Accepted version on Author's Personal Website: C. R. Koch

Article Name with DOI link to Final Published Version complete citation:

A. Audet and C. R. Koch. Experimental HCCI cyclic variations using camshaft phasing. In *Combustion Institute/Canadian Section (CI/CS) Spring Technical Meeting*, page 6, May 2008

See also:

https://sites.ualberta.ca/~ckoch/open_access/CICS2008a.pdf

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Proceedings of Combustion Institute-Canadian Section
Spring Technical Meeting
University of Toronto, Ontario
May12-14, 2008

Experimental HCCI Cyclic Variations Using Camshaft Phasing

A. Audet[1], C.R. Koch

Mechanical Engineering Department, University of Alberta, Edmonton, AB T6G 2G8

Abstract

The effect of valve timing on cyclic variation of Homogeneous Charge Compression Ignition (HCCI) combustion is experimentally investigated. The understanding of the effect engine parameters have on the cyclic variations of HCCI is needed to control the combustion phasing of HCCI. Cyclic variation control is necessary since the load range of HCCI is partially limited by high cyclic variations seen at the misfire limit. A one-cylinder Ricardo engine with a Mercedes 4-valve camshaft phasing cylinder head is operated at a constant speed, intake temperature and intake pressure. Using camshaft phasers the timing of both the intake and exhaust valves are independently changed at different levels of engine load and fuel octane number.

Phasing the intake valves primarily effects the effective compression ratio of the engine, and it is observed that decreasing the effective compression ratio increases the cyclic variation of the timing of HCCI but has little effect on the cyclic variation of Indicated Mean Effective Pressure (IMEP). Phasing of the exhaust valves is used to obtain negative valve overlap which results in trapped residuals. Increases in these trapped residuals is seen to effect both cyclic variation of combustion phasing as well as IMEP mainly through charge temperature.

Introduction

HCCI has many potential benefits over conventional combustion modes, such as lower soot emissions, higher efficiencies, and very low NO_xemissions [1]. HCCI also has a set of problems that need to be solved before it can be commercially implemented. Included in these problems is the lack of direct control over the start of combustion [1]. While spark timing and injection timing initiate combustion in normal spark ignition and compression ignition engines, the start of combustion for HCCI is dictated by the pressure-temperature history of the fuel air mixture, the chemical properties of the fuel, and many other factors [1]. Another problem with HCCI is the limited load range [2]. Typical HCCI operation is contained between the knock limit on the high load side and the misfire limit at low load side [2]. To both extend the low load range of HCCI and incorporate reliable control of the combustion phasing there is a need to understand the effects of engine parameters on the cyclic variation of HCCI.

This study builds on our previous work which focused on the effects of intake manifold temperature, pressure, external exhaust gas recirculation, equivalence ratio, coolant temperature, and engine speed on the cyclic variation of ignition, IMEP, and P_{max}[3, 4, 5, 6]. In this current study the effects of both the intake and exhaust valve timing on combustion timing and stability of IMEP for three different fuel octane numbers is examined.

Changes in the intake and exhaust valve timing have been shown as effective ways of controlling the combustion timing of HCCI [7, 8, 9, 10]. The use of modified exhaust valve timing to trap residuals has also been shown as an effective method to initiate HCCI combustion [11, 12]. If either of these two methods of initiating and controlling HCCI are to be used it is important to understand their effect on the cyclic variation of HCCI.

