

MODEL-BASED FEEDBACK CONTROL FOR AN AUTOMATED TRANSFER OUT OF SI OPERATION DURING SI TO HCCI TRANSITIONS IN GASOLINE ENGINES

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ABSTRACT

This paper takes a first step towards model-based feedback control for the transition between spark ignition (SI) and homogeneous charge compression ignition (HCCI) combustion modes by approaching the transfer out of SI operation during the SI into HCCI transition in a closed-loop control framework. The combustion mode switch is taken to be directly from SI to HCCI without an intermediate combustion mode between the two, and the HCCI phase of the transition is not addressed. The transfer out of SI operation is formulated as a multi-input, multi-output control problem with input and output constraints. A baseline feedback controller for the transfer is designed using linear quadratic regulator methods, and is tested in simulation on a nonlinear mean value engine model. A simple open-loop transition based on look-up table set points is included as well for comparison. The feedback controller shows the ability to complete the SI phase of the transition in a short number of cycles, while maintaining a minimal disturbance to the engine torque in comparison to the open-loop controller.

INTRODUCTION

The incorporation of homogeneous charge compression ignition (HCCI) combustion into the operating regime of gasoline engines is a topic of great interest in the automotive industry. Operation in HCCI offers substantial improvements in fuel economy over traditional spark ignition (SI) combustion, as well as the potential for ultra-low NO_x emissions [1]. Several methods exist for enabling HCCI operation, such as intake air heating [2, 3], exhaust gas recirculation (EGR) [4, 5], or variable compression ratio [6]. This paper focuses on the EGR method, in which hot exhaust gases are mixed with the fresh charge to manipulate the charge's thermodynamic state, so that when

compressed, the mixture auto-ignites with desirable combustion phasing. Recycling of exhaust gases can be achieved by internal EGR [7], wherein a variable valve timing (VVT) mechanism is used to transfer exhaust gas from one cycle to the next (residual gas), and also by external EGR [4], wherein the exhaust gas is routed back to the engine intake through piping in the air path. Recompression internal EGR is the assumed method of HCCI operation in this paper, which employs an early closure of the

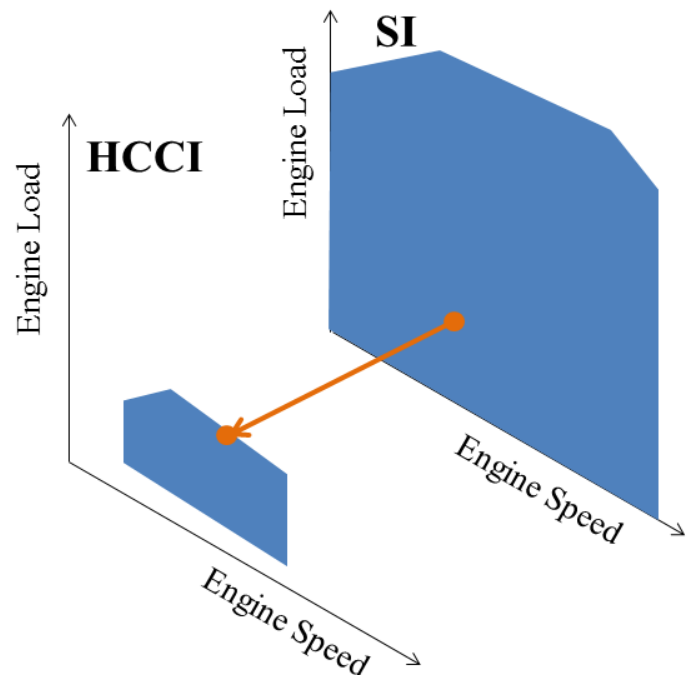


Figure 1. REPRESENTATIVE SI AND HCCI SPEED-LOAD MAPS WITH TRANSITION ILLUSTRATED.

exhaust valve to trap residuals in the cylinder [8], though internal EGR by reinduction of exhaust gases can be used as well [9]. For a detailed description of HCCI combustion, see [1].

A major hurdle to the implementation of HCCI is that its operating range is limited to low and intermediate engine speeds and loads, giving an operating envelope similar to that shown in Figure 1. At low loads, recompression HCCI has problems with late auto-ignition timing and/or misfire due to the lower exhaust gas temperature and hence lower heat transfer from the residual gases to the fresh charge. Conversely, at mid to high loads, the larger fuel amount can cause very high pressure rise rates with associated audible noise or hardware damage. These drawbacks are amplified by the advance of the combustion phasing caused by the higher residual gas temperature. Thus, although typical drive cycles operate mostly at low loads where HCCI is feasible, an HCCI engine will be forced to revert to traditional SI combustion in regimes where HCCI is not feasible, and so the engine must be capable of switching between these combustion modes.

The difficulty in switching between SI and HCCI combustion modes lies in the drastically different engine settings in which each mode functions (at a given speed and load). SI traditionally runs throttled with a stoichiometric air-fuel ratio (AFR) and positive valve overlap (PVO). Recompression HCCI runs unthrottled with a lean AFR and negative valve overlap (NVO), and has lower lift cams, a larger in-cylinder residual gas fraction, and a lower exhaust gas temperature than standard SI. This means that in order to transition from one mode to the other, each mode must be exposed to operating conditions that are significantly outside its normal range. Thus, the necessity of switching between SI and HCCI combustion modes poses a difficult control problem.

