

Model-Based Control of Combustion Phasing in an HCCI Engine

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ABSTRACT

Robust control of combustion phasing in Homogenous Charge Compression Ignition (HCCI) engines is a well-recognized challenge limiting the automotive industry for exploiting HCCI benefits in mass production vehicles. Real-time model-based control of combustion phasing is the key to tackle this daunting challenge. In this paper, a new control oriented model is developed for predicting HCCI combustion phasing over a range of engine operation. The model is validated against the experimental data from a single cylinder Ricardo engine. A model-based integral state feedback controller is designed to control HCCI combustion phasing by modulating the ratio of two Primary Reference Fuels (PRFs). The controller's performance is compared with a manually tuned proportional integral controller. The simulation results demonstrate that while both controllers have proper tracking performance, the model-based state feedback controller is more effective in rejecting the external physical disturbances and keeping the engine operation within the desirable range.

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INTRODUCTION

High fuel efficiency and negligible nitrogen oxide (NO_x) and particulate matter emissions have made HCCI engines attractive to main automotive OEMs particularly in the past decade [1, 2]. But lacks of direct means to trigger combustion and consequent control complexity have been the main obstacles limiting application of HCCI engines in the automotive industry. Undesirable combustion phasing deteriorates HCCI operation and leads to high cyclic variation and also high unburned hydrocarbon (uHC) and carbon monoxide (CO) emissions [3, 4]. A number of strategies have been used to closed-loop control of HCCI combustion phasing including: altering the fueling mixture ratio [5], changing the effective compression ratio by variable intake valve closing (IVC) timing [6], adjusting intake charge temperature by methods of fast thermal management [7] and changing inducted gas composition by modulating amount of

the in-cylinder trapped residual gas through variable exhaust valve closing (EVC) timing [8].

HCCI combustion phasing controllers in the literature are divided into two main categories including experiential controllers and model-based controllers. The first category is related to manually tuned controllers without any need for modeling. Examples of experiential HCCI controllers are found in [5, 9] where a Proportional-Integral-Differential (PID) controller with experimentally tuned gains is used to control the combustion phasing by adjusting the mixture ratio of two PRFs [5] or by adjusting both the negative valve overlap (NVO) and IVC timing [9]. The second category refers to model-based controllers, in which an accurate control model for predicting combustion phasing is needed. The model-based controllers can be divided into two groups depending what type of control model is used in the controller design. The first group includes model-based controllers derived from empirical models such as system identification black box models. Examples of HCCI

controllers based on system identification models are found in [6] and [10]. The second group of model-based controllers is derived from physical models such as thermodynamic models of HCCI engine cycle. Studies in [11] and [12] exhibit examples of model-based controllers derived from physical models for cycle-to-cycle control of HCCI combustion phasing. Physical model-based controllers have the potential to be more reusable on future generation models since a new controller can be designed by changing parameters of HCCI powertrain model to reflect the new vehicle.

In this work, a new discrete nonlinear control oriented model (COM) is designed to dynamically predict cycle-to-cycle HCCI combustion phasing. The model is validated against both experimental data and simulation results from a more detailed physical model from the previous work [13]. The new model is fast enough for real-time control while maintaining a desirable level of accuracy. The control model is linearized around a nominal operating point. Then an integral state feedback controller coupled with an optimal state observer is designed based on the linearized COM. The designed controller adjusts the combustion phasing by manipulating the fuel mixing ratio of two PRFs, i.e. iso-Octane (PRF100) with the octane number of 100 and n-Heptane (PRF0) with the octane number of 0. Performance of the designed controller is tested for retaining operation of an engine within a desired HCCI region in the presence of load and speed disturbances. In addition, the results from the designed controller are compared with those from a tuned PI controller to evaluate the benefits of a model-based controller versus an experimental controller for control of HCCI combustion phasing.

HCCI MODEL FOR CONTROL

In this section, a COM is developed by approximating the whole stages of an HCCI engine cycle. They include: induction stroke and mixing process of charge mixture with residual gases, polytropic compression of combustion mixture to the point of SOC, combustion period, polytropic expansion till the EVO moment and finally polytropic exhaust process through the exhaust valves. The following model states are chosen to be linked to thermodynamics of engine cycle.

1. Temperature at the start of combustion (SOC) moment (T_{soc})
2. Pressure at SOC (P_{soc})
3. Crank angle where 50% of the fuel is burnt (θ_{50})
4. In-cylinder mass at EVC moment (m_{evc})
5. Residual gas temperature (T_{rg})

These five states are dominant parameters which determine main dynamics affecting HCCI combustion phasing. T_{soc} and P_{soc} are the thermodynamic states of the in-cylinder gases at SOC. m_{evc} and T_{rg} are two of the main factors which determine temperature of the next cycle's charge at IVC moment after mixing with residual gas and consequently influence HCCI combustion phasing.

