## MODEL BASED CONTROL OF EXHAUST RECOMPRESSION HCCI

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Abstract: This paper presents an approach to model-based controller development for Homogeneous Charge Compression Ignition (HCCI). The control strategy for HCCI is based on a physical two state discrete time model of the process. As the trapped exhaust is pivotal in setting up the cyclic coupling, its temperature and the amount of oxygen present in it are selected as the states of the system. The model is linearized about an operating point in order to generate simple linear controllers. The states of the model, however, are not directly measurable. Hence an observer is used to generate estimates of the states, which are then used by a controller to track a desired system trajectory. Results from simulation show that the system can be controlled with some error about an operating point. Experimental results demonstrate comparable tracking, along with a reduction in the cycle-to-cycle variability of HCCI.

Keywords: HCCI, model based control, HCCI model states

### 1. INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) provides significantly better fuel efficiency and emissions characteristics than today's engines, specifically extremely low $NO_x$ emissions. There are several ways of achieving HCCI, all of which involve increasing the sensible enthalpy of the reactant mixture, so as to promote its auto-ignition. One possibility is to heat or pre-compress the inducted air (Tunestal, 2001). Alternately, hot exhaust gases can be trapped and recompressed in the engine by closing the exhaust valve early in the exhaust stroke. The trapped exhaust can then be mixed with fresh charge to obtain a mixture with higher sensible energy (Law, 2001).

This paper presents the development of a control strategy for exhaust recompression HCCI with direct injection of liquid fuel (such as gasoline) into the cylinder. The controller is synthesized on the basis of a simple physical model of HCCI described in previous work (Ravi, 2006). A variety of models for HCCI have been developed in the past. These include simple zero-dimensional models (Najt, 1983), as well as ones with more detailed chemical kinetic and thermodynamic descriptions (Smith, 1997). For the synthesis of model-based controllers, however, simple models capturing the key dynamics of HCCI are required. The model used for controller synthesis in this paper is such a model. It is based on a description of the fundamental thermodynamics of HCCI. The model states are chosen so as to represent physical quantities critical in determining the nature of HCCI combustion - reactant concentrations and temperature (Shaver, 2004). Since the role of the trapped exhaust in determining the phasing and nature of combustion is critical, the states for the model are associated with the trapped exhaust.

This model can used to develop a variety of control laws for HCCI. In this paper, the model is linearized about an operating point to enable the use of simple linear controllers. The states as chosen are not directly measurable on an actual engine setup, and so an observer that generates an estimate of the states based on the available measurements must be developed. This estimate is then used by an LQR controller to track a desired output trajectory, which is fed to the controller as a reference input. When implemented, the controller shows comparable behavior in both simulation and experiment around the desired operating point. Additionally, experimental results indicate that the controller is also useful in reducing the cyclic variability of HCCI, thereby achieving greater combustion stability at points otherwise unstable.

The value of the proposed framework can also be seen in the insight it provides into the control problem. For instance, the output matrix of the linearized system is poorly conditioned. This is because one of the two model outputs - peak pressure during a cycle - is not a strong function of the states except through the other output, the angle of peak pressure. Once this is known, therefore, the value of peak pressure itself does not provide much more information about the states. With a model-based approach, therefore, such effects can be clearly seen and an observer designed accordingly.

## 2. MODEL DESCRIPTION

### 2.1 Model inputs and outputs

The model used to synthesize a controller for HCCI assumes the following inputs:

(1) *Moles of fuel injected* in the current cycle, $n_f$
(2) *Volume at intake valve closure*, or the point at which instantaneous mixing between air, fuel and trapped exhaust is assumed to occur, $V_{IVC}$
(3) *Volume at exhaust valve closure*, or the point at which the states of the system are determined, $V_{EVC}$ and
(4) *Fraction of external EGR*, $\psi$. This assumes the presence, in addition to exhaust retention, of an external EGR mechanism where some portion of exhaust is routed from the exhaust manifold into the main incoming air stream. External EGR enables control of the oxygen concentration independent of the mixture temperature, as the exhaust cools down while it is recirculated externally. It is particularly useful during high-load operation of HCCI.

