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MODEL-BASED

Internal combustion (IC) engines have undergone a series of regulatory constraints owing to their enormous environmental and energy impacts. The fossil fuel dependence of conventional IC engines also raises concerns about their sustainability and is leading to a gradual adoption of alternative and renewable fuels. Another negative impact of IC engines is carbon-dioxide (CO_2) emissions, which are an important contributor to climate change. Consequently, regulations are in place for greenhouse gas (GHG) emissions that require IC engines to be more efficient. Finally, toxic engine emissions include nitric-oxides and nitrogen-dioxides (NO_x), particulate matter (PM) emissions, and products of incomplete combustion such as hydrocarbons (HC). Regulations on these emissions have been enforced for several decades with increasingly stringent requirements and have led to significant progress in new technology development for modern IC engine systems with refined control systems.

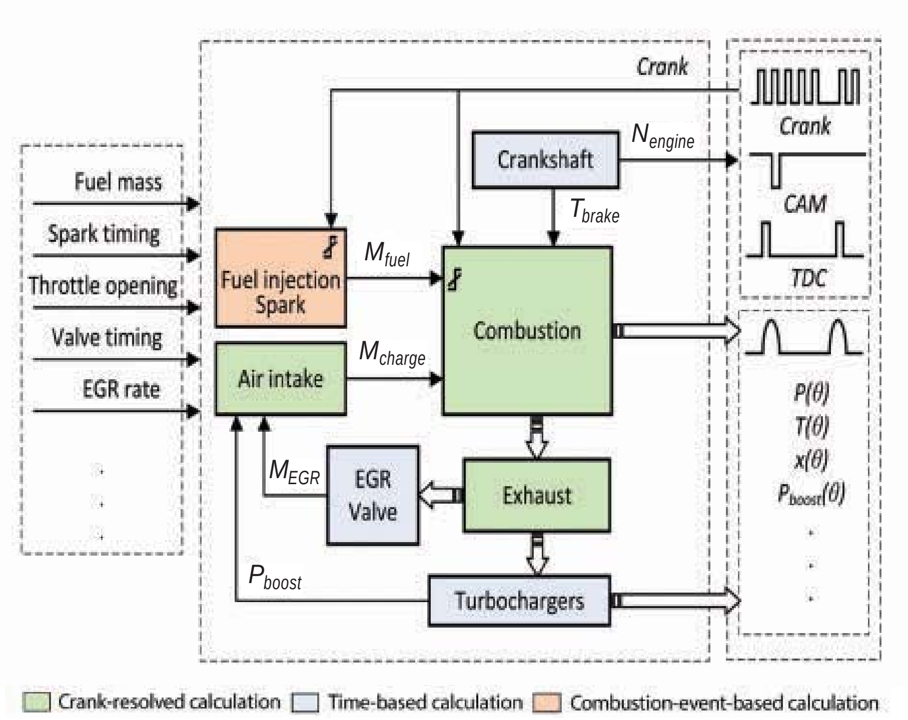

FIGURE 1 **Crank-resolved control-oriented engine model.**

Engine system dynamics involve chemical reactions, heat transfer, thermodynamics, fluid and mechanical dynamics and are highly nonlinear, multivariable, parameter dependent, and heavily coupled with subsystems. The discrete combustion events, multiple operational modes (such as cold start, aftertreatment regeneration, different combustion modes, etc.), and actuation constraints introduce additional complexity in control system design and calibrations. Traditionally, most of the engine control parameters, such as fuel injection, ignition timing, engine boost pressure, and the rate of exhaust gas recirculation (EGR), are open-loop controlled, and these open-loop set points are obtained through the so-called engine mapping process during engine development, and multi-dimensional interpolations (or lookup tables) are used during the deployment stage. As the emission regulations tighten, some control parameters, for example, the engine air-to-fuel ratio and intake manifold pressure, are now controlled in a closed loop.

One of the challenges for combustion control is the smooth transition between different combustion modes, such as the mode transition between spark ignition (SI) and homogeneous charge compression ignition (HCCI) combustion. To improve the transient performance, a model-based feedforward control structure is often preferred. For example, during the combustion mode transition, model-based sensitivity feedforward control is often necessary to have fast compensation of engine cycle-to-cycle variations, along with optimal tracking of desired air charge [1],[2].

