

Understanding the Dynamic Evolution of Cyclic Variability at the Operating Limits of HCCI Engines with Negative Valve Overlap

Erik Hellström, Anna Stefanopoulou, Jiri Vavra, Aristotelis Babajimopoulos and Dennis Assanis
University of Michigan

Li Jiang and Hakan Yilmaz
Robert Bosch LLC

ABSTRACT

An experimental study is performed for homogeneous charge compression ignition (HCCI) combustion focusing on late phasing conditions with high cyclic variability (CV) approaching misfire. High CV limits the feasible operating range and the objective is to understand and quantify the dominating effects of the CV in order to enable controls for widening the operating range of HCCI. A combustion analysis method is developed for explaining the dynamic coupling in sequences of combustion cycles where important variables are residual gas temperature, combustion efficiency, heat release during re-compression, and unburned fuel mass. The results show that the unburned fuel mass carries over to the re-compression and to the next cycle creating a coupling between cycles, in addition to the well known temperature coupling, that is essential for understanding and predicting the HCCI behavior at lean conditions with high CV.

CITATION: Hellström, E., Stefanopoulou, A., Vavra, J., Babajimopoulos, A. et al., "Understanding the Dynamic Evolution of Cyclic Variability at the Operating Limits of HCCI Engines with Negative Valve Overlap," *SAE Int. J. Engines* 5(3):2012, doi:10.4271/2012-01-1106.

INTRODUCTION

It was early noted that homogeneous charge compression ignition (HCCI) combustion can have low cyclic variability (CV) compared to spark ignited (SI) combustion [1]. However, at certain operating conditions, even if all inputs are held constant, the combustion phasing can alternate between early and late timing in an oscillatory manner or run away towards increasingly advanced conditions. The process can exhibit significant oscillations at late phasing [2, 3] and runaway-knock phenomena at high loads [4]. This behavior points to the fact that there is considerable nonlinear feedback between cycles that can stabilize or destabilize the process. The coupling has been attributed to the fast variations from cycle to cycle in combustion inefficiency and residual gas temperature [5, 6] and the slower variation of the cylinder wall temperature [4, 7, 8].

HCCI can be achieved with variable valve timing and the focus here is on the negative valve overlap (nvo) strategy. This re-compression strategy is also known as controlled autoignition (CAI) and means that autoignition is induced by raising the charge temperature of the next cycle by trapping

and compressing residual gases from earlier cycles through an early closing of the exhaust valve [9].

To investigate the nature of CV, we perform a set of experiments and then analyze the data in order to arrive at an interpretation of the dynamical behavior of the process. To enable the analysis, standard combustion analysis methods are extended with a novel method for determining the unburned fuel mass and combustion efficiency on cycle basis. The main contribution in our work is this thermodynamic analysis of the experimental data which explains and quantifies the dominating effects governing the process. Specifically, the analysis shows that lean HCCI combustion with high CV is mainly the result of the recycled thermal energy in the residual gas and the unburned fuel passing over to the re-compression and to the next cycle.

Experiments with the re-compression strategy at conditions with noticeable CV were performed and analyzed in [5, 10]. Linear correlation coefficients were also computed to study the relationships between subsequent cycles. Control of valve timings and injection timings for reducing the CV was developed in [11, 12] based on model linearizations. Our analysis corroborates the trends found in [5, 10] and used in



[11, 12] but also show that, for high CV, the relationships are more complicated than can be captured with a linear analysis. Specifically, there are sequences of cycles, characterized by combustion phasing and heat release, that occurs with increasing probability when the CV increases.

