Closed-Loop Combustion Control of HCCI Engines

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Abstract

The HCCI engine, with its excellent potential for high efficiency and low NO_X emissions, is investigated from a control perspective. Combustion timing, *i.e.*, where in the thermodynamic cycle combustion takes place, is identified as the most challenging problem with HCCI engine control. A number of different means for controlling combustion timing are suggested, and results using a dual-fuel solution are presented. This solution uses two fuels with different ignition characteristics to control the time of auto-ignition. Cylinder pressure measurement is suggested for feedback of combustion timing. A simple net-heat release algorithm is applied to the measurements, and the crank angle of 50 % burnt is extracted. Open-loop instability is detected in some high-load regions of the operating range. This phenomenon is explained by positive feedback between the cylinder wall heating and ignition timing processes. Closed-loop performance is hampered by time delays and model uncertainties. This problem is particularly pronounced at operating points that are open-loop unstable.

14.1 Homogeneous Charge Compression Ignition (HCCI)

The Homogeneous Charge Compression Ignition (HCCI) engine with its excellent potential for combining low exhaust emissions with high efficiency gained substantial interest towards the end of the 20th century. The HCCI engine combines features of the traditional spark ignited (SI) Otto-cycle and compression ignited (CI) Diesel-cycle engines into something that must be characterized as a separate engine concept.

The HCCI Engine Concept

The HCCI engine combines features of the traditional SI and CI engines into a new engine concept. The HCCI engine features homogeneous charge like the SI engine, and compression ignition like the CI engine. HCCI operation can be two-stroke or four-stroke, and the first studies [19], [15] were performed on two-stroke engines. Later studies [23], [3] on four-stroke engines show that high efficiency can be combined with low NO_X emissions for HCCI engines running with a high compression ratio and lean operation. This text will focus exclusively on the four-stroke version of the HCCI engine.

The HCCI Cycle

The four-stroke HCCI cycle can be described by its four strokes: *intake*, *compression*, *expansion*, and *exhaust*. During the *intake stroke*, a more or less homogeneous mix of fuel and air is inducted into the cylinder. During the *compression stroke*, this charge is compressed by the upward motion of the piston. Towards the end of the *compression stroke*, temperature and pressure have reached levels where pre-combustion reactions start to take place. Somewhere near the TDC (top dead center), actual combustion starts. During the initial part of the *expansion stroke*, the bulk of combustion takes place during the course of a few crank-angle degrees. During the rest of the *expansion stroke*, the high pressure caused by combustion forces the piston down towards BDC (bottom dead center). During the *exhaust stroke*, the upward motion of the piston forces the exhaust gas to leave the cylinder through the exhaust valve.

Ignition. An HCCI engine, contrary to SI and diesel-cycle CI engines, has no direct means for controlling ignition timing. The SI engine has spark timing, and the diesel-cycle CI engine has the start of fuel injection, which both directly control the onset of combustion. However, for an HCCI engine, ignition timing is dictated by the conditions of the charge and the cylinder walls at the time when the intake valve closes. This is one of the biggest challenges with practical implementation of HCCI engine technology. Ignition timing can only be controlled indirectly through adjustments in the cylinder charge preparation. The following paragraphs will describe the most important parameters that control ignition timing for an HCCI engine.

The temperature of the air when it enters the cylinder has a large influence on the charge temperature towards the end of the compression stroke. With a compression ratio of 18:1, a change in intake temperature by 30 K will result in a change in temperature at TDC by almost 100 K. Since temperature is a very important factor in auto-ignition, an increase in intake temperature will have a very strong advancing influence on ignition timing.

The portion of the exhaust gas that is not expelled during the exhaust stroke, the residual gas, is particularly important for HCCI operation. The thermal energy provided by the residual gas contributes in heating the charge of the following cycle, and affects the crank angle at which ignition takes place. On an engine with variable valve timing, the residual-gas fraction can be controlled—*e.g.*, by early closing of the exhaust valve, which will trap a larger amount of exhaust gas in the cylinder for the following cycle. It is necessary to remember though that exhaust gas also acts as a diluent, and thereby slows down the combustion chemistry. This will tend to retard ignition timing, and with a very high residual-gas fraction this effect will dominate.

