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Actuator comparison for closed loop control of HCCI combustion timing

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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) is an emerging combustion technology due to its increased efficiency and decreased NO_x emissions. One of the most challenging aspects of HCCI is the regulation of the combustion timing. Unlike conventional combustion modes there is no direct control over the start of combustion. Autoignition timing is a function of the temperature, pressure and composition of the mixture, so to adjust the combustion timing of HCCI changes have to be made to these. Both variable valve timing and variable fuel octane number are effective inputs to achieve cycle-to-cycle combustion control of HCCI combustion timing. The application of these control methods are investigated in this paper.

A one-cylinder Ricardo engine is fitted with a 4-valve spark ignition cylinder head equipped with camshaft phasers. These phasers independently adjust both the intake and exhaust camshaft phasing. By modifying the intake valve timing the effective compression ratio is changed, which affects the temperature-pressure condition of the mixture. Variable fuel octane is realized using two independent fuel injector systems, one equipped with iso-Octane and the other with n-Heptane. The CA50 (crank angle of 50% mass fraction burned) is regulated using feedback control and two separate actuators for combustion timing are implemented; intake camshaft phasing and variable fuel octane. These actuators are compared according to their range of operation and ability to reject system disturbances. The different combustion controllers are subjected to disturbances of both engine speed and engine load (changes in injected fuel energy). The results show the benefits and limitations of each actuator.

INTRODUCTION

HCCI has many potential benefits over conventional combustion modes, such as lower soot emissions, higher efficiencies, and very low NO_x emissions [1]. HCCI also has a set of problems that need to be solved before it can be commercially implemented for automotive applications. Included in these problems is the inability to directly control 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 condition of the fuel air mixture, the chemical properties of the fuel, and other factors [1]. Changes in the engine operating condition can drastically change the combustion timing of the HCCI combustion event, which in some cases leads to misfire or engine knock. Two potentially large variations in operating condition are engine speed or engine load. Both these variables are commanded by the driver of a car. To maintain optimal combustion timing throughout the engine operating range and to compensate for changes in engine load and engine speed, compensation needs to be performed using either the mixture composition or the pressure-temperature condition.

HCCI ignition timing can be controlled by varying the charge conditions (intake temperature, intake pressure, charge dilution level, etc.). Only some of charge conditions can be practically changed quickly enough to respond to large fast transient changes to engine conditions typical when driving. The three main methods for transient control of HCCI operation are: Variable Compression Ratio (VCR) [2, 3], Dual Fuel Modulation (DFM) [4, 5], and Variable Valve Actuation (VVA) [6, 7]. Changes in the intake and exhaust valve timing using VVA have been shown as effective ways of controlling the combustion timing of HCCI [8, 9, 10, 11], where Intake Valve Closing (IVC) timing changes are shown to be very effective and are therefore used in this study. VVA used to modify the IVC timing changes the effective compression ratio of the engine. These adjustments in IVC timing change the amount of mixture trapped in the cylinder for the compression stroke. A later IVC timing after bottom dead center results in less trapped air-fuel mixture, and a lower effective compression ratio which results in lower temperatures and pressures at the end of the compression stroke. VVA is also used to modify the HCCI ignition timing by adjusting the amount of trapped exhaust gas (residual gas fraction) [7, 12], or by modulating exhaust rebreathing [13, 14] or a combination of these two methods [15]. Other studies have shown the suitability of using dual fuel modulation to regulate the combustion timing [16]. In this study two independent port fuel injection systems have been implemented, and by varying the volumetric ratio of each fuel the autoignition characteristics of the ingested mixture is changed during engine operation.

Feedback control of HCCI is implemented using a variety of sensors. In-cylinder pressure sensors [17] and ion sensors [18, 19] are alternatives to obtaining the quality of combustion for feedback. Controllers used in [14, 16, 20] are typical examples of this. Model-based controllers which use physical models to predict HCCI ignition timing have also been developed [8, 21, 22, 23]. Other control strategies include: Model Predictive Control [8], Linear Quadratic Gaussian control [22], as well as state estimation control methods [11]. These studies show the performance of different control methods using a variety of actuators, but with the exception of [24], there is no direct comparison of the different actuators. This investigation provides a direct comparison of the effectiveness of intake camshaft phasing compared to using a dual fuel injection system for feedback control of HCCI combustion timing.

