SI AND HCCI COMBUSTION MODE TRANSITION CONTROL OF A MULTI-CYLINDER HCCI CAPABLE SI ENGINE VIA ITERATIVE LEARNING

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ABSTRACT

The combustion mode transition between spark ignition (SI) and homogeneously charged compression ignition (HCCI) combustions of an internal combustion (IC) engine is challenging due to the distinct engine operational parameters over these two combustion modes and the cycle-to-cycle residue gas dynamics of the HCCI combustion. The control problem becomes even more complicated when multi-cylinder operation is involved. This paper studies the combustion mode transition problem of a multi-cylinder IC engine with dual-stage valve lifts and electrical variable valve timing systems. A control oriented engine model was used to develop a multistep mode transition control strategy via iterative learning for combustion mode transition between SI to HCCI with minimal engine torque fluctuations. The hardware-in-the-loop (HIL) simulations demonstrated the effectiveness of the developed control strategy for the combustion mode transition under both constant load and transient engine operational conditions.

NOMENCLATURE

- $\lambda$: Normalized air-to-fuel ratio
- $x_{EGR}$: EGR gas fraction in engine intake manifold
- $\theta_{ST}$: Spark timing in degree after combustion TDC (top dead center)
- $\theta_{SOHCCI}$: Start of HCCI combustion timing
- $N_{TPO}$: Throttle pre-opening timing before intake/exhaust valve lift switch (quarter cycle)
- $F_{DI}$: Direct injection (DI) fuel pulse duration (ms/cycle)
- $F_{FB}$, $F_{FF}$: Feedback/feedforward portions of $F_{DI}$ (ms/cycle)
- $F_{ILC}$: ILC (iterative learning control) portion of DI pulse duration (ms/cycle)
- $\phi_{\text{throttle}}$: Engine throttle position (%)
- $\phi_{\text{acc}}$: Acceleration pedal position (%)
- $\phi_{EGR}$: External EGR valve opening (%)
- $H_{\text{lift}}$: Peak lift of intake/exhaust valve (mm)
- $T_{\text{IVC}}$: In-cylinder gas temperature at IVC ($^\circ$K)
- $M_{\text{EVC}}$: Residue gas mass at EVC (mg)
- $\theta_{\text{INTM}}$: Intake valve timing at peak lift (degrees after gas exchange TDC)
- $\theta_{\text{EXTM}}$: Exhaust valve timing at peak lift (degrees before gas exchange TDC)
- $\text{MAP}$: Intake manifold absolute pressure (bar)
- $\text{IMEP}$: Indicated mean effective pressure (bar)
- $\text{IMEP}_{\text{des}}$: Desired IMEP due to driver need (bar)
- $\text{IMEP}_{\text{ref}}$: IMEP learning reference used in ILC (bar)
- $i, k, n$: Indexes of ILC iteration, engine cycle, and cylinder

INTRODUCTION

Homogeneously charged compression ignition (HCCI) combustion has the potential for internal combustion (IC) engines to meet the increasingly stringent emissions regulations with improved fuel economy. The flameless nature of the HCCI combustion and its high dilution operation capability lead to low combustion temperature. As a result, the formation of NOx (Nitrogen Oxides) can be significantly reduced. Furthermore HCCI engine is capable of un-throttled operation that greatly reduces pumping loss and improves fuel economy ([1] and [2]).

On the other hand, HCCI combustion has its own limitations. It is limited at high engine load due to the audible
knock, and at low load due to misfire caused by the lack of sufficient thermal energy to trigger the auto-ignition of the gas-fuel mixture late in the compression stroke [3]. In fact, HCCI combustion can be regarded as a type of engine operating mode rather than a type of engine [4]. In order to take the advantage of the HCCI combustion mode in an IC engine, other combustion mode, such as spark ignited (SI) combustion, is required at high load, at high speed, at ultra-low load (such as idle), and at certain operating conditions (such as cold start).

