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Decoupled Control of Combustion Timing and Work Output in Residual-Affected HCCI Engines

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Abstract— **Homogeneous Charge Compression Ignition (HCCI) is a promising low temperature combustion strategy for internal combustion engines. However, when HCCI is achieved with variable valve actuation (VVA) the lack of a direct combustion initiator and cycle-to-cycle dynamics complicate control of the process. This work outlines a strategy for the simultaneous control of both in-cylinder pressure and combustion timing through the use of an approximately decoupled timing/pressure controller. The decoupling is achieved by controlling peak pressure and combustion timing on separate time scales with different VVA-induced control inputs, inducted gas composition and effective compression ratio, respectively. The paper also details how the strategy can be extended to handle decoupled timing/work output control. Experimental results show that in-cylinder pressure or work output can be controlled on a cycle-to-cycle basis, while combustion timing is slowly varied.**

I. Introduction

Homogeneous charge compression ignition represents a promising approach for reducing emissions and increasing efficiency in internal combustion engines. There are various ways to achieve HCCI, including heating or pre-compressing the intake air ([1], [2]) or varying the compression ratio [3]. The method considered in this paper, residual-affected HCCI, uses variable valve actuation (VVA) to induct the desired amount of fresh reactants and previously exhausted combustion products ([4], [5]). In the case of residual-affected HCCI, several challenges exist including cycle-to-cycle coupling through the exhaust gas and the lack of a direct combustion trigger. Closed-loop control can help address these challenges.

Several approaches to closed-loop control of HCCI engines have been demonstrated [6], [7], [8], [9]. Agrell et al. [6] used valve timings to effectively alter the compression ratio and control combustion timing. Haraldsson et al. [7] modulated the fuel amount to vary IMEP while altering the mixture ratio of two fuels to control combustion timing, a timing control strategy also adopted by Bengtsson et al. [9]. Olsson et al. [8] took a similar approach but used compression ratio instead of fuel mixture to shift combustion timing.

In previous work [10], the authors demonstrate cycle-to-cycle peak pressure control using a physics-based approach. Due to the self-stabilizing nature of the residual-affected HCCI, the combustion timing remains fairly constant for the engine studied. In [11], an additional control input - intake valve closing (IVC) - is introduced as a way to control both combustion timing and peak pressure simultaneously. IVC is used to dictate the effective compression ratio (ECR) by determining the start of compression. While the peak pressure control strategy developed in [10] is easily implemented in experiment, the cycle-to-cycle control methodology for both pressure and combustion timing, presented in [11], is computationally intensive and difficult to practically implement in real time. This paper develops a simpler experimentally implementable approach to dual peak pressure/combustion timing control, and shows how this technique can be extended to direct work output control.

The simplified strategy outlined in this paper is to approximately decouple the control of peak pressure and combustion timing by controlling them on separate time scales with different control inputs (inducted composition and IVC, respectively). By designing the combustion timing controller to be notably slower than the cycle-to-cycle control of peak pressure, the effect of cycle-to-cycle combustion timing variation on peak pressure can be neglected. Combustion timing and IVC become slowly varying parameters in the peak pressure dynamics. This simplifies the peak pressure control problem, allowing the use of an approach that varies inducted gas composition to control peak pressure (similar to that implemented in [10]). Since combustion timing in residual-affected HCCI is more dependent on IVC than inducted gas composition, IVC is used to modulate combustion timing. The combustion timing control gains are intentionally selected to achieve transient responses that are slightly slower than the cycle-to-cycle pressure control. While modulation of IVC does influence in-cylinder pressure, the cycle-to-cycle pressure controller compensates for it.

Experimental results show accurate tracking of both combustion timing and peak in-cylinder pressure. Controlled combustion timing and peak pressure response times are on the order of 30 (2 seconds) and 5 engine cycles (0.3 seconds), respectively, validating the claim that these two system outputs can be controlled simultaneously on different time scales. Experimental control results also show the capability of direct, cycle-to-cycle regulation of work output.