EXPERIMENTAL SETUP

For this study, a Ricardo Hydra Mark 3 engine is fitted with a Mercedes E550 camshaft phasing cylinder head [25]. This engine represents a typical spark ignition engine, except that the head of the engine contains a piezo-electric in-cylinder pressure transducer. The engine specifications can be seen in Table 1. A schematic of the test cell can be seen in Figure 1. The fuel scheduling of the two injectors, as well as the control of the camshaft phasing system, is performed using a dSPACE-MicroAutobox ECU. This processor also incorporates the combustion timing controllers. The real-time combustion information is calculated with A&D Baseline CAS [26]. The cylinder pressure trace is recorded 3600 times per crank revolution, and then analyzed for the pertinent combustion metrics, such as Indicated Mean Effective Pressure (IMEP), CA50 and burn duration. Similar to previous studies the CA50 value is used as the measurement for combustion timing [27]. The value of CA50 is sent to the MicroAutoBox via an analog voltage. CA50 [1] – the crank angle when 50% of the mass fraction of fuel has burned is used as a measure of combustion timing. This is calculated by performing a 1st law energy analysis on the measured pressure trace. The method estimates the net energy heat release since no correction is made for the heat lost through heat transfer at the cylinder walls.

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

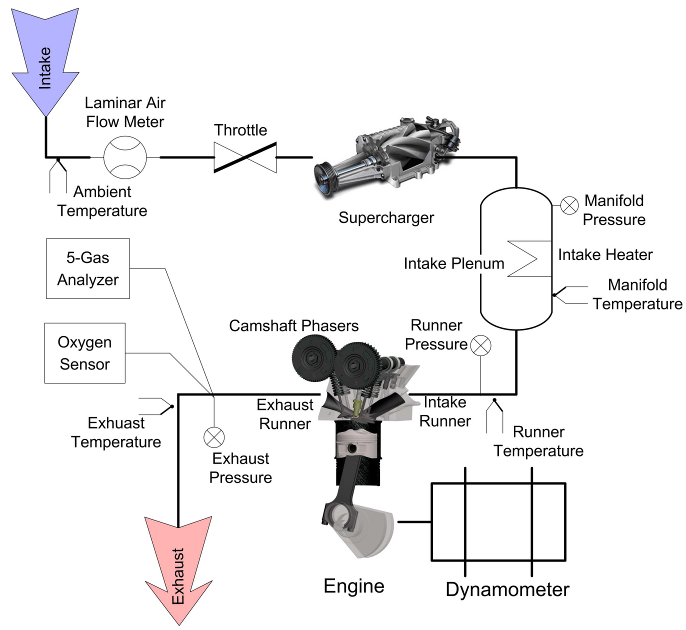

Figure 1: HCCI engine with camshaft phasing cylinder head.

Camshaft phasing is done with a hydraulic vane-type camshaft phaser for both the intake and exhaust camshafts. These phasers provide a 40^o range 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. A plot of the valve timing ranges obtained by camshaft phasing used in this study can be seen in Figure 2. The control of these phasers is also done with the MicroAutoBox. The maximum speed of these phasers is measured to be 130CAD/s. Variable fuel octane is accomplished with two independent port fuel injection systems. The fuel injectors are calibrated so that an estimate of the injected mass of fuel can be made from the injector pulse width. Using this, the fuel octane, which is the volumetric ratio of iso-Octane to the total amount of injected fuel, is modulated with the MicroAutoBox while maintaining a constant injected fuel energy based on the lower heating values of the fuel.