Closely related to residual gases is EGR (exhaust gas recirculation). This refers to exhaust gas that is routed back from the exhaust manifold to the intake manifold. Combined with an EGR cooler, this can be used for diluting the charge and thus lowering the reaction rate. An increase in EGR rate will retard ignition timing.

Another important factor is the cylinder-wall temperature. Hot cylinder walls will heat the charge throughout the intake and compression strokes, and will advance ignition timing.

The fuel-air equivalence ratio affects both fuel concentration and oxygen concentration. However, since HCCI engines operate lean, the equivalence ratio has a stronger influence on fuel concentration than on oxygen concentration. The dominating effect of increasing the equivalence ratio, thus, is an increase in fuel concentration, which will result in a higher reaction rate. Thus, increasing the equivalence ratio serves to advance ignition timing.

Another possible way to control ignition timing is by changing the fuel composition. Addition of a second fuel with higher reactivity will serve to advance ignition timing. Examples are the addition of hydrogen to natural gas and n-heptane to iso-octane.

A variable compression ratio provides an effective means of controlling the temperature towards the end of the compression stroke. A higher compression ratio increases the charge temperature near the TDC, and tends to advance ignition timing.

Charge stratification—*i.e.*, inhomogeneous charge distribution—can be used to locally increase the equivalence ratio, and thus the reaction rate, in order to advance the ignition timing. Charge stratification can be achieved through late fuel injection. The drawback is locally high temperatures, causing an increase in NO_X production.

Evidently, there are many parameters that affect ignition timing, but they all do so in non-trivial ways, and furthermore, many of the parameters affect each other as well. Some of the parameters are even affected by ignition timing itself. The cylinder wall temperature, *e.g.*, increases with advanced ignition timing. When ignition timing is advanced, the peak cylinder temperature increases which, in turn, causes an increase in cylinder wall temperature. It follows that ignition timing is very sensitive to operating conditions

14.2 Closed-loop Control of Ignition Timing

It is evident from above that ignition control is much more of a challenge for an HCCI engine than for an SI or diesel-cycle CI engine. The most readily available means of controlling ignition timing is by adjusting the fuel composition. This does not require any novel mechanical design like variable valve timing or variable compression ratio. It merely requires a doubling of the port fuel injection system.

Selection of feedback. The sensitivity of ignition timing to operating conditions does not allow an open-loop solution in the form of *e.g.*, a look-up table. Furthermore, the system becomes unstable for some operating conditions at high load. Thus, closed-loop control is an absolute necessity, which poses the question of what to use for feedback. Cylinder pressure is the natural choice, since ignition is an in-cylinder phenomenon. What characteristic of the cylinder pressure trace reflects when combustion takes place, though?

The crank angle of maximum pressure gives some information about when the bulk of combustion is taking place, but for combustion timing before or near the TDC, this angle tends to gravitate towards TDC due to the dependence on volume in the ideal gas law. Furthermore, for very late combustion timing, the pressure maximum from compression dominates the one from combustion. Another problem is that a maximum has a certain flatness to it, which makes it non-unique.

The crank angle of maximum pressure derivative suffers from the same problems as the crank angle of maximum pressure in addition to the inherent noise problems with numerical differentiation. Another possibility is to search for the inflection point, where the pressure trace transitions are from negative to a positive second derivative due to the onset of combustion. This also suffers from the problems with numerical differentiation.

It turns out that a first-law analysis based on the pressure measurements and heat release analysis provides a very robust source of feedback. Heat release analysis applies the first law of thermodynamics to the combustion chamber during the entire combustion event in order to estimate the rate at which chemical energy is converted to thermal energy. If no adjustments are made for heat transfer or flow into and out of crevices, the net heat release is obtained. Integration with respect to the crank angle yields the cumulative heat release, which roughly reflects the mass fraction burned.

Combustion in an HCCI engine is usually very fast. The mass fraction burned usually goes from 10 % to 90 % in about 5 crank angle degrees, which means that the crank angle of 50 % heat release, CA50, provides a very accurate measure of when combustion is taking place. In the following, CA50, combustion timing, and ignition timing will be used interchangeably to denote the crank angle of 50 % heat release.