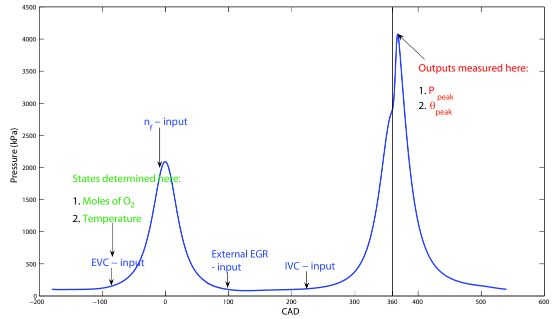


Fig. 1. States, inputs and outputs in the HCCI model

Though the effect of all these inputs is modeled, the work presented here focuses on primarily two inputs - the quantity of fuel injected into the cylinder and the volume at exhaust valve closure. The intake valve is held at a constant timing, and there is no external EGR.

In terms of the outputs of the model, what is ultimately desired is that the engine produce the amount of work required, and that combustion occur at the desired phasing. The outputs of the model are therefore chosen as quantities that are representative of these values, but are also easily measurable on an actual engine test-bed. These are

(1) *Peak pressure*, $P_{peak}$, which serves as a proxy for the net work output of the engine
(2) *Angle of peak pressure*, $\theta_{peak}$, which represents the phasing of the combustion event

### 2.2 Model states

Several possible state choices exist for a simple HCCI model. In (Shaver, 2004), the desired outputs - peak pressure and angle of peak pressure - are chosen as the states. This model has been shown to be useful for control of exhaust-reinduction HCCI. However a fundamental set of states, directly linked to the thermodynamic state of the engine, is more desirable.

The process of combustion is basically dictated by two characteristics of the reactant mixture: reactant concentrations (which determine whether the reactant molecules are close enough to have sufficient number of collisions), and mixture temperature (which determines whether the collisions are energetic enough to cause a reaction to happen). Choosing a set of state variables, therefore, that in some way represent these quantities, gives a fundamental basis for an HCCI model.

Specifically in HCCI, the pivotal role of the trapped exhaust can be recognized. The exhaust retained from the previous engine cycle is used to heat (and dilute) the fresh charge for the current cycle, and induces cycle-to-cycle coupling. This trapped exhaust, therefore, is essentially what carries information about combustion in one engine cycle through to the next.

Based on this, the states of the HCCI system can be chosen as those variables that capture the temperature and amounts of reactants in the trapped exhaust. Of the two reactants (fuel and $O_2$), the fuel is an input to the system, and is directly controlled through the fuel injector. Therefore the states are chosen as:

(1) Moles of oxygen in the products of combustion at EVC, $n_{O_2,k}$
(2) Temperature of the trapped exhaust at EVC, $T_{e,k}$

Figure 1 shows a typical HCCI pressure trace, with the locations of the different inputs, outputs and states within an engine cycle. Top dead center after recompression is taken as the 0 crank angle reference.

### *2.3 Modeling assumptions*

A typical HCCI engine cycle with exhaust recompression and direct fuel injection can be described as follows. The intake valve is opened during the induction stroke, drawing in fresh air that mixes with the trapped exhaust from the previous engine cycle to form a homogeneous mixture. This mixture is compressed during the upward stroke of the piston. The compression ends in a fast, uniform combustion process close to the top dead center position of the piston. This combustion is homogeneous in nature, unlike typical SI combustion which is characterized by a propagating flame. Subsequent to this, useful work is extracted from the engine as the piston moves down. Close to bottom dead center, the exhaust valve is opened, and a portion of the products of combustion is pushed out into the exhaust manifold during the upward stroke of the piston. The exhaust valve, however, is closed early so as to trap a significant amount of the combustion products in the cylinder. The trapped exhaust is then recompressed as the piston reaches top dead center. During this recompression stroke the fuel is injected into the cylinder.