The two methods of CA50 regulation, IVC timing and fuel octane, are tested at five different steady state engine conditions as outlined in Table 2. The different engine conditions are chosen to test the different actuation methods for different intake manifold temperatures and pressures, as well as different engine speeds. The first three test points are at 1000RPM but the intake manifold temperature and pressure are increased for each point. For these three cases the intake manifold conditions (temperature and pressure) are varied using the supercharger and air heater while adjusting the fuel octane number. For the last two test points the manifold conditions are identical to that of the second test point, but the engine speed is increased to 1250RPM and then 1500RPM. At each different engine condition the fuel octane is adjusted so that combustion timing is 5^oaTDC for a similar IVC timing. For each engine conditions the injected fuel energy (or nominal engine load) is maintained constant. The 12:1 compression ratio, EVC timing for internal EGR, low octane number fuels, and boosted manifold pressure contribute to stable HCCI operation.

Table 2: Five engine conditions that are used as a base for all subsequent tests.

Test	Engine Speed [RPM]	Injected Energy [kj]	Manifold Press. [kPa]	Manifold Temp. [^{o}C]
BP1	1000	0.718	110	60
BP2	1000	0.718	125	80
BP3	1000	0.718	140	105
BP4	1250	0.718	125	80
BP5	1500	0.718	125	80

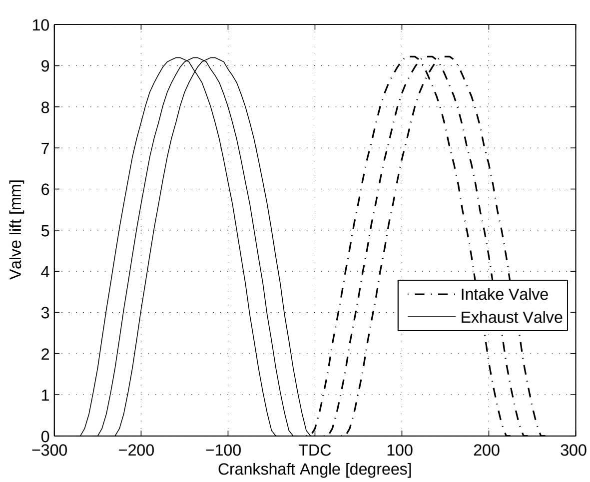

Figure 2: Schematic of the minimum, maximum and normal valve profiles used in this study.

RESULTS

SENSITIVITY AND LINEARITY

Steady state open loop tests are performed to find the sensitivity of the combustion timing to the two different methods of actuation. The sensitivity of the combustion timing to changes in IVC timing and fuel octane is plotted in Figures 3 and 4 respectively. These figures result from varying either IVC or fuel octane while keeping all other inputs constant around each of the five basepoints, listed in Table 2. Then for each of the tests the actuator is adjusted from the base point so that timing moves to the knock limit at the early side of combustion to the misfire limit and to the late side of combustion from the basepoint. Knock is determined when the root mean squared value of bandpass filtered pressure trace is above 6kPa, and misfire is when the coefficient of variation of the Indicated Mean Effective Pressure (IMEP) is above 5%. The original basepoints are chosen so that the injected fuel energy, IVC timing, and CA50 timing is identical and fuel octane is set manually (the manual setting of octane number results in the slight offsets in Figure 3).

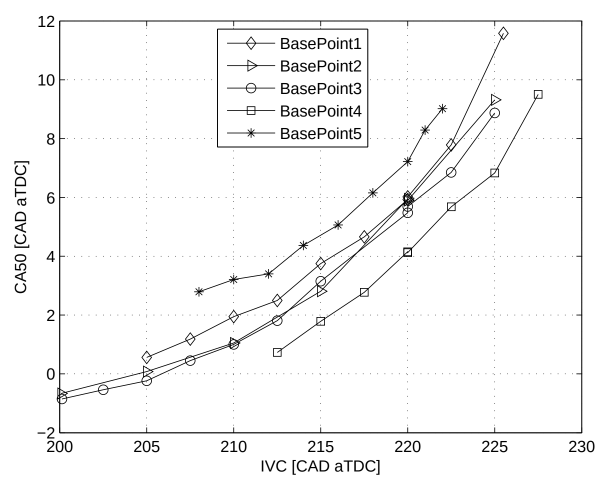

Figure 3: Sensitivity of steady state CA50 to IVC timing for the 5 engine basepoint conditions.The Octane for Basepoint 1 through 5 is: 10, 28, 43, 11 and 6

