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Mode-Switching Development for a Natural Gas SI-HCCI Engine

Jason Boddez, David Checkel, and Charles Robert Koch

Dept. of Mechanical Engineering, University of Alberta, Edmonton, AB, T6G 2G8

Abstract

A modified CFR engine is used to explore the transition between idling in spark ignition (SI) to mid-load homogeneous charge compression ignition (HCCI) when fuelled with port-injected compressed natural gas (CNG) at a constant engine speed. Mechanical complexity is minimized by utilizing only throttle and fuel control during the mode-switch. This method could be of utility to cold start an engine in SI mode that is designed to operate primarily in HCCI without requiring advanced control such as variable valve technology.

Open-loop fuel control is found to be insufficient to provide stable HCCI operation immediately following the mode-switch from SI to HCCI, so feedback is implemented using the measured maximum rate of pressure rise. Several cycles of reduced power output are experienced during initial HCCI operation due to advanced combustion timing and the lower fuel delivery necessary to avoid knock. Hot residual gas and a hot exhaust valve from prior SI operation are thought to be the cause. This understanding may be applied to develop a more stable mode-switch in future work.

1. Introduction

The Homogeneous Charge Compression Ignition (HCCI) engine cycle has shown promise of providing the desirable combination of high thermodynamic efficiency, similar to a compression ignition (CI) engine, and low NO_x and soot emissions, similar to a spark ignition (SI) engine. There remain, however, difficulties that hinder implementation in production engines such as cold starting, idling, and limitations in the high load and speed-range. One solution is to allow operation in another engine mode, SI or CI, to satisfy these requirements. Although this provides an appealing solution, it involves the added difficulty of controlling the necessary and possibly frequent transitions between the two modes. To accomplish this, a method must be developed that offers flexible, rapid, and repeatable mode-switching using existing engine inputs when possible.

Several studies have already presented viable mode-switching strategies with variable valve lift and timing systems [1-*5*] and fully variable valves [2]. Initial testing has also been carried out utilizing spark assist and variable compression ratio [6] while fast thermal management (FTM) may also offer potential. Each approach has certain tradeoffs in terms of switch duration, smoothness, mechanical/control complexity, and flexibility. While these methods are capable of providing a large degree of control over operating conditions to achieve a mode-switch, a simpler approach may be desirable for applications that do not require the added complexity of these controls during normal operation.

The purpose of this study is to explore meeting the mode-switching requirement through the use of only throttle and fuel control. By avoiding advanced mechanical methods, it becomes possible to develop an engine to operate primarily in HCCI while allowing for cold starting with the addition of a spark plug. Compressed natural gas (CNG) is chosen for testing as it is a common fuel for stationary applications where power fluctuations are slower and advanced control for fast engine response becomes less desirable due to increased maintenance and cost.

The main focus in this study is power produced during mode-switching, measured as indicated mean effective pressure (IMEP), and acceptable maximum rate of pressure rise (ROPR). Satisfactory engine operation is defined between two conditions: positive IMEP to rule out misfire, and an ROPR less than the knock threshold of 10 bar per crank angle degrees (bar/CAD) [7, *8*]. Combustion quality and emissions are generally worse than optimum during a mode-switch, so the change will be made as quickly as possible. When implemented in a production engine, a quick transition to a more stable operating point will occur soon after the mode-switch is complete [*9-12*]; however, for this study the testing is carried out between two steady-state conditions to allow a systematic study of only the variables of interest.

Transient emissions measurements such as those experienced during mode-switching are difficult to measure and are beyond the scope of this study,

2. Experimental Setup

Experimental testing is carried out on a Waukesha Cooperative Fuels Research (CFR) engine as described in Table 1. An optical research head provides several access ports to the combustion chamber for installation of a spark plug and Kistler 6043B piezoelectric pressure transducer with a Kistler 507 charge amplifier. Compression ratio is variable through adjustment of the cylinder sleeve height, but remains constant for all tests in this study. Effective compression ratio is evaluated by measuring the maximum pressure during motoring with wide open throttle and compared with values achieved in a single zone engine model [7]. Average cylinder wall temperature is measured using a plug drilled to 5mm from the inner surface and installed in a combustion chamber access port with a Type T thermocouple. Engine air mass flow is measured using a TSI 4235 meter installed upstream of a 210L pulsation dampening tank and a Woodward 1 inch L-series electronic throttle. Supercharging is obtained by pressurizing the intake system with the building air supply. The system is controlled by a program developed in LabVIEW running on a 2.4 GHz dual-core Pentium 4 desktop computer, with fuel injection timed by a remote PXI 1010 chassis. All data collection and controls are synchronized on a cycle basis with analysis parameters (IMEP, ROPR, CA50) calculated in real-time.