Processing cylinder pressure measurements. Cylinder pressure measurements are normally performed with either piezoelectric elements combined with charge amplifiers or with fiber-optical sensors. Both methods fail to measure the DC component of the cylinder pressure. A thermodynamically-based method of estimating the DC component is detailed in [26], and amounts to estimating an initial pressure and a measurement offset based on pressure measurements during the compression stroke. It is essential to select the crank-angle interval for estimation between the intake-valve closing and the start of combustion for the thermodynamic assumptions to hold.

The cylinder pressure measurements, p_m, can be decomposed into the actual pressure, p, and a sensor offset, Δp, according to (14.1).

$$p_m = p + \Delta p \tag{14.1}$$

The real pressure can be modeled with polytropic compression:

$$p = CV^{-\kappa} \tag{14.2}$$

where V is the combustion chamber volume, κ is the polytropic exponent, and C depends on the initial pressure according to:

$$C = p_0 V_0^{\kappa} \tag{14.3}$$

If the polytropic exponent is assumed to be known (normally between 1.3 and 1.4), two parameters need to be estimated from the measurements: C and Δp. Thus, the vector of estimates is given by:

$$\theta = \begin{pmatrix} \Delta p \\ C \end{pmatrix} \tag{14.4}$$

The output vector, assuming n measurements, is given by:

$$Y = \begin{pmatrix} y_1 \\ \vdots \\ y_n \end{pmatrix} = \begin{pmatrix} (p_m)_1 \\ \vdots \\ (p_m)_n \end{pmatrix} \tag{14.5}$$

and the regression matrix is given by:

$$\Phi = \begin{pmatrix} \varphi_1 \\ \vdots \\ \varphi_n \end{pmatrix} = \begin{pmatrix} 1 & V_1^{-\kappa} \\ \vdots & \vdots \\ 1 & V_n^{-\kappa} \end{pmatrix} \tag{14.6}$$

The least squares estimate of θ is then given by (14.7).

$$\hat{\theta} = \left(\Phi^T \Phi\right)^{-1} \Phi^T Y = \Phi^+ Y \tag{14.7}$$

where Φ^+ is the Moore-Penrose pseudo inverse of Φ.

In [26], a method of estimating the polytropic exponent is also provided. In cases where the polytropic exponent is thought to vary significantly from cycle to cycle, this method can be used. Intake temperature and fuel composition as well as the equivalence ratio affect the polytropic exponent.

Heat release analysis. An analysis of the combustion chamber, based on the first law of thermodynamics, relates the rate at which chemical energy is converted to thermal energy to the pressure in the combustion chamber. This type of analysis is conventionally called *heat release analysis*, and can be used to determine when combustion is taking place. This term stems from the simplification that is normally done,in which the charge composition is assumed to be constant, and that the increase in internal energy is interpreted as heat. If the actual heat transfer to the cylinder walls as well as crevice flow is neglected, equation (14.8) relates heat release to cylinder pressure:

$$\delta Q_{ch} = \frac{c_v}{R} V \mathrm{d}p + \frac{c_p}{R} p \mathrm{d}V \tag{14.8}$$

This equation is integrated over a crank angle interval which includes the whole combustion event. The parameters c_v and c_p are the specific heats at constant volume and pressure, respectively, and technically depend on temperature. However, if the only objective is to determine combustion timing, they can be assumed to be constant. R is the universal gas constant.

The result of the integration of the heat release equation is the cumulative net heat release as a function of crank angle, $Q_{ch}(\alpha)$. A typical heat release trace is plotted in Figure 14.1 together with some definitions. The most important definition in this context is CA50, the crank angle of 50 % heat release. Since combustion is very fast, CA50 can be used as a robust source of feedback for combustion timing.