The increasing on-board computing power and memory also motivate

ENGINE CONTROL FOR Improved Fuel Economy WITH Reduced Emissions

model-based engine control since many of the engine control problems can be formulated as constrained multi-input-multi-output (MIMO) optimal control problems. Therefore, control-oriented engine modeling becomes very important since it forms the foundation for model-based control. This article mainly focuses on control-oriented engine modeling and model-based engine control techniques.

CONTROL-ORIENTED ENGINE MODELING

The engine modeling research is centered on the engine combustion process. Multi-zone, three dimensional CFD (computational fluid dynamics) models [3] with detailed chemical kinetics are able to precisely describe the thermodynamics, fluid and flow dynamics, heat transfer, and pollutant formation of the combustion process. The simplified one-dimensional combustion models have also been implemented into commercial codes such as GT-Power and Wave. However, these high fidelity models cannot be used for model-based control since they are too complicated to be used for real-time computing. Traditionally, zero-dimensional mean-value engine models [4] are widely used for engine control validation and hardware-in-the-loop (HIL) simulations. The main drawback of the mean-value engine model is that it does not provide detailed combustion information, such as crank-resolved in-cylinder pressure and temperature that have been widely used for closed-loop combustion control [5]. Crank-resolved engine air handling system modeling is also important for describing the in-cylinder charge-mixing process [6]. Therefore, for model-based control and real-time HIL simulations, it is necessary to have a crank-resolved engine model with its complexity intermediate between the time-based mean-value and one-dimensional CFD models. **Table 1** compares the capability of different modeling methods and this article focuses on the zero-dimensional crank-resolved model.

In order to make real-time simulation possible for the crank-resolved control-oriented engine model, typically the engine model has three different simulation steps [6],[7], where the engine combustion process, including

DIFFERENT COMBUSTION MODELS	0-D mean-value combustion model	0-D crank-resolved combustion model	1-D crank-resolved combustion model
Implementation tool	Matlab/Simulink	Matlab/Simulink and HIL simulator	GT-Power, Wave and Fluent
Time cost per cycle	Micro seconds	Real-time	Minutes to hours
One-zone	Yes [4]	Yes [6], [7]	Yes
Two-zone, three-zone	No	Yes [6], [7]	Yes
Multi-zone	No	No	Yes
Chemical concentration	No	Yes [8]	Yes
Charge mixing model	No	Yes [6]	Yes
In-cylinder flow dynamics	No	Yes [6], [7]	Yes
IMEP	Yes [4]	Yes [6], [7]	Yes
In-cylinder pressure and temperature	No	Yes [6], [7]	Yes

TABLE 1 **Features for different combustion models.**

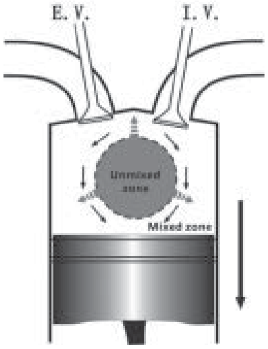

FIGURE 2
A simplified charge mixing model.

intake charge mixing, is crank-resolved and updated every crank degree; air-to-fuel ratio and fuel injection are updated every combustion event; and crank shaft dynamics, turbocharger, and EGR flow are updated every millisecond; see **Figure 1** for the model architecture and color coded update rates. The inputs to the engine model are the actual control signals to the engine actuators (see the left side of **Figure 1**) and the outputs are the engine sensor signals (such as in-cylinder pressure $P(\theta)$, boost pressure $P_{boost}(\theta)$, crank and cam positions) and engine physical variables (such as in-cylinder temperature $T(\theta)$ and mass fraction burned $x(\theta)$). Also any variables in the model (center block of **Figure 1**) can be accessed during the real-time simulations.

In order to model the combustion process accurately, the control-oriented model also needs to contain the charge mixing model; see **Table 1**. Reference [6] assumes that during the charge mixing process the unmixed zone (formed by residual gas) remains in the center of the cylinder and the shape is assumed to be spherical. It also assumes that the fresh charge surrounds the unmixed zone with velocity tangent to the sphere and mixes with the residual gas gradually to form the mixed zone. The mass transport from the unmixed zone to mixed zone is caused by gas diffusion (dominated by molecular diffusions); see **Figure 2**.