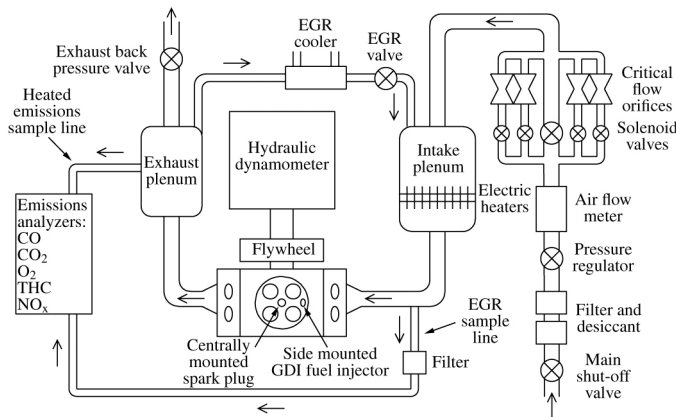


Figure 1. Schematic of the FFVA engine experimental setup.

The focus here is on understanding lean HCCI combustion operated with nvo. Wagner et al. [13, 14] report experimental observations and analysis of transitions, using variable valve timing, between SI and spark-assisted HCCI combustion with stoichiometric mixtures where the process transitions through regions with high CV. These conditions give qualitatively different dynamical behavior than the lean case without spark assist. HCCI engines operating with intake heating and a low amount of residuals also show different behavior, as shown in [15].

Experimental work investigating the effect of different mixtures of primary reference fuels on CV in HCCI are found in [16, 17, 18]. An analysis of the stability of HCCI with large amounts of residual gas was done in [19] where, by studying the thermal dynamics of the charge, unstable behavior with limit cycles at late phasing and runaway knock at early phasing were predicted. Similar analysis is also shown in [20]. Our work show that, for higher CV, the effect of unburned fuel creates an additional coupling that must be taken into account. The use of injecting fuel during the nvo period for control was mentioned in [9] and experimentally investigated in [21, 22] and used in [23, 24] to extend the low load limit of HCCI. In the present work, the injection parameters are constant for each operating point and the effect of the unburned fuel, recycled from incomplete combustion in previous cycles, is investigated.

The objective here is to identify, understand, and quantify the most influential properties for the CV to enable control-oriented modeling. A simulation model for studying misfire in HCCI using chemical kinetics with 31 species and air flow model was developed in [25] and reduced to a eight state model in [26]. Our aim is to find the dominant factors and a minimum set of variables for describing CV.

The paper is organized as follows. The experimental setup is described first and the data analysis method used is outlined. After that, the results from the methodology are interpreted in order to explain the dynamic behavior of the combustion process. Finally, the conclusions from the work are summarized.

EXPERIMENTAL SETUP

Experiments were performed at the Automotive Laboratory at University of Michigan on a single-cylinder gasoline direct-injected engine with a Ricardo Hydra crankcase. The engine is equipped with an electro-hydraulic fully-flexible valve actuation (FFVA) system from Sturman Industries, which allows lift, timing, and duration of each valve event to be controlled independently. Fuel is delivered via a gasoline injector mounted between the two intake valves and aimed into the piston bowl. The engine geometry is listed in Table 1, while a detailed schematic of the engine setup is seen in Fig. 1. More details for the experimental setup can be found in [27]. The usable range of

Table 1. FFVA engine geometry.

Component	Characteristic
Cylinders	1
Displaced volume	550 cm ³
Stroke	94.6 mm
Bore	86.0 mm
Connecting rod	152.2 mm
Compression ratio	12.5:1
Number of valves	4
Piston shape	Shallow bowl
Head design	Pent-roof
Fueling method	Direct injection

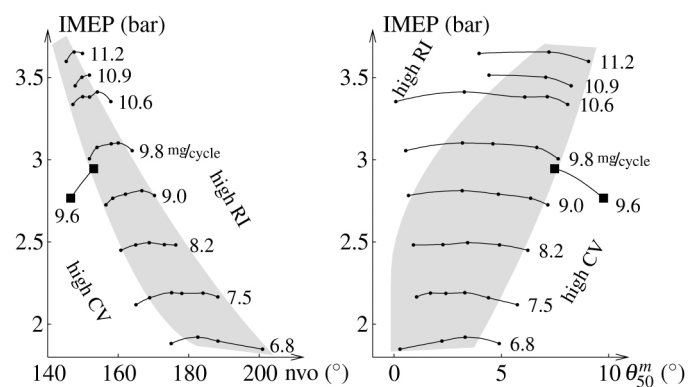


Figure 2. Usable HCCI operating range (shaded region) at 2000 rpm as determined in [27]. The data points analyzed in the present work are marked with squares.