The development of control strategies for the SI-HCCI transition has been investigated in numerous studies through open-loop experimentation, see, e.g., [10–14]. Some approaches utilize variable valve actuation systems for greater flexibility in controlling the residual gas and air charge [10, 11], while others use a more practical VVT system with a two-stage cam profile to switch between high lift and low lift cams and achieve the necessary residual amount required for recompression HCCI within a single engine cycle [12–14]. Studies employ a variety of modifications to nominal SI/HCCI operation to aid in the transition, such as spark assisted compression ignition (SACI) [10, 11, 13, 14], split fuel injection [12, 13], external EGR [14], and late intake valve closing timing [10, 11]. The specifics of each open-loop strategy vary, but most are able to achieve a SI-HCCI transition at one or a few operating points after careful optimization, with each strategy displaying a differing degree of disturbance to the engine torque. While some of these studies incorporate nominal engine control unit feedback control to assist in their open-loop strategies, the SI-HCCI transition is never approached as closed-loop control problem, and no model-based feedback controller is applied to regulate the engine through the transition. Model-based feedback control may substantially aid

in carrying out the SI-HCCI transition, because the switching process is complex and involves many actuator-output couplings. As such, even if an open-loop transition strategy is developed that is successful in one operating condition, it may be difficult to schedule the strategy so that it functions robustly across the entire range of possible switching conditions, as noted in [15].

This paper takes a first step towards model-based feedback control for the SI-HCCI transition by applying state feedback control to transfer out of SI operation in the SI to HCCI transition. The HCCI phase of the SI to HCCI transition, as well as the transition in the direction of HCCI to SI, are left for future work. Naturally aspirated conditions are assumed, and more stochastically influenced dynamical effects such as cyclic variability and cylinder to cylinder variations (see, e.g., [16, 17]) are neglected. The work is carried out using a control-oriented mean value engine model (MVEM) with zero-dimensional manifold filling dynamics that has the ability to simulate both SI [18] and HCCI [8] combustion. The SI model of [18] is extended to predict combustion behavior at the extreme operating conditions experienced during the transfer out of SI operation. The problem of transferring out of SI operation is formulated assuming a direct switch from SI to HCCI, meaning that there is no intermediate combustion mode between the two. This represents a worst-case scenario for transferring out of SI, because by the end of the SI phase, the engine conditions must be sufficient for the execution of HCCI combustion on the subsequent cycle. In the case where SACI is introduced between SI and HCCI combustion, the spark may be used to assist in controlling the combustion phasing in order to smooth the transition into HCCI. Thus, the use of SACI or other potentially helpful modifications, such as split fuel injection during the HCCI phase of the transition, would only give greater flexibility in transferring out of SI operation as compared to the approach utilized here. The HCCI model used as a reference during the transfer out of SI operation is parameterized for speeds near 2000 RPM and loads up to approximately 3 bar net mean effective pressure (NMEP); thus, the transition region of focus will be near this range, idealized as a high load entry point into HCCI from SI similar to that depicted in Figure 1.

The paper proceeds by first describing the control objectives for the SI phase of the SI-HCCI transition and then formulating the control problem based on these objectives. The model used to simulate the SI phase of the SI-HCCI transition is covered next, and a description of state feedback controller applied for the transition follows. Closed-loop simulation results for the controller are presented and compared with a simple set point based open-loop scheme. Finally, the work is summarized and directions for future research are stated.

CONTROL OBJECTIVES

In order for a direct SI-HCCI switch to be successful, several goals must be met during the SI phase of the transition. As a first and most obvious goal, the throttle must be opened (dethrottled) as far as possible while in SI, so that the HCCI combustion has

sufficient air to operate properly during the cycles immediately after HCCI is engaged. This goal is approached by specifying an intake manifold pressure reference command, as the intake manifold pressure largely determines the flow into the cylinders and is a convenient variable for feedback control. The manifold pressure reference is taken to be specified externally by a look-up table. Ideally, the manifold pressure reference should be reached as quickly as possible, so that the duration of the switching process can be minimized. It is desirable to keep the switching process short because during the switch, the SI combustion must be placed in extreme operating conditions that are highly undesirable for fuel efficiency and emissions (explained below). Additionally, the engine may enter and exit the HCCI range of feasibility on a time scale on the order of a few seconds, as observed in the work of [10], which necessitates a fast transition process.

The next goal, which is common to all engine operating regimes, is to maintain the driver specified torque throughout the course of the transition. In this paper, the driver specified torque is quantified through a reference NMEP, which is assumed to be available for feedback through in-cylinder pressure sensors. These sensors greatly aid in monitoring the combustion phasing of HCCI and are therefore expected to be present on many HCCI engines. Throughout all simulations, it is assumed that the switching process happens fast enough that no change in the driver torque command occurs during the switching process, so that the reference NMEP stays constant throughout the entire duration of the switch. When the engine torque is maintained at some constant value throughout the transition, the transition is said to be torque neutral.