Table 1: Engine specifications

Cylinders	1
Bore (mm)	82.6
Stroke (mm)	114.3
Displacement (L)	0.612
Effective Compression Ratio	17.3
Valves per Cylinder	2
Valve Timing	
IVO (°aTDC)	10
IVC (°aBDC)	34
EVO (°bBDC)	40
EVC (°aTDC)	15

Compressed natural gas (CNG) is port injected in both SI and HCCI modes near the intake valve in a large (~11 L) electrically heated intake manifold. Type T thermocouples measure intake air temperature downstream of the heater and mixture temperature immediately outside the intake valve where room temperature CNG is mixed with hot intake air. The high auto-ignition temperature of natural gas (approximately 1100 K) requires the use of a high compression ratio along with intake air heating to achieve HCCI combustion, both of which increase the tendency of knock in SI. Exhaust gas recirculation (EGR) can be used to reduce knock in SI mode, but this added complexity is not implemented. Instead, late spark timing and reduced intake pressures are used to avoid knock during SI operation.

3. Results and Discussion

Preliminary testing was carried out to find the limits of operation in both SI and HCCI when only intake pressure and CNG fuel rate are varied. Although intake temperature is generally very different in the two modes, being around room temperature for SI and higher than 100 °C in HCCI when additional internal EGR is not used, two points with overlapping conditions were found as shown in Table 2. For this engine, the range of acceptable inputs at these conditions is very limited in both modes. Brake power output is small in SI under these conditions due to the significant friction inherent in the CFR engine. While this represents an idle condition for the test engine, a typical production engine would have much less friction and this would thus be producing positive power output.

Table 2: Initial and final SI-HCCI steady-state operating conditions

	SI	HCCI
Engine Speed (RPM)	700	700
Lambda	1.05	2.55
Air Mass Flow Rate (g/s)	1.39	3.67
CNG Mass Flow Rate (mg/s)	76.9	83.4
CNG Injection Timing (°bTDC)	200	200
CNG Pulse Width (ms)	11.4	12.2
Cylinder Head Temperature (°C)	111	114
Exhaust Temperature (°C)	424	238
Intake Air Temperature (°C)	110	113
Mixture Temperature (°C)	92	96
IMEP (bar)	2.6	4.2
Intake Pressure (kPa)	53	120
Spark Timing (°bTDC)	-5	N/A

The procedure to mode-switch between the two states combines open and closed loop control. Intake pressure is controlled using throttle position in a PID manually tuned for a pressure rise over 3 engine cycles with a 10 kPa overshoot. Faster response time is possible using open loop throttle control, but this method is sensitive to building air variations. Fuel was adjusted on an open loop cycle-by-cycle basis with a desire to achieve combustion without misfire or knock. The maximum achievable IMEP was thus developed on each cycle to approach the new steady-state HCCI operating point as quickly as possible.

Figure 1 shows a set of 10 tests conducted following this approach. The last SI cycle is the point where the mode-switch has initiated and indicated as cycle 0 in all subsequent plots. The IMEP values have a large variation that develops after the first 8 cycles of HCCI as several tests experience partial combustion. For the low intake temperatures used, the HCCI operating region is narrow and unstable, so achieving repeatable results with the same input values is not always possible. To rectify this situation and allow for more consistent power output, closed loop feedback was adopted to fine-tune the fuel sequence. Using a moving average of 5 ROPR values as the reference input, a PID controller adjusts the CNG pulse width by a maximum of ±1 ms towards a target ROPR of 6 bar/CAD. Figure 2 shows the results of 10 tests with this control and more stable HCCI operation is achieved as expected.