It is fairly challenging to operate the engine in two distinct combustion modes, and it is even more difficult to have smooth combustion mode transition between SI and HCCI combusions because the favorable thermo conditions for one combustion mode are always adverse to the other. For example, high intake charge temperature is required in the HCCI mode to initiate the combustion, while in the SI mode it leads to reduced volumetric efficiency and increased knock tendency. For this reason, engine control parameters, such as intake and exhaust valve timings and lifts, throttle position and EGR (exhaust gas recirculation) valve opening, are controlled differently between these two combustion modes. During the combustion mode transition, these engine control parameters need to be changed rapidly. However, the physical actuator limitations on response time prevent them from completing their transitions within the required duration, specifically, within one engine cycle. The multi-cylinder operation makes it even more difficult. Accordingly the combustion performance during the transition cannot be maintained unless proper control strategy is applied.

The control of the HCCI combustion process has been widely studied in past decades. Robust HCCI combustions can be achieved through model based control as described in [5], [6], and [7]. To implement the HCCI combustion in a practical IC engine, the challenge of the combustion mode transition is inevitable. In recent years, more and more attentions have been paid to the mode transition control between SI and HCCI combustions. In [8] and [9], smooth mode transitions between SI and HCCI combustions are realized for a single cylinder engine equipped with the camless VVA (Variable Valve Actuation) system. However, high cost prevents the implementation of the camless VVA system in production engines. In [10] a VVT (Variable Valve Timing) system with dual-stage lifts was used on a multi-cylinder engine. Mode transition results show that smooth mode transition can be achieved by controlling the throttle opening timing and the direct injection (DI) fuel quantity. However, satisfactory mode transition is not achieved due to the lack of the robust mode transition control strategy.

In this paper, the combustion mode transition control strategy is investigated through the hardware-in-the-loop (HIL) engine simulations. The studied four-cylinder engine is equipped with external cooled EGR, dual-stage valve lifts and electrical VVT valvetrain system. The SI-HCCI hybrid combustion mode, as discussed in [1], was implemented during the multistep mode transition and iterative learning control (ILC) strategy was used for regulating the throttle opening timing and the individual cylinder DI fuel quantity. Smooth combustion mode transitions were achieved when the engine is operated at both stead constant load and transient conditions.

THE HCCI CAPABLE SI ENGINE

Figure 1 shows the target HCCI capable SI engine configuration. It is a four-cylinder 2.0 liter engine with compression ration of 9.8. The key feature of this engine is its valvetrain system. It has two stages of lifts for both intake and exhaust valves. The high lift stage has 9 mm maximum lift and is used for SI combustion mode; the low lift stage has 5 mm maximum lift and is used for HCCI combustion mode. The ranges of both intake and exhaust valve timing were extended to ±40 crank degrees to improve the controllability of the internal EGR fraction, effective compression ratio, and engine volumetric efficiency during the HCCI operating condition. In addition to the modification of the valvetrain, external cooled EGR is added to the engine to enable high dilution charge, resulting in low charge mixture temperature. The engine throttle can be electronically controlled to obtain the desired engine charge in both SI and unthrottled HCCI operations. Each cylinder of the test engine will be fitted with piezoelectric pressure transducer. Four fast response UEGO (Universal Exhaust Gas Oxygen) sensors are also assumed to be connected to each exhaust pipe, corresponding to each cylinder.

FIGURE 1. HCCI CAPABLE SI ENGINE CONFIGURATION

An HIL simulation station, shown in Fig. 2, is used for the purpose of initial control strategy development and validation. The dSPACE based engine HIL simulator provides all measureable engine signals (such as in-cylinder pressure and MAP signals) in real-time. These signals can be directly fed into the Opal-RT based real-time prototype engine controller. Some immeasurable engine states (such as in-cylinder temperature) are also generated by the simulator to help with the control strategy development. On the other hand, the control signals generated by the prototype engine controller are also fed back into the dSPACE based engine simulator. These signals are also monitored and recorded for all the simulations that will be discussed in the rest of this paper. The proposed control strategy discussed in this paper had been validated in
the HIL simulation environment, and will be verified against measurements obtained from engine dynamometer tests after the test engine is ready.

The HIL simulation is a useful tool for control strategy development and validation, but the effectiveness of the HIL simulation relies on the engine model that is executed in the dSPACE simulator. A control oriented real-time engine model is required. It must reflect all dominant engine dynamics with relatively high fidelity. Next section introduces the engine model used in this study.