As a last goal, at the end of the SI phase, the conditions should be such that when HCCI is engaged on the subsequent cycle, the combustion is safe from abnormalities such as ringing or misfire. To translate this goal into a performance objective for the control system, the HCCI model is employed to estimate the combustion phasing that would result if HCCI was executed on the next cycle. Quantifying the combustion phasing by the crank angle at which 50% of the mass has burned, θ_{50} , the controller can adjust the high lift valve timings while operating in SI mode to indirectly affect the low lift valve timings so that the expected HCCI θ_{50} is maintained within some acceptable limits, and the control system can refrain from switching into HCCI if these limits are not met (see model development for a full description of the cylinder valve hardware). The predicted HCCI θ_{50} , which will serve as an output for the control system, will be denoted as $\hat{\theta}_{50}^H$.

CONTROL PROBLEM FORMULATION

Meeting the above objectives necessitates several substantial adjustments to be made during the SI phase of the transition. As the AFR of SI operation is typically fixed at stoichiometry, opening the throttle implies increasing the fuel injection amount, and therefore increasing the torque output of the engine. This ex-

Table 1. CONTROL PROBLEM FORMULATION

Control Inputs	$u = [\theta_t \quad m_f \quad \theta_{sp} \quad PVO \quad \theta_{SOI}]^T$
Controlled Outputs	$y = [p_{im} \quad NMEP \quad \lambda \quad \hat{\theta}_{50}^H]^T$
Reference Inputs	$r = [p_{im}^* \quad NMEP^*]^T$
Input Lower Bounds	$u_{low} = [0 \quad 0 \quad -50 \quad 0 \quad 300]^T$
Input Upper Bounds	$u_{high} = [100 \quad \infty \quad 0 \quad 60 \quad 400]^T$
λ Bounds	$0.9 \leq \lambda \leq 1.4$
$\hat{\theta}_{50}^H$ Terminal Bounds	$0 \leq \hat{\theta}_{50}^H \leq 10$

tra torque needs to be eliminated so that a torque neutral switch can be maintained. A natural choice is to retard the spark timing so that less work is extracted from the combustion process. Another option is to allow the AFR to be leaned during the SI portion of the transition, so that more air can be aspirated without necessitating an increase in fuel. This may be undesirable because lean mixtures are typically prohibited during nominal SI operation for NOx emissions and catalyst efficiency reasons. However, the transition process consists of only a small number of cycles, so the cumulative NOx emissions due to the transition may not be exceedingly severe. As an additional benefit, leaning the mixture reduces the exhaust gas temperature, which aids in achieving acceptable HCCI combustion phasing immediately after HCCI is engaged. The temperature issues stem from the fact that the exhaust gas temperatures of SI are substantially higher than those of HCCI, so that the residual gas is exceedingly hot during the first cycle on which HCCI is engaged, which tends to advance the auto-ignition timing even if the amount of trapped residual has been tuned for steady-state HCCI operation. The reduction in exhaust gas temperature also helps to counterbalance the rise in exhaust temperature that is observed when the spark is retarded. A last benefit of leaning is that it lengthens the burn duration and retards the combustion phasing, which can be useful in reducing the torque output of the engine. Taking these advantages into account, leaning of the mixture is incorporated into the switching strategy.

In addition to opening the throttle, retarding the spark, and leaning the mixture as required to meet the first two goals, the valve overlap must adjust so that the $\hat{\theta}_{50}^H$ value is within some safe limits as per the third goal. The motion of the valves will be highly dependent on the valve train hardware design, i.e. how the high lift valve timings relate to the low-lift valve timings. For the valve train modeled in this paper, even modest amounts of high lift PVO correspond to very low amounts of low lift NVO. This set-up makes regulating $\hat{\theta}_{50}^H$ easier because a lower NVO will trap less residuals and prevent $\hat{\theta}_{50}^H$ from becoming too advanced, which is a major risk due to the higher exhaust temperature of SI than HCCI as stated. For other valve train hardware

designs, a more extensive manipulation of the valve timings may be necessary before transitioning into HCCI. The HCCI combustion phasing is also affected by the fuel injection timing (see, e.g., [19]), which can be chosen by the controller in SI mode in order to assist the cylinder valves in the regulation of $\hat{\theta}_{50}^H$.

