Controlling Cyclic Combustion Variations in Lean-Fueled Spark-Ignition Engines

L. I. Davis Jr., L. A. Feldkamp, J. W. Hoard, F. Yuan, and F. T. Connolly
Ford Research Laboratory, Dearborn, Michigan 48121

C. S. Daw and J. B. Green Jr.
Oak Ridge National Laboratory, Oak Ridge, Tennessee 37831-8088

Copyright © 2001 Society of Automotive Engineers, Inc.

ABSTRACT

This paper describes the reduction of cyclic combustion variations in spark-ignited engines, especially under idle conditions in which the air-fuel mixture is lean of stoichiometry. Under such conditions, the combination of residual cylinder gas and parametric variations (such as variations in fuel preparation) gives rise to significant combustion instabilities that may lead to customer-perceived engine roughness and transient emissions spikes. Such combustion instabilities may preclude operation at air-fuel ratios that would otherwise be advantageous for fuel economy and emissions. This approach exploits the recognition that a component of the observed combustion instability results from a noise-driven, nonlinear deterministic mechanism that can be actively stabilized by small feedback control actions which result in little if any additional use of fuel. Application of this approach on a test vehicle using crankshaft acceleration as a measure of torque and fuel pulse width modification as a control shows as much as 30% reduction in rms variation near the lean limit.

INTRODUCTION

The existence of cyclic combustion variations in spark-ignited, internal combustion engines has long been recognized. Such variations can be particularly severe for lean air-fuel mixtures (i.e., when the ratio of air to fuel is greater than that required by chemical stoichiometry). The analysis of these variations is made difficult by the existence of several contributing phenomena that can act separately or in concert. One of the most important factors is variation in the delivery of air and fuel into the cylinder. The fuel injection process has attracted great attention for many years, deservedly so in view of the spray pattern variations which have been observed by modern imaging techniques. The effect of variations in either fuel mass or fuel distribution is exacerbated under lean conditions, when the total mass of fuel is relatively smaller. Grunefeld et al. (SAE 941880) discuss fluid dynamic effects during engine intake and exhaust strokes and conclude that they are dominant contributors to cyclic variation. The importance of the residual gas, both content and amount, has also been recognized and is generally regarded as the cause of the frequently observed alternating pattern of high and low work output cycles, although other mechanisms have been proposed (Hancock et al., SAE 860321). Stevens, Shayler, and Ma (ImechE, C448/058/92, and C465/021/93, SAE 950687) considered cyclic variations from the standpoint of understanding the mechanism well enough to effect a reduction in the variation by imposing control. They found that significant correlation exists between consecutive firings of a particular cylinder, and that various relevant measurable quantities, such as indicated mean effective pressure, are subject to reasonable prediction one cycle in advance. These authors also considered, in theory at least, various means of imposing control such as through changes of spark timing and fuel delivery.

Because lean combustion is extremely sensitive to small changes in local conditions (i.e., it is highly nonlinear), it is a candidate for exhibiting the complex behaviors associated with nonlinear systems including bifurcations and deterministic chaos, or just chaos for short. If chaotic behavior takes place in a system with many important state variables (e.g., more than ten), it is termed high-dimensional chaos. While high-dimensional chaos is in principle still deterministic, it is usually so complex that as a practical matter (at least with current understanding), it can only be treated with methods applicable to stochastic (random) systems. Hence to be of present practical importance (e.g., for real-time control), it is necessary for the identified chaotic behavior to be low-dimensional, (e.g., have a number of important state variables that is less than ten).

The possibility of chaotic behavior in spark-ignited engines was suggested at least as far back as 1984 (Kantor, Science vol. 224) and has been a continuing source of investigation (Daily, Combust. Sci. and Tech.,...
vol. 57, 1988; Foakes and Pollard, Combust. Sci. and Tech., vol. 90, 1993). Chew et al. (SAE 942486) claim to have identified chaotic behavior in a production internal combustion engine, though they are not rigorous in distinguishing their observations from stochastic behavior. Further, they do not discuss implications for practical application of their findings. Finney, Nguyen, and Daw (Japanese Combustion Symposium, Sendai, 1994) make use of chaotic time series analysis to analyze data from a one-cylinder engine, concluding that the variations observed are not consistent with purely random behavior and hence that short-term predictability might be possible.