![Engine simulation setup](image)

FIGURE 2. HIL SIMULATION STATION

THE HCCI CAPABLE SI ENGINE MODEL

The engine model used for the HIL simulation of this paper has been presented in the early work ([1] and [15]). It models the physical characteristics of the engine system. And it consists of two main sub-system models. One is the crank based model for combustion process; and the other is the time based models for engine air-handling system, crankshaft dynamics, and engine actuators.

The crank based combustion model has the resolution of every crank degree. It models the SI-HCCI hybrid (or spark assistant) combustion process that starts with the SI combustion and ends with the HCCI combustion, where the SI combustion phase is modeled under two-zone assumption and the HCCI combustion phase is modeled under one-zone assumption. The SI and HCCI combustion modes are actually special cases of the SI-HCCI hybrid combustion mode. Accordingly this combustion model is applicable for any operations during the combustion mode transition. The combustion model of the cycle-to-cycle dynamics can be expressed in the nonlinear state space form as

\[ x(k+1) = f(x(k), u(k)) \]
\[ y(k) = h(x(k), u(k)) \]  

(1)

where \( x(k) \), \( u(k) \) and \( y(k) \) are states, inputs and outputs, respectively. They are given by

\[
\begin{align*}
    x(k) &= [M_{AVC}(k) T_{EGR}(k)]^T \\
    u(k) &= [\theta_{ST} F_{DI} \theta_{SOOT} \theta_{SO} \overset{\circ}{\lambda}_{SO} MAP \ \Pi_{or}]^T(k) \\
    y(k) &= IMEP(k)
\end{align*}
\]

(2)

The state and output functions, \( f \) and \( g \), in Eq. (1) are composed by the governing equations of the engine combustion process. The details of the combustion model are described in [1].

The engine crankshaft model and air handling system sub-models, such as the gas filling dynamical model of intake manifold, are time based and updated every millisecond, since the dynamics of these sub-systems are continuous. The complete description of these models is presented in [15].

In the HIL simulation of combustion mode transition, the engine control actuator dynamics, such as engine throttle, VVT actuator and EGR valve, are found to affect the combustion performance significantly. They must be reflected in the engine model. In this study, the actuator response \( Y(s) \) with respect to the control input \( U(s) \) is approximated as the first order system in the engine model as

\[
\frac{Y(s)}{U(s)} = \frac{1}{1+rs}
\]  

(3)

where \( s \) is the Laplace operator and \( r \) is the associated actuator time constant. For the HIL simulations of this paper, \( r \) equals to 0.15, 0.12 and 0.1 second for engine throttle, VVT actuator and EGR valve, respectively. The transient responses of these actuators can be found in Fig. 4, Fig. 6 and Fig. 12. Their influences to the combustion mode transition will be discussed.

ONE-STEP COMBUSTION MODE TRANSITION

The combustion mode transition was first studied for constant load engine operating condition. At 2000 rpm engine speed the transition is assumed to occur at 4.5 bar IMEP. Table 1 lists the engine parameters (actuator positions) associated with the SI and HCCI combustions. These parameters were optimized for the steady state engine performance with the best fuel economy that satisfies the engine knock limit. That is, the maximum pressure rise \( (dP/d\theta) \) is less than or equal to 3 bar per crank degree. It can be seen in Tab. 1 that the engine control parameters are quite different between the SI and HCCI combustions. Among these control parameters, some of them can be adjusted to the target positions within one engine cycle, such as \( \theta_{ST} \), \( F_{DI} \) and \( H_{Ipp} \); but the others cannot, due to the response delays of these engine actuators.

The combustion characteristics are also quite different between these two combustion modes. For example, the HCCI combustion has higher peak in-cylinder pressure comparing with that of the SI combustion, due to the faster fuel burn rate as illustrated in Fig. 3. Most likely, it also has a recompression phase but the SI combustion does not. The goal of the combustion mode transition is to transfer the combustion mode
without detectable engine torque fluctuation by regulating those engine control parameters, or in other words to maintain the engine IMEP during the combustion mode transition.