The control problem can then be formulated according to Table 1. The control inputs are throttle θ_t (%), fuel amount m_f (milligrams), spark timing θ_{sp} (CAD aTDC), PVO (CAD), and the HCCI start of fuel injection timing θ_{SOI} (CAD bTDC), and the outputs are the intake manifold pressure p_{im} (bar), NMEP (bar), the ratio of the AFR to stoichiometry λ , and $\hat{\theta}_{50}^H$ (CAD aTDC). The θ_{SOI} value specified by the controller is used only as a virtual control input to alter the predicted $\hat{\theta}_{50}^H$, assuming HCCI operates with a single injection during recompression. Due to the large delay associated with standard λ sensors, the λ value is assumed to be obtained from the known fuel mass and a model-based estimate of the cylinder air charge. To keep the model causal, unit cycle delays are introduced on values of the NMEP, λ , and $\hat{\theta}_{50}^H$ outputs used for feedback. This is necessary because the combustion actuators of fuel and spark timing have a direct feedthrough to these outputs, and so the delays ensure that the feedback controller calculates the the fuel mass and spark timing inputs based on feedback of the outputs from the previously occurring cycle, and not the current cycle. As a result of the delays, each of the outputs are also states, with manifold pressure coming from a continuous-time conservation of mass equation in the intake manifold, and the other outputs coming from a discrete-time unit delay. The manifold pressure and NMEP are controlled to some reference values, while $\hat{\theta}_{50}^H$ and λ are kept between some reasonable bounds. The manifold pressure reference is chosen to attain as high an air flow as possible in SI at the given speed and load, and the NMEP reference is assumed to come from the driver and stays constant over the transition duration. The λ bounds are chosen to prevent the combustion from operating too lean in order to avoid the sharp decrease in combustion stability observed with very lean mixtures [17] or operating too rich where the excess fuel does not contribute to the work extracted from combustion. The $\hat{\theta}_{50}^H$ bounds are chosen so that the first cycle of HCCI is expected to be free of combustion abnormalities such as ringing or misfire. Note that the $\hat{\theta}_{50}^H$ bounds represent terminal constraints, as $\hat{\theta}_{50}^H$ only must satisfy these bounds on the last cycle of SI to give safe combustion phasing on the first cycle of HCCI. The input bounds are chosen based on actuator saturation and knowledge of the range of validity of the model. Special attention must be given to the spark upper bound to prevent the SI combustion from retarding too much and risking a misfire.

MEAN VALUE MODEL FOR SI-HCCI TRANSITION CONDITIONS

Simulating a controlled transfer out of SI in an SI-HCCI transition requires the use of both SI and HCCI combustion models, where the HCCI model is necessary to produce the predicted HCCI value $\hat{\theta}_{50}^H$. The approach used here is to separate the com-

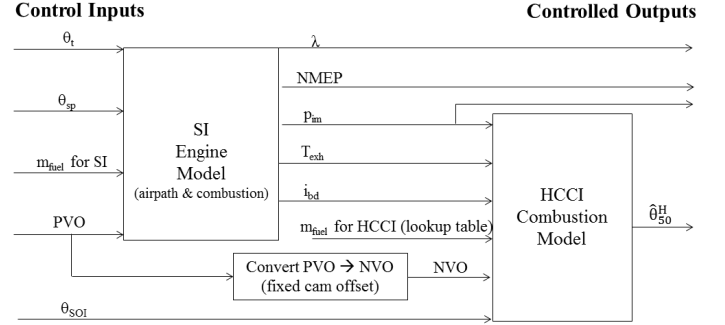


Figure 2. MODEL STRUCTURE WITH METHOD TO PREDICT THE HCCI 50% BURN POSITION FOR SUBSEQUENT CYCLE, $\hat{\theta}_{50}^H$.

Combustion modes into two distinct models, with the SI combustion model and air path model coming from [18] and the HCCI combustion model coming from [8]. Both models are parameterized to 2 liter, 4-cylinder engines. The throttle of the air path model retains the first order actuator dynamics with a time constant of .04 seconds from [18]. This separated modeling approach appears to show reasonable trends considering that hybrid combustion between SI and HCCI modes is not utilized in this work, though there was no data available to validate that the model does indeed capture all important transient phenomena during a SI-HCCI transition. Future work will evaluate the adequacy of the mean value model and its subsequent model-based controller in experiment. Note that the effects of boosting from the turbocharger in the air path model of [18] are omitted from this study by keeping the wastegate 100% open so that the turbine is majorly bypassed. Hence, the model corresponds to a naturally aspirated SI-HCCI engine.

An important issue for the SI-HCCI transition concerns the modeling of the cylinder valve actuation on the SI-HCCI switching engine. The model utilizes VVT hardware that contains a set of high lift and low lift cams that are switched between with an electro-hydraulic mechanism. The switching action is fast enough to occur within one cycle while the valves are closed, so no adverse disturbances to the valve profiles during the breathing phase of the switched cycle are modeled or compensated for in the algorithm to transfer out of SI. The crank angle offset between each set of cams is fixed, so that the position of the high lift cams can be explicitly calculated from the position of the low-lift cams during a switch, and vice-versa. The VVT mechanism is an electronic cam phaser whose dynamics are modeled with a first order lag with a time constant of .1 seconds.

The predicted HCCI 50% burn position $\hat{\theta}_{50}^H$ is generated by using the exhaust temperature T_{exh} and burned gas fraction i_{bd} output by the SI model as inputs to the HCCI model. As a conservatively high estimate of the residual gas temperature, T_{exh} is set equal to the temperature after blowdown, which excludes any heat transfer to the walls or exhaust manifold during the exhaust stroke. The valve timings for the $\hat{\theta}_{50}^H$ calculation are obtained by calculating the low-lift valve timings corresponding to

