Control of HCCI Engines through Variable Valve Actuation

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Comparison of HCCI, SI, and Diesel engine cycles

A look at variable valve actuation (VVA)

Motivation of HCCI, SI, and Diesel engine cycles

Proposed model

Model comparison with experiment

Control approach

Control simulation

Conclusion

Future work
A comparison of 4-stroke Internal Combustion (IC) Engine Strategies

Atypical fourstroke SI or Diesel engine cycle consists of:

1. Induction
2. Compression or reactant charge
3. Combustion near top dead center volume
4. Expansion
5. Exhaust or combustion products

Diesel - Initiation via in-cylinder injection of fuel near top dead center

SI - Initiation via in-cylinder spark

Combustion initiation occurs due to differing mechanisms:

For SI and Diesel:

1. Induction of reactant charge
2. Compression or reactant charge
3. Combustion near top dead center volume

A typical four stroke SI or Diesel engine cycle consists of:
A comparison of 4-stroke Internal Combustion (IC) Engine Strategies

A four stroke HCCI engine cycle w/ VVA consists of:

1. Induction of reactant charge AND exhaust
2. Compression or reactant charge
3. Combustion near top dead center volume
4. Expansion
5. Exhaust of combustion products

for HCCI with VVA:

NO DIRECT INITIATOR OF COMBUSTION
HCCI with VVA

So what’s a key difference about HCCI?

So VVA does affect combustion initiation, BUT it is indirect, unlike the case for spark in SI and fuel injection in Diesel.

HCCI achieved via VVA essentially allows modification to these three parameters:

- increase initial reactant temperature
- adjust the amount of mixing residence time
- modify initial reactant concentration
- to:

HCCI is achieved via VVA used.

Depending on the amount and the temperature of re-induced product, VVA is used.

 combustions dominated by chemical kinetics, which depends on:

- ensuring combustion occurs with acceptable timing (or at all) is more complicated

So what’s a key difference about HCCI?

- combustions dominated by chemical kinetics, which depends on:
VVA Valve Timings

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Motivation for Modeling and Control

Challenge: Combustion phasing and load tracking

No direct combustion initiator (like spark for SI or fuel injection for Diesel)

Combustion phasing dominated by chemical kinetics, which depends on:
- Initial temperature of reactants
- Amount of inducted reactants
- Combustion phasing

Engine cycle work output, $\mathcal{W}_P$, depends on (to first order):
- Combustion phasing
- Amount of inducted reactants

Residence time for mixing

$\theta P \left[ \frac{\pi O}{2} \times \frac{\pi H C}{3} \right] \left( \frac{\pi R}{\pi F} \right)^{\frac{\pi E}{\pi D}} dx \in \left[ \frac{\pi R}{\pi \theta} \right] \left( \frac{\pi O_{\infty}}{\pi T_{\infty}} \right) \int = \text{thres} \Leftrightarrow$

These parameters are controllable with the VVA system

CONTROL STRATEGY (maintain desired phasing as load is varied):

- Hold phasing constant
- Vary amount of inducted reactant to get desired work output
- Use peak pressure as a proxy for work output

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Modeling approach: Schematic

Model input: molar ratio of re-induced products to reactants, \( \alpha \).

Model output: peak in-cylinder pressure (proxy for work output at constant phasing).

Re-induced product temp. directly related to previous cycle’s exhaust temp.

Exhaust: Isentropic exhaust to atmospheric pressure condition in exhaust manifold

Exhaust: Isentropic exhaust to point where exhaust valve opens

Expansion: Constant volume combustion to major products w/ heat transfer

Compression: Isentropic to the point where combustion initiates

Induction: Constant pressure, adiabatic induction process

Assumptions:

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mixture at end of induction

The first law of thermodynamics is applied to determine the thermodynamic state of the reactant and product at end of induction. Important assumptions include:

- Constant atmospheric pressure during induction
- No significant heat transfer during induction
- Reactant and product are well mixed at end of induction
- Assumption that the concentration of reactant decreases the concentration of reactant
- Heats the induced reactant

Re-induction of hot products from the previous cycle:
The relations for an isentropic process can be used to determine the thermodynamic state:

Assumption: compression is isentropic

Compression to pre-combustion point:

Compression process
Combustion of reactants:
results in products at elevated temperature and pressure

Assumptions:

- HCCI combustion is fast \( \Rightarrow \) constant volume combustion assumed
- major products assumed \((CO_2, H_2O, N_2, \text{and } O_2)\)
- heat transfer modeled as a percentage of available reactant chemical energy