Based on this description, a single HCCI cycle can be broken down into several well defined, and easily modeled stages. In the following description, standard thermodynamic variables such as pressure ($P$), temperature ($T$), volume ($V$) and number of moles ($n$) are used to relate the thermodynamic properties from one stage to the next.

(1) *Adiabatic induction at atmospheric pressure* culminating in *instantaneous mixing* of fuel, air and trapped exhaust at point of intake valve closure (IVC). On mixing, we have

$$n_{total} = n_{air}(1+\psi) + n_{tr.exh} \tag{1}$$

where $\psi$ is the fraction of external EGR. Also, applying the 1st law to mixing process, and the ideal gas law at the point of mixing, we get two additional relationships

$$h_{sensible,before} - h_{vaporization} = h_{sensible,after}$$
$$n_{total} = \frac{P_1 V_{IVC}}{R_u T_1} \tag{2}$$

where $P_1$ and $T_1$ are pressure and temperature of the mixture at IVC. The first relation accounts for the enthalpy of vaporization of the fuel.

(2) *Isentropic compression* from IVC to point of combustion. Therefore at the end of compression

$$T_2 = \left(\frac{V_{IVC}}{V_{23}}\right)^{\gamma-1} T_1$$
$$P_2 = \left(\frac{V_{IVC}}{V_{23}}\right)^{\gamma} P_i \tag{3}$$

where $V_{23}$ is the cylinder volume at end of compression, and start of combustion.

(3) *Isochoric combustion* occurring instantaneously and uniformly. As HCCI combustion is typically very fast and homogeneous, it can be approximated as an instantaneous, constant volume process. Applying the first law,

$$U_2 = U_3 + Q_{comb} \tag{4}$$

based on which an expression for temperature at the end of combustion, $T_3$ can be obtained. Also, based on ideal gas law, and the recognition that the number of moles of reactants $\approx$ number of moles of products, we have

$$P_3 = P_2 \frac{T_3}{T_2} \tag{5}$$

which gives an expression for the peak pressure in the cycle, $P_3$. This is one of our model outputs.

(4) *Isentropic expansion* from point of instantaneous combustion to the moment the exhaust valve opens (EVO)

$$T_4 = \left(\frac{V_{23}}{V_4}\right)^{\gamma-1} T_3$$
$$P_4 = \left(\frac{V_{23}}{V_4}\right)^{\gamma} P_3 \tag{6}$$

(5) *Isentropic exhaust* from EVO to when the exhaust closes (EVC). Assuming atmospheric pressure at valve closure, temperature at the point of exhaust valve closure is:

$$T_5 = \left(\frac{P_{atm}}{P_4}\right)^{\frac{\gamma-1}{\gamma}} T_4$$

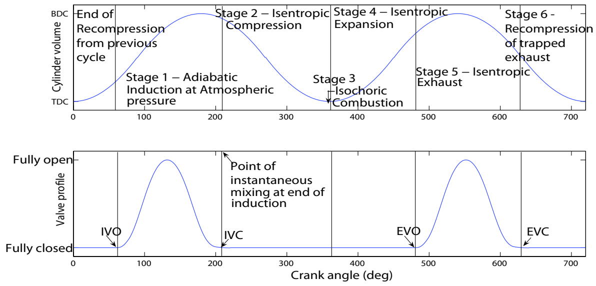


Fig. 2. Stages in the HCCI cycle

The fraction of trapped exhaust, $\beta$ can be obtained from applying the ideal gas law at the beginning and end of the exhaust process.

$$\beta = \frac{N_5}{N_4} = \frac{V_{EVC} P_{atm} T_4}{V_4 P_4 T_5}$$

The total amount of trapped exhaust is given by

$$n_{tr.exh} = \frac{P_{atm} V_{EVC}}{R_u T_5} \tag{7}$$

(6) *Recompression* of trapped exhaust from EVC till intake valve opening (IVO) in the next cycle. Only the temperature of the cylinder contents changes during recompression. The heat transfer in this process is accounted for by relating the temperature of the mixture after recompression to the temperature before with a simple algebraic factor. This is the temperature of the trapped exhaust as it mixes instantaneously with fresh air on the next cycle.

$$T_{1,tr.exh} = \chi T_{EVC} \tag{8}$$

The above relation has been used by Shaver et al. (Shaver, 2004) to model heat transfer during recompression, and shown to be fairly accurate around an operating point.