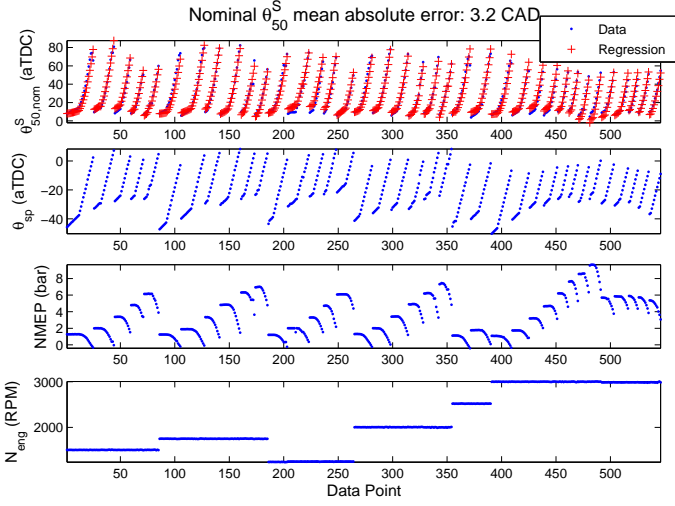


Figure 3. NOMINAL θ_{50}^S REGRESSION AT STOICHIOMETRIC CONDITIONS.

the current high-lift valve timings, which consists of a simple shift by the fixed offset between the two cam sets. Thus, the $\hat{\theta}_{50}^H$ value produced by the HCCI model estimates the θ_{50} position that would occur if HCCI was switched into on the next cycle (i.e. following the SI cycle that just occurred), with the current valve settings and state of the residual gases. A diagram illustrating the calculation of $\hat{\theta}_{50}^H$ is depicted in Figure 2. It may be possible to improve the model by including cylinder wall temperature dynamics in order to account for the fact that during the initial cycles of HCCI, the wall temperature will correspond to hotter SI conditions. Inclusion of wall temperature dynamics will be investigated in future work. Note that the $\hat{\theta}_{50}^H$ calculation is carried out with a fuel mass that is assumed to be given by a look-up table for HCCI operation at the current speed and load, and with an intake manifold pressure that is saturated from below at 0.8 bar, because the HCCI model is not conditioned on highly throttled data.

As described in the control problem formulation, during the SI-HCCI transition, the SI combustion model must be placed into extreme regions with a lean mixture and a large degree of spark retard in order to dethrottle the engine to the greatest extent before switching to HCCI. Capturing the combustion behavior in these extreme regions is important for the accuracy of the model simulations and model-based controller design. As such, the SI combustion model of [18] was modified to include an extended regression for the 50% burn position, denoted θ_{50}^S . The θ_{50}^S position was the main focus of the model development because it defines the point of instantaneous combustion in the SI model, as in the approach of [20], and so the effects of combustion phasing enter the model through the θ_{50}^S position. The extended θ_{50}^S regression has the following form:

$$\theta_{50}^S = \theta_{50,nom}^S(N_{eng}, m_f, \theta_{sp}) + \Delta\theta_{50,lean}^S(\lambda) \quad (1)$$

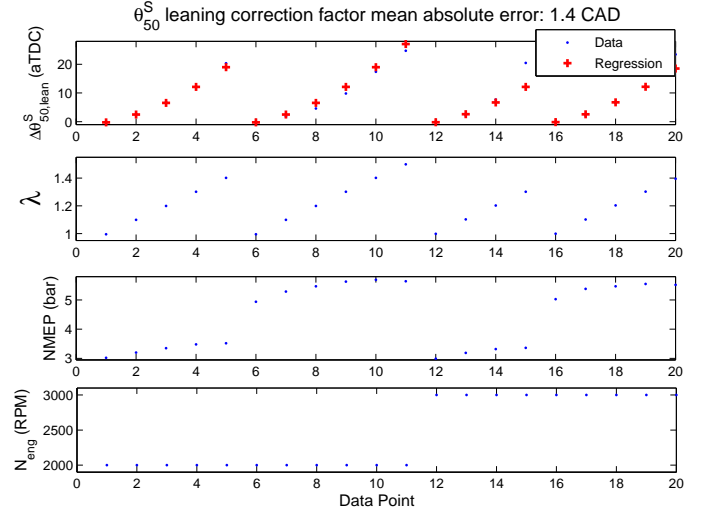


Figure 4. θ_{50}^S LEANING CORRECTION FACTOR REGRESSION.

where $\theta_{50,nom}^S$ captures the variation with engine speed, load, and spark timing at nominal stoichiometric conditions, and $\Delta\theta_{50,lean}^S$ captures the increase in combustion duration due to leaning. It should be noted that this model structure makes a significant approximation in separating the effect of AFR from the rest of the dependencies, since combustion is a nonlinear process and so superposition does not hold. However, the data available contained only retarded spark timings and leaned AFRs separately, and so each effect had to be accounted for independently. Despite this fact, the extended SI model with Eq. (1) has correct trends and adequate quantitative accuracy, so its outputs can be expected to be at least reasonable. The fitting results as well as the operating conditions for both regressions are shown in Figures 3 and 4.

