is enabled as shown by the reduction in standard deviation from 6.89 to 5.3 ° CA or a reduction of 23 %.

Before the controller is activated there are eight cycles with a combustion phasing later that 30° CA aTDC (considered a misfire). After the controller is activated there are no cycles with a combustion phasing greater than 30° CA aTDC, showing the controller is able to completely eliminate misfire cycles. The number of early combustion phasing cycles have been greatly reduced even though the spark does not have a direct effect on these cycles. This reduction in cycles with a very advanced combustion phasing is due to the fuel being burnt in the desired cycle instead of being transferred to subsequent cycles. This shows that the proposed control method is able to decouple subsequent cycles.

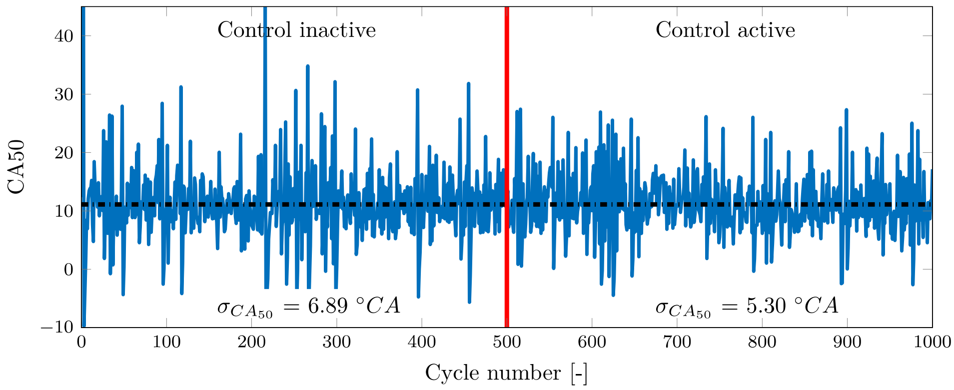

Fig. 6: Combustion phasing, CA_{50}, stability improvement with in-cycle controller based on heat release at 5°CA aTDC. The controller is activated after 500 cycles. Here, n = 1500 1/min, NVO= 133°CA, IMEP = 4.0 bar.

Table 1 presents a summary of the tested control strategies at various operating points. At all of the operating points the controller showed an increase in the average IMEP when the controller is activated. This shows that the control strategy is able to improve the combustion efficiency by delivering more power out of the same fuel. There is very little change in the average combustion phasing which is expected as the controller is preventing both very retarded and advanced combustion phasing cycles. Therefore, there is almost no change of the average combustion phasing which is desired.

Control input	IMEP [bar]		CA_{50} [°CA]		$\Delta\sigma_{IMEP}$ [%]	$\Delta\overline{dp/d\varphi}$ [%]	Control Interactions [cycles]
Controller	Off	On	Off	On			
Pres	2.86	2.89	9.4	9.6	-21.2	-12.9	-1.0
Pres	4.03	4.04	11.0	11.1	-13.8	-2.5	-1.4
HR	2.71	2.72	9.4	9.4	-5.0	-1.9	-1.6
HR	3.92	3.96	11.0	10.8	-25.2	-34.3	-0.8
HR	4.02	4.05	11.1	11.1	-28.9	-23.0	-1.9

Tab. 1: Combustion stability improvement due to feedforward controller. Control input of cylinder pressure (Pres) and heat release (HR) are both shown.

The standard deviation of both IMEP and combustion phasing are significantly reduced when the controller is activated. This large reduction shows that the controller is successfully able to stabilize HCCI combustion which help to provide a smoother running engine. The reduction in peak pressure rise rate is also shown in Table 1. While this improvement is relatively small, the proposed controller does not directly reduce the pressure rise rate and the reduction seen is due to the reduction of unburnt fuel being transferred between cycles and the overall improved combustion stability.

8 Emission benefits

The proposed control strategy is aimed at reducing the uHC emissions related to misfire by providing a spark to help the combustion process begin, while preserving the benefit of not having a spark when not needed. The proposed ignition control strategy was compared with the cases of no spark interaction (HCCI) and having a spark every cycle at 5° CA aTDC (SACI). For each ignition strategy three measurements were performed with each measurement consisting of averaging the emissions produced over 30 s. The results of these tests can be seen in Table 2.

Emission species [ppm]	NO_x	NO	HC	CO
HCCI	29.2±1.79	26.6±0.58	1709±23	2449±105
Controller	29.0±0.04	27.6±0.66	1597±5	2354±13
SACI	33.5±2.72	30.4±1.99	1882±108	2826±74

Tab. 2: Single cylinder research engine parameters

Table 2 shows that the expected increase NO_x and NO emission was observed when the spark was always used. The nitrogen oxide emissions remain constant between the pure HCCI case and the controller operation. This result is expected as the spark is used in approximately 10% of cycles. In the cycles when the spark is used, the combustion occurs very late and as the cylinder volume is increasing which helps to keep the combustion temperature below the NO_x formation temperature.

The unburnt hydrocarbon emissions show that with the controller activated the lowest HC emissions are seen. Then compared the pure HCCI case (no spark) an improvement of 6.6% is achieved. However, the case where the spark occurs every cycle actually has the highest HC emissions. When a spark is always used it is expected that a hotter more complete combustion occurs every cycle, however, the very late spark timing leads a very high cyclic variability[10],[17]. This high cyclic variability leads to unstable combustion and increased HC emissions. Similarly, the carbon monoxide emissions are also increased in the SACI case.

The high cyclic variability of the SACI case as seen by the large standard deviation of the emissions presented. When the spark was used every cycle the combustion stability is decreased compared to the pure HCCI operating condition. This decrease in combustion stability is due to an increase in the fuel burnt through flame propagation compared to autoignition. The large standard deviation is also due to the limited number of tests performed, however, the basic benefit of the proposed control strategy is shown. To better understand the impact to the engine emission output, further statistical research is important.

9 Conclusions and future work

The rapid ignition system was combined with the fast calculation rate of an FPGA based engine controller to experimentally test an ignition controller. This controller utilizes either the cylinder pressure or heat release to determine if the auto-ignition process of HCCI has started. The proposed controller was able to successfully reduce the standard deviation of combustion phasing and IMEP at multiple operating points. The controller required a spark in approximately 10% of cycles at the operating point tested. This lead to nitrogen oxide levels similar to the pure HCCI case while showing an improvement of 6.6% in unburnt hydrocarbon emissions over pure HCCI.

The tests performed showed the potential of the controller to improve HCCI combustion stability, however, to fully understand the emission benefits further testing is required to statistically evaluate the emission improvement of the proposed controller. Combining this controller with direct water injection to help control the thermal energy transferred between cycles could further improve the HCCI combustion stability and is the focus of our future work.

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