II. HCCI MODELING

The framework for modeling HCCI combustion (originally outlined in [11]) is based on linking six distinct stages that occur during a given engine cycle: induction, compression, combustion, expansion, exhaust and residence in the exhaust manifold. The cylinder volume at the beginning and end of each stage is either known or modeled (see Figure 2).

The inducted gas composition is the first VVA-controllable model input. For this study, exhaust valve closing (EVC) variation is used to modulate the inducted gas composition. As EVC is delayed, the exhaust valve is open for an increased amount of time during the induction stroke, increasing the amount of reinducted exhaust. This trend is demonstrated with steady-state experiment values in Figure 1, where the metric used to describe the inducted composition is the molar ratio of re-inducted products to inducted reactant, $\alpha \equiv N_p/N_r$.

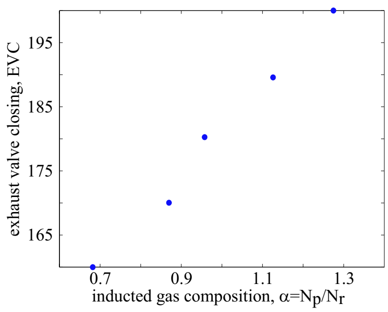

Fig. 1. Relationship between exhaust valve timing and experimental estimate of molar ratio of inducted exhaust products and reactants, $\alpha \equiv N_p/N_r$

The second control input is the intake valve closing (IVC). Since the intake valve is the last to close (as shown in Figure 2, IVC determines the amount of compression (i.e. ECR). In Section II-C it will be shown that IVC has direct influence on combustion timing. The model outputs are the peak in-cylinder pressure, P, and the combustion timing, θ_{comb}, both of which are measurable quantities to be controlled.

A. Induction

The induction process is modeled as a constant pressure, adiabatic process. The first thermodynamic state considered follows the stage 1 induction process at IVC, and is evaluated through use of the 1st law of thermodynamics and the ideal gas assumption. At this point the reactant charge and re-inducted product gases are assumed to be homogeneously mixed throughout the combustion chamber.

B. Isentropic Compression

During stage 2, isentropic compression is assumed. The thermodynamic state of the system is expressed at the end of this stage and is related directly to the post-induction condition.

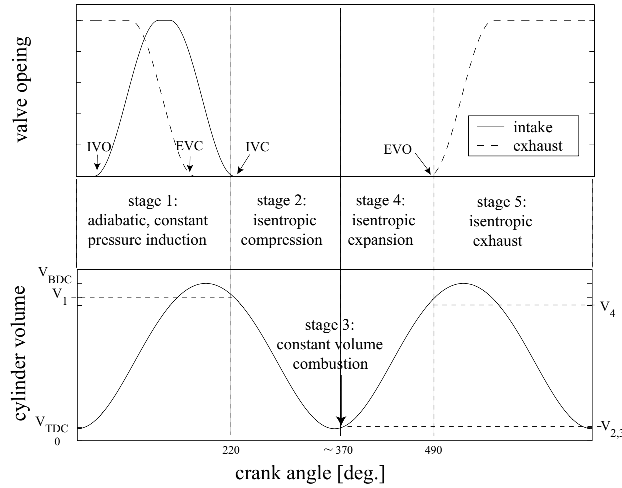

Fig. 2. HCCI cycle partitioning

C. Constant Volume Combustion

The combustion event is modeled as a constant volume process at an in-cylinder volume of V_{23}, producing the major products of combustion at an elevated temperature and pressure. The thermodynamic state at this point is evaluated with respect to the previous ones by again invoking the ideal gas assumption and 1st law of thermodynamics. Cylinder gas to wall heat transfer is included to approximate the total amount of in-cylinder heat transfer that occurs during the engine cycle. The heat transfer is simply modeled as a percentage, ϵ, of the chemical energy available during the conversion from reactant to products, $LHV_{C_3H_8}$, the lower heating value of the fuel.