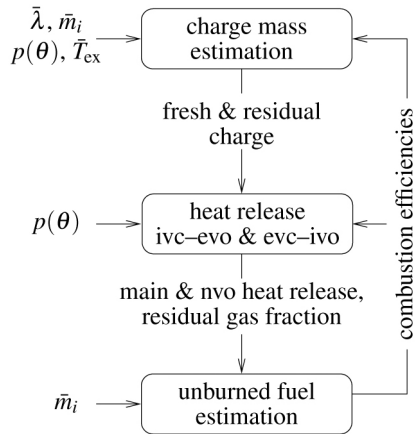


Figure 3. Block diagram for the iterative estimation routine using measurements of λ , m_i , $p(\theta)$, and T_{ex} .

HCCI for the engine was determined by Manofsky et al. [27] and is shown by the shaded region in Fig. 2. Experiments were performed at various fueling rates, indicated by lines in Fig. 2, and nvo was varied to control combustion phasing. Fuel was delivered with a single injection, starting at a crank angle 330° before top dead center combustion (TDCm). The fuel used was research-grade gasoline (90.5 RON, 82.6 MON). The usable HCCI operating range was limited by high CV (measured by the coefficient of variation, CoV, of IMEP) at late phasing and high ringing intensity (RI) at early phasing. In [27], these limits were chosen to a CoV of IMEP of 5% and a RI of 4MW/m². Note that the focus in the current work is on individual engine cycles and that these values are averages over many cycles only used to depict the feasible operating range. As indicated in Fig. 2, at low loads, a large range of nvo could be spanned before the limits of combustion were reached. However, at higher loads, slight changes in nvo had a greater effect on combustion. As fueling was increased, the limits of ringing intensity and CV converged and the operating range became narrower.

The analysis in the present work is focused on the two square points in Fig. 2, at fuel injection of 9.6mg/cycle, that transition from acceptable CV to higher CV. These conditions are a typical medium load operating point corresponding to net IMEP is approximately 2.8 bar with $\lambda = 1.7$. To fully observe and analyze the patterns of cycle-to-cycle coupling, in-cylinder pressure was sampled at each crank angle degree for 3000 consecutive cycles. Nominal operating conditions for the experiments were 2000 rpm, controlled by a hydraulic dynamometer, and a coolant temperature that was controlled at 90°C . Analysis of points at lower and higher load were also done revealing that the same characteristics appear at these loads with the addition of runaway knock phenomena at higher loads.

DATA ANALYSIS

To analyze the experimental data an iterative estimation routine, depicted in Fig. 3, based on conservation of mass and energy is used. A novel method for estimating the mass of unburned fuel on a cycle basis is developed and combined with standard combustion analysis methods. The goal is to calculate the thermodynamic state, pressure and temperature, as well as variables important for explaining CV, such as unburned fuel and combustion efficiencies. The process is iterated until the values of combustion efficiencies converge at which point the governing relations of mass and energy conservation are satisfied.

The definitions of the engine cycle and important variables are given in Fig. 4 together with typical curves of measured pressure $p(\theta)$, calculated heat release $Q(\theta)$, and known valve lift. Note that the heat release is calculated for both the main combustion event, between