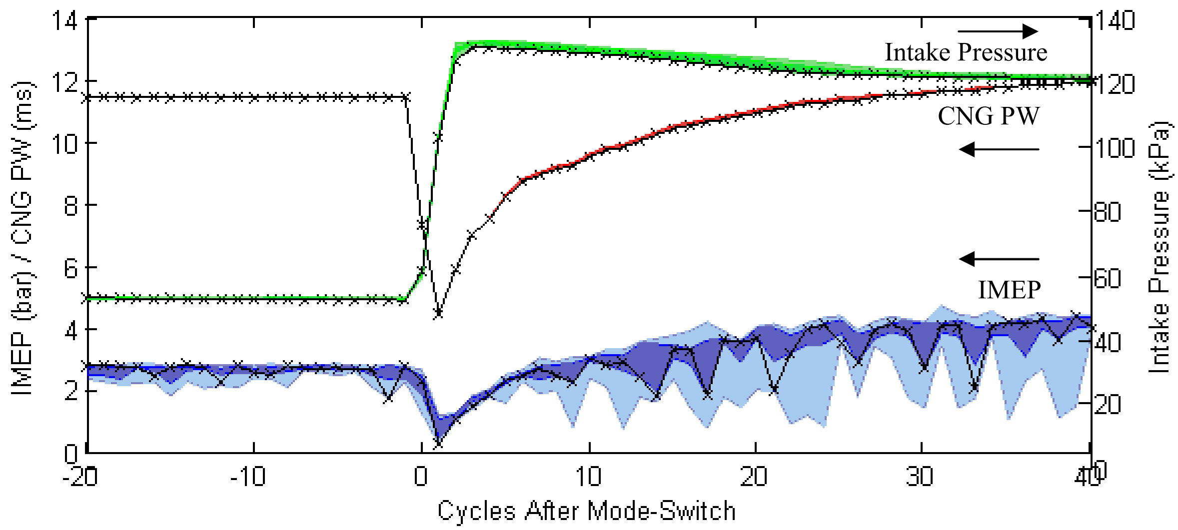

Figure 1: Ten repeated mode-switches with open loop control of CNG pulse width (PW). Outer limits for all tests are shown in a lighter color, while darker regions exclude outliers. Black lines show one test.

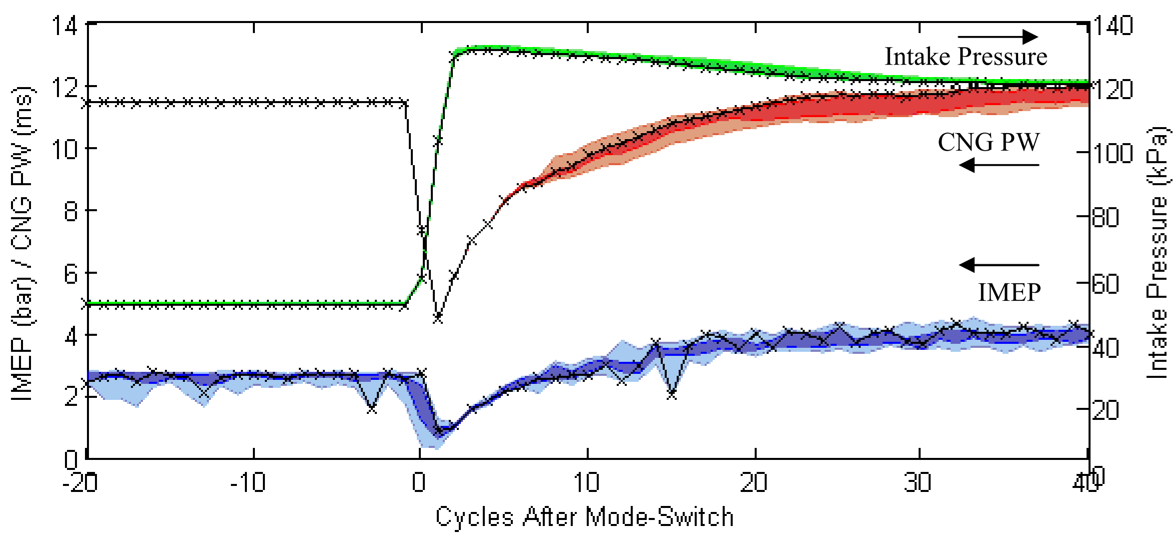

Figure 2: Ten mode-switch tests using the same fuel sequence as in Figure 1, but with PID feedback on CNG pulse width. Black lines provide one test.