Figure 2 shows these discrete stages with respect to the cylinder volume at any given position of the crank. The figure also shows typical valve profiles for the intake and exhaust valves. As seen, the exhaust valve is closed well before the piston reaches top dead center, to allow part of the exhaust to be trapped in the cylinder.

### 2.4 Combustion phasing modeling

A physical model for combustion would need to incorporate the effects of chemical kinetics on the process. Here a simple global Arrhenius rate model is used, which has been shown to be adequate to capture the dynamics affecting phasing (Shaver, 2004). The reaction rate for the overall combustion reaction is given as

$$RR = A_{th} e^{\left(\frac{E_a}{R_u T}\right)} [fuel]^a [O_2]^b \tag{9}$$

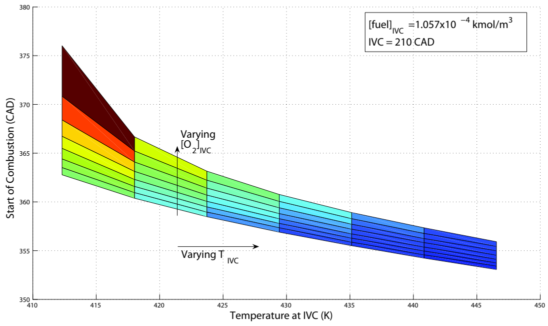


Fig. 3. Effect of temperature and oxygen concentration on combustion phasing

where $E_a$ is the activation energy, $A_{th}$ is a pre-exponential factor, $T$ is temperature, and $[fuel]$ and $[O_2]$ are concentrations of fuel and oxygen respectively. Integrating this global Arrhenius rate equation from IVC to the point of combustion gives an expression of the form

$$\int RR = \int_{\theta_{IVC}}^{\theta_{th}} A_{th} e^{\left(\frac{E_a}{R_u T}\right)} [fuel]^a [O_2]^b dt \tag{10}$$

Combustion can then be modeled to begin when this integral crosses a certain threshold value, $K_{th}$. The point at which the peak in-cylinder pressure is reached, $\theta_{23}$ is related to the point at which the threshold is crossed as $\theta_{th} = \theta_{23} - \delta\theta$, where $\delta\theta$ is assumed to be constant as a consequence of approximating the combustion event to be a function of the crank angle. This simplification is valid around an operating point, especially as the engine speed is assumed to be constant. The effect of oxygen concentration and temperature at the beginning of compression on the phasing of combustion can be seen in Fig. 3, which simulates the Arrhenius integral for a particular set of initial conditions, across a range of initial mixture temperatures and oxygen concentrations.

### 2.5 Model summary

Based on the assumptions described above, and the state description, a two state nonlinear state space model is developed that maps the inputs on one engine cycle to the outputs on that cycle through the states.

$$\begin{aligned} x_{k+1} &= F(x_k, u_k) \\ y_k &= G(x_k, u_k) \end{aligned} \tag{11}$$

The states, inputs and outputs are given by

$$x_k = \begin{bmatrix} n_{O_2,k} \\ T_{e,k} \end{bmatrix}, u_k = \begin{bmatrix} n_{f,k} \\ V_{IVC,k} \\ V_{EVC,k} \\ \psi_k \end{bmatrix}, y_k = \begin{bmatrix} P_{pk,k} \\ \theta_{pk,k} \end{bmatrix} \tag{12}$$

where $n_{O_2,k}$ and $T_{e,k}$ represent the number of moles of oxygen in the trapped exhaust, and the temperature of the trapped exhaust at EVC, $n_{f,k}$ represents the quantity of fuel injected into the cylinder, $V_{IVC,k}$ and $V_{EVC,k}$ are the cylinder volumes at IVC and EVC respectively, $\psi_k$ is the amount of external EGR (as a fraction of the total air entering the cylinder), $P_{pk,k}$ is the peak pressure in a cycle, and $\theta_{pk,k}$ is the crank angle at which the peak occurs.