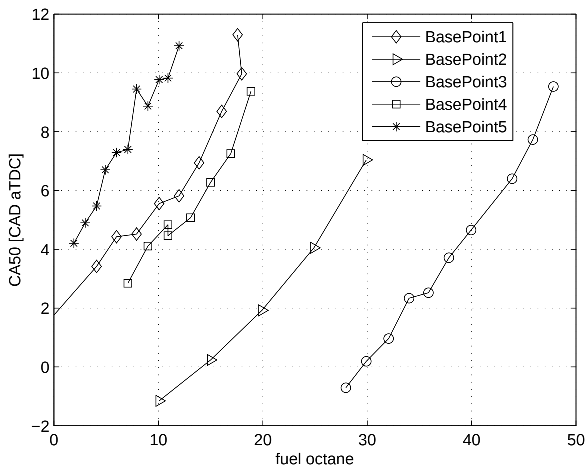

Figure 4: Sensitivity of steady state CA50 to fuel octane number for the 5 engine basepoint conditions

Linear fits are applied to these data sets as a metric of the sensitivity of the combustion timing to each actuator. Table 3 shows slope of the calculated linear fits. It is apparent from the figures, as well as the calculated values of the linear fits, that the sensitivity does not change drastically between the different engine conditions used in this study. In Figure 3 a non-linear trend appears between the IVC timing and measure CA50 value. Other research have used input linearization techniques to correct for the obvious non-linearity [28], but this is not investigated in this study. In Figure 4 it is apparent that the mean values of fuel octane is very different for each test point.

Table 3: Sensitivity of CA50 to IVC timing and fuel octane at all five engine conditions.

	Sensitivity	
Operating Point	$\frac{^oCA50}{^oIVC}$	$\frac{^oCA50}{ON}$
$BP1$	0.48	0.48
$BP2$	0.39	0.41
$BP3$	0.39	0.49
$BP4$	0.39	0.50
$BP5$	0.45	0.66

SYSTEM DYNAMICS

To characterize the input-output dynamics, the combustion timing response (CA50) to step changes of IVC timing or fuel octane are input as shown in Figure 5 and 6 respectively. For these open loop tests, the setpoint of either fuel octane or IVC timing is manually stepped. The measured combustion timing includes the pure time delay of two cycles that is caused by calculating the combustion timing from the recorded pressure trace.

Figure 5 shows the changes in combustion timing as a result of the change in IVC timing. The IVC timing is retarded by 10^o's from the 2nd engine condition listed in Table 2. Figure 6 shows a similar test but here the octane number is decreased by 10. The change in combustion timing is faster (≈ 2 cycles) for the IVC timing step compared to the fuel octane step (≈ 6 cycles).

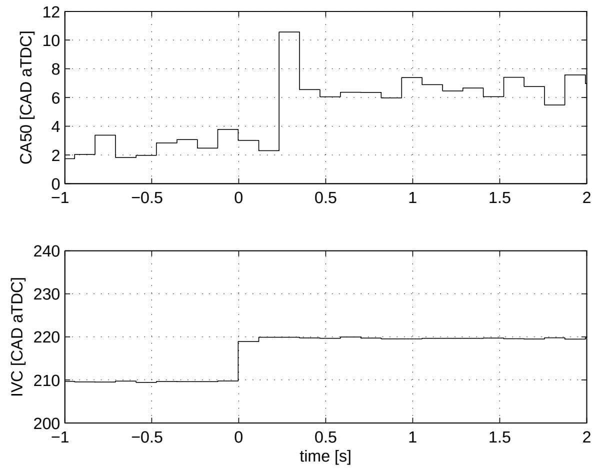

Figure 5: Response of CA50 to a step change ($210 \rightarrow 220$) in the IVC timing. (test point $BP2$)