[1]Corresponding author: aaudet@ualberta.ca

Experimental Setup

A Ricardo Hydra Mark 3 engine is fitted with a Mercedes e550 camshaft phasing cylinder head [13]. The engine specifications can be seen in Table 1. This engine represents a typical spark ignition engine, except that the head of the engine contains a piezo-electric pressure transducer. A schematic of the test cell can be seen in Figure 1. The fuel scheduling of the two injectors is done with a dSpace-MicroAutobox ECU. Real time combustion analysis is done with A&D Baseline CAS [14]. With this system the cylinder pressure trace is recorded 3600 times per crank revolution (every tenth of a degree), and then analyzed for the pertinent combustion metrics, such as Indicated Mean Effective Pressure (IMEP), 50% mass fraction fuel burned (CA50) and burn duration. Similar to the previous studies the CA50 value is used as the measurement for combustion timing [3, 4]. For each test point the pressure trace is recorded for 300 engine cycles. All other parameters are logged on a time basis using A&D Baseline DAC for the entire duration of the 300 engine cycles.

Camshaft phasing is done with vane-type camshaft phaser for both the intake and exhaust camshafts. These phasers provide 40°of phase change for each camshaft, and they are independent of each other. The lift profiles of the camshafts are not adjustable, so the maximum lift and durations are fixed. This poses problems when the exhaust valve is advanced earlier to achieve more trapped residuals because the valve begins to open too early in the power stroke of the engine causing a large decline in thermal efficiency. The earliest exhaust valve opening time used in this study is 90°after Top Dead Center (aTDC). A plot of the valve timing ranges obtained by camshaft phasing used in this study can be seen in Figure 2.

The engine operating conditions are seen in Table 2. Three different octanes of fuel were ran for both the intake valve timing sweeps and the exhaust valve timing sweeps. For these different octanes HCCI combustion is stable for different amounts of injected fuel, which means that each octane is ran at a different IMEP, or engine load.

Table 1: Configuration of the Ricardo single-cylinder engine.

Parameters	Values
Bore $\times$ stroke [mm]	97 $\times$ 88.9
Compression Ratio	12
Displacement [L]	0.657
Valves	4
IVC [aTDC]	202 - 242
EVC [bTDC]	53 - 13

Table 2: Engine operating conditions

Parameter	**Range**
Coolant Temperature [$^\circ$C]	70
Oil Temperature [$^\circ$C]	50
Oil Pressure [*psi*]	50
Manifold Pressure [*kPa*]	115
Manifold Temperature [$^\circ$C]	80
External EGR [%]	0
Fuel Octane [PRF]	0 - 40

Factors that Effect Cyclic Variation

It has been described previously [3] that the contributors to cyclic variation can be grouped into five main categories: thermal stratification, mixture composition inhomogeneities, in-cylinder turbulence, air-fuel fluctuations, and EGR fluctuation. For tests in this study the timing of both intake and exhaust valves are changed which alters the gas exchange process of the engine. These fluctuations in the gas exchange process affect all the five major contributors to cyclic variation.

Thermal stratification is the result of heating from the cylinder walls and other hot surfaces, as well as from fluctuations in residual gas. Changes in valve timing have a strong effect on the residual gas fraction [15], so thermal stratification is expected to change as the valve timing is varied. Mixture composition inhomogeneities are spatial variations in the amount of air, fuel and residual gas, and can arise from imperfect mixing of the constituents. In-cylinder turbulence, which aids in-cylinder mixing, has been shown to change with different valve profiles [16]. In-cylinder turbulence also directly influences cyclic variation of combustion through its role in moving the flame to unburned areas of the combustion chamber. Air-fuel fluctuations are cycle to cycle changes in the overall ratio of air and fuel. The valve timing has strong effects on the volumetric efficiency of the engine [15]; therefore, valve timing changes will effect the cyclic variation of the overall air-fuel ratio. Exhaust Gas Recirculation (EGR) fluctuations also contribute to cyclic variation and are due to the dynamics of the gas exchange process. Internal EGR is largely controlled by Exhaust Valve Closing (EVC) timing, so it is expected to have an effect on the cyclic variation of EGR.