The control strategy developed in this paper starts with one-step transition strategy. That is to directly change the control references of all engine control parameters from the SI to the HCCI engine parameters, as listed in Tab. 1, in one engine cycle. The HIL simulation of the one-step transition was conducted and key engine variables are plotted in Fig. 4.

**TABLE 1. STEADY STATE ENGINE PARAMETERS FOR SI AND HCCI COMBUSTION MODES**

<table>
<thead>
<tr>
<th>Engine parameter</th>
<th>SI</th>
<th>HCCI</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \theta_{ST} ) (deg ACTDC)</td>
<td>-36</td>
<td>none</td>
</tr>
<tr>
<td>( \phi_{EGR} ) (%)</td>
<td>3</td>
<td>26</td>
</tr>
<tr>
<td>( \phi_{throttle} ) (%)</td>
<td>16.6</td>
<td>100</td>
</tr>
<tr>
<td>( \theta_{IN} ) (deg AGTDC)</td>
<td>70</td>
<td>95</td>
</tr>
<tr>
<td>( \theta_{EX} ) (deg BGTDC)</td>
<td>100</td>
<td>132</td>
</tr>
<tr>
<td>( I_{lift} ) (mm)</td>
<td>9</td>
<td>5</td>
</tr>
</tbody>
</table>

**FIGURE 3. STEADY STATE COMBUSTION CHARACTERISTICS OF SI AND HCCI COMBUSTION MODES**

As illustrated in Fig. 4, during the one-step mode transition, once the step references are applied to the corresponding engine actuators some engine parameters change in one engine cycle, such as the intake/exhaust valve lifts and DI fuel quantity. Whereas the other engine parameters, such as the intake/exhaust valve timings and the throttle opening, response much slow. It takes them several engine cycles to reach their reference values, as seen in Fig. 4. When the gas mixture filling dynamics of the intake manifold is coupled with the first order dynamics of corresponding engine actuators, higher order dynamical response can be observed on the engine MAP signal, see Fig. 4. Furthermore, the intake/exhaust valve lift switch imposes a significant impact to the engine respiration process. It reduces the charge mass and increases residue gas mass. As a result, the volumetric efficiency of each cylinder for the engine cycle right after the valve lift switch drops significantly, leading to rich combustions for all four cylinders. For the succeeding engine cycles the volumetric efficiency is increased due to the increment of MAP.

**FIGURE 4. ONE-STEP TRANSITION ENGINE RESPONSES**

The HIL simulation results indicate that the smooth combustion mode transition cannot be achieved by the one-step transition strategy due to the response delay of the engine actuators and the flow dynamics of engine respiration process. Multiple engine cycles need to be inserted between the two distinct combustion modes. This is the main motivation of the proposed multistep combustion mode transition strategy that will be discussed in the next section.

**MULTISTEP COMBUSTION MODE TRANSITION**

In the one-step mode transition, rich combustions are found for the engine cycle right after the valve lift switch. This implies the lack of fresh in-cylinder charge. Without enough fresh charge the combustion will inevitably be degraded. There are two ways to increase the charge during the transient condition. One is to advance the intake valve timing and retard the exhaust valve timing; the other is to open the engine throttle before the valve lift switch. Using the first approach, both intake and exhaust valve timings need to be moved to one direction first and then to the opposite direction. This prolongs the settling time of the valve timings, and complicates the engine flow dynamics during the combustion mode transition. The approach of pre-opening the throttle does not have this issue. As illustrated in Fig. 5 two engine cycles (cycles 1 and 2) are inserted between the SI and HCCI combustion modes. Control events are scheduled at different timings during the two cycles. At the beginning of cycle 1 the engine controller computes \( NTPO (0 \leq NTPO \leq 8) \). Then at \( NTPO \) quarter-cycles before the end of cycle 2, the control reference of throttle opening is set to 100%, and the throttle plate starts to move to the wide open throttle (WOT) position. At the end of cycle 2, the control references of the other engine parameters are set to their target values as given in Tab. 1; the spark ignition is eliminated and
the engine operation is transited into the HCCI combustion mode. Notice that the SI-HCCI hybrid combustion cycles in Fig. 5 are not used in this case and they will be used in the next proposed control strategy.