The nominal θ_{50}^S is regressed to spark timing sweep data at various speeds and loads with stoichiometric operating conditions. The form of this regression is founded on the logic that at a given speed and load, θ_{50}^S follows a clear quadratic trend with spark timing. Thus a quadratic function of spark timing is used whose coefficients are linear functions of speed and fuel mass,

$$\theta_{50,nom}^S = a_2\theta_{sp}^2 + a_1\theta_{sp} + a_0 \quad (2)$$

$$a_i = \alpha_{i1}m_f + \alpha_{i2}N_{eng} + \alpha_{i3}, \quad i = 0, 1, 2 \quad (3)$$

Other regression forms, such as the classic form proposed by Kantor [21] or forms such as those in [22] for the 10 to 90% burn duration, were also employed to regress θ_{50}^S (or to first regress the burn duration then regress θ_{50}^S to the burn duration). Such forms were found to have comparable or better accuracy than that in Eq. (2), but the form of Eq. (2) has the benefit that it is based on variables that are all readily available, while the other forms rely on calculated variables such as the residual gas fraction or unburnt charge temperature, and therefore tend to amplify modeling error.

The leaning correction factor is regressed to lean AFR sweep data at constant combustion phasing across several speeds and loads. The “ Δ ” in $\Delta\theta_{50,lean}^S$ indicates the amount to which spark had to be advanced to maintain constant combustion phasing as the mixture was leaned. The leaning correction factor takes the form of a simple quadratic regression to λ , which captures the trend well but does not account for the variations of the leaning effect with speed and load.

LQR STATE FEEDBACK CONTROL

The SI phase of the SI-HCCI transition as formulated above is a multi-input multi-output (MIMO) control problem with a complicated input-output connectivity. As such, centralized state feedback control is a logical approach to the problem, due to its inherent ability to work with MIMO systems and account for input-output cross couplings. As a baseline for the effectiveness of state feedback control when designed through common linear methods, infinite horizon linear quadratic regulator (LQR) methodology is applied to the problem. The tuning is carried out using a linearization of the MVEM, and is implemented in discrete time in order to circumvent the MVEM’s hybrid nature. The sample period is chosen as the time for one cycle at an engine speed of 2000 RPM, assuming that the engine speed remains relatively constant throughout the duration of the transition. Full state feedback is assumed in order to first investigate the effectiveness of LQR control with the complication of an observer excluded. In naturally aspirated engines equipped with an in-cylinder pressure sensor, obtaining the states may not pose a large problem because a simple air path model with only intake manifold filling dynamics may suffice, and the combustion output states may be obtained through the in-cylinder pressure sensor. However, in a boosted scenario and/or without an in-cylinder pressure sensor, obtaining the states may require a substantial observer design effort.

Integral action is included on each of the outputs to ensure zero-error tracking to their references. Note that for the outputs of λ and $\hat{\theta}_{50}^H$, integral action is not absolutely necessary because these outputs are only required to stay within some bounds, instead of tracking some exact references as the intake manifold pressure and NMEP must. However, output constraints are not directly enforced with LQR methodology, and simulations showed that it is very difficult to tune the LQR controller weights to maintain λ and $\hat{\theta}_{50}^H$ within their respective bounds. Adding integral action on λ and $\hat{\theta}_{50}^H$ significantly eases this task by guaranteeing that these outputs are tracked to some reference, which can be chosen within the output bounds. For λ , the reference is set slightly below the λ upper bound, to allow for overshoot in the λ response without violating the constraints. For $\hat{\theta}_{50}^H$, the reference is taken to be equivalent to the reference supplied to the HCCI combustion phasing controller, assuming that the HCCI combustion controller is driven by a reference θ_{50} value, as in, e.g., [19, 23].

With the discrete integration carried out through a simple

rectangle rule, the dynamics of the integrator states w follow

$$w(k+1) = w(k) + T_s(y(k) - \bar{r}(k)) \quad (4)$$

where k is the time step index, T_s is the sample period, and $\bar{r} = [p_{im}^* \quad NMEP^* \quad \lambda^* \quad \hat{\theta}_{50}^{H*}]^T$ is the reference vector extended to include reference commands for λ and $\hat{\theta}_{50}^H$ to satisfy the constraints. The augmented system then takes the form

$$\begin{bmatrix} x(k+1) \\ w(k+1) \end{bmatrix} = \begin{bmatrix} A & 0 \\ T_s C & I \end{bmatrix} \begin{bmatrix} x(k) \\ w(k) \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u(k) + \begin{bmatrix} 0 \\ -T_s I \end{bmatrix} \bar{r}(k) \quad (5)$$

$$y(k) = \begin{bmatrix} C & 0 \end{bmatrix} \begin{bmatrix} x(k) \\ w(k) \end{bmatrix} \quad (6)$$

where A , B , and C follow the usual notation for state variable models and I is the identity matrix. The feedback control input is then calculated using the LQR state feedback gain K and combined with the operating point of the linearization u_0 ,

$$u(k) = -K \begin{bmatrix} x(k) \\ w(k) \end{bmatrix} + u_0 \quad (7)$$

The linearization point u_0 is chosen towards the end of the transition out of SI (i.e. high throttle opening, retarded spark, etc.). Linearizing in this region was found to be helpful for a fast execution of the transition because when u_0 is combined with the linearized feedback via Eq. (7), it pushes the actuators further towards their end points and hence the transition can be completed faster. When applying this operating point strategy, the fuel mass set point may cause the mixture on the first cycle of the transition to become rich, because the fuel mass set point will be higher than the nominal fuel mass since it is chosen towards the end of the transition out of SI. To give the air charge sufficient time to build so that it can accommodate the higher fuel mass set point, the controller fuel mass command is not applied on the first cycle; that is, the fuel mass control is not activated until the second cycle of the transition. The same modification is applied to the spark timing, because retarding the spark before the fuel increases will cause a drop in the the engine torque.