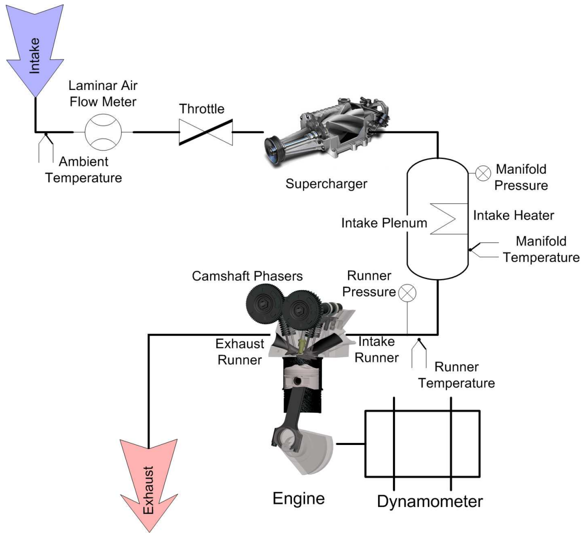

Figure 1: HCCI engine with camshaft phasing cylinder head.

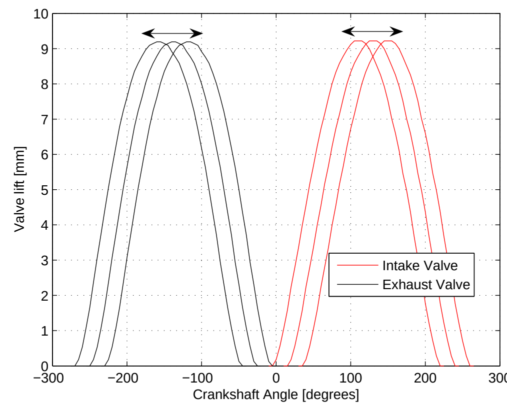

Figure 2: Schematic of the minimum, maximum and normal valve profiles used in this study.

Results & Discussion

For all the test points taken the exhaust valve is closed before the gas exchange Top Dead Center (TDC). Valve timing of this nature is termed Negative Valve Overlap (NVO) [15]. Early exhaust valve closing traps a certain portion of the exhaust gas, and this gas is then mixed with the new charge of fresh fuel and air. An estimation of the trapped residual can be performed with a measurement of the cylinder pressure at exhaust valve closing timing, the exhaust temperature, and the mass flow rate of fresh air and fuel. The trapped mass is estimated by:

$$m_{tr} = \frac{P_{evc}V_{evc}}{RT_{exh}} \tag{1}$$

The mass fraction of internal residual r is then estimated by (no external EGR for tests reported here):

$$r = \frac{m_{tr}}{m_{tr} + m_{freshair} + m_{fuel}} \tag{2}$$

Figure 3 shows the estimated internal residuals for the exhaust valve timing sweeps. For this study the trapped residual percent is changed from 9 to 28 %. As seen in the figure there is a different EGR percentage for each different engine load, but all three loads have the same trend. The lowest load has the highest internal EGR fraction for a given valve timing.

For the experiments the Intake Valve Close (IVC) point is changed from 202°to 242°aTDC. This changes the effective compression ratio, which is calculated using:

$$CR_{eff} = \frac{V_{tdc}}{V_{tdc} + V_{IVC}} \tag{3}$$

With a mechanical compression ratio of 12:1 the effective compression ratio is changed from 11.75 to 9.75 as shown in Figure3.

The relationship between **exhaust valve** timing and the cyclic variation of the IMEP is shown in Figure 5 for three fuel octane numbers. Retarding the exhaust valve timing decreases the trapped residual fraction causing an increase in the cyclic variation of IMEP. Decreasing internal residuals decreases the mixture temperature at the start of compression because hot residuals are replaced with cooler intake air. Since auto-ignition is very sensitive to temperature [5] HCCI combustion becomes unstable at lower temperatures and high cyclic variation in IMEP is observed. There is not a large change in the IMEP for most of the sweep since mass flow rate of fuel is held constant for these tests. The load does decrease at very retarded exhaust valve timings, low internal residual fraction, due to the large cyclic variation of IMEP seen at these points ($COV > 5\%$). The combustion starts to become inefficient at this point as there are a large

