EXPERIMENTAL STUDY OF HCCI CYCLIC VARIATIONS FOR LOW-OCTANE PRF FUEL BLENDS

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ABSTRACT

The operating range of Homogeneous Charge Compression Ignition (HCCI) engines is limited by the knock boundary on one side and by high cyclic variations (misfire) on the other side. A challenging problem for HCCI engines is achieving cycle-by-cycle ignition control due to these variations. To control combustion in HCCI engines, it is essential to understand how parameters affect the cyclic variations of HCCI combustion. This paper investigates cyclic variability of HCCI combustion using experimental data collected at 360 operating points from a single cylinder Ricardo engine. The engine is fueled with four different blends of Primary Reference Fuels PRFs (iso-octane and n-heptane) at octane values of 0, 10, 20 and 40 over a wide range of equivalence ratios, intake temperatures, intake pressures, Exhaust Gas Recirculation (EGR) rates, and engine speeds.

The experimental results show there are three main distinct patterns of cyclic variations for combustion peak pressure (P_{max}), Indicated Mean Effective Pressure (IMEP) and ignition timing. These patterns include normal cyclic variations, periodic cyclic variations and cyclic variations with weak/misfired ignitions. The results also show cyclic variation of HCCI combustion is highly dependant on the location of the start of combustion and there is less cyclic variation in the first stage of HCCI combustion compared to that of the second stage for the PRF blends studied.

INTRODUCTION

HCCI has emerged as a promising engine technology with the potential to combine fuel efficiency and improved emissions performance with extremely low levels of NOx and PM emissions[1]. HCCI combustion relies on the auto-ignition of a compressed air-fuel mixture at local hot spots distributed through the mixture [2]. Cyclic variations of HCCI engines are known to be small compared to those in an SI engine, but they can be more significant under certain operating conditions [3]. High cyclic variations limit the range to which HCCI engines can operate and also make it difficult to achieve accurate ignition timing control, which is the most challenging problem in HCCI engines [4]. Characteristics of these cyclic variations need to be understood in order to extend the HCCI operating limits and to achieve better ignition timing control.

Few experimental studies [3, 5, 6, 7] have been done on the cyclic variations of HCCI engines compared to those of SI (Spark Ignition) engines. These studies include investigating the influence of HCCI cyclic variations on the cycle resolved values of gas temperature and unburned hydrocarbons [3], exploring HCCI boundaries for different levels of cyclic variations and influence of applying some additives [6], and also a study investigating the cyclic variability of HCCI combustion during the transition between SI mode and HCCI mode [5].

This study is to extend our previous work [7], where cyclic variations of HCCI ignition timing were analyzed and the influence of intake manifold temperature, EGR rate, and equivalence ratio was investigated. In this study, patterns of cyclic variations for P_{max}, IMEP, and Start of Combustion (SOC) are shown and the effect of coolant temperature, intake manifold pressure and engine speed are discussed.

Sources of Cyclic Variation in HCCI Engines

There are generally five major sources causing the cyclic variations in HCCI engines[7]:

• Temperature inhomogeneity & thermal stratification. A temperature gradient exists in the unburned charge resulting from heat transfer with walls (piston, valves, cylinder head) with different temperatures and also from imperfect mixing of fuel, air and residual gas [8]. Due to fluctuations in the temperature of cylinder walls and residual gas, thermal stratification inside the cylinder can vary cycle by cycle that can lead to cyclic variations in the HCCI combustion.

• Mixture compositional inhomogeneity. Mixture composition inside the cylinder is spatially variant because of the imperfect mixing of fuel, air and residual gas and incomplete vaporization of liquid fuel. Cyclic variation in the mixture compositional inhomogeneity can lead to the cyclic variations in HCCI combustion. Compared to the charge inhomogeneity, the temperature inhomogeneity has a stronger effect on the spatial variations of HCCI combustion [9].

• Fluctuations in Air Fuel Ratio (AFR). A considerable variation exists in the AFR of the engine charge from cycle to cycle particularly in port fuel injected engines [10]. AFR variations can happen due to the fluctuations in the gas exchange process during intake and exhaust strokes and also incomplete vaporization of liquid fuel particularly for very lean fuel operation. Since HCCI combustion varies when AFR of the charge changes [4], cyclic variations of HCCI combustion can be promoted by the cyclic fluctuations in the AFR.