The first law of thermodynamics is applied to determine thermodynamic state of mixture at end of combustion
The relations for an isentropic process can be used to determine the thermodynamic state. Expansion/exhaust is isentropic to atmospheric pressure in the exhaust manifold. Assumption: decreases product temperature. Expansion to EVO and then through exhaust valve:

\[
\frac{\gamma}{1-\frac{\gamma}{k}} \left( \frac{\dot{V}}{\dot{m}_p} \right) = \left( \gamma \right) \frac{\dot{V}}{L} \quad \text{for exhaust:}
\]

\[
\frac{\gamma}{\frac{\gamma}{k}} \left( \frac{\dot{V} \Lambda}{\dot{e} \Lambda} \right) = \left( \gamma \right) \frac{\dot{V}}{d} \quad \frac{\gamma}{\frac{\gamma}{k}} \left( \frac{\dot{V} \Lambda}{\dot{e} \Lambda} \right) = \left( \gamma \right) \frac{\dot{V}}{L} \quad \text{for expansion:}
\]
Although simple, this expression matches experimental observations well:

\[ \left( 1 - \gamma \right) \gamma L \chi = \gamma_{in} \gamma_{ex} L \]

a simple model for the exhaust gas coupling is:

causes cycle-to-cycle coupling

temp. from previous cycle

the temperature of the re-inducted products is directly related to the exhaust gas

A portion of the exhaust gas from one cycle is re-inducted during the next
Assembling the expressions for the in-cylinder temperature following each of the stages of HCCI combustion, yields:

\[
T_k = C_4 + \frac{1}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\left[1 + \frac{C_1 \chi_k (1 + C_2 \chi_k \alpha_k \beta_k^{1/K})}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\right] + C_3 \chi_k \alpha_k \beta_k^{1/K} \left[\frac{\Delta \chi_k \alpha_k \beta_k^{1/K}}{\chi_k \alpha_k \beta_k^{1/K}}\right] + \frac{1}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\left[1 + \frac{C_1 \chi_k (1 + C_2 \chi_k \alpha_k \beta_k^{1/K})}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\right]
\]

Cycle-to-cycle dynamics are evident:

\[
P_k = C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}
\]

Since the desired model output is a measurable quantity related to work output, some fairly extensive algebraic manipulation yields:

Cycle-to-cycle dynamics are evident:

\[
T_k = \frac{C_2 \chi_k^{1/K} \alpha_k \beta_k^{1/K}}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}} + \frac{1}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\left[1 + \frac{C_1 \chi_k (1 + C_2 \chi_k \alpha_k \beta_k^{1/K})}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\right] + \frac{1}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\left[1 + \frac{C_1 \chi_k (1 + C_2 \chi_k \alpha_k \beta_k^{1/K})}{C_1 + C_2 \chi_k \alpha_k \beta_k^{1/K}}\right]
\]

of HCCI combustion, yields:

Assembling the expressions for the in-cylinder temperature following each of the stages of the model model...
Model comparison with experiment

- Model is a good candidate for model-based controller syntheses
- Motivation for feedback control

But model is not exact

Good correlation between experiment and model:

<table>
<thead>
<tr>
<th>Case</th>
<th>IVO/EVC</th>
<th>$P_{\text{max, model}}$ [atm]</th>
<th>$P_{\text{max, exp}}$ [atm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>25/165</td>
<td>57.4</td>
<td>44.4</td>
</tr>
<tr>
<td>2</td>
<td>45/185</td>
<td>52.4</td>
<td>45.5</td>
</tr>
<tr>
<td>3</td>
<td>65/205</td>
<td>59.5</td>
<td>57.4</td>
</tr>
</tbody>
</table>

Comparison between the experiment and model predicted values of peak pressure can

"fold gas temperature and volumetric flow through the intake

Experimental estimates of $a$ can be calculated given measurements of exhaust manif-

Model comparison with experiment
\[
\left( \frac{d}{d - \gamma d} \right) = \gamma \theta
\]

where \( \theta \) is the normalized difference between desired and actual pressure as:

\[
\frac{\gamma \ddot{z} + \gamma \dot{z} \frac{d}{d - \gamma d}}{1 - \gamma \ddot{d}} = \frac{(z)\dot{\nu}}{(z)\theta}
\]

Applying these, and neglecting cross terms of fluctuation yields the transfer function:

\[
\gamma \ddot{d} + \gamma \dot{d} \approx \gamma \dot{d}
\]
\[
\gamma \dot{\nu} + \nu = \gamma \nu
\]

Using the linearizations:

\[
\frac{1 - \gamma \nu}{\gamma} \frac{C_1 \gamma \dot{\nu} + (1 - \gamma \nu) \gamma \nu \gamma}{\gamma} \frac{C_1 \gamma \nu + (1 - \gamma \nu) \gamma \nu \gamma}{\gamma} \frac{C_1 \gamma \nu + (1 - \gamma \nu) \gamma \nu \gamma}{\gamma} \frac{C_1 \gamma \nu + (1 - \gamma \nu) \gamma \nu \gamma}{\gamma} = \gamma \dot{d}
\]

Recall the model for peak pressure:
A map from desired composition, \( a \), to required valve timing (IVO/EVC) is needed:

\[
(\forall) f + O \Delta I = 9 + O \Delta I = (\forall) O V\Delta I
\]

Experimental and simulation show that this scheme yields fairly consistent combustion phasing so choice of map \( \text{gives} \) open loop control of combustion phasing with an LQR controller:

Let: \( P = p \), max, desired
This control scheme was simulated in closed loop with a more complete model of HCCI developed in ongoing work.