14.3 Closed-Loop Combustion Control of HCCI Engines

Feedback structure. Two fuel injectors and one cylinder pressure sensor per cylinder allows separate control loops for each cylinder. Thus, the

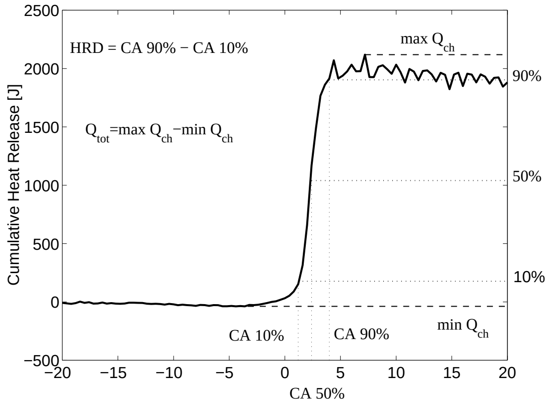

Figure 14.1 Definitions of some heat-release based cycle parameters

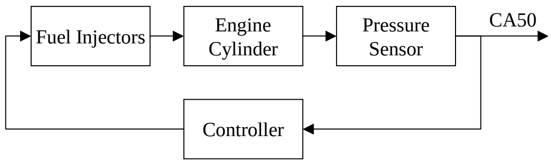

Figure 14.2 Control structure for combustion timing control. The CA50 based on cylinder pressure measurements provides feedback about combustion timing, and octane ratio provides a control input.

control structure indicated in Figure 14.2 can be used for the combustion timing control of each cylinder. Fuel octane number is a measure of a fuel's resistance to auto-ignition, and can be used as the control input for combustion timing control of an HCCI engine cylinder. When using a mixture of iso-octane and n-heptane, the octane number is, by definition, the percentage of iso-octane.

The plant. The sensitivity of CA50 to changes in fuel octane number varies by orders of magnitude for different operating points, see Figure 14.3. Each line in the plot represents a specific intake temperature and load. Within each line, the fuel octane number has been varied to achieve an interval of combustion timings. The strong nonlinearity of the plant makes a linear controller unsuitable for the task. The situation can be remedied, however, if the sensitivity is mapped over the multi-dimensional space of operating conditions. This map can be used for gain-scheduling the otherwise linear controller.

In [18], a multivariable function is fitted to measurements of the sensitivity of CA50 to changes in octane number for a multitude of operating conditions. In order to get a simple, computationally inexpensive model, the

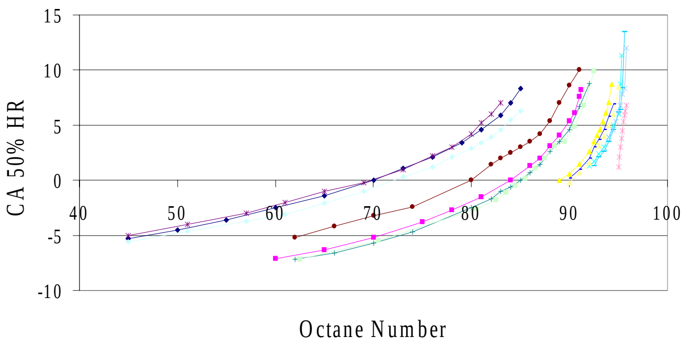

Figure 14.3 Combustion timing versus fuel octane number for various operating points. Measurements on a Scania D12 6-cylinder engine converted for HCCI operation. Octane number varied through fueling with a variable mixture of iso-octane and n-heptane

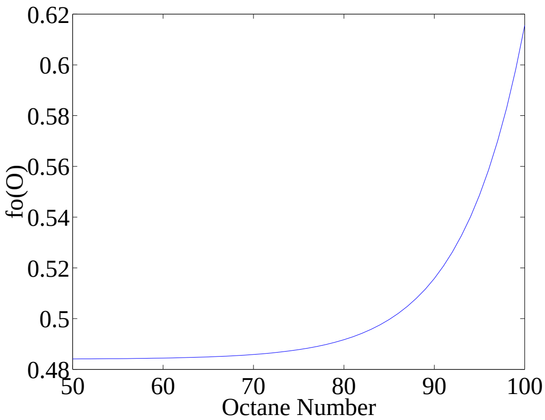

Figure 14.4 The octane number component of the sensitivity function

sensitivity is modeled as a product of functions of one variable each. This approach is entirely empirical, but yields a sensitivity model with acceptable residuals (within 3 %). The variables that are included in this model are engine speed, inlet air temperature, fuel octane number, fuel mass per cycle, and CA50. A later model revision includes inlet pressure as well. Figure 14.4 shows the octane number component of the sensitivity function. The sensitivity model is used for scaling the controller gains, which implicitly assumes that the dynamic behavior of the plant is independent of the operating point. Only the DC gain of the plant changes. For the case of hydrogen enrichment of natural gas, the sensitivity change with respect to operating condition is more modest. Figure 14.5 shows the hydrogen requirement as a function of CA50 for two different engine loads, and the