Combustion is often modeled using the Weibe mass-fraction burned function [6],[7] and can also be modeled based on the chemical reaction process [8]. With the proper charge-mixing model, the Weibe-based combustion model is capable of providing comparable simulated in-cylinder pressure and temperature in real-time. **Figure 3** [6] shows the simulation results of the homogeneous charge compression ignition (HCCI) combustion process modeled by a Weibe function with a two-zone combustion model, where the start of combustion is modeled using the Arrhenius integral.

MODEL-BASED ENGINE CONTROL

Model-based engine control has been widely used in engine actuation subsystems [9][14], engine boost system, and engine combustion process including engine air-to-fuel ratio control [10], [11]. Due to the high degree of freedom of engine systems, single-input-single-output closed-loop control limits the engine performance and cannot meet future fuel economy and emission requirements. For instance, shifting from single fuel injection to multiple fuel injections in one engine cycle for both diesel and gasoline engines requires multi-input-multi-output (MIMO) control. On the other hand, the linear MIMO control theory may not be applicable due to the high nonlinearity nature of engine systems, therefore, utilizing nonlinear-systems-oriented approaches such as LPV (linear parameter varying) control [12],[13], MPC (model predictive control) [11], and extremum seeking (ES) [16],[17] seems inevitable and promising to achieve the improved robust performance. Two control examples, actuation and combustion control, are described in detail below.

Engine actuation system control

Variable valve actuation (VVA) is able to significantly improve the fuel economy with reduced emissions and increase the engine power output. Reference [14] presents a model reference control of an electro-pneumatic VVA aimed to improve the lift repeatability of the intake VVA. The model parameters are first identified using a model reference adaptive scheme and then a closed-loop valve lift and closing timing control are formulated to generate the feedforward control of valve timings based on the identified parameters. The closed-loop control is used to eliminate the steady-state error and its control block diagram can be found in **Figure 4** and also in [14].

Bench tests were conducted using the model reference control strategy shown in **Figure 4** with a target valve lift of 9 mm at the corresponding engine speed of 1200 rpm. Two hundred cycles of data were collected for each experiment. The valve lift experimental data was plotted as histograms in **Figure 5**, where the left plot is associated with the open-loop control and

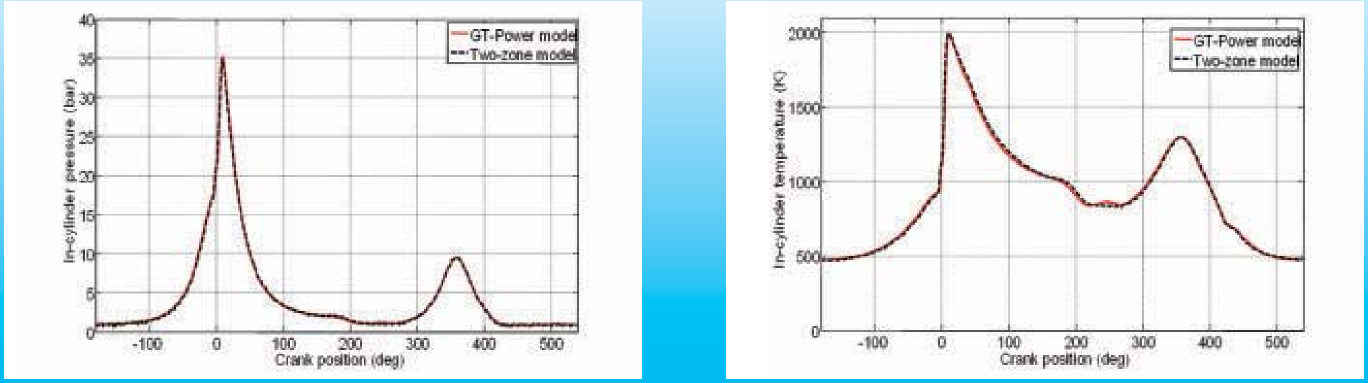

FIGURE 3 Comparison of GT-Power and crank-resolved control oriented combustion model.