θ_{50} is a robust feedback indicator of HCCI combustion phasing due to the steep heat release in the main stage of HCCI combustion [14]. Here, θ_{50} is chosen as the main output of plant (HCCI model). The octane number (ON) of the blended fuel mixture is the main control input to the plant. In addition variations in engine load (fuel equivalence ratio, ϕ) and fluctuations in engine speed (N) are treated as disturbances to the plant.

MODEL DESCRIPTION

The HCCI model consists of a set of equations describing consecutive events starting from calculating the residual gas from a previous cycle until calculating the in-cylinder gas properties at EVC moment when the engine cycle is complete. Values of model parameters are shown in [Appendix A](#).

RESIDUAL GAS PROPERTIES

Residual gas has heating effects on the charge properties of the next cycle. The residual gas temperature and residual gas fraction at EVC are the two prominent parameters which control the thermal coupling between two consecutive engine cycles. Indeed, these two parameters link the model states from each cycle to the next one. So, it is needed to obtain these linking parameters as functions of the model states.

At each cycle (k^{th} cycle), residual gas fraction is obtained by [Eq. \(1\)](#).

$$RGF_k = \frac{m_{evc,k}}{m_{t,k}} \quad (1)$$

where, "k" is the cycle number index, RGF is the residual gas fraction, m_{evc} is mass of the trapped in-cylinder gas at EVC and m_t is mass of the total combustion mixture which can be found by the ideal gas state equation at each definite point after IVC and before the exhaust valve opening (EVO) moment.

$$m_{t,k} = \frac{P_{soc,k} V_{soc,k}}{\bar{R}_{soc} T_{soc,k}} \quad (2)$$

$\bar{R}_{soc}$ and V_{soc} are the average gas constant and the cylinder volume at SOC, respectively. The crank-slider mechanism relation [15] is used to calculate cylinder volume at different crank angle positions including SOC.

The crank angle at SOC (θ_{soc}) is calculated as a function of θ_{50} and combustion duration ($\Delta\theta_{comb}$), assuming a constant fuel burn rate.

$$\theta_{soc,k} = \theta_{50,k} - 0.5 \Delta\theta_{comb} \quad (3)$$

By stepping through the Eqs. (1), (2), (3), the residual gas fraction can be obtained as a function of the model states at the same engine cycle.

$$RGF_k = f(T_{soc,k}, P_{soc,k}, \theta_{50,k}, m_{evc,k}) \quad (4)$$

INDUCTION STROKE

Temperature and pressure at IVC are obtained by two semi-empirical correlations from [13].

$$P_{ivc,k+1} = \frac{(N_k^{0.027} \phi_k^{0.046})}{T_{man}^{0.005}} P_{man} \quad (5)$$

$$T_{ivc,k+1} = (a T_{man}^2 + b T_{man} + c) \frac{\phi_k^d N_k^e}{(1+EGR)^f} \quad (6)$$

where P_{man} and T_{man} are the pressure and temperature of the intake manifold in Kilopascals (kPa) and Celsius ($^{\circ}\text{C}$), respectively. N is the engine speed in rpm and EGR is the rate of external exhaust gas recirculation.

THERMAL EFFECTS OF THE TRAPPED RESIDUAL GAS ON THE NEW FRESH CHARGE

Mixing process of the residual gas and the fresh charge yields to Eq. (7).

$$T_{ivc,mix,k+1} = (1 - RGF_k) \frac{\bar{c}_{vnc}}{\bar{c}_{vt}} T_{ivc,k+1} + RGF_k \frac{\bar{c}_{vrg}}{\bar{c}_{vt}} T_{rg,k} \quad (7)$$

This correlation is based on the ideal gas assumption and the energy conservation in the mixing process. $\bar{c}_v$ is the average specific heat capacity of the mixture and $T_{ivc,mix}$ is modified temperature of the mixture at IVC after including the heating effect from mixing the new fresh charge with the residual gas from a previous cycle (k). Subscripts “rg”, “t” and “nc” represent the residual gas, the total mixture and the new air-fuel charge, respectively.

POLYTROPIC COMPRESSION

The compression of the unburned mixture prior to combustion can be considered as a polytropic process [15]. Therefore the temperature and pressure can be obtained at each definite point between IVC and SOC moments.

$$T_{soc,k+1} = T_{ivc,mix,k+1} \left(\frac{V_{ivc}}{V_{soc,k+1}} \right)^{n_c-1} \quad (8)$$

$$P_{soc,k+1} = P_{ivc,k+1} \left(\frac{V_{ivc}}{V_{soc,k+1}} \right)^{n_c} \quad (9)$$

where n_c is the specific heat capacity ratio for compression.