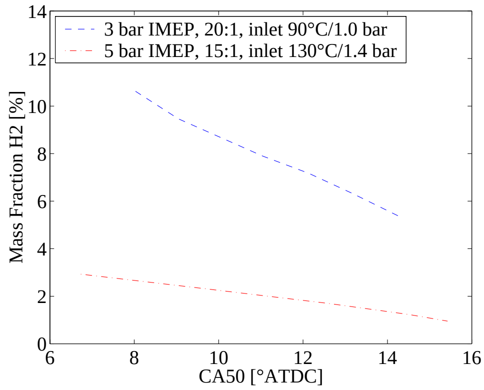

Figure 14.5 Dependence of CA50 on H_2 fraction for light load (3 bar IMEP) and medium load (5 bar IMEP)

dependence is roughly linear within each engine load.

Open-loop stability. An interesting phenomenon that appears in some regions of the operating space of an HCCI engine is open-loop instability. This phenomenon results when wall temperature effects dominate the ignition dynamics. Figure 14.6 shows the open-loop behavior at a stable and an unstable operating point, respectively. All control inputs are held constant in both cases. The effect of open-loop instability under closed-loop operation is non-minimum-phase behavior, *i.e.*, the control input starts out in the “wrong” direction after a setpoint change (compare Figures 14.7 and 14.8).

The cause of instability is the positive thermal feedback provided through the interaction between ignition timing and cylinder wall temperature. A small increase in cylinder wall temperature results in a hotter cylinder charge, which advances ignition timing. Advanced ignition timing, however, results in higher gas temperature and more heat transfer to the walls, thus higher wall temperature. The reversed case is a small drop in cylinder wall temperature, which results in cooler cylinder charge. Ignition timing is retarded, which results in lower gas temperature, which in turn reduces the heat transfer to the walls, and thus the wall temperature. It is evident that operating points where this effect dominates are unstable.

The positive feedback mentioned above is always present, but not all operating points are unstable. The stabilizing negative feedback responsible is closely related to the destabilizing positive feedback. Early ignition leads to high peak temperature and heat transfer, but this results in lower gas temperature towards the end of the cycle, which means both colder residual gas and more of it. This reduces the reactivity of the charge for the next cycle, and retards ignition timing. The opposite holds for late ignition. Thus,

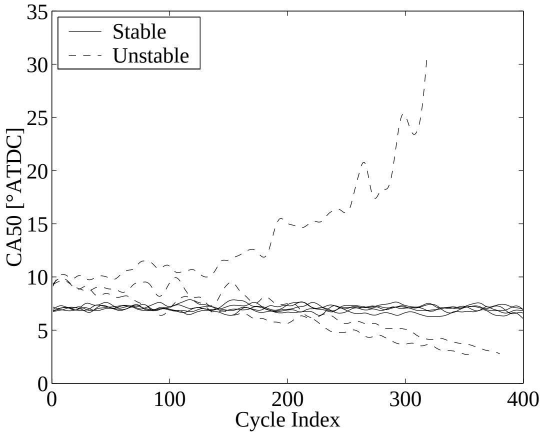

Figure 14.6 Repeated open-loop operation at one *stable* and one *unstable* operating point

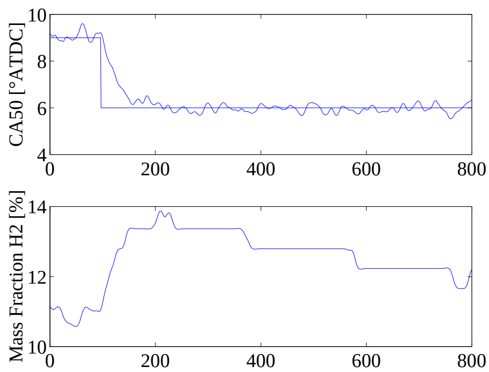

Figure 14.7 Change of CA50 setpoint at a stable operating point. The independent axis indicates number of engine cycles

the residual gas provides the stabilizing negative feedback.