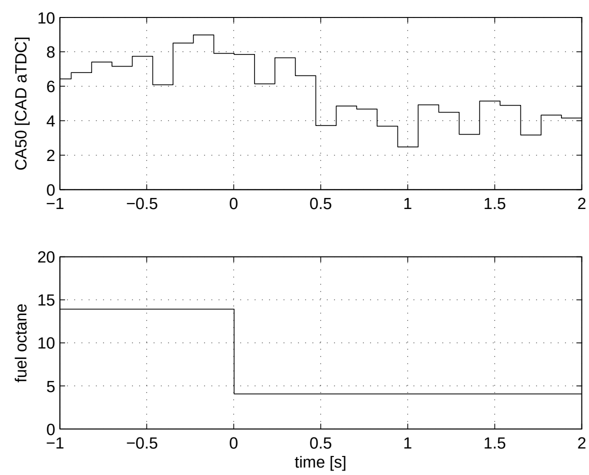

Figure 6: Response of CA50 to a step change ($14 \rightarrow 4$) in the octane number. (test point $BP2$)

To further characterize the input-output response a Pseudo Random Binary Sequence (PRBS) is applied to either IVC timing or the fuel octane number. In [29], the PRBS inputs are individually implemented around test condition 2 with the levels of the input such that CA50 does not reach either the misfire or knock limit. Using this data and system identification ARX models including pure time delay are identified. It is found that the pure time delay between the fuel octane and CA50 is one cycle larger than the delay between the IVC timing and CA50 (three cycle versus two cycles). This input-output model can be used for control and details of a feedforward control using model inversion is detailed in [29].

CONTROL

Closed loop control of combustion timing using a single input, either IVC timing or fuel octane, is performed. A PI controller is implemented and the other input is held fixed. A schematic of the control system is shown in Figure 7 where *U* is the control input, either IVC timing control or fuel octane. Combustion timing (CA50) is regulated using a Proportional Integral (PI) controller in a once per engine cycle event based control. Different Proportional, k_P, and Integral, k_I, gains values are used at each of the five different engine conditions, with the values shown in Table 4. The disturbance rejection performance to engine speed and load range of each of the two separate controllers is discussed next. These two controllers are tested at the five different engine base conditions shown in Table 2.

Using the **intake valve close timing** (IVC) control the load disturbance rejection is illustrated in Figure 8 for the 1st engine condition listed in Table 2. In this case the load disturbance is stepped in increments of 50J (of injected fuel energy). The combustion timing (CA50) is maintained to within 5CAD of the set point for this moderate increase in injected fuel energy. No misfire or knock is present when the amount of fuel is changed, and the deviations in combustion timing are short lived and coincide with the step in injected fuel energy. Engine speed disturbance rejection is shown in Figure 9 for the same engine condition. Here the engine speed is incremented in steps of 100RPM as quickly as the dynamometer's internal speed controller allowed. Again CA50 is regulated close enough to the set point so that no abnormal combustion events occur.

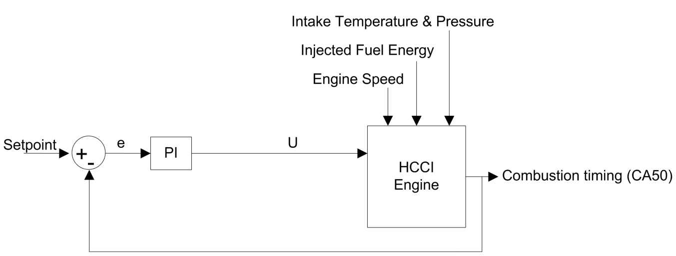

Figure 7: Schematic of the CA50 PI controller where *U* is either IVC timing or fuel octane.