A complete description of the model with validation is presented in (Ravi, 2006), with analytical expressions for all equations in Eqn. 11.

## 3. LINEARIZATION

This model is now used to synthesize a controller for HCCI. However, due to its highly nonlinear nature, it is first linearized about an operating point, so as to allow the synthesis of linear controllers. Linear control around an operating point is a first step towards more global HCCI control. The linearization is performed analytically, with linear expressions being taken for any quantity $a_k$ of the form

$$a_k = \bar{a}_k + \tilde{a}_k \tag{13}$$

where $\bar{a}_k$ represents the value of the quantity $a$ at the nominal operating condition, and $\tilde{a}_k$ represents its deviation from that operating point. These expressions are then substituted in the nonlinear state and output update equations, and Taylor expansions to the first term give linear system equations around the particular operating condition.

$$\begin{aligned} \tilde{x}_{k+1} &= A\tilde{x}_k + B\tilde{u}_k \\ \tilde{y}_k &= C\tilde{x}_k + D\tilde{u}_k \end{aligned} \tag{14}$$

where $A$, $B$, $C$ and $D$ are matrices. As the linearization is performed analytically, this gives the form of a general linear model. Expressions for the matrices are functions of the operating point at which the system is linearized, engine parameters, and physical properties of the cylinder constituents.

## 4. CONTROLLER SYNTHESIS AND IMPLEMENTATION

The states of the system as chosen are not directly measurable. Therefore the first step in the development of a control strategy is the synthesis an observer that can be used to estimate the states. The state estimate can then be used by the controller to track a desired system trajectory.

### *4.1 Observer design*

An estimator for the system can be designed using the linearized model. If the state estimate for the state $\tilde{x}_k$ is $\hat{x}_k$, the estimator dynamics are represented as

$$\hat{x}_{k+1} = A\hat{x}_k + B\tilde{u}_k + L(\tilde{y}_k - \hat{y}_k) \tag{15}$$

where $L$ is the estimator, $\tilde{u}_k$ is the input (obtained from the controller), $\tilde{y}_k$ is the measured value of the output, and $\hat{y}_k$ is the estimated value of the output.

From the output equation in Eqn. 14, $\hat{y}_k = C\hat{x}_k + D\tilde{u}_k$, and therefore

$$\begin{aligned} \hat{x}_{k+1} &= A\hat{x}_k + B\tilde{u}_k + L(\tilde{y}_k - C\hat{x}_k + D\tilde{u}_k) \\ &= (A - LC)\hat{x}_k + (B - LD)\tilde{u}_k + L\tilde{y}_k \end{aligned} \tag{16}$$

The estimation is however complicated by the fact that the output $C$ matrix for the linear system is poorly conditioned. This is due to the fact that of the two outputs, the peak pressure value is a strong function of the amount of fuel injected in the cylinder, which is an input. This is particularly true once the phasing of combustion (the other output) is fixed. Therefore once the angle of peak and amount of fuel injected are known, the value of peak pressure does not give much additional information. A measurement of the net work output (IMEP) would suffer a similar limitation, as it is directly related to the amount of fuel injected. Hence the observer uses only the angle of peak measurement to estimate both states.

This redundancy of the peak pressure value as a measurement for this particular control strategy has several implications. Peak pressure has been used in past work (Shaver, 2004) as a reasonable substitute for work output. Control of work output, therefore, is possible through controlling peak pressure and phasing. However, it would appear that from the standpoint of estimation, peak pressure is not a very useful *measurement*, given other information (such as the angle of peak pressure). This limits the role of peak pressure as an indicator of HCCI combustion.