• Fluctuations of diluents. The amount of diluents (EGR and residual gases) inside the cylinder varies cycle by cycle due to the fluctuations in the gas exchange process, particularly for the cases when VVT (Variable Valve Timing) is used to change the amount of the trapped exhaust gases [11]. Since the amount of diluents inside the cylinder influences HCCI combustion [4], cyclic fluctuations in EGR and residual gas can cause cyclic variations in HCCI combustion.

• Turbulence intensity. HCCI combustion is influenced by the turbulence through effectively transporting hot combustion products and radicals from the burned zone to the cold unburned zone, similar to the flame propagation mechanism in SI engines [12]. Fluctuations in the gas exchange process can cause cyclic variations in the turbulence intensity which can result in the cyclic variations of HCCI combustion particularly for the conditions with a high temperature inhomogeneity.

ENGINE SETUP

A single cylinder Ricardo Hydra Mark III engine with a Rover K7 head is used to carry out HCCI experiments.

A schematic of the experimental apparatus is shown in Figure 1 and the configuration of the engine is given in Table 1. The engine setup is briefly introduced here and more details can be found in [7]. Fuel is injected into the intake port only when the intake valves are closed. The charge entering the engine is heated by the electric air heater positioned downstream of the throttle body. The intake manifold pressure is increased using a supercharger driven by an electric motor. Typically, pressure traces from 200 consecutive engine cycles with 0.1 CAD (Crank Angle Degree) resolution are recorded for each experimental point.

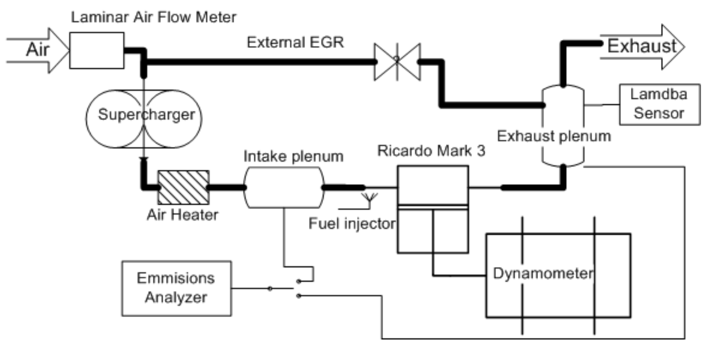

Figure 1: Schematic of the testbench used to obtain experimental data

Table 1: Configurations of the Ricardo single cylinder engine

Parameters	Values
Bore × Stroke [mm]	80 × 88.9
Compression Ratio	10
Displacement [L]	0.447
Number of Valves	4
IVC [aBDC]	55°
EVO [aBDC]	−70°

Table 2: Engine operating conditions

Parameters	Range
Engine Speed (rpm)	760 - 1340
Manifold Temp. (^oC)	60 - 140
EGR(%)	0 - 28
Equivalence Ratio	0.29 - 0.95
Manifold Pressure (kPa)	88 - 156
Fuel (PRF)	0, 10, 20, 40
Coolant Temp. (^oC)	25 - 84
Oil Temp. (^oC)	48 - 80

Table 2 shows the 360 experimental engine running conditions used for this study. To compare cyclic variations it is necessary to define Start of Combustion (SOC). Saturated compounds including paraffins such as n-heptane and iso-octane have two-stage ignition [13]. Either start of the 1^{st} stage (cool flame) or start of the 2^{nd} (main) stage of combustion can be used as the definition of SOC. Here, Start of the 2^{nd} stage of combustion is defined as being the point at which the third derivative of the pressure trace with respect to the crank angle (θ) in CAD exceeds a heuristically determined limit [14]. ($\frac{d^3P}{d\theta^3}\big|_{ign} = 0.25\frac{bar}{CAD^3}$)

The start of cool flame combustion (SOC-CF) is defined as the crank angle where the net Heat Release Rate (HRR) is 10% of the peak of HRR for Low Temperature Reactions (LTR) region [7]. The usual heat release method [15], that applies the first law analysis on the engine charge assuming ideal gas properties, is used to determine net HRR. The Rassweiler method [15] is used to calculate CA50, which is the crank angle for 50% of the fuel has been burnt.