Table 4: Summary of gains used with IVC timing control and fuel octane control.

	fuel octane		IVC timing	
Operating Point	k_P	k_I	k_P	k_I
BasePoint1	0.27	0.1	0.6	0.15
BasePoint2	1.5	0.3	0.6	0.2
BasePoint3	1.5	0.2	0.6	0.1
BasePoint4	1.6	0.3	0.8	0.2
BasePoint5	1	0.3	0.9	0.2

Using **fuel octane** as the input in Figure 7 the disturbance rejection is examined. Fuel energy and engine speed disturbance rejection are shown in Figures 10 and 11 respectively. As seen in these figures the octane control is able to adequately compensate for both the changes in engine speed as well as fuel energy changes. There are short lived deviations in the combustion timing when the disturbances occur, but the controller quickly regulates the combustion timing back to the setpoint of 5CAD aTDC. There is no apparent difference between the performance of this controller using fuel octane when compared to the controller using IVC timing.

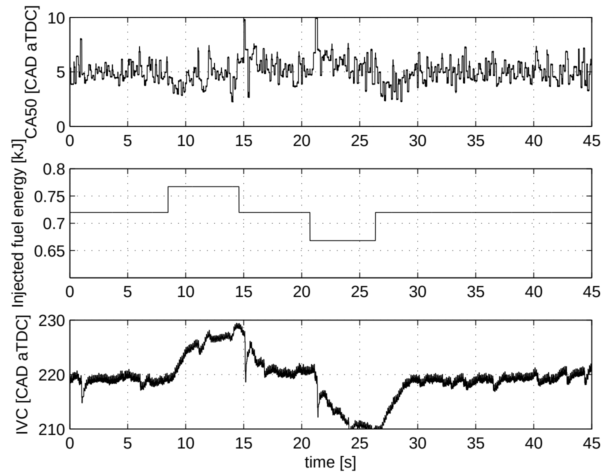

Figure 8: Load disturbance rejection performance for the controller using IVC timing for $BP1$.

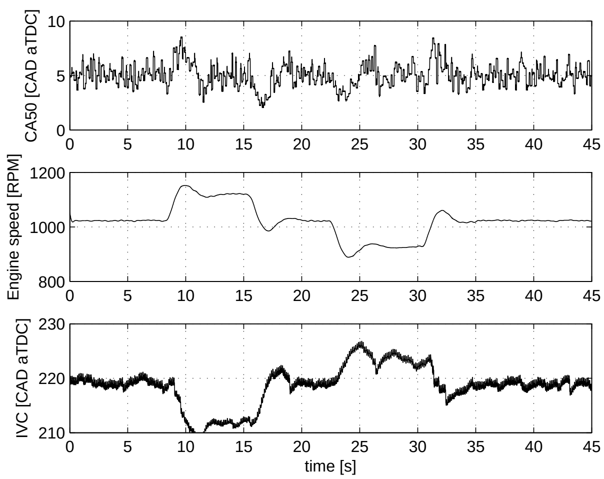

Figure 9: Engine speed disturbance rejection performance for the controller using IVC timing for $BP1$.

A more detailed comparison the variance of CA50 for each of the two controllers at each of the five test cases is tabulated in Table 5. A no-control case at each test base point is also performed to show the variance when there is no control of the combustion timing. In Table 5, E denotes the CA50 variation for the injected fuel energy disturbance (comparable to Figure 8) while ω denotes the CA50 variation the disturbance for the engine speed dis-

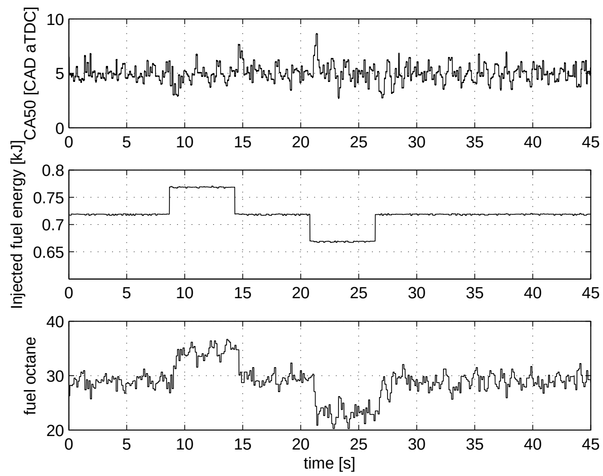

Figure 10: Load disturbance rejection performance for the controller using fuel octane control for $BP2$.