### *4.2 Output controller design*

To track a desired output trajectory, a reference input is used as described in (Franklin, 1994). The control input, then, is of the form

$$u = -K_x x + (N_u + K_x N_x)r \tag{17}$$

where $r$ is the reference input (representing the desired output trajectory) and $K_x$ is the controller. $N_u$ and $N_x$ are matrices obtained from the following relation that specifies that the system

Table 1. Engine parameters

| Parameter | Value | Units |
|---|---|---|
| Engine speed | 1800 | rpm |
| Stroke | 93.2 | mm |
| Connecting rod length | 147 | mm |
| Bore diameter | 81 | mm |
| Compression ratio | 13 | |

Table 2. Operating point at which continuous simulation and simple control model are compared

| Parameter | Value | Units |
|---|---|---|
| IVO | 80 | CAD |
| IVC | 220 | CAD |
| EVO | 480 | CAD |
| EVC | 640 | CAD |
| Mass of fuel injected per cycle | 7 | mg |
| Peak pressure | 4075 | kPa |
| Angle of Peak Pressure | 367.7 | CAD |

respond with a zero steady state error to any constant input.

$$\begin{bmatrix} A-I & B \\ C & D \end{bmatrix} \begin{bmatrix} N_x \\ N_u \end{bmatrix} = \begin{bmatrix} 0 \\ I \end{bmatrix} \tag{18}$$

where $I$ represents the identity matrix, and 0 the zero matrix.

Combining Eqn.s 16 and 17, we obtain the complete representation of the controller-observer system.

$$\begin{aligned} \tilde{u}_k &= -K_x\hat{x}_k + (N_u + K_xN_x)r_k \\ \hat{x}_{k+1} &= (A - LC)\hat{x}_k + (B - LD) \times \\ &\quad \{-K_x\hat{x}_k + (N_u + K_xN_x)r_k\} + L\hat{y}_k \\ &= (A - LC - BK_x + LDK_x)\hat{x}_k + L\hat{y}_k \\ &\quad + (B - LD)(N_u + K_xN_x)r_k \end{aligned} \tag{19}$$

The particular control strategy to be used is an arbitrary choice, as the real motivation is to test the applicability of the approach to experimental control of HCCI. Here an LQR controller is chosen as a representative controller.

### 4.3 Implementation in simulation

The controller-observer pair is implemented on a more complex continuous time simulation model that is validated against experiment. This model is described in (Shaver, 2004). It is a continuous time, ten-state model that includes much of the complex thermodynamics of the HCCI process. Valve flows are modeled using compressible flow equations, heat transfer occurs continuously throughout the engine cycle and is modeled by the extended Woschni correlation (Chang, 2004), and the combustion event is of finite duration and captured by a Wiebe function.

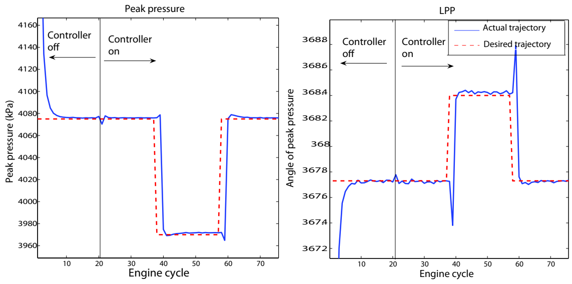


Fig. 4. LQR output controller implemented in simulation - tracking of outputs - small step change