FIGURE 5. SCHEDULE OF ENGINE CONTROL EVENTS FOR MULTISTEP MODE TRANSITION STRATEGIES

By comparing the simulation results in Fig. 4 and Fig. 6, one can see that the throttle pre-opening helps to increase the fresh charge at the start of transition since the engine MAP is increased before the valve lift switch; and it also reduces the engine torque fluctuation, see Fig. 11 for the comparison with the one-step mode transition. However the performance is still not acceptable since the large engine torque fluctuation indicates degraded combustions during the mode transition.

FIGURE 6. ENGINE RESPONSE OF MULTISTEP MODE TRANSITION WITH THROTTLE PRE-OPENING (NTPO=4)

In order to investigate the cause of the degraded combustions during the mode transition, additional combustion related engine variables are plotted in Fig. 7. One can see that $T_{IVC}$ is increased after the valve lift switch, but still far below the temperature required for steady state HCCI combustion. This is mainly due to the response delays of the intake/exhaust valve timings, which lead to insufficient residue gas charge. The reduced charge temperature leads to retarded HCCI combustion timing and degraded IMEP. Furthermore, as discussed in [11], late combustion timing is accompanied with large variation of $\theta_{SOHCCI}$ and IMEP variables for the HCCI combustion mode. This means unstable combustions. Misfire might happen when the combustion is not stable. And misfire is disastrous for HCCI combustion mode since it might trigger misfire for the succeeding engine cycles.

FIGURE 7. COMBUSTION PERFORMANCE OF MULTISTEP MODE TRANSITION WITH THROTTLE PRE-OPENING

In order to avoid misfire and unstable combustions during the mode transition, three engine cycles of the SI-HCCI hybrid combustions are applied in addition to the throttle pre-opening. As illustrated in Fig. 5 the spark ignition is kept for three engine cycles after the valve lift switch and it is eliminated at the end of cycle 5. With the spark ignition, the combustions of cycles 3, 4 and 5 are initiated by spark ignition in the SI mode but ended in the HCCI mode, and they are called SI-HCCI hybrid combustions. More details about this hybrid combustion mode can be found in [1] and [12].

The SI-HCCI hybrid combustions only occur under certain thermo conditions. The in-cylinder gas-fuel mixture needs to be lean, but extreme lean condition could lead to misfire since the spark ignition might not be able to initiate the SI combustion. The charge temperature needs to be higher than what SI combustion requires but lower than that of the HCCI combustions. Only under this condition the unburned gas-fuel mixture during the initial SI combustion phase can meet the start of HCCI combustion criterion to initiate the HCCI combustion in the unburned zone. These thermo conditions are satisfied for cycles 3, 4, and 5. Accordingly the combustions during these engine cycles are in the SI-HCCI hybrid combustion mode. Once the hybrid combustion is engaged, misfire will not occur even with very late SOHCCI timing, since a portion of gas-fuel mixture has already been burned in the SI mode before $\theta_{SOHCCI}$. Moreover, under these special thermo conditions, engine IMEP output is highly correlated with the DI fuel quantity, so it is possible to regulate the engine IMEP (or torque) by controlling the DI fuel quantity. Whereas for the pure HCCI combustion mode the increased cooling
effect due to increment of the DI fueling reduces the charge temperature, leading to degraded HCCI combustions.

The HIL simulation results of the multistep combustion mode transition are plotted in Fig. 10. One can see that $T_{\text{ave}}$ of the first engine cycle after the valve lift switch is still low, but $\theta_{\text{SOHCCI}}$ is advanced for a few crank degrees, and the engine torque fluctuation is smaller than those without the hybrid combustions. In the HIL simulations, the DI fuel quantity and spark timing are controlled using the values for steady state engine operating condition. They are not optimal since the engine flow dynamics and the cycle-to-cycle residue gas dynamics make the thermo conditions different from those of the steady state. More advanced control strategy is required to cope with the thermo dynamics.