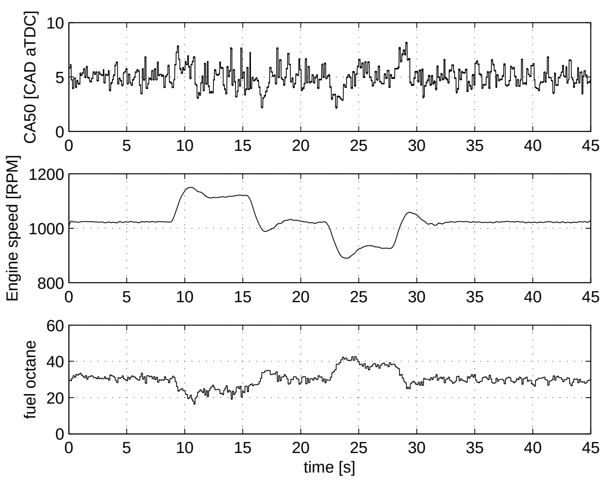

Figure 11: Engine speed disturbance rejection performance using fuel octane number control for $BP2$.

turbance (comparable to Figure 9). Similar sized disturbances are applied at each each of the five different engine base conditions listed in Table 2. At all five engine conditions, lower variance in CA50 timing is obtained with control for both types of actuation when compared to the case of no compensation. The variance of CA50 for either input, IVC timing or fuel octane number, perform similarly. However, better performance for the rejection of the injected fuel energy disturbance compared to engine speed is seen for both types of actuation, but this attributed to the chosen size of the engine speed disturbance versus fuel energy disturbance.

Table 5: Calculated variance of combustion timing (CA50 $[CAD^2]$) for the two combustion timing controllers rejecting different disturbances at the five different engine conditions.

	No Control		Fuel octane		IVC timing	
Dist.	E	ω	E	ω	E	ω
$BP1$	3.18	7.47	1.54	2.07	1.37	1.32
$BP2$	2.56	6.99	0.69	1.06	0.62	1.35
$BP3$	2.81	6.01	1.12	1.39	1.35	2.07
$BP4$	2.21	2.64	0.76	1.08	0.79	0.96
$BP5$	4.40	4.18	1.32	2.04	1.02	1.82

LOAD RANGE

The load range is defined to be the amount that the engine load output can be changed (by changing the injected fuel energy) between the misfire and knock limits of the engine. By using feedback control to maintain the combustion timing, the load range can be extended from the no compensation case. For the five engine conditions shown in Table 2 the injected fuel energy is adjusted from the knock limit to the misfire limit of the engine. No combustion timing control, or either IVC timing or fuel octane is used to regulate the combustion timing to 5^o aTDC while the injected fuel energy is changed. For each of these cases at test point three, the fuel conversion efficiency is calculated and shown as a function of engine load (IMEP) in Figure 12 for the steady state points. The efficiency decreases at high load due to the increased knock intensity and at low load due to the lower combustion efficiency.

The load range when combustion timing is regulated by the fuel octane controller is slightly wider than when timing is regulated by the IVC controller. This is seen for all the conditions where the fuel octane controller does not saturate at fuel octane of 0. This larger range obtained by the fuel octane controller is attributed to the adverse effect that changing IVC has on the dilution. The exhaust oxygen percent can be seen in Figure 13, and it is seen that the slope for the IVC controller is different that the other two slopes. IVC timing affects the dilution whereas changing the fuel octane does not. When the load is increased, more fuel is added, the combustion timing advances. To regulate the timing to the desired value the IVC timing becomes later, decreasing the effective compression ratio and decreasing the amount of trapped air (because of the smaller volume at IVC). This decreases the air to fuel ratio, increasing the engine tendency towards knock [1]. A similar effect happens when the engine load is decreased. This decreases the range obtained the IVC timing controller compared to the fuel octane controller.