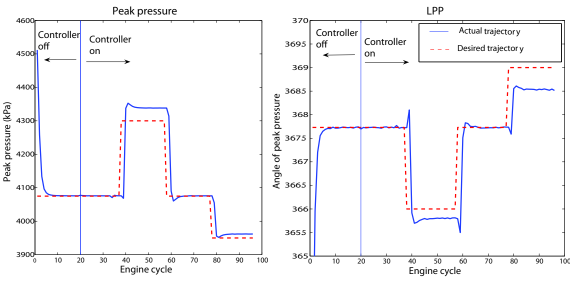


Fig. 5. LQR output controller implemented in simulation - tracking of outputs

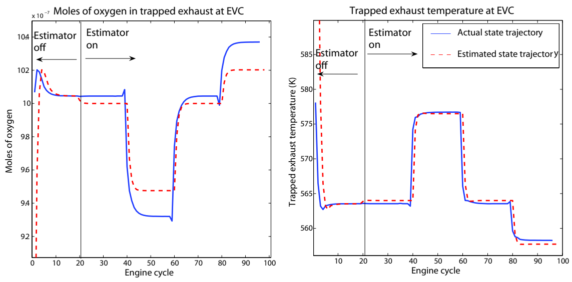


Fig. 6. Estimator performance in simulation - evolution of states

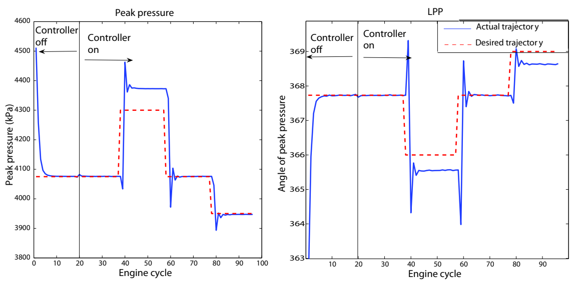


Fig. 7. LQR output controller implemented in simulation - tracking of outputs, higher control gains

The simulation model is parameterized and validated against a single cylinder engine testbed with a gasoline direct inject system and a fully flexible variable valve actuation system. Engine parameters are shown in Table 1.

Table 2 shows the characteristics at the particular operating condition about which the nonlinear control model has been linearized for the results shown. The linearization about this point is used

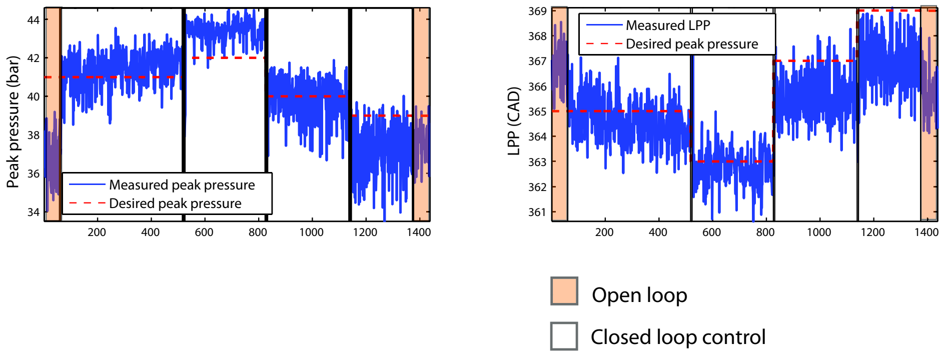


Fig. 8. Control of HCCI - output tracking

as the basis for the observer and controller synthesized and implemented in simulation. First, the controller is used to track a small step change about the operating point. Figure 4 shows the tracking achieved in simulation. Subsequently a larger change in operating point is commanded, and the controller performance is shown in Fig. 5. Figure 6 shows the observer performance, comparing the actual state trajectory obtained in simulation with the trajectory estimated by the observer for the same case.

### *4.4 Discussion of simulation results*

The controller-observer pair is used to track desired trajectories in simulation. First an estimate is generated for the states, based on the angle of peak measurement. The estimate obtained from the observer is then used by the controller to track a desired output trajectory. An LQR controller is used here, with large weights on the IVC and external EGR inputs. Therefore the only two inputs used to control the system here are the exhaust valve closure timing, and the quantity of fuel injected. The desired output trajectory is fed to the controller as a reference input, $r$.

For a small step about the operating condition, it can be seen in Fig. 4 that the controller-observer pair works well, with accurate tracking being achieved. There is no steady state error for small step changes, as the linearization is valid in a small region around the operating point. Therefore, as expected, the performance of the controller is as desired.