The IMEP traces shown in Figure 1 and Figure 2 reveal a reduction in power output for the initial HCCI cycles for all tests. As mentioned previously, the fuel delivered each cycle was adjusted to avoid knock while still supporting ignition. The lower fuel values used during the initial HCCI cycles dictate that they are unable to produce higher levels of output, but further investigation is necessary to determine why this fuel limitation exists during the mode-switch. Figure 3 shows the cylinder pressure traces for the cycles that occur during the mode switch. Intake pressure has already started to increase during cycle 0, indicated by the overall higher pressure curve, while fuel delivery has been reduced to avoid knock with the higher compression temperatures experienced. The next cycle is HCCI combustion since the spark has been deactivated, and this is supported by a sharp increase in pressure near top dead center (TDC) characteristic of HCCI. This is unexpected as the intake pressure has not yet reached the required level. Hot residual gas from the previous SI cycle must assist ignition of the lean mixture since compression temperature alone is not sufficient at this point. After this cycle, the intake stroke occurs at or above the desired intake pressures and are expected to produce sufficient compression temperature to initiate auto-ignition.

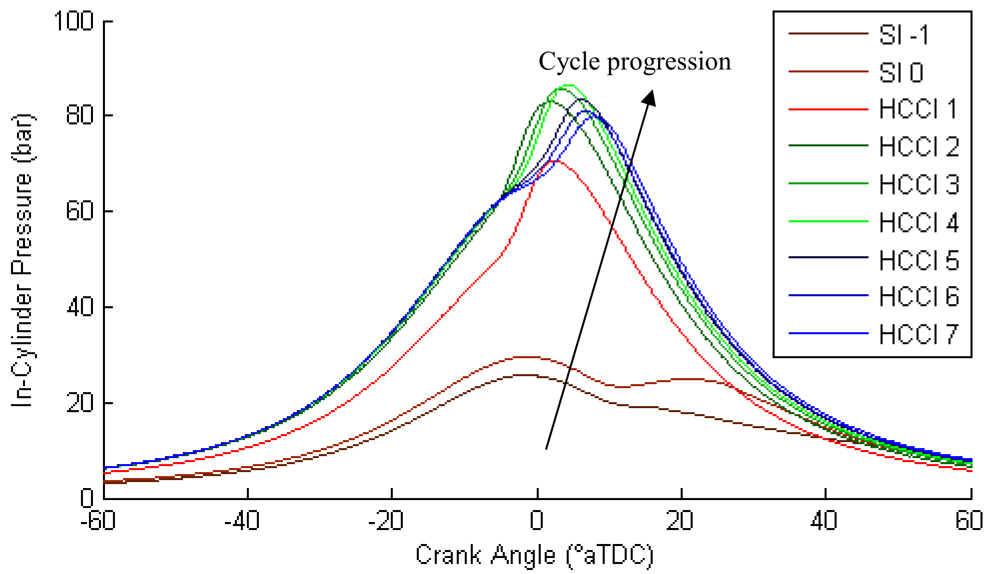

Figure 3: Cylinder pressure traces during SI-HCCI mode-switch for the test point shown in Figure 2.

Another observation from the pressure traces is variable HCCI combustion timing. This property is evaluated with the crank angle at which 50% of the fuel energy has been released (CA50). Heat release is determined from the pressure traces and cumulatively integrated to determine the crank position where half the maximum value is reached, while allowing for heat loss using the modified Woschni heat transfer model and the measured cylinder head temperature [13]. Figure 4 shows the resulting CA50 values for the test presented in Figure 2 and Figure 3 indicating the expected trend of advanced but slowly retarding combustion timing during the initial 10-15 HCCI cycles following the mode-switch. ROPR values are included showing that combustion severity has been maintained at a reasonable level throughout the mode-switch. HCCI operation approaches misfire as the ROPR decreases to approximately 2 bar/CAD, as governed by the compression stroke, but improves as the fuel controller increases CNG injection pulse width.