Previous work by the authors ([12], [13]) demonstrates that an integrated Arrhenius model of combustion is a simple and accurate way to mathematically describe HCCI combustion timing. The model takes the form:

$$K_{thresh} = \int_{IVC}^{\theta_{comb}} exp(-E_a/(R_u T))[fuel]^a[O_2]^b d\theta/\omega \quad (1)$$

where ω is the engine speed and θ_{comb} corresponds to the combustion timing, such that $V_{23} = V(\theta_{comb})$. The values A, E_a/R_u, a, b and n are empirical parameters determined from combustion kinetics experiments for the particular fuel. K_{thresh} is set at one experimental operating condition. Note the dependence of combustion timing on in-cylinder temperature, reactant concentration and the start of compression. A very interesting aspect of residual-affected HCCI, captured by Equation 1 and originally reported by the authors in [12], is the self-stabilizing nature of the process due to the competing affects of reactant concentration and mixture temperature. For increasing amounts of hot reinducted exhaust, the reactant concentration drops while the mixture temperature increases. This trend causes little

$$P_k = \frac{\left((1-\epsilon)LHV_{C_3H_8} + c_v\left(\frac{V_1}{V_{23}}\right)^{\gamma-1} T_{in}\right)(1+\alpha_{k-1})\left(P_{k-1} - \left(\frac{V_1}{V_{23}}\right)^{\gamma}\right)\left(\frac{V_1}{V_{23}}\right) + \chi\alpha_k(1-\epsilon)LHV_{C_3H_8}(V_1/V_{23})^{\gamma} P_{k-1}^{1/\gamma}}{c_v T_{in}(1+\alpha_{k-1})\left(P_{k-1} - \left(\frac{V_1}{V_{23}}\right)^{\gamma}\right) + \chi\alpha_k(1-\epsilon)LHV_{C_3H_8} P_{k-1}^{1/\gamma}} \quad (2)$$

to no change in combustion timing as inducted composition is varied. On the other hand, effective compression ratio via IVC has a direct and substantial affect on combustion timing. As the lower limit of integration is decreased (i.e. as IVC occurs earlier), the integration satisfies Equation 1 for lower values of the upper limit, the combustion timing. In other words, earlier IVC leads to earlier combustion timing. Then, one way of directly affecting the combustion timing, with very little change to the inducted gas composition, is through modulation of IVC. For this reason IVC is used as a direct control input for combustion timing.

D. Isentropic Expansion and Exhaust

Post-combustion, the stage 4 expansion process is assumed isentropic. It progresses until exhaust valve opening (EVO) occurs. The major product gases are then assumed to exhaust isentropically to the exhaust manifold. The thermodynamic state of the system is evaluated at this point by assuming atmospheric pressure in the exhaust manifold.

E. Cyclic Coupling: Exhaust Manifold Heat Transfer

The reinducted product temperature on a cycle is related to the temperature of the gas exhaust on the previous cycle. The reinducted product temperature is modeled as a percentage, χ, of the temperature of the products exhausted on the previous cycle. Although simple, this approach works well in predicting the behavior of the experimental engine.

By linking the thermodynamic states together for an engine cycle, a dynamic model for the peak pressure can be developed (Equation 2 given at the top of the page). Together with the model for combustion timing, Equation 1, the peak pressure model completes the mathematical description of the HCCI process. Here c_v is the constant volume specific heat (assumed to be the same, for the inducted reactant and reinducted product gases), T_{in} is the temperature of the reactant gas in the intake manifold, γ is a mean specific heat ratio, P_k and P_{k-1} are the in-cylinder peak pressures on the current and previous cycles, V_{23} is the cylinder volume at the combustion event and V_1 is the cylinder volume at IVC. Equation 2 clearly shows the existence of cycle-to-cycle dynamics, as the current cycle peak pressure is related to that of the previous cycle.