Closed-loop Performance

The bandwidth of the closed-loop system is limited by time delays. In the case of pulse width modulated fuel injection using solenoid injectors, the time between fuel injection command and actual fuel injection is very small. For hydrogen enrichment of natural gas however, a mass flow controller with its own very conservative control system is used, which increases the lag from command to actual change in fuel injection.

The most severe time delay is, however, in the control system itself.

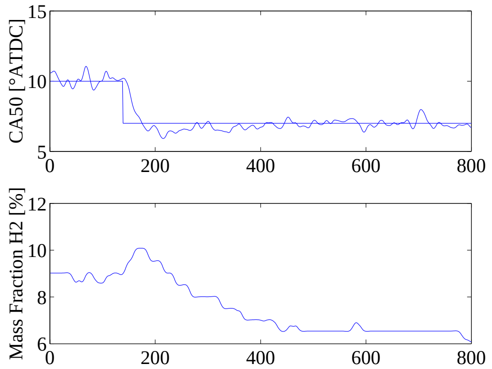

Figure 14.8 Change of CA50 setpoint at an unstable operating point. The independent axis indicates number of engine cycles

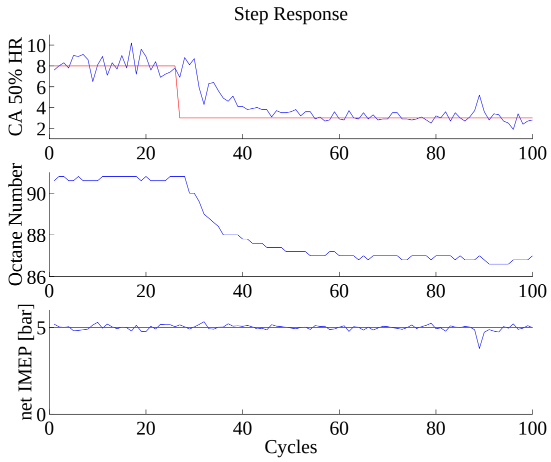

Figure 14.9 Change of combustion timing setpoint. Operation with iso-octane and n-heptane. The time delay is clearly seen in the response.

Data handling and communication with fuel-injector drivers causes a delay between a change in measurement and a corresponding change in control input of approximately four engine cycles, see Figure 14.9. With this delay present, it is, of course, impossible to achieve closed-loop bandwidth better than a few multiples of the delay.

14.4 Conclusion and Discussion

The field of closed-loop control of HCCI engines is a very new one, and a lot remains to be done. This text aims to show the potential and to highlight the difficulties. It is shown that closed-loop control of ignition timing can be achieved using a set-up with a secondary fuel as ignition improver. Closing the loop does, however, require a measurement also. The preferred measurement is the crank angle of 50 % heat release, CA50, which offers a robust measure of when the rapid HCCI combustion is taking place. In lieu of a CA50 sensor, CA50 has to be computed from crank-angle based cylinder pressure measurements.

Both the choice of control input and the choice of measurement can be questioned. A secondary fuel is impractical on a vehicle, unless it can be produced on-board. For a stationary application, it may be acceptable, however. Using cylinder pressure as a measurement is expensive, but cheaper cylinder pressure sensors can be expected in the future. For control input, other solutions exist; *e.g.*, variable compression ratio and variable valve timing. For feedback, however, there is no good alternative to cylinder pressure measurements.

Transient performance is limited at present. It is the authors' opinion, however, that most of this problem is due to time delays in the control system. A physical change in CA50 takes in the order of four engine cycles to propagate through the control system to a physical change in fuel injection. This delay is mostly due to inefficient communication and data handling, and should be possible to cut down to around one engine cycle.

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