In all five base cases the use of feedback control extends the load range at each test condition compared to no-control. When either IVC timing or fuel octane is used to control CA50 timing the load range is approximately twice as large as the no- control case.

The detailed effect of boost pressure on load range using IVC timing control is beyond the scope of this paper but is documented in [29].

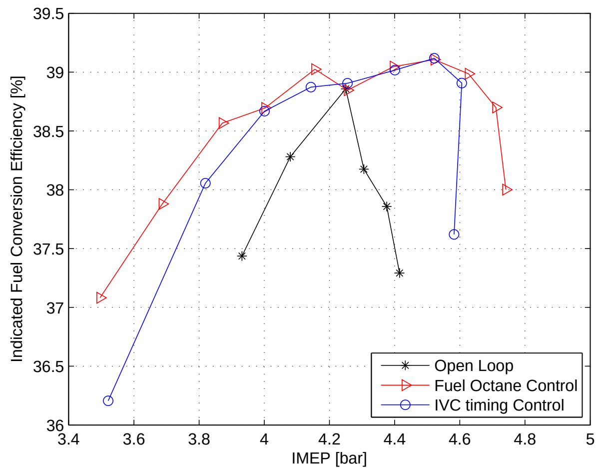

Figure 12: Indicated efficiency for the load range sweep for open loop, fuel octane control and IVC control for $BP3$.

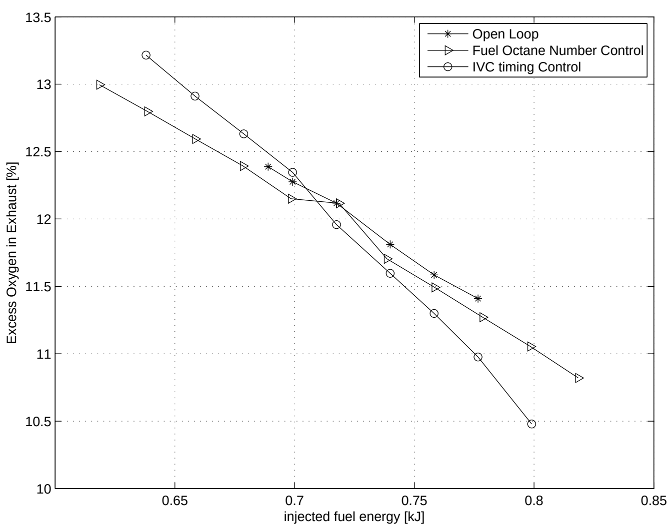

Figure 13: Exhaust oxygen percent for load range sweep for open loop, fuel octane control, and IVC control for $BP3$.

SUMMARY

Single input single output closed loop control of HCCI combustion timing is performed using PI control. IVC timing and fuel octane are individually used to control the combustion timing and their ability to reject injected fuel energy or engine speed disturbances are directly compared.

- Closed loop control of HCCI combustion timing can be achieved by either IVC timing or fuel octane. Proportional Integral controllers provide good performance at all five engine conditions tested in this study. During modest step disturbances in engine speed and injected fuel energy the combustion is kept within the stable range, no knocking combustion or high cyclic variation.
- Fuel octane control provides a larger load range. The use of IVC timing to control the combustion timing has adverse effects on the dilution of the mixture which affects both the knock and misfire limits of the engine. The use of fuel octane to control combustion timing does not effect the dilution.
- For the engine speeds tested the dynamics of the fuel system seem to be slower than the dynamics of the camshaft phaser – slower combustion timing response is seen for step change in fuel octane than for IVC timing step changes. Thus faster control is achievable with IVC timing if no compensation is done for the fuel dynamics

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