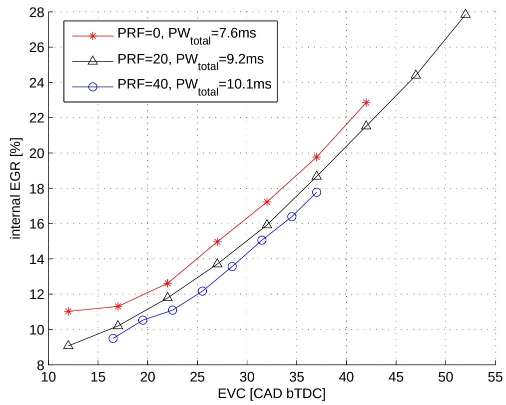

Figure 3: Estimated internal EGR for the exhaust valve timing ranges.

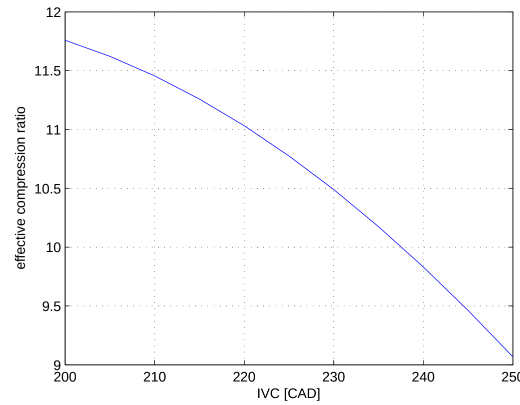

Figure 4: Effective compression ratio for the intake valve timing ranges.

amount of unburned fuel and carbon monoxide in the exhaust emissions. A similar trend is seen for all three different octane values but a steady increase of cyclic variation of IMEP is prominent in the low octane, high load case.

Changes in CA50 and the standard deviation of CA50 as a function of the exhaust valve timing are plotted in Figure 6 for three fuel octane numbers. As the amount of trapped residual is increased the timing of the HCCI combustion is advanced, which is a result of high mixture temperatures from the retention of the hot residuals. While cooled external EGR can be seen to retard HCCI combustion timing because of the heat capacity effect of EGR [1], it is seen here that the primary effect of internal EGR is the advance of CA50. This is the result of the charge heating effect of internal EGR. The standard deviation of CA50 decreases as the trapped residual fraction increases. This trend is seen for all three fuel octane numbers and agrees with our other results that show as the timing of the combustion retards the cyclic variation will increase.

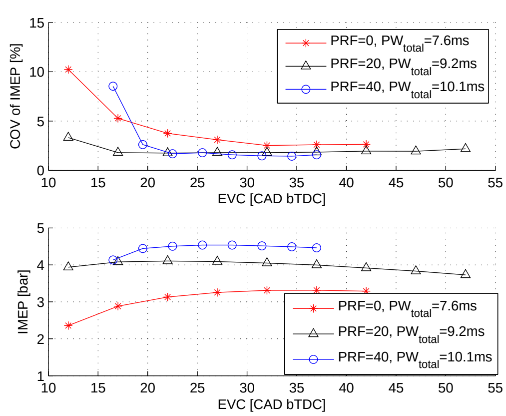

Figure 5: Influence of the timing of the exhaust valve on cyclic variation of IMEP.

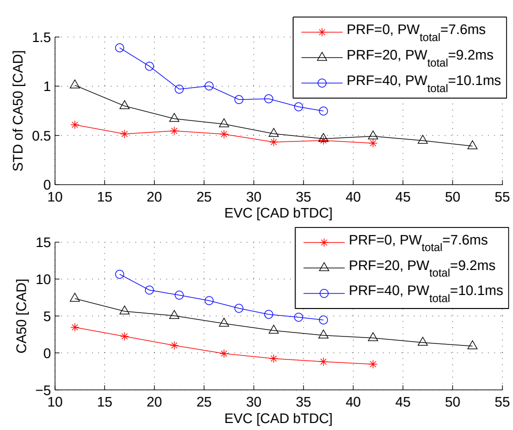

Figure 6: Influence of the timing of the exhaust valve on cyclic variation of CA50.