MODELING CYCLIC VARIABILITY

In an attempt to explain how residual gas could lead to low-dimensional nonlinear deterministic coupling between cycles, Daw et al. (SAE 962086) developed a simple model for variations of fuel and air in an engine cylinder:

\[
m(i+1) = m(i) \times (1 - CE) \times F + (1 - F) \times [MF + dMF(i)] \tag{1}
\]

\[
a(i+1) = [a(i) - R \times CE \times m(i)] \times F + (1 - F) \times AF \tag{2}
\]

where the main variables are defined as:

\[
m(i) = \text{mass of fuel before ith burn} \\
a(i) = \text{mass of air before ith burn} \\
dMF(i) = \text{small change in mass of fresh fuel per cycle, dictated by control;}
\]

constant, or slowly varying variables are:

\[
MF = \text{mass of fresh fuel per cycle} \\
AF = \text{mass of fresh air fed per cycle;}
\]

parameters whose values are indicated by engine or fuel characteristics are:

\[
F = \text{fraction cylinder gas remaining} \\
R = \text{stoichiometric air-fuel ratio, 14.6;}
\]

and a key variable which may, for example, be a function of air-fuel ratio:

\[
CE = \text{combustion efficiency of ith burn.}
\]

With no control imposed ( \( dMF(i) = 0 \) ) and no stochastic perturbation of the parameters, this model exhibits unstable behavior in the form of period-doubling bifurcations and chaotic behavior at lean conditions, depending on particular values of the variables and parameters and the functional form and characteristics of the combustion efficiency curve. Given simple functional forms for CE, the equations may be solved for fixed points in the variables (i.e., values where the behavior is at least marginally stable), and a control equation may be developed that will force the system to a fixed point and keep it there. For practical application, the functional form of CE is not known a priori and must be developed from heuristic arguments and experience. Nevertheless, it is expected from combustion physics that the functional form of CE should include the effect of a strong nonlinear dependence of flame speed on the in-cylinder fuel and air content at the time of ignition.

The period-2 bifurcation sequence exhibited by the Daw et al. model results from a dynamic instability driven by the nonlinear coupling between present and future cycles. As the air-fuel ratio increases, the instability leads initially to a period-2 bifurcation, which produces an alternating high-low heat release pattern. Further increases in air-fuel ratio lead to a cascade of additional bifurcations to period-4, period-8, etc., ultimately culminating in chaos. This is a classic instability process exhibited by many nonlinear systems [Strogatz, *Nonlinear Dynamics and Chaos*, Addison-Wesley, 1994].

The period-2 instability in the engine is physically explained by considering that the residual mass fraction for a slow burn or partial misfire increases the fuel-air ratio for the next burn. Similarly, a strong burn will leave no fuel in the residual gas, leading to the possibility of a leaner than average mixture and lower output for the next burn. Near stoichiometry, such small changes would have little impact, but as the strongly nonlinear lean combustion boundary is approached, small changes in cylinder inventory produce big consequences. When the alternating strong and weak burns occur, we expect them to show up as an anti-correlation in time series of combustion indices such as heat release and IMEP. As additional bifurcations happen, we expect the detailed structure of the oscillations to become more and more complex.

Additionally, there is considerable uncertainty or "noise" associated with the engine parameters on each cycle. While such noise is actually deterministic, the large number of contributing sources (e.g., injector dynamics, fuel droplet evaporation, in-cylinder flow patterns) make it effectively high-dimensional, and the resulting complex variations can be described in terms of stochastic (typically Gaussian) variations in the nominal engine parameters. For the Daw et al model, the strong combustion nonlinearity amplifies the effect of these stochastic variations, and the tendency to bifurcate and transition to chaos is increased. Although the presence of such noise complicates the cyclic variation patterns, their global features continue to be dominated by the characteristics of the unperturbed nonlinear system.
As long as the nonlinear determinism continues to dominate, it should be possible to use an adaptive control approach of the following form to reduce cyclic variations:

\[ dMF(i) = Gain \times (CE(i) - tgtCE(i)) \] (3)
\[ tgtCE(i+1) = tgtCE(i) + scalar \times dMF(i) \] (4)

where:

\[ tgtCE = \text{desired or target CE}, \]

and Gain and scalar are parameters to be determined experimentally. This approach may also work when CE is not available but we have some quantity that is well correlated with it, such as heat release or even crankshaft acceleration:

\[ dMF(i) = Gain \times (accel(i) - tgtaccel(i)) \] (5)
\[ tgtaccel(i+1) = tgtaccel(i) + scalar \times dMF(i) \] (6)

where:

\[ tgtaccel = \text{desired or target accel}. \]