For a larger step change, however, the performance deteriorates slightly. Though the controller is still fairly effective, some steady state error can be observed in Fig. 5. This can be attributed to the fact that the linearization of the model is a little more inaccurate as we move away from the nominal operating point. Consequently, the estimation of the states is not completely accurate. This can be seen in Fig. 6. It is also seen that the trapped exhaust temperature is estimated more accurately than the oxygen content. This is because the phasing of combustion is a stronger function of temperature than oxygen concentration. This is evident in Eqn. 10, where it can be seen that combustion phasing has an exponential dependence on temperature, and a polynomial dependence on fuel and oxygen concentrations. Therefore the angle of peak measurement, which is essentially a proxy for combustion phasing, gives a more accurate estimate of the trapped exhaust temperature. Some of the error in tracking also arises due to the fact that both states are being estimated using just one measurement.

Figure 7 shows the tracking achieved with higher gains on the controller. As seen, the overshoot increases, and there is some ringing effect with a higher controller gain.

Therefore, the control strategy developed here is very reliable for small changes about the nominal operating point. As we move further away from it, however, the linearization becomes less valid, and errors creep in. Though these errors can be eliminated with a more aggressive control strategy, the aim of this work is really to prove the validity of the model-based approach to HCCI control.

### *4.5 Experimental implementation*

The controller tested in simulation was implemented on the single cylinder HCCI engine testbed described earlier. Figure 8 shows the tracking achieved on the engine, when a series of step changes around the nominal operating point are commanded. As seen the engine responds as soon as the controller is switched on, and then subsequently follows the desired system trajectory. The performance in experiment is similar to that in simulation, displaying the same magnitude of steady-state error.

The controller also enables a significant reduction in cycle-to-cycle variability of HCCI, particularly when the phasing of combustion is late. Figure 9 shows the net work output of the engine (IMEP) at such a point, where the controller is switched

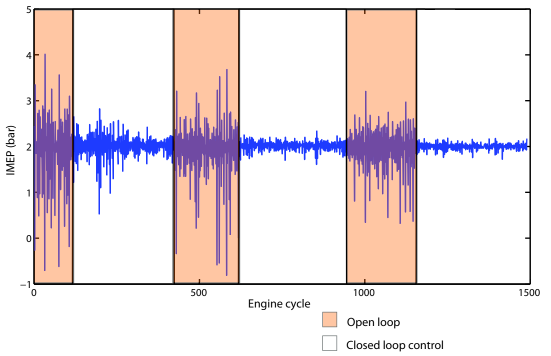


Fig. 9. Reduction in cyclic variability - IMEP

on and off several times. As seen, there is a marked increase in the stability of combustion when the controller is switched on, even though this is not an output being controlled for directly.

## 5. CONCLUSIONS

The results described in this paper suggest that a simple physical two state model can be used to develop control strategies for HCCI about an operating point. The states for the model are chosen as the concentration of oxygen in the trapped exhaust at EVC, and the temperature of the trapped exhaust at EVC. The model is linearized and used to synthesize a state estimator and controller for HCCI. This controller is seen to be effective at tracking a desired output trajectory in both simulation and experiment. It also enables stabilization of combustion at open-loop unstable points on the engine. Performance is similar in both simulation and experiment, validating the use of model-based control techniques. Additionally, the model provides an insight into the redundancy of one of the measurements used by the observer - the peak pressure in an engine cycle.

The results indicate that a measure of work output of the engine does not provide much information for state estimation once we have a measure of the phasing of the combustion and the quantity of fuel injected. Also, as phasing is a stronger function of the mixture temperature, the temperature state can be estimated more accurately than the oxygen concentration. Future work will investigate the extent to which these estimation challenges impact controller performance. If necessary, the inaccuracy in the estimation of oxygen concentration could be addressed with the inclusion of additional sensors, such as an exhaust oxygen sensor.

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