III. Controller Development

A variety of closed-loop controllers can be synthesized to track desired values of peak pressure and combustion timing using the models developed in the previous section. A fairly simple approach is to try to approximately decouple the regulation of peak pressure and combustion timing by controlling them on separate time scales with different control inputs, inducted composition and ECR, respectively (Figure 3). The approach outlined here is to control the peak pressure on a cycle-to-cycle basis by using EVC to modulate the inducted gas composition. Combustion timing control is achieved on an intentionally slower time scale through use of IVC to vary the effective compression ratio. This is justified by the observation in Section II-C that combustion timing is directly and most profoundly affected by the amount of compression.

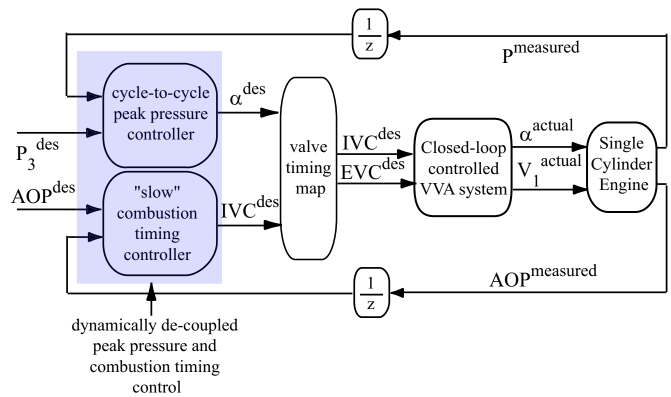

Fig. 3. Control Strategy for Simultaneous Control of Peak Pressure and Combustion Timing

A. Combustion Timing Control

Previous results [6], have shown that IVC timing can effectively control combustion timing. In that study, a simple proportional-integral (PI) control scheme controlled combustion timing with responses on the order of 10 cycles. Due to the successful implementation of this approach elsewhere, a slow PI combustion timing controller has been adopted here as well, with the form:

$$u_{tc,k} = u_{tc,k-1} + K_p(e_{tc,k} - e_{tc,k-1}) + K_I e_{tc,k} \quad (3)$$

where the intake valve closing time on cycle k is:

$$IVC_k = I\bar{V}C + \Delta IVC = I\bar{V}C + u_{tc,k} \quad (4)$$

and:

$$e_{tc,k} = AOP_k^{measured} - AOP_k^{desired} \quad (5)$$

is the error in the combustion timing (angle of peak pressure (AOP)) on cycle k. The IVC to combustion timing dynamics in open loop are simplified to a static gain. The PI combustion timing control gains are selected via pole placement to achieve a response time of 2 seconds.

B. Peak Pressure Control

In order to address the need for mean tracking and a reduction in cycle-to-cycle variation of peak in-cylinder pressure while bounding the “energy” of the control input, a local H_2 controller is synthesized from a linearized version of the peak pressure model. The following sections outline the linearization and control synthesis approaches.

C. Linearization of Pressure Relation

The peak pressure model, Equation 2, can be linearized about an operating point $(\bar{\alpha}, \bar{P}, \bar{V_{23}}, \bar{V_1})$. At this point, the combustion timing (V_{23}) and IVC (V_1) are considered slowly varying parameters in the peak pressure model since they are modulated at slower time scales. Straightforward linear expansions for α and P and the Taylor expansion of $P_{k-1}^{1/\gamma}$ taken to two terms, yields:

$$\alpha_k = \bar{\alpha} + \tilde{\alpha_k} \tag{6}$$

$$P_k = \bar{P} + \tilde{P_k} \tag{7}$$

$$P_{k-1}^{1/\gamma} \approx \bar{P}^{1/\gamma} + \tilde{P}_{k-1}\frac{\bar{P}^{(1-\gamma)/\gamma}}{\gamma} \tag{8}$$