COMBUSTION PERIOD: COMBUSTION PHASING PREDICTION

The HCCI auto-ignition timing (θ_{soc}) is mainly dominated by the octane number of the fuel, the mixture temperature at IVC moment and the amount of the input fuel energy [3,16]. A Modified Knock Integral Model (MKIM) was used in the previous work [13] to predict θ_{soc} . While the MKIM is accurate in predicting θ_{soc} , it has a highly non-linear structure which limits its direct application for real-time control of HCCI combustion phasing. Here, the MKIM is condensed into a fitted correlation including the dominant factors affecting θ_{soc} .

$$\theta_{soc,k+1} = f(T_{ivc,mix,k+1}, ON_k, \phi_k) \quad (10)$$

HCCI combustion duration is a function of θ_{soc} and the charge temperature and composition [3, 16]. θ_{50} is calculated assuming a constant fuel burning rate.

$$\theta_{50,k+1} = \theta_{soc,k+1} + 0.5 \Delta\theta_{comb} \quad (11)$$

Combining Eqs. (10) and (11) yields to Eq. (12).

$$\theta_{50,k+1} = g(T_{ivc,mix,k+1}, ON_k, \phi_k) \quad (12)$$

where:

$$g = c_1 \phi_k T_{ivc,mix,k+1} + c_2 \phi_k + c_3 ON_k + c_4 \quad (13)$$

Values of the constant coefficients and more discussion about the correlation are found in Appendix A.

THERMODYNAMIC STATES AT EOC

According to Eq. (11), the crank angle at the end of combustion (EOC) can be predicted by Eq. (14).

$$\theta_{eoc,k+1} = \theta_{soc,k+1} + \Delta\theta_{comb} \quad (14)$$

By neglecting the heat transfer to the engine cylinder walls, the temperature increase due to the combustion process (ΔT_{comb}) can be found.

$$\Delta T_{comb,k+1} = \frac{m_{f,k} LHV \overline{CoC}}{m_t \bar{c}_v} \quad (15)$$

where m_f is mass of the injected fuel in the premixed mixture and $\overline{CoC}$ is the average completeness of combustion coefficient. LHV is the low heating value of the blended mixture of PRF fuels and is evaluated by Eq. (16).

$$LHV = \frac{\%V_n \rho_n LHV_n + \%V_{iso} \rho_{iso} LHV_{iso}}{\%V_n \rho_n + \%V_{iso} \rho_{iso}} \quad (16)$$

where $\%V$ is the volume percentage and ρ is the density of the fuel. The subscripts “iso” and “n” denote iso-Octane and n-Heptane, respectively.

By considering the definitions of the residual gas fraction (Eq. (1)) and also the fuel equivalence ratio, the temperature increase due to the combustion is evaluated.

$$\Delta T_{comb,k+1} = \frac{LHV CoC}{(1+RGF_k)(\phi_k^{-1} AFR_{st} + 1) \bar{c}_v} \quad (17)$$

where AFR_{st} is the stoichiometric air-fuel ratio of the charge.

Eventually, the temperature at EOC (T_{eoc}) is obtained by Eq. (18).

$$T_{eoc,k+1} = T_{soc,k+1} + \Delta T_{comb,k+1} \quad (18)$$

The mixture pressure at EOC is found by the ideal gas state equation ($PV = m\bar{R}T$) and the mass conservation law between EOC and SOC.

$$P_{eoc,k+1} = \frac{P_{ivc,k+1} V_{ivc}}{V_{eoc,k+1}} \frac{T_{eoc,k+1}}{T_{ivc,k+1}} \frac{\bar{R}_{eoc}}{\bar{R}_{ivc}} \quad (19)$$

POLYTROPIC EXPANSION

Expansion of the burned gases after EOC can be considered as a polytropic process [15]. Therefore the temperature and pressure of the burned gas at EVO are calculated based on the available data at EOC.

$$T_{evo,k+1} = T_{eoc,k+1} \left(\frac{V_{eoc,k+1}}{V_{evo}} \right)^{n_e - 1} \quad (20)$$

$$P_{evo,k+1} = P_{eoc,k+1} \left(\frac{V_{eoc,k+1}}{V_{evo}} \right)^{n_e} \quad (21)$$

where n_e is the specific heat capacity ratio for the expansion process.