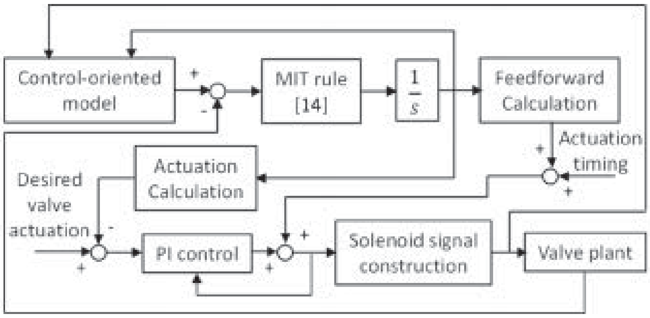

FIGURE 4 MPC of engine variable valve timing actuation.

right associated with the closed-loop model reference control. Note that the valve lift histogram reflects the valve lift repeatability. The mean and the standard deviation for open-loop control are 9.16 mm and 0.85 mm, respectively, and the mean and standard deviation for closed-loop model reference control are 9.10 mm and 0.39 mm. Note that standard deviation reduction of 55% is achieved.

Engine combustion control

Closed-loop combustion control using in-cylinder ionization (or pressure) signal is investigated in [5] for engine retard ignition timing control to maintain combustion stability during cold start, for optimal ignition timing control to maximize engine brake torque during normal operation, and for spark advance limit control to operate the engine at its knock ignition timing limit and maximize the engine output torque. The stochastic knock control is used to accurately operate at its knock limit.

Homogeneous charge compression ignition (HCCI) combustion has the potential of providing higher thermal efficiency and lower emissions than those of the conventional spark ignition (SI) combustion due to its unthrottled operation, flameless and lean burn nature [15]. However, HCCI combustion also has limited operational range due to possible misfires at low load and knock at high load. An HCCI-capable SI engine, operating under HCCI combustion at low load and under SI combustion at high load, is necessary for practical applications. Therefore, smooth mode transition between the HCCI and SI combustion is the key for combustion control of an HCCI-capable SI engine.

It is fairly challenging to achieve smooth mode transitions between SI (spark ignition) and HCCI combustion for an HCCI-capable SI engine, for the reason that the in-cylinder thermal and charge mixture properties are quite different due to the distinct combustion characteristics. A model-based mode transition strategy between SI and HCCI combustion over the entire SI and HCCI mode transition boundary is presented in [1],[2]. The mode transition strategy is experimentally validated on an HCCI-capable SI engine equipped with electric variable valve timing (EVVT) actuation, dual-lift valves and electronic throttle control systems. The throttle was opened gradually to its wide open position during the SI to HCCI transition or closed directly to the target position during the HCCI to SI transition, where the hybrid combustion mode is employed with the spark assistant to gradually transit to the desired HCCI (or SI) combustion mode, respectively. This is accomplished by increasing (or decreasing) the percentage of the HCCI combustion cycle-by-cycle under the hybrid combustion mode. **Figure 6** shows the experimental mass fraction burned (MFB) curves during the transitional cycles 1 to 8 at 2000 rpm with 4.5 bar net mean effective pressure (NMEP), where the red dots denote the auto-ignition (start of HCCI combustion in the unburned zone) locations for the hybrid combustion mode. It can be observed that during the mode transition from the SI to HCCI combustion, the hybrid combustion starts at the 4th cycle and it lasts for four cycles. The percentage of the HCCI combustion gradually increases

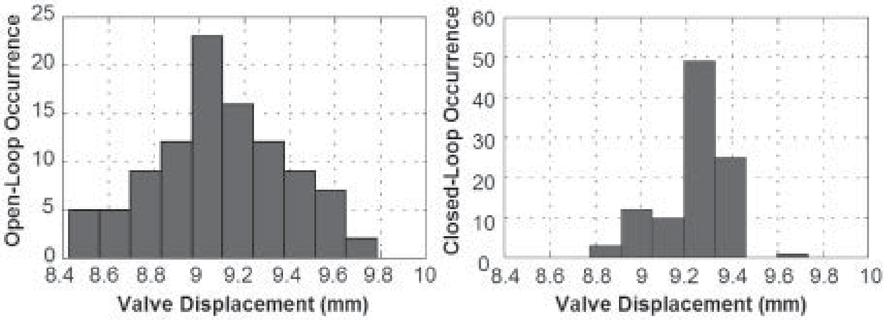

FIGURE 5 OL and CL valve lift histogram at 1200 rpm.