Applying these to Equation 2, and neglecting second order terms of fluctuations (i.e. $\tilde{\alpha_k}\tilde{P_k}, \tilde{\alpha_k}\tilde{\alpha_k}, \tilde{P_k}\tilde{P_{k-1}}, \tilde{P_k}\tilde{\alpha_{k-1}}\tilde{\alpha_k}$, etc) leads to:

$$c_1\beta_k = c_2\beta_{k-1} + c_3\tilde{\alpha_k} + c_4\tilde{\alpha_{k-1}} \tag{9}$$

where $\beta_k \equiv (P_k - \bar{P})/\bar{P}$ is the normalized difference between desired and actual pressure, and:

$$c_1 = \frac{c_v T_{in}}{(1+\bar{\alpha})^{-1}}\left(\bar{P} - \frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right) + \chi(1-\epsilon)LHV_{C_3H_8}\bar{\alpha}\bar{P}^{1/\gamma} \tag{10}$$

$$c_2 = \frac{-c_v T_{in}}{(1+\bar{\alpha})^{-1}}\left(\bar{P} - \frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right) - \frac{LHV_{C_3H_8}}{(1-\epsilon)^{-1}\gamma}\left(\frac{\chi\bar{\alpha}}{\bar{P}^{(\gamma-1)}}\left(\bar{P} - \frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right) + \frac{\bar{V}_1}{\bar{V}_{23}}(1+\bar{\alpha})\right) \tag{11}$$

$$c_3 = -c_v T_{in}\bar{P} - \chi(1-\epsilon)LHV_{C_3H_8}\bar{P}^{(1-\gamma)/\gamma}\left(\bar{P} - \frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right) \tag{12}$$

$$c_4 = \frac{c_v}{T_{in}^{-1}}\frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}} + \left(\frac{LHV_{C_3H_8}}{(1-\epsilon)^{-1}}\frac{\bar{V}_1}{\bar{V}_{23}} + \frac{c_v}{T_{in}^{-1}}\frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right)\frac{1}{\bar{P}}\left(\bar{P} - \frac{\bar{V}_1^{\gamma}}{\bar{V}_{23}^{\gamma}}\right) \tag{13}$$

Equation 9 can also be written as a low-order discrete linear transfer function:

$$\frac{\beta(z)}{\alpha(z)} = \frac{c_3 + c_4 z^{-1}}{c_1 - c_2 z^{-1}} \tag{14}$$

or in state space form:

$$x_{k+1} = Ax_k + Bu_k \tag{15}$$

$$y_k = Cx_k + Du_k \tag{16}$$

where:

$$A = \begin{bmatrix} 0 & 0 \\ \frac{c_4}{c_1} & \frac{c_2}{c_1} \end{bmatrix}, B = \begin{bmatrix} 1 \\ \frac{c_3}{c_1} \end{bmatrix}, C = \begin{bmatrix} 0 & 1 \end{bmatrix}, D = 0 \tag{17}$$

$$x_k = \begin{bmatrix} \tilde{\alpha}_{k-1} \\ \beta_{k-1} \end{bmatrix} \qquad u_n = \tilde{\alpha_k} \qquad y_n = \beta_k \tag{18}$$

D. H_2 Control Formulation

From the linearized model, Equations 15-18 an H_2 control strategy can be developed. The general feedback control problem formulation is shown in Figure 4, where the weights on state noise, measurement noise, and performance noise are depicted in detail. The standard H_2 optimal control problem is to find a stabilizing feedback controller which minimizes the H_2 norm from the system "noise" inputs, w, to "performance" outputs, z.