POLYTROPIC EXHAUST: STATES at EVC

For the sake of simplicity, a polytropic process is also assumed for the exhaust stroke [11, 17]. The exhaust manifold pressure is considered to be the atmospheric pressure (P_0). By applying the polytropic correlation of temperature during the exhaust stage, T_{rg} is evaluated by Eq. (22)

$$T_{rg,k+1} = T_{evo,k+1} \left(\frac{V_{evo}}{V_{evc}} \right)^{n_e - 1} \quad (22)$$

Eventually, m_{evc} is evaluated by using the ideal gas state equation at EVC moment.

$$m_{evc,k+1} = \frac{P_0 V_{evc}}{\bar{R}_{evc} T_{rg,k+1}} \quad (23)$$

MODEL SUMMARY

The temperature and mass fraction of residual gas link the states of two consecutive engine cycles. By stepping through the model equations in the $k+1^{\text{th}}$ cycle, the model states and output are obtained as functions of model states (X), input (u) and disturbances (W) of the previous cycle.

$$X_{k+1} = F(X_k, u_k, W_k) \quad (24)$$

$$y_k = G(X_k, u_k, W_k) \quad (25)$$

where the state and disturbance vectors along with model input and output are shown in (26).

$$\begin{aligned} X &= [\theta_{50} \quad T_{soc} \quad P_{soc} \quad T_{rg} \quad m_{evc}]^T & W &= [\emptyset \quad N]^T \\ X \quad u &= ON & y &= \theta_{50} \end{aligned} \quad (26)$$

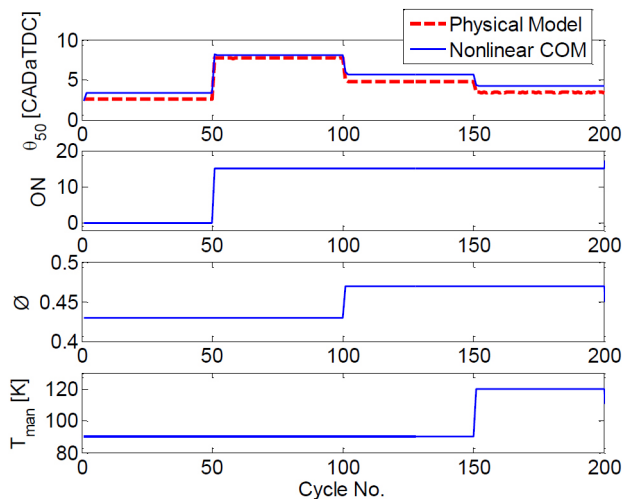
SIMULATION-BASED VALIDATION OF MODEL

The designed nonlinear HCCI model is parameterized for a single cylinder Ricardo engine with the specifications listed in Table 1.

Table 1. Specifications of Single Cylinder Ricardo Engine

Parameter	Value
Bore	80 mm
Stroke	88.90 mm
Compression ratio	10:1
Displacement volume	0.447 L
Number of Valves	4
IVO/IVC	-175/+55 CADaBDC
EVO/EVC	-70/-175 CADaBDC

The new parameterized COM is tested against a detailed physical model from [13]. Simulated θ_{50} by these two models are compared in Fig. 1 for three step changes in ON, ϕ and T_{man} . The θ_{50} results in Fig. 1 show the new COM can capture the main transient dynamics and it predicts θ_{50} with an average error of 0.7 CAD comparing with θ_{50} from the detailed physical model.

**Figure 1. Performance of New Nonlinear COM vs. Physical Model at Transient Operating Conditions**

While having similar accuracy in predicting θ_{50} , the new COM requires significantly less computation requirements. The processing time for running the models in Fig. 1 on a 2.67 GHz Intel processor is about 9 seconds for the physical model and only 0.07 second for the new COM. The new COM is fast enough for real-time control applications on HCCI engines.

MODEL LINEARIZATION

The nonlinear COM is linearized around a nominal operating point. Conditions of the selected operating point are listed in Table 2. The selection of the operating point is based on the experimental data in [3] to ensure this point is located in a desirable operating range with low cyclic variations. State and output equations of the linear model are shown in Eqs. (27) and (28). Elements of the linear model's matrices are found in Appendix A.

$$X_{k+1} = A X_k + B u_k + E W_k \quad (27)$$

$$y_k = C X_k \quad (28)$$

Table 2. Nominal Operating Point around which the COM is linearized

Parameter	Value
θ_{50}	8 CADaTDC
T_{soc}	785 K
P_{soc}	2088 kPa
T_{rg}	590 K
m_{evc}	0.033 g
ON	15
ϕ	0.45
T_{man}	348 K
P_{man}	100 kPa
External EGR	0 %
N	800 rpm

EXPERIMENTAL VALIDATION OF MODEL

Both nonlinear and linear COMs from this work have used a number of assumptions to simplify HCCI cycle for combustion phasing control. These models should be experimentally validated to build up confidence in using them for predicting HCCI combustion phasing. Experimental data from [13] for the Ricardo engine is used to test performance of the designed COMs for transient fueling conditions.

In Fig. 2 and Fig. 3, the control models are validated against the results of the transient fueling experiments. A quantitative comparison between accuracy of the models is shown in Table 3. While the nonlinear model has a slightly better accuracy in predicting θ_{50} , both models have desirable performance in predicting HCCI combustion phasing.