More complete model:

HCCI has steady state compressible flow relations used to model the mass flow through the valves. Based on an open-system first-law analysis of both in-cylinder and exhaust gases, it has 10 states.

The result of a series of step changes in desired peak pressure yields:
Desired control objectives met:

- Tracking of desired peak pressure obtained
- Steady combustion phasing obtained

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As expected, the work output correlates well with peak pressure at constant phasing.
Experimental Implementation

Peak in-cylinder pressure during HCCI (atm) during engine cycles

<table>
<thead>
<tr>
<th>Time [seconds]</th>
<th>Engine cycles</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>150</td>
</tr>
<tr>
<td>40</td>
<td>300</td>
</tr>
<tr>
<td>90</td>
<td>450</td>
</tr>
<tr>
<td>140</td>
<td>600</td>
</tr>
<tr>
<td>190</td>
<td>750</td>
</tr>
</tbody>
</table>

Improvements made with control

<table>
<thead>
<tr>
<th>6.45</th>
<th>1.11</th>
<th>35.96</th>
<th>36</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.37</td>
<td>5.95</td>
<td>39.75</td>
<td>40</td>
</tr>
<tr>
<td>16.8</td>
<td>2.71</td>
<td>50.93</td>
<td>40</td>
</tr>
</tbody>
</table>

No controller OFF controller ON

Note presence of cycle-to-cycle variation in peak pressure (cyclic dispersion) attributable to variation in gas exchange/mixing process increases likelihood of misfire and decreases efficiency. A low order model does not predict it, but controller appears to reduce it. An enhanced understanding of dispersion should be helpful in determining stability. Improvements made with control.

Characteristics of process

- Improved peak pressure
- Decreased cyclic dispersion

Presence of cycle-to-cycle variation in peak pressure (cyclic dispersion) attributable to variation in gas exchange/mixing process increases likelihood of misfire and decreases efficiency. A low order model does not predict it, but controller appears to reduce it. An enhanced understanding of dispersion should be helpful in determining stability.
Conclusion

HCCI achieved via VVA results in low emissions and improved efficiency. Cycletocycletodynamics and chemical kinetics make VVA-induced HCCI a complex process with no direct combustion initiator. Cycle to cycle dynamics and chemical kinetics make VVA-induced HCCI achievable via VVA results in low emissions and improved efficiency.

Preliminary results of experimental closed-loop control are quite good. Research engine results of simulation are very promising, prompting implementation on a research engine. Preliminary results of experimental closed-loop control are quite good. Preliminary results of experimental closed-loop control are quite good. Preliminary results of experimental closed-loop control are quite good. Preliminary results of experimental closed-loop control are quite good. Preliminary results of experimental closed-loop control are quite good.

Regulating the peak in-cylinder pressure (a proxy for work output at constant phasing) is a closed loop controller to stabilize combustion phasing and achieve desired work output. By: holding phasing relatively constant.

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Recap: Controller Synthesis Strategy

A low-order, physics-based model of HCCI combustion through Variable Valve Actuation (VVA) has been formulated. Its accuracy and simplicity make it amenable to controller synthesis. A controller synthesized was tested in simulation on a more accurate, higher-order model. Results are promising.

Control technique is implemented on actual system implementation successful.

Controller synthesized through Variable Valve Actuation (VVA)
Implement control strategy on single cylinder research engine in Stanford engine lab (DONE!!)

Future Work

Control of HCCI engine through Variable Valve Actuation

- This could pursue model-based (integrated Arrhenius rate) phasing control to achieve
- May be necessary during mode switching

Control of variable combustion phasing

- Care must be taken during transitions, due to cycle-to-cycle coupling
- Could be used for cold start (i.e. use SI to warm engine)
- Likely required since HCCI w/ VVA is a low-load strategy (i.e. use SI at high load)

Mode switching

- Likely utilizing much improved valve controller (J.P., Jasim and Aleks)
- Done utilizing much improved valve controller (J.P., Jasim and Aleks)

Future Work
Future Work (cont.)

Characterization of cyclic dispersion should increase understanding of conditions which lead to misfire and robust control strategy for conventional control near limits identified via dispersion characteristics.

Extending model and control methodology to varying engine speed and multi-cylinder operation for different fuels.

HCCI with other fuels: Hydrogen, Gasoline, and diesel.

Identified via dispersion characteristics.

A robust control strategy is thus prepared with which back off on conventional control near limits should increase understanding of conditions which lead to misfire.