RESULTS & DISCUSSION

Similar to other studies [3, 6] COV (Coefficient of Variation) is used to measure cyclic variability of IMEP, P_{max}, and HRR_{max} while STD (Standard Deviation) is applied to measure cyclic variability of crank angle based parameters such as SOC and CA50 [7]. Figure 2 shows three main distinct patterns of cyclic variations observed for SOC, P_{max}, and IMEP. Normal variations in Figure 2(a) can be caused by the normal variations described above. Figure 2(b) shows a nearly periodic pattern of oscillations within two limits. This can happen when the mixture condition inside the cylinder has a periodic variation between two limits. An ignition timing at the earlier threshold produces a relatively low in-cylinder pressure and gas temperature at exhaust valve closing. This will cause the combustion of the following cycle to occur at a crank angle later than the previous cycle. This trend can continue to shift (delay) ignition timing until a threshold that the late enough combustion produces a relatively higher cylinder pressure and temperature at exhaust valve closing compared to the previous cycle that can trigger an earlier combustion for the next cycle and so on. Figure 2(c) shows a cyclic variation pattern with many weak/misfired ignitions. The weak/misfired ignition cycles in this pattern results in significant amounts of unburned fuel that can be accumulated for the next cycles. For a weak HCCI combustion ignition usually occurs very late which can cause a higher gas temperature at exhaust valve closing. Results in [3] also show that a late combustion timing generates a high level of unburned hydrocarbons in the residuals that can auto ignite during the following gas exchange phase which can increase the temperature of the residual gases. The higher gas temperature in combination with the extra unburned fuel from the previous cycles can promote a relatively strong combustion for the next cycle. It can be noted that a small amount of IMEP is

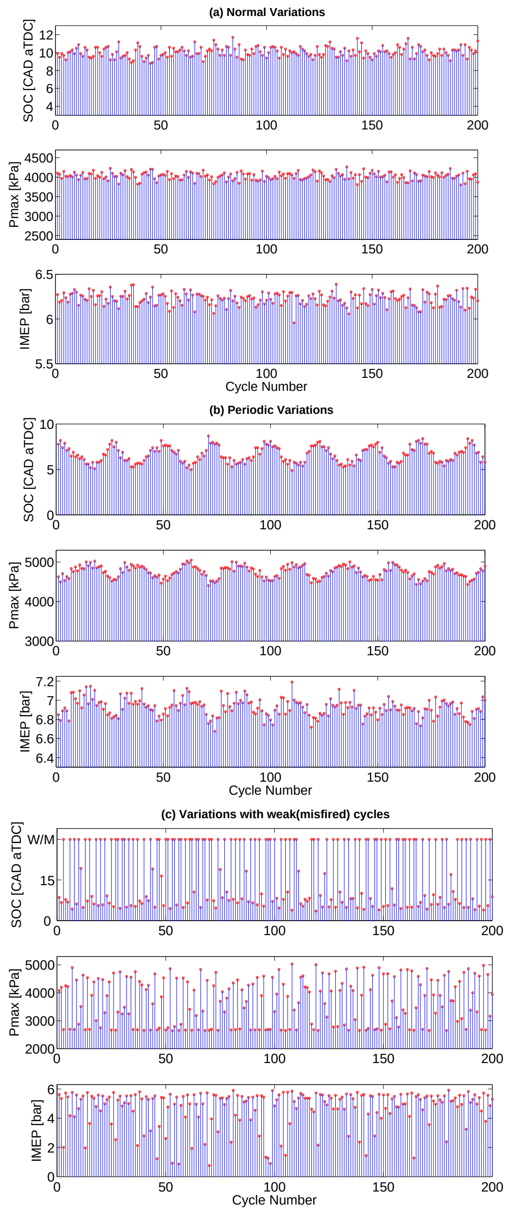

Figure 2: Different observed cyclic variations patterns of SOC, P_{max}, and IMEP. (The symbol W/M in part "c" represents Weak/Mifired ignitions whose value of $\frac{d^3P}{d\theta^3}$ never goes higher than the defined threshold for SOC.)

still produced even for the weakest ignitions in Figure 2-c though the peak pressure in these cases is close to that of motoring condition. A weak and late combustion in the expansion stroke that it is also evident in the pressure trace data for those cycles seems to be present.

Figure 3(a) shows the values of CA50 are almost the same as those of the crank angle of the maximum heat release rate ($\theta_{HRR_{max}}$). There is also a good correlation between the cyclic variation of CA50 with that of $\theta_{HRR_{max}}$ particularly for low cyclic variations as shown in Figure 3(b). Similarly, a good correlation was found in our previous study [7] between the cyclic variations of SOC with those of CA50, P_{max}, burn duration and IMEP (when cyclic variation of SOC is low STD $<$ 1.5 CAD).