**MULTISTEP COMBUSTION MODE TRANSITION WITH ITERATIVE LEARNING CONTROL**

By investigating the simulation results plotted in Fig. 7 and Fig. 10. One can see that significant cycle-to-cycle and cylinder-to-cylinder variations occurred in most combustion related variables, such as $T_{\text{ave}}$ and $\theta_{\text{SOHCCI}}$. This is mainly contributed by the fact that the change of the engine actuator positions affects the individual cylinder thermo condition differently. As a result, the IMEP and engine torque output also show significant variations. To reduce the variations of engine IMEP and torque, a few control schemes were tried. Feedback control was firstly tried, but since the mode transition only happens for several engine cycles, feedback controls might not have time to improve the performance. Feedback control was then considered. Through an iterative learning control (ILC) approach, the feedforward DI fuel quantity can be regulated by the thermo conditions different from those of the steady state. More advanced control strategy is required to cope with the thermo dynamics.

As illustrated in Fig. 8, the feedforward controller is initialized using the steady state control values as described in the last section. Two control parameters are directly adjusted by the ILC during each learning iteration. They are $N_{\text{TPO}}$ and $F_{\text{DI}}$. For $N_{\text{TPO}}$ the learning algorithm is

$$
N_{\text{TPO}}(i+1)=N_{\text{TPO}}(i)+N_{\text{IC}}(i)=N_{\text{TPO}}(i)+\delta(\lambda(i,k,n))
$$

$$
N_{\text{FP}}(i+1)=N_{\text{FP}}(i+1)
$$

where

$$
\delta(\lambda(i,k,n))=-1 \quad \text{if:} \quad \max(\lambda(i,k,n))>\lambda_{\text{high}}
$$

$$
\delta(\lambda(i,k,n))=1 \quad \text{if:} \quad \min(\lambda(i,k,n))<\lambda_{\text{low}}
$$

$$
\delta(\lambda(i,k,n))=0 \quad \text{otherwise}
$$

Through this learning algorithm, the normalized air-to-fuel ratios of each cylinder during the SI-HCCI hybrid combustion cycles are limited to between $\lambda_{\text{low}}$ and $\lambda_{\text{high}}$ ($\lambda_{\text{low}}=1$ and $\lambda_{\text{high}}=1.4$ in this case). Under this air-to-fuel ratio, IMEP is highly correlated to $F_{\text{DI}}$. Accordingly IMEP can be regulated by controlling $F_{\text{DI}}$ through the following formula,

$$
F_{\text{FP}}(i+1,k,n)=F_{\text{FP}}(i,k,n)+F_{\text{IC}}(i,k,n)
$$

$$
F_{\text{FP}}(i+1,k,n)=F_{\text{FP}}(i+1,k,n)
$$

The iterative learning control term ($F_{\text{ILC}}$) in Eq. (6) is calculated by a typical P type self learning algorithm [13]. Notice that in Eq. (6) $F_{\text{DI}}$ is controlled with respect to each individual cylinder, due to the cylinder-to-cylinder variations as discussed. The trained control values are stored in memory after each learning iteration. They will be used as feedforward terms for next learning iteration or torque tracking control that is going to be discussed in the next section. Figure 9 shows that the iterative learning control variables converge after a few iterations of learning. The first engine cycle (cycle 3) after the intake/exhaust valve lift switch experiences the largest improvement, leading to significant correction of $F_{\text{DI}}$. The largest cylinder-to-cylinder variation of $F_{\text{DI}}$ can also be seen at cycle 3. The variation of $F_{\text{DI}}$ counteracts the variance of IMEP, and makes IMEP smooth.
Some engine variables are plotted in Fig. 10 for both with and without the iterative learning control. With ILC of \( N_{TPO} \), the normalized air-to-fuel ratios of all cylinders during the hybrid combustion cycles are limited in the desired range, while without it rich gas-fuel mixture still exists during the combustion mode transition. Same as in the mode transition without ILC, the variables \( \lambda \), \( T_{IVC} \) and \( \theta_{SOHCCI} \) during the mode transition with ILC also show large cycle-to-cycle and cylinder-to-cylinder variations. The only signal with the reduced variation is the IMEP signal for the mode transition with ILC. This demonstrates the effectiveness of the ILC strategy.