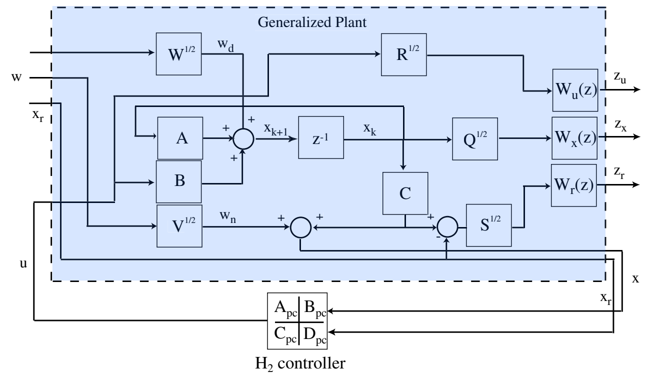

Fig. 4. General control configuration considered for the synthesis of the in-cylinder peak pressure controller

The noise weights, W and V, depict the variances of the state (i.e. "process") and output (i.e. measurement) noise. The system control input, states and tracking error are each weighted with a constant (R, Q and S, respectively) and frequency dependent transfer function ($W_u(z)$, $W_x(z)$ and $W_r(z)$, respectively). In order to stress the desire for tracking, the transfer function for tracking error, $W_r(z)$, is a low pass filter. This introduces the same sort of effect that the integrator portion of a PID controller achieves. In order to reduce cyclic dispersion, the portion of the state weighting transfer function which corresponds with the normalized peak pressure, $W_x(z)$, is a high pass filter. This weights the higher frequency components of the peak pressure, yielding a control law which attenuates cycle-to-cycle variation. In a similar manner, the control input transfer function weighting, $W_u(z)$, is a high pass filter, emphasizing the fact that a rapidly changing control input is not desirable. Overall, the formulation shown in Figure 4 allows a tradeoff to be made between mean tracking, cycle-to-cycle variation reduction and control effort. Figure 5 shows the performance weights ($RW_u(z)$, $QW_x(z)$ and $SW_r(z)$) used in this study.

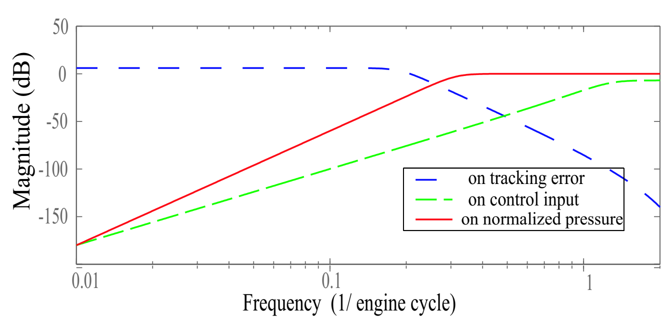

Fig. 5. The frequency dependent weights used for synthesis of the peak pressure control

From the solution of the H_2 synthesis problem, the controller gains (A_{pc},B_{pc},C_{pc}) are found. The general form of the H_2 peak pressure controller is:

Bore/Stroke	97/92 mm
Engine speed	1800 rpm
Coolant temperature	100 C
Fuel	Propane
Intake air conditions	Ambient temp., unthrottled
Compression ratio	15.5:1

TABLE I

ENGINE SPECIFICATIONS

$$x_{pc,k+1} = A_{pc}x_{pc,k} + B_{pc}\begin{bmatrix} \tilde{\alpha}_k \\ \beta_k \\ \beta_{desired,k} \end{bmatrix} \quad (19)$$

$$\tilde{\alpha}_k = C_{pc}x_{pc,k} \quad (20)$$

As shown in Equations 19 and 20, the inducted gas composition, $N_p/N_r \equiv \alpha = \bar{\alpha} + \tilde{\alpha}$ is the output of the peak pressure controller. The inputs to the controller are the inducted gas composition from the previous cycle, the normalized peak pressure from the previous cycle β_k and the desired normalized peak pressure on the current cycle.