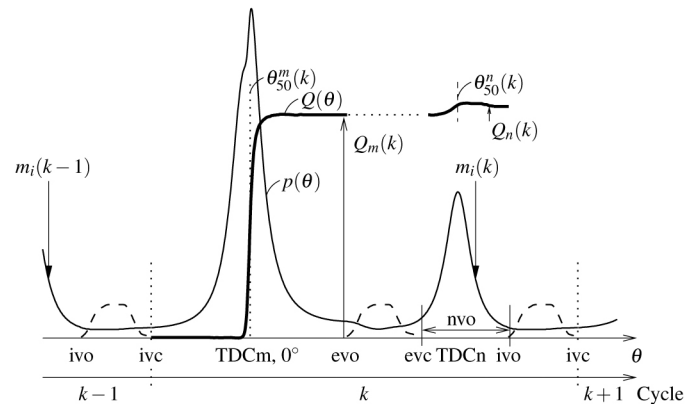


Figure 4. Definition of the engine cycle and important variables for typical curves of measured pressure $p(\theta)$, calculated heat release $Q(\theta)$, and known valve lift (dashed line).

intake valve closing (ivc) and exhaust valve opening (evo), and the nvo period, between exhaust valve closing (evc) and intake valve opening (ivo). Top dead center main combustion is 0° and a new cycle starts at ivc. The total heat release in cycle k during main combustion and the nvo period are denoted by $Q_m(k)$ and $Q_n(k)$ respectively. In cycle k , the 50% burn angle where half the Q_m has occurred during main combustion is denoted by $\theta_{50}^m(k)$ and the corresponding angle during the nvo period is denoted by $\theta_{50}^n(k)$.

In the remainder of this section, the method for estimating the mass of unburned fuel on a cycle basis is derived. The other steps shown in Fig. 3 are based on standard methods and are described in detail in the Appendix.

UNBURNED FUEL

In order to model the mass of unburned fuel on cycle basis, a difference equation is derived for the unburned fuel mass that depends on known quantities and an initial unburned fuel mass. It is observed that the equation converges to the correct value given any initial guess. The model is subsequently used to estimate the combustion efficiencies.

The main combustion consumes a fraction $\eta_m(k)$ of the fuel mass and after the exhaust phase a fraction $x_r(k)$ of the residual gases is trapped. During re-compression a fraction $\eta_n(k)$ is further consumed so that, with homogeneous residuals, the unburned fuel mass carried over to the next cycle $k + 1$ is

$$m_u(k+1) = x_r(k)(1 - \eta_m(k))(1 - \eta_n(k))m_f(k) \quad (1)$$

where the fuel mass $m_f(k)$ in the beginning of cycle k is

$$m_f(k) = m_i(k-1) + m_u(k) \quad (2)$$

with $m_i(k-1)$ being the injected fuel mass. The unknown combustion efficiencies in (1) are now related to the accumulated heat release. The gross heat released during main combustion Q_m is modeled by

$$Q_m(k) = m_f(k)\eta_m(k)q_{lhv} \quad (3)$$

with q_{lhv} as the lower heating value for the fuel. As explained above, the fraction $\eta_m(k)$ of the fuel mass is consumed during main combustion and then the fraction $x_r(k)$ of the residual gases is trapped. Thus, the gross heat released during re-compression Q_n is given by

$$Q_n(k) = m_f(k)(1 - \eta_m(k))x_r(k)\eta_n(k)q_{lhv}. \quad (4)$$

Combining Eqs. (1), 2, 3, (4) yields a difference equation for $m_u(k)$ that can be iterated given an initial condition $m_u(0)$. The average value is used here for the fuel injection, i.e., $m_i(k) = \bar{m}_i$, since constant fueling was used in the experiments. The initial condition $m_u(0)$ is unknown but it can be shown that the series is convergent since x_r is between zero and one. The initial condition can be guessed and the result will be unique, after a transient, whatever the guess. The convergence rate depends on the value of x_r and, in this work, the value is about 40% which means that the influence of the true initial condition is reduced by a factor of $0.4^8 < 1/1000$ after 8 cycles. The results are practically indistinguishable after this number of cycles for all cases starting from different initial conditions. This is demonstrated in Fig. 5 which shows

the first iterations, starting from a range of initial conditions, for two cases. The approach adopted here is to guess $m_u(0) = 0$ and discard the first cycles in the analysis. Finally, when the unburned fuel mass is computed, (3) and (4) give

$$\eta_m(k) = \frac{Q_m(k)/q_{lhv}}{m_f(k)} \quad (5a)$$

$$\eta_n(k) = \frac{Q_n(k)/q_{lhv}}{x_r(k)(m_f(k) - Q_m(k)/q_{lhv})} \quad (5b)$$