Intake valve phasing effects on IMEP and cyclic variation of IMEP are plotted in Figure 7 for three fuel octane numbers. The timing of the intake valve has little effect on the cyclic variation of IMEP for most of the intake timing sweep. Cyclic variation does not change appreciably until the misfire limit is reached. At this point the engine power is seen to slightly decrease resulting from the decrease in combustion efficiency, whereas it is constant for most of the intake valve sweep. This results shows that adjustments of the phasing of the intake valve can be made within a ceratin range and there will be no effect on the IMEP or the cyclic variation of IMEP, at constant injected fuel. This result was seen in all three octane cases.

The intake valve phasing effects on CA50 and standard deviation of CA50 are shown in Figure 8 for three fuel octane numbers. As the intake valve timing is retarded the HCCI combustion is retarded, and the standard deviation of CA50 is increased. The timing of the intake valve closing point has a very strong effect on the timing of CA50 due to the change in the effective compression ratio of the engine. Increasing the effective compression ratio increases the temperature and pressure at the end of the compression stroke, which advances the timing of HCCI. This is seen for all three octane cases. This effect is also seen in Figure 6 where the advancement in the combustion timing decreases the cyclic variation of the CA50.

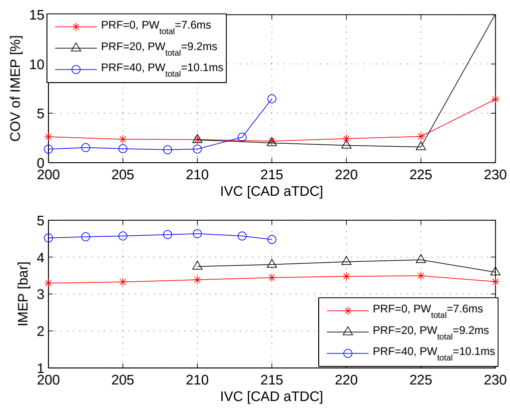

Figure 7: Influence of the timing of the intake valve on cyclic variation of IMEP.

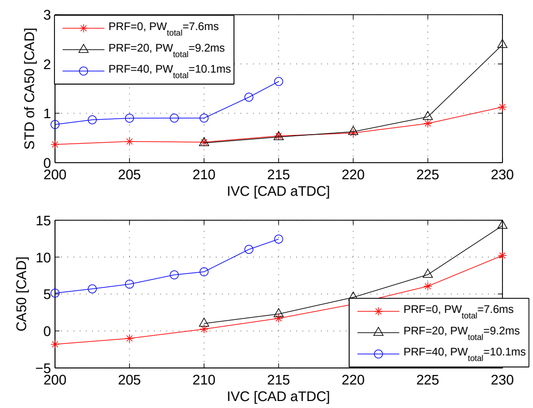

Figure 8: Influence of the timing of the intake valve on cyclic variation of CA50.

Conclusions

Retarding the exhaust valve phasing increases amount of trapped residual. It is seen that changes in residual effect both the cyclic variation of HCCI timing as well as IMEP. This is mainly attributed to the effect of charge heating by the hot residuals and was similar for all three fuel octane numbers tested. Intake valve phasing is seen to have a direct effect on the timing of the HCCI combustion, and the standard deviation of CA50. This change in timing did not effect the cyclic variation of the IMEP until the misfire limit is reached, where loss of combustion efficiency is noticed for all three fuel octane numbers.

Acknowledgments

This work was supported by the AUTO21 Network of Centres of Excellence, the Natural Sciences and Engineering Research Council of Canada (NSERC) and Daimler.

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