Note that the combustion timing (i.e. V_{23}) and IVC (i.e. V_1) are slowly varying parameters in the peak pressure dynamics, Equation 2. In this study, these dynamics have been linearized about a particular operating point $(\bar{\alpha}, \bar{P}, \bar{V_{23}}, \bar{V_1})$. It is likely that a parameter-varying control strategy directly accounting for variation in V_{23} and V_1 will improve the control response, albeit at the expense of additional complexity.

IV. Controller Implementation on Research Engine

The controller has been experimentally implemented on a single cylinder research engine, with general engine characteristics given in Table I. Experimental results are shown in Figures 6 and 7. Figure 6 shows the benefit of simultaneous control. If only control of combustion timing is considered (plot on left), then there is no regulation of peak in-cylinder pressure. In order to hold a desired peak pressure while modulating the desired timing, simultaneous control of both timing and peak pressure (plot on right) must be used. Figure 7 shows the decoupled controller performance over a range of desired pressures and combustion timings. Peak pressure and combustion tracking responses occur within 5 and 25 engine cycles, respectively. Using this scheme rapid modulation of work output is possible while maintaining combustion timing within a desired region.

V. Extension to Work Output Control

In the previous sections, the HCCI model motivates a decoupled pressure and combustion timing control strategy. In practice however, it may be more useful to directly

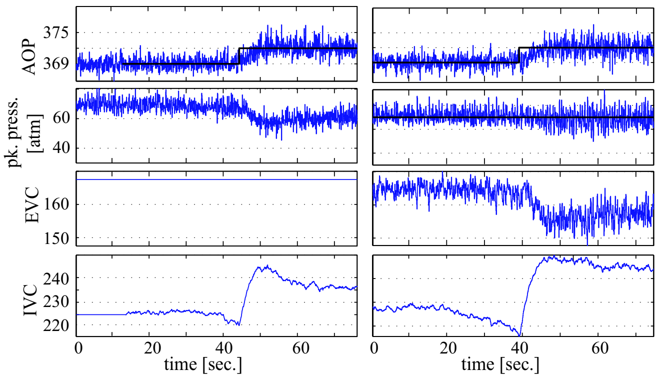

Fig. 6. Comparison of system response with combustion timing control only (a) and both combustion timing and peak pressure control simultaneously (b)

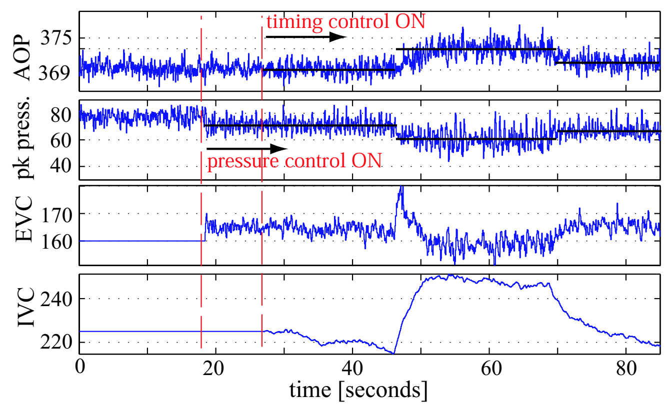

Fig. 7. Simultaneous control of both peak pressure and combustion timing

regulate work output and combustion timing. This section outlines a simple model of work output for the HCCI process and shows how work output control is a simple extension of peak pressure control in the case where ECR and combustion timing are slowly varying parameters.

In a piston engine, the work output is due to boundary work, $W = \int PdV$, which can be split into contributions before combustion (BC) and after combustion (AC) (maintaining the assumption of a constant volume combustion process):

$$W = \int PdV = \int_{V_{IVC}}^{V_{comb}} P_{BC}dV + \int_{V_{comb}}^{V_{EVO}} P_{AC}dV \quad (21)$$

Assuming isentropic compression and expansion processes, the pressure before and after the combustion event can be modeled as:

$$P_{BC}(\theta) = \frac{P_{atm}V_{IVC}^{\gamma}}{V(\theta)^{\gamma}} \qquad P_{AC}(\theta) = \frac{P_{pk}V_{comb}^{\gamma}}{V(\theta)^{\gamma}} \quad (22)$$