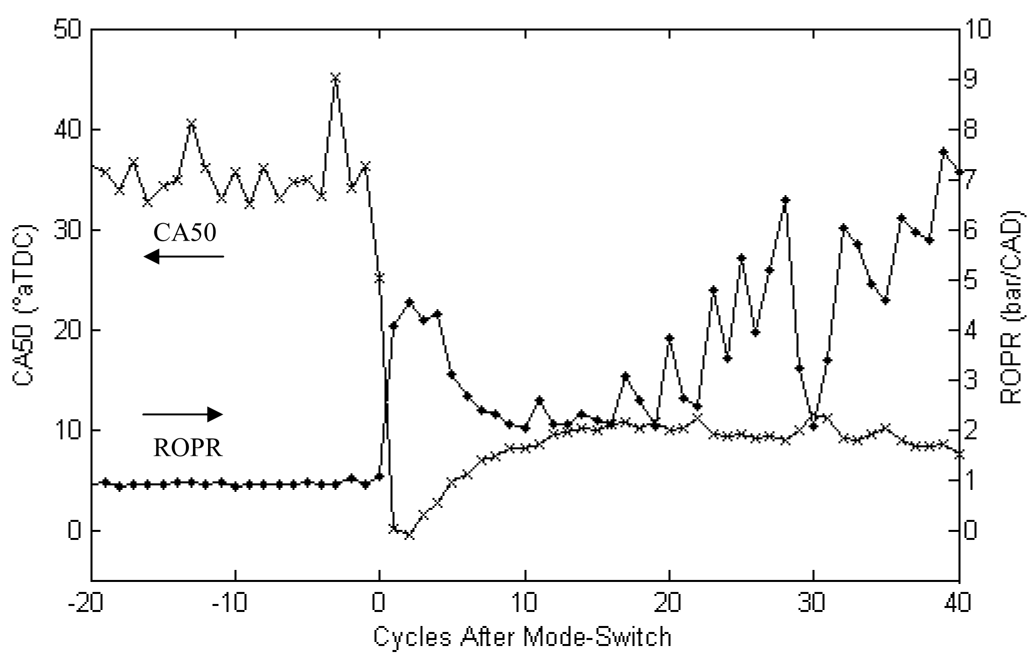

Figure 4: Combustion timing and maximum rate of pressure rise for the test shown in Figure 2 and Figure 3.

While SI residual gas is the cause for advanced ignition during the first HCCI cycle, it becomes increasingly unlikely that hot SI exhaust gases are present in the residual on subsequent cycles. On the other hand, SI exhaust also has the effect of heating the exhaust valve. This allows for a hot spot in the combustion chamber that triggers early auto-ignition of the bulk mixture [3]. The 10-15 cycles of advanced combustion timing experienced then correspond to the time required for the exhaust valve to cool with the lower HCCI exhaust temperature. Mode-switch development can benefit from this understanding to coordinate the available inputs. Adopting methods to retard combustion timing for the initial HCCI cycles following a mode-switch should be the priority rather than approaching the desired steady-state conditions as quickly as possible. For the tested scenario, a slower increase in intake pressure without overshoot may retard combustion timing by reducing compression temperatures, and higher fuel delivery could then provide higher power output to lessen or remove the drop in IMEP. External EGR may also be considered to help prevent knock in SI, then stabilize and retard combustion in HCCI; however, additional consideration of the changing EGR composition following the mode-switch will be necessary due to its influence on combustion.

4. Concluding Remarks

Experimental testing was carried out on a modified CFR engine to develop a mechanically simplified method to mode-switch between SI and HCCI operation. Compatible steady-state operating points in both SI and HCCI modes were found with the same intake temperature such that only intake pressure and CNG pulse width differed, both of which can be changed in a few engine cycles. A PID controller was developed to control intake pressure step commands using the throttle while fuel was controlled open loop at each cycle to allow for reasonable combustion during all cycles without misfire or knock. Then, a feedback controller was augmented to adjust the fuel sequence 5 cycles after transition to achieve a target maximum rate of pressure rise of 6 bar/CAD in order to stabilize operation following the mode-switch. Indicated power output drops for the first 5 cycles due to advanced combustion timing requiring a significant reduction in fuel, but then steadily increases to the higher steady-state HCCI value.

Advanced combustion ignition timing (CA50) during the first 10 cycles of HCCI operation is found to be the limiting factor immediately following the mode-switch. Hot SI residual gases allow for HCCI combustion during the first HCCI cycle, then a hot exhaust valve explains the subsequent slow reduction of advanced combustion timing as the valve cools due to lower HCCI exhaust temperatures. Methods to retard combustion should be a priority during this period rather than approaching the desired steady-state operating conditions as quickly as possible. One way to do this with these actuators is to slow the rise of intake pressure and increase the fuel delivery.

Acknowledgements

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