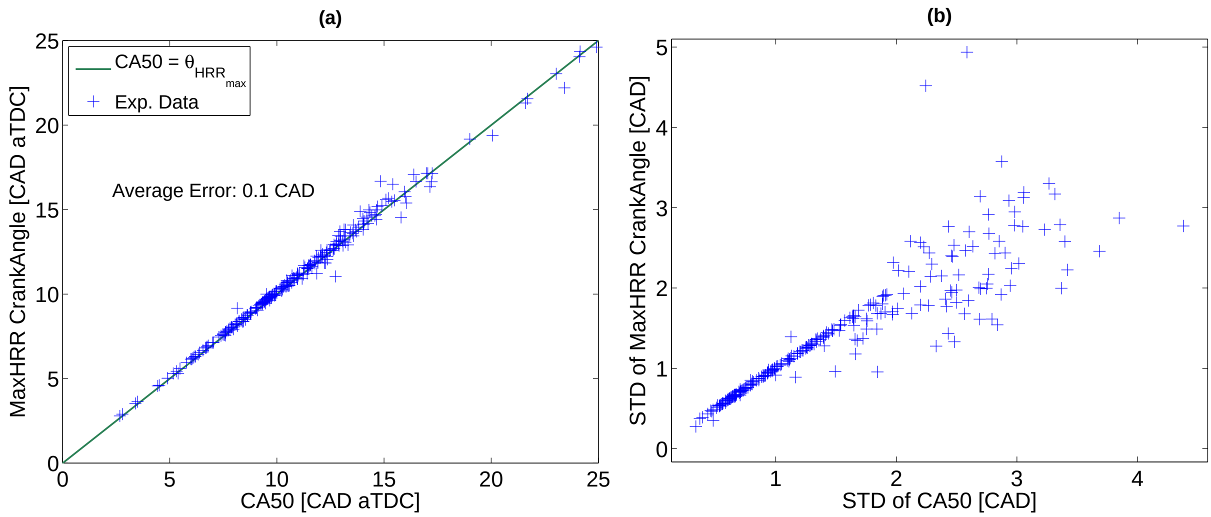

Figure 3: Trend of change in CA50 and cyclic variations of CA50 with those of the crank angle of the maximum heat release.

Table 3 indicates the average of cyclic variations of ignition timing and heat release parameters in terms of STD/COV for 360 experimental points. The table shows the SOC of the first stage combustion (cool flame) has considerably less cyclic variation compared to that of the second (main) stage combustion for the low octane fuels studied. Furthermore, both HRR_{max} from LTR region and $\theta_{HRR_{max}}$ from LTR region have less cyclic variations compared to those from High Temperature Reactions (HTR) region. This can happen as LTRs in the first stage of HCCI combustion are slower than HTRs in the second stage of HCCI combustion and LTRs are mainly a function of the n-heptane to oxygen mole ratio [13].

Table 3: Comparing the average cyclic variations of ignition timing and heat release parameters

Ignition timing parameters	STD [CAD]
SOC-CF	1.1
SOC	2.1
Heat release parameters	**STD/COV**
$\theta_{HRR_{max}}$ in LTR	0.6 [CAD]
$\theta_{HRR_{max}}$ in HTR	1.3 [CAD]
HRR_{max} in LTR	10 [%]
HRR_{max} in HTR	14.6 [%]

Figure 4 to Figure 6 show the influence of coolant temperature, intake pressure and engine speed on the cyclic variations of SOC and P_{max} while keeping the other variables constant. Figure 4 shows that the cyclic variation of SOC and P_{max} increase with a decrease in the coolant temperature. This can happen as the stability of HCCI combustion is highly dependant on the cylinder wall temperature and more stable HCCI combustion can occur when the cylinder walls are warmer [16]. Due to a high temperature difference between cylinder walls and in-cylinder gas when the coolant temperature is low, the heat transfer between cylinder walls and in-cylinder gas becomes more significant in this condition. Fluctuations in the heat transfer process can increase the possibility of higher cyclic variation when the coolant temperature is low.

Figure 4 also indicates a 8 CAD delay in SOC when the coolant temperature is lowered from 80^oC to 31^oC. HCCI combustion that occurs too late has high SOC cyclic variations [7]. Figure 5 and Figure 6 show that the influence of engine speed and intake pressure on the cyclic variations of both SOC and P_{max} is highly dependant on the position of SOC. Cyclic variation of SOC is low when SOC occurs immediately after TDC, but cyclic variation increases when SOC happens late or before TDC. The same trend is seen for cyclic variations of P_{max} except that the cyclic variation is still low for the cases with SOC before TDC. Figures 5 and 6 show that SOC retards with an increase in the engine speed, but it advances with an increase in the intake pressure. This can be explained by the fact that HCCI combustion is a time based process dominated by the time scales of the reactions leading to auto-ignition and the higher intake pressure causes higher gas pressure in the compression stroke that makes the mixture auto-ignites earlier.