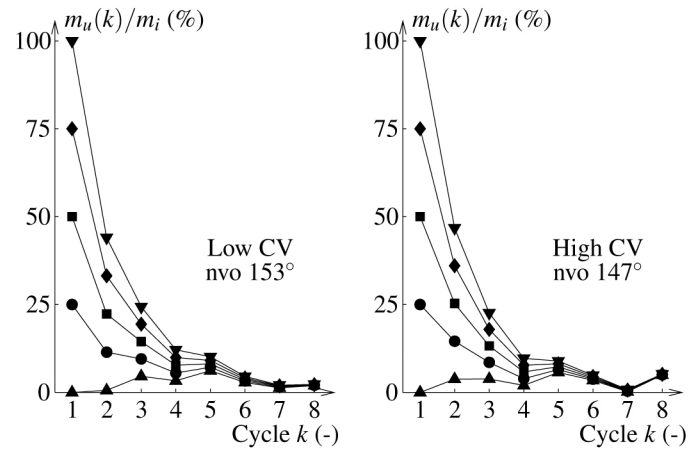


Figure 5. Convergence of the unburned fuel model is typically obtained in a few cycles. Shown here are the first cycles of one low-CV case (left) and one high-CV case (right).

as approximations for the efficiencies which depends on the total gross heat releases (Q_m , Q_n) and residual gas fraction x_r . The computation of these quantities are based on standard methods and are described in the Appendix.

The proposed estimation of unburned fuel, based on the model (1), includes possible heat release during re-compression, the term $Q_n(k)$, which is neglected in previous low-order models associated with higher CV in HCCI-SI transients [28]. Simulation models with higher fidelity, including the effect of unburned fuel, are used in [25, 26] to study the effect of misfire in HCCI. In contrast, the aim here is to estimate the unburned fuel mass from experimental pressure data.

RESULTS

Observations from the data processing described in the previous section, applied on the measurements, are interpreted here. Results are shown for different nvo settings transitioning from lower to higher values of CV. The observations show that lean HCCI combustion with high CV is the result of unburned fuel passing over to the re-

compression and to the next cycle which is supported by the following three important findings:

- There is a strong coupling and complementarity of the fuel burned during the main combustion and the re-compression events.
- Despite the complementarity, the available fuel does not necessarily burn during one cycle meaning that unburned fuel may carry over to the next cycle.
- Following a partial misfire, injection after TDC of re-compression does not necessarily cause combustion.

To explain and quantify these findings we highlight results from experiments with two nvo settings.

All the measured cylinder pressure traces for the two cases are shown in the upper half of Fig. 6. The lower part of the figure shows the charge temperatures computed from the heat release analysis. When reducing the overlap from 153° to 147° the average residual gas fraction reduces from 46% to 44% (see Fig. 17 in the Appendix). The average combustion phasing θ_{50}^m is pushed from 7.4° to 9.7° and the standard deviation is increased from 0.9° to 2.4°. The CoV of IMEP raises from a moderate 2.4% to a value of 13.3% which is typically considered above the acceptable limit. A few consecutive cycles are marked in Fig. 6, with the numbers 6 to 9, for the high CV case with nvo of 147°. These show, in the upper part of the figure, the peak pressure locations and, in the lower half, they identify the corresponding temperature traces. The same cycles are also shown in Fig. 7, 9, and in more detail in 13, 14, 15.