The tuning of the state weighting matrix is carried out focusing only on the output and integrator states, and setting the rest of the weights to zero. This approach is equivalent to using an output weighting matrix to generate the weights for the states x in conjunction with a diagonal weighting matrix for the integrator states w . The weights are chosen to achieve a reasonable trade-off between torque regulation, the speed of the intake manifold pressure response, and flexibility in responding to disturbances to $\hat{\theta}_{50}^H$. It is also important to choose the λ output weight to ensure that any overshoot in the λ response does not violate its upper

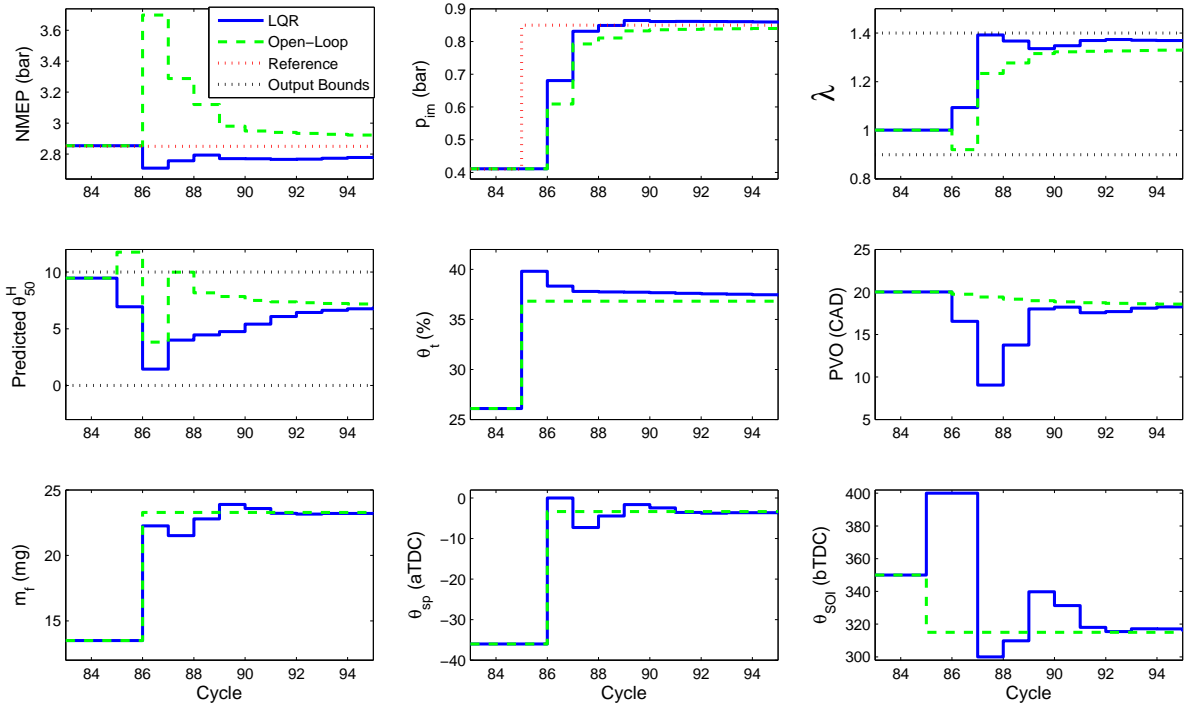


Figure 5. SIMULATION OF SI PHASE OF SI-HCCI TRANSITION WITH LQR AND OPEN-LOOP CONTROLLERS.

bound. The input weighting matrix is kept diagonal with very small weights, in order to create a cheap control scenario where the actuators are free to move quickly for a fast transfer out of SI. The weights on the PVO and θ_{SOI} inputs are kept lower than the others, in order to allow these inputs more freedom to compensate for the various disturbances to $\hat{\theta}_{50}^H$ throughout the transition, such as the changing exhaust temperature. The higher weights on the other actuators were still found to give sufficient intake manifold pressure response speed and torque regulation. The weighting matrices used are given below, with only the diagonal entries shown because all weighting matrices are diagonal:

$$\text{diag}\{Q_y\} = [1 \quad 1000 \quad 150 \quad 0.1] \quad (8)$$

$$\text{diag}\{Q_i\} = [1000 \quad 1000 \quad 10 \quad 40] \quad (9)$$

$$\text{diag}\{R\} = [0.1 \quad 0.01 \quad 0.1 \quad 0.1 \quad 0.001] \quad (10)$$

where Q_y is the output state weighting matrix, Q_i is the integrator state weighting matrix, and R is the input weighting matrix. The input and output ordering is the same as that in Table 1.