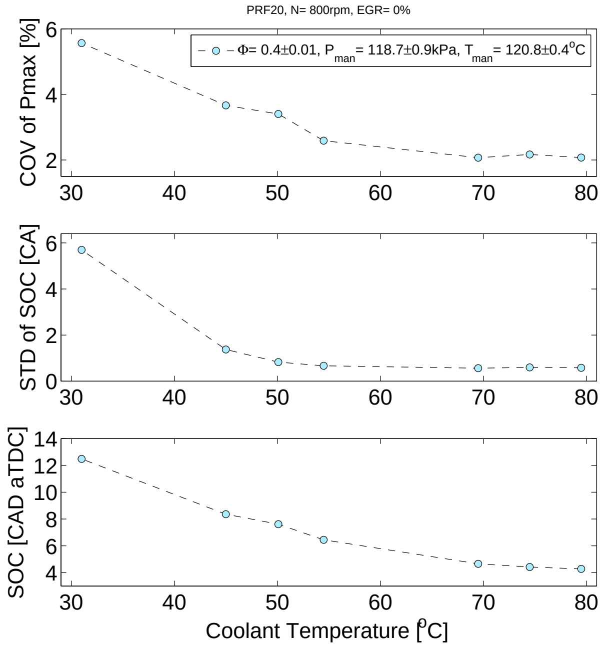

Figure 4: Influence of the coolant temperature on the cyclic variations of SOC, and P_{max}.

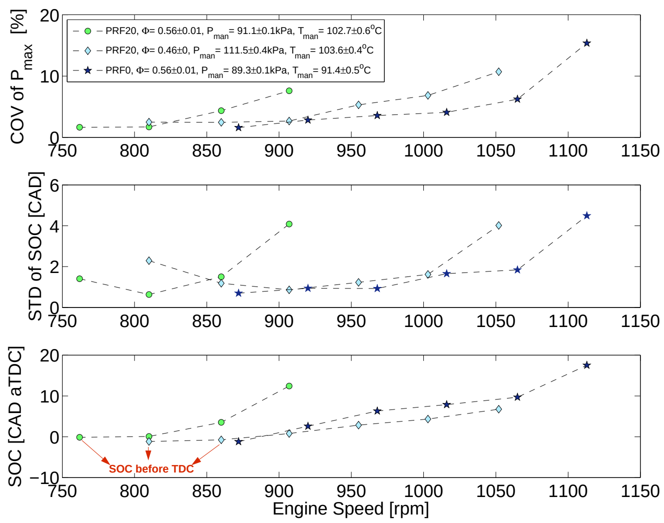

Figure 5: Influence of engine speed on the cyclic variations of SOC, and P_{max}.

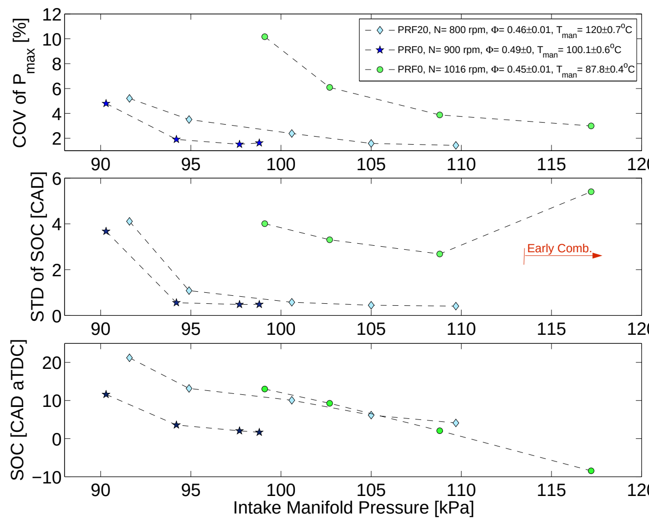

Figure 6: Influence of intake manifold pressure on the cyclic variations of SOC, and P_{max}.

CONCLUSIONS

An experimental study with 360 operating points indicates that HCCI cyclic variation can have a deterministic periodic/normal/misfired oscillation pattern based on the physics occurring inside the cylinder. For the PRF fuels studied, the main (second) stage of HCCI combustion has higher cyclic variation compared to that of the first stage. Higher coolant temperatures advance SOC and less cyclic variations are observed. In general, changing the engine speed or lowering the intake pressure changes the location of SOC which influences the cyclic variation of HCCI combustion.

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