The spread of the attained pressures and temperatures generally increases when reducing nvo as shown in Fig. 6. Both lower and higher peak pressures and pressure rise rates are seen. The temperature during the re-compression does, however, mostly increase which suggests that if the temperature at evo is low, there is unburned fuel that will result in heat release during nvo. This connection will be made more clear later when discussing Fig. 9. The injection, starting at 30° after TDCn as marked in Fig. 6, has no visible effect on neither temperature nor pressure on the contrary to the simulation results in [25, 26] where, with a similar injection timing, combustion of all the available fuel is initiated.

The gross heat release traces for the two nvo settings are shown in Fig. 7. The traces for the case with low CV, in black, are tightly gathered and appear as one thick line whereas the larger variability is evident for the case with smaller nvo, in gray. The cycles with a low total heat release during main combustion, indicating a poor burn, have noticeable heat release occurring in the subsequent re-compression phase. The burn rates during re-compression for these cycles are rather slow, so a low efficiency is expected. Also, a number of cycles in the high variability case have higher total heat release than the ones observed in the low variability case. This indicates that unburned fuel carry over to the nvo period and to the next combustion cycle. Looking

at cycles marked 6-9 in Fig. 6 and 7, reveals the same features. The sequence starts and ends with cycles with an evolution that looks similar to cycles in the low CV case. In between these well behaved cycles are two extreme cycles showing the effect of unburned fuel. Cycle 7 burns poorly and is followed by heat release and high temperatures in the nvo period. The eighth cycle has high peak pressure and high heat release during main combustion. Finally, there is no observable heat release in Fig. 7 associated with the injection. The results indicate that endothermic processes, due to vaporization and possibly pyrolysis, take place rather than combustion as suggested by the simulation results in [25, 26].

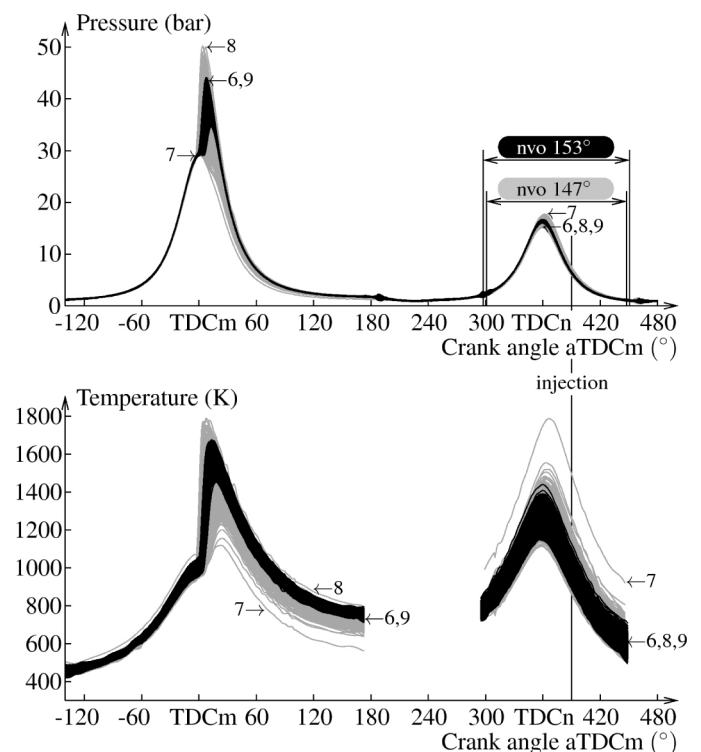


Figure 6. Measured pressure and calculated temperature for two different nvo settings with high and low CV. The numbered cycles are also marked in Fig. 7, 9, 13, 14, 15.

The estimates of the residual gas fraction x_r were found to be close to normally distributed (see the normal probability plot in Fig. 18 in the Appendix) indicating that the variability in x_r is random and without deterministic structure. The result in Fig. 8 further indicates that the variability in x_r has no clear influence on the heat release during main combustion Q_m . For the high CV case the cloud of points is moved to the left, since the mean x_r is reduced, and stretched vertically, due to the higher variability in Q_m , but no trends appear.