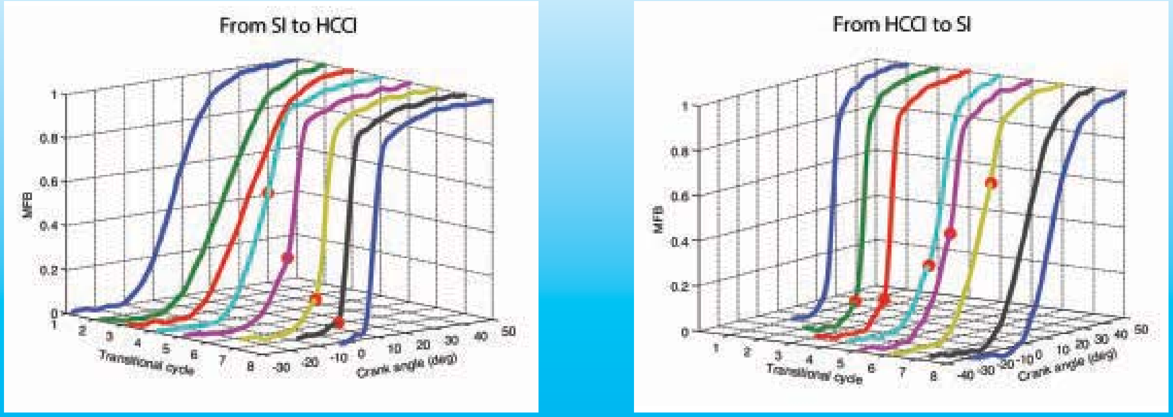

FIGURE 6 Cycle-to-cycle MFB responses during mode transition (2000 rpm and 4.5 bar NMEP).

over the four hybrid combustion cycles. For the case of mode transition from HCCI to SI combustion, the hybrid combustion lasts for five cycles, and the percentage of the HCCI combustion gradually decreases over the five hybrid combustion cycles.

One of the challenges to have smooth mode transitions is dealing with cycle-to-cycle combustion variations. Closed-loop control is often not fast enough to react to these variations due to the short transition period (5 to 8 engine cycles). Model-based sensitivity compensation is used to reduce the cycle-to-cycle variations during the mode transition. The experimental results in **Figure 7** show that the developed strategy is able to achieve smooth combustion mode transition with the NMEP and combustion phase fluctuations at the level of stable SI and HCCI combustion.

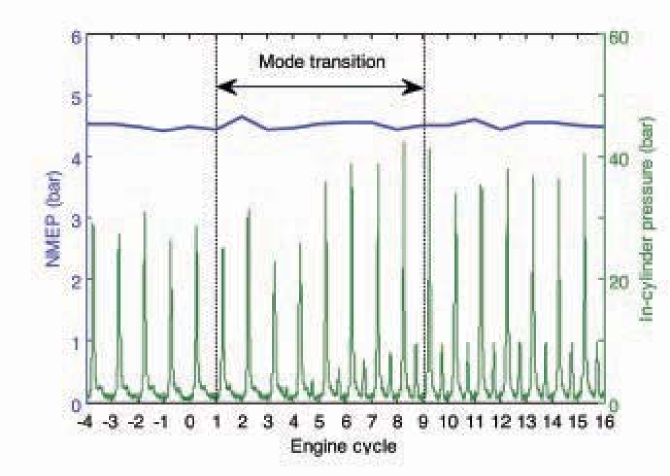

FIGURE 7 In-cylinder pressure and NMEP during the SI to HCCI mode transition.

FUTURE TRENDS

Model-based engine control is the future direction for engine actuation subsystems, engine boost and EGR systems, and engine combustion systems. The main challenge is the nonlinear and parameter-varying nature of engine systems. Model-based control methodologies enable systematic control design, such as MPC, model reference and LPV (gain-scheduling) control, to cope with the nonlinear nature of engine systems and will be the center of focus for future engine control development. LPV control methodology, in particular, provides MIMO gain-scheduling controllers with guaranteed robust stability and performance. Another potentially promising avenue for robust engine control is to integrate the extremum seeking (ES) approach into model-based control and optimization to develop data-driven on-line feedforward and feedback control mechanisms for engines, enjoying the tremendous robust performance of ES while significantly enhancing its convergence speed to facilitate real-time application. ■

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