SIMULATION RESULTS

The LQR controller was tested in simulation on the nonlinear MVEM. To assess the benefits of employing model-based

feedback control to transfer out of SI operation, a simple open-loop control was also tested. In this open-loop control scheme, actuator positions at the end of the transition out of SI are stored in a look-up table as set points, and when the switch is requested the actuators are commanded to these set points. The set points are taken at the end point of the LQR controller simulation where all references are tracked and constraints satisfied by virtue of the integral action. This open-loop controller is not meant to portray a fully optimized open-loop strategy, but simply to show how feedback control may be helpful in shaping the system response beyond just specifying set points derived from actuator settings at the desired steady-state. The same policy of preventing the activation of the fuel and spark commands on the first cycle of the transition used by the LQR controller is used for the open-loop controller as well.

Figure 5 shows the closed-loop response during a transition out of SI operation for both of the controllers at an engine speed of 2000 RPM and load of 2.85 bar NMEP. The linearization point for the LQR controller for this operating condition is chosen at $u_0 = [35.5 \quad 22.6 \quad -4 \quad 25 \quad 320]^T$ (see Table 1 for input ordering). The transition is commanded and the controllers activated at cycle 85, and the outputs are plotted over the next several cycles so that the response can be seen. Both controllers attain a 95% rise time for the intake manifold pressure of 2-4 cycles, and both are able to regulate $\hat{\theta}_{50}^H$ inside its bounds within the same or fewer

amount of cycles as well. This implies that the SI phase of the transition can be completed within 2-4 cycles, because the air charge has been increased sufficiently and the combustion phasing on the first cycle of HCCI is expected to be satisfactory. The intake manifold pressure response of the LQR controller is slightly faster than that of the open-loop, which results from the feedback controller compensating for the manifold dynamics by commanding an overshoot in the throttle input from its steady-state value. The LQR controller also aggressively uses the valve overlap and θ_{SOI} in an attempt to regulate $\hat{\theta}_{50}^H$ to its reference value. If too aggressive, however, the controller may momentarily advance $\hat{\theta}_{50}^H$ below its lower bound when compounded with the effect of an increasing exhaust gas temperature as the spark retards throughout the transition. The open-loop $\hat{\theta}_{50}^H$ controller should always yield a $\hat{\theta}_{50}^H$ that is within its bounds in around the same time frame that it takes to track the manifold pressure reference (at a given operating point). This is because, once the fuel, spark timing, and valve overlap actuators are stepped to their end points, the only remaining effect on the predicted HCCI combustion phasing is through the manifold dynamics, and once these settle out, the system should be at the open-loop controller set point where $\hat{\theta}_{50}^H$ is within its bounds.

The main differentiating factor between the controllers is the severity of the disturbance to the torque that each one causes. Clearly, the open-loop control causes a torque spike that is noticeably large, with the peak deviation from the reference being .84 bar NMEP, or 29.8%. The LQR controller sees milder torque fluctuations, with a peak deviation of approximately .15 bar NMEP, or 5.1%. This value is close to the standard limits on the indicated mean effective pressure coefficient of variation (IMEP COV) of 5-6%. The LQR controller is able to achieve this smoother transition because it shapes the transient path of the system to its end point, instead of simply commanding all actuators to the end point immediately.

CONCLUSION

This paper approached the transfer out of SI operation during the SI to HCCI transition from a model-based feedback control standpoint, as a first investigation into the use of model-based feedback control to switch between SI and HCCI combustion modes. The transfer out of SI operation was formulated as a MIMO control problem with input and output constraints, with the objective being to reach a state in SI operation that is favorable for switching into HCCI, while maintaining the driver load command as the operating condition is changed. The target end state for the SI phase is characterized by an increased intake manifold pressure to dethrottle the engine as much as possible, as well as a predicted HCCI combustion phasing that is within some reasonable bounds so that the first cycle of HCCI is expected to be free of combustion abnormalities. An LQR state feedback controller was applied to this problem based on a linearization of a nonlinear MVEM that was extended to lean and late SI combustion regimes. The controller met all criteria nec-

essary for the completion of the SI phase of the transition within 2-3 cycles in closed-loop simulations on the MVEM. The LQR controller compared favorably for drivability and response speed with a simple open-loop control approach that stepped all actuators to stored set points, demonstrating that feedback control may substantially improve the transient response over open-loop methods that are not carefully optimized. Feedback control may therefore be useful as an alternative to stringent open-loop optimization. In future work, the HCCI phase of the transition will be included in the model-based feedback control framework, and other options for closed-loop transition control, such as SACI, will be explored. The effectiveness of the finalized controller will then be tested in experiment.

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NOMENCLATURE

(a/b)TDC	(After/Before) top dead center combustion
AFR	Air-fuel ratio
CAD	Crank angle degree(s)
EGR	Exhaust gas recirculation
i_{bd}	Burned gas fraction
λ	Ratio of AFR to Stoichiometry
LQR	Linear quadratic regulator
m_f	Fuel mass
MVEM	Mean value engine model
N_{eng}	Engine speed
NMEP	Indicated net mean effective pressure
NVO	Negative valve overlap
p_{im}	Intake manifold pressure
PVO	Positive valve overlap
T_{exh}	Exhaust gas temperature
θ_{50}	50% mass fraction burn angle
$\hat{\theta}_{50}^H$	Predicted HCCI θ_{50}
θ_{50}^S	θ_{50} for SI combustion
θ_t	Throttle angle
θ_{sp}	Spark timing
θ_{SOI}	Start of fuel injection timing (HCCI mode)
VVT	Variable valve timing

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