Substituting these expressions into Equation 21, and evaluating yields the following model of work output:

$$W = \left(\begin{array}{c} P_{atm}V_{IVC}^{\gamma}\left(V_{comb}^{1-\gamma} - V_{IVC}^{1-\gamma}\right) \\ + \; P_{pk}V_{comb}^{\gamma}\left(V_{EVO}^{1-\gamma} - V_{comb}^{1-\gamma}\right) \end{array} \right)(1-\gamma)^{-1} \quad (23)$$

As expected the work output depends on the system states (peak in-cylinder pressure and combustion timing) and control input, IVC. Again, combustion timing and ECR (via IVC) are considered slowly varying parameters in Equation 23, as any timing control via IVC modulation is done at an intentionally slower time scale than the work output control. Then, by inspection of Equation 23, the work output is strongly dependent on peak in-cylinder pressure in a linear fashion. With this linear dependence on peak pressure, the work output control problem is simply an extension of peak pressure control. Figure 8 shows successful experimental results of direct work output control utilizing the same H_2 control framework outlined in Section III-B. Desired step responses are achieved within about 5 engine cycles. In addition, both positive and negative load transients (via a sine function) are tracked. In each case the combustion timing remains nearly fixed, with a modest amount of additional cyclic dispersion.

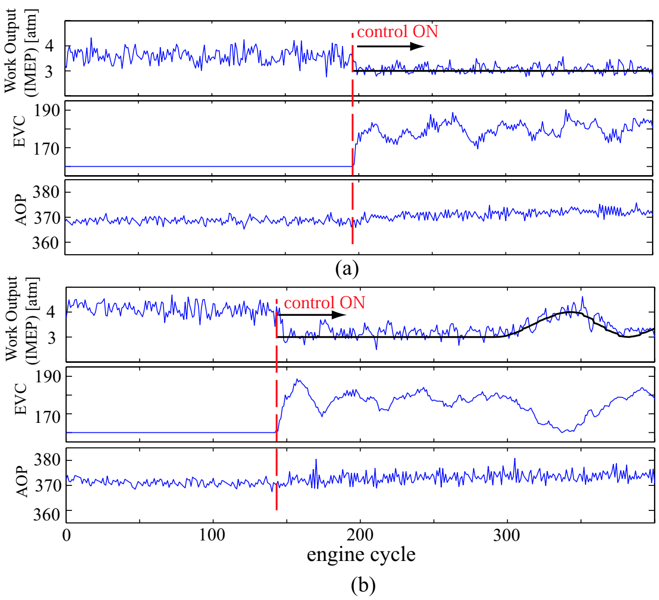

Fig. 8. Direct control of work output, (a) - step response, (b) - step followed by a sine

VI. Conclusion

HCCI is a promising combustion methodology for future engines. However cycle-to-cycle dynamics and the lack of a direct combustion trigger complicate control of the process. These challenges are addressed in this paper with a strategy for the simultaneous control of in-cylinder pressure and phasing. The paper also outlines a technique for extending this strategy to direct work output control, which may be more desirable in practice. The scheme approximately decouples the cycle-to-cycle dynamics of combustion timing and peak in-cylinder pressure (or work output) by controlling them on separate time scales with different control inputs - inducted composition and ECR, respectively. The cycle-to-cycle control is based on a physics-based H_2 control framework. Timing controller gains are selected via pole placement to achieve a response time that is slightly slower than the pressure controller. Although in-cylinder pressure depends on the combustion timing and ECR, a benefit of slowly varying them is that their influence on peak pressure is compensated for by the cycle-to-cycle peak pressure controller. Experimental results show that this simple framework effectively controls peak pressure or work output on cycle-to-cycle basis while desired combustion timing is achieved over a slightly slower time scale.

VII. Acknowledgments

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