Engine torque responses during the combustion mode transitions using different strategies are plotted in Fig. 11. Their statistics are listed in Tab. 2 for comparison. Smooth combustion mode transition can be realized by combining the throttle pre-opening, the SI-HCCI hybrid combustions and the iterative learning control strategy. Engine torque fluctuation reduces to 4.5\% during the engine combustion mode transition from the SI to HCCI combustion mode with constant load. A significant reduction of engine knock index, \( dP/d\theta \) (a good indicator of engine knock [14]), can also be found during the mode transition. This indicates that the proposed strategy can suppress engine knock during the mode transition.

Notice that the iterative learning control is only activated when engine is operated at 4.5 bar IMEP constant load condition. Once the ILC is activated, the feedback control needs to be deactivated, as illustrated in Fig. 8. Accordingly the ILC is only conducted in engine calibration during the product development phase, or during the constant load operation, for instance, the vehicle is under cruise operation. This also makes the engine mode transition robust against the engine aging. Under transient engine conditions, the ILC is deactivated and the engine will be controlled in different way.

**COMBUSTION MODE TRANSITION DURING ENGINE TIP OUT OPERATION**

During the transient engine operation, the ILC term is deactivated and the feedback control will be activated, as illustrated in Fig. 8. The combustion mode transition process is controlled by a typical feedforward combined with feedback control scheme. It is represented as

\[
F_{fb} = F_{feed} + F_{fb}
\]

The feedforward term \((F_{feed})\) in Eq. (7) is the trained value after \( i^{th} \) iterations of learning. The feedback term \((F_{fb})\) in Eq. (7) is the output of a PI controller, and it is used not only during the mode transition but also in the HCCI combustion mode.

In order to validate the control strategy under the transient engine operating condition, an HIL simulation was conducted to simulate the engine throttle tip out operation. In the simulation, the engine was initially operated in the SI mode. As a step input is applied to the accelerator pedal, the pedal position decreased from 40\% to 15\%. As a result, the engine IMEP reduced from 8.1 bar to 4 bar, which crosses the combustion mode transition threshold (4.5 bar). Accordingly the mode transition was triggered. As shown in Fig. 12, both \( \phi_{trottle} \) and MAP signals were decreased at first, and increased due to the combustion mode transition. Slightly rich combustions can be seen during the early stage of the tip out operation, since the throttle opening was decreased in the SI mode; once the mode transition started, the in-cylinder gas mixture was lean. In Fig. 12, one can also see that the engine IMEP tracks the desired IMEP during the entire tip out operation with the feedforward control trained by the ILC. In contrary, the IMEP signal shows about ±10\% fluctuation during

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**TABLE 2. ENGINE KNOCK INDEX AND MAXIMUM ERROR OF IMEP AND ENGINE TORQUE SIGNALS**

<table>
<thead>
<tr>
<th>Signal Type</th>
<th>IMEP Error (%)</th>
<th>Max ( dP/d\theta ) (bar/deg)</th>
<th>Torque Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>One-step</td>
<td>51.6</td>
<td>3.4</td>
<td>81.8</td>
</tr>
<tr>
<td>TPO</td>
<td>26.9</td>
<td>2.82</td>
<td>42.5</td>
</tr>
<tr>
<td>TPO + Hybrid</td>
<td>12</td>
<td>2.65</td>
<td>18.6</td>
</tr>
<tr>
<td>(TPO + Hybrid) with ILC</td>
<td>3.3</td>
<td>2.55</td>
<td>4.5</td>
</tr>
</tbody>
</table>
the combustion mode transition when the feedforward term was not trained. It can be concluded that the combination of feedback control and ILC trained feedforward control is capable of smooth combustion mode transition even under transient engine operating conditions.

![Graph](https://example.com/graph.png)

**FIGURE 12. ENGINE RESPONSES DURING TIP OUT**

**CONCLUSION**

In this paper, different control strategies are studied to make the SI to HCCI combustion mode transition smooth. It is gradually discovered that the smooth SI to HCCI combustion mode transition can be realized by combining the throttle pre-opening, SI-HCCI hybrid combustion phases, and the iterative learning control (ILC) of the throttle pre-opening timing and DI fuel quantity. The entire control strategy is validated for the fixed engine load condition in a hardware-in-the-loop simulation environment. Furthermore, the ILC control strategy can also be used during engine calibration process. When the trained feedforward control is combined with the PI feedback control, smooth SI to HCCI combustion mode transition can also be realized during transient engine operations.

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