IDENTIFYING BIFURCATED COMBUSTION

In controlled test cell experiments with an eight-cylinder, 4.6L, 2-valve engine we have demonstrated that the general patterns predicted by the above simple model are actually produced under induced lean fueling conditions (Daw et al., SAE 962086 and Daw et al., Phys Rev E, Vol. 62, No. 2, 2000). Specifically, we monitored dynamic combustion variations in standard indicators such as heat release and IMEP over several thousand cycles by recording and processing the in-cylinder pressure. Analyzing these data with techniques from nonlinear time series analysis (e.g., time delay embedding, return maps, and symbolization), we find strong evidence that the combustion becomes unstable with increasingly lean operation via a period-2 bifurcation sequence that leads to alternating low- and high-power strokes. This bifurcation pattern is clearly visible in our test cell measurements in spite of the stochastic parameter perturbations that are known to be occurring in our experiments. At extremely lean fueling it appears that the engine becomes fully chaotic, although this condition is so erratic it would not seem to be of interest for passenger automobiles. As predicted by the model, increased stochastic perturbations tend to accelerate the onset of the bifurcations (i.e., they begin to occur at higher equivalence ratios). Comparisons of our experimental return map patterns with model predictions show strong similarities that confirm the basic correctness of this model (see Figs. 9-12, Daw et al., SAE 962086 and Figs. 6-8, Daw et al Phys Rev E, Vol. 62, No. 2, 2000).

We have also looked for bifurcations in individual cylinder combustion events of an eight cylinder 4.6L 2-valve engine on a '94 Grand Marquis vehicle. Because in-cylinder pressure transducers were not available on this test vehicle (as is the case for almost all current passenger automobiles), we recorded the engine torsional acceleration as an indicator of combustion event energy output. We derived the engine torsional acceleration from standard timing interval measurements (PIP interval) available in the electronic engine control (EEC) computer. For lean conditions (equivalence ratio, \( \phi \cong 0.77 \)), our observations indicate a definite anti-correlation for all cylinders between consecutive combustion events on the same cylinder. The anti-correlation may be seen in plots of the acceleration associated with a particular cylinder versus the acceleration for that cylinder's prior combustion event (Figure 1).

![Figure 1: Phase Plot](image)

The anti-correlation is indicated by a distribution of the points spread out along a negative slope, indicating a tendency for the accelerations to alternate back and forth between relatively large and small values. For operation under more usual conditions of stoichiometry (equivalence ratio, \( \phi = 1.0 \)), no anti-correlation or other pattern is seen in such plots; rather the distribution is spread in a uniform and symmetric blob.
Figure 2: All Cylinders Lean, $\phi = 0.77$, $G_p = 0.0$
The anti-correlation between successive burns at lean conditions is a manifestation of the bifurcation predicted by the simple model. Even though our torsional acceleration measurements are less accurate than more direct combustion indicators derived from in-cylinder pressure measurements (e.g., heat release), they appear to be quite adequate for detecting the onset of combustion bifurcations in an automobile. The anti-correlation is readily seen on all eight cylinders for lean conditions and low engine speed (400 to 700 rpm). Figure 2 shows phase plots similar to those in Fig. 1, but for each cylinder, arranged in a circle around the phase plot for the average acceleration. The labels, "Sync 0", "Sync 1", etc. refer to the temporal firing order of the cylinders. The engine is operated at idle, with an overall (there is no assurance for individual cylinders – indeed, some of the cylinders, Sync 4 and Sync 7 in particular, appear to be operating at a somewhat leaner point) equivalence ratio of 0.77, as indicated. No control is applied as indicated by the overall gain factor Gp (discussed later), being zero, and the time series of the control signal being flat. The time series (in engine cycles) of the average acceleration is also shown at the bottom of Fig. 2.

CONTROLLING TO REDUCE VARIATION

A direct approach to reducing the anti-correlation is evident: if the current acceleration is low, anticipate that the next will be high and do something -- such as reducing fuel -- to reduce it. Similarly, if the current acceleration is high, increase the fuel for that cylinder’s next event. This approach sounds fairly straightforward, but if it is applied without further consideration, it does not serve to reduce overall cycle to cycle variations. Figure 3 shows results for the ’94 Grand Marquis of applying individual cylinder positive feedback (modifying the as-injected fuel-to-air ratio) as more and more feedback is applied. As the gain, Gp, of the overall feedback is increased, the amount of variation in crankshaft acceleration changes, but the minimum occurs for zero gain.

Figure 4 shows the effect that varying fuel in one cylinder has on subsequent cylinders. Eight traces are shown, one for each cylinder. Each point represents the correlation between fuel injection variation that nominally affects the acceleration of cylinder (i) and the acceleration on other cylinders (i+j). Index in Figure 4 refers to the relative index, (j). All the cylinders follow a similar pattern, so we simplify by using the average response.

Figure 4: Regression Coef. vs. Index

Given the dependence on other cylinder's fuel and the anti-correlation in cylinder accelerations, we may predict a cylinder's next acceleration based on its previous acceleration. Its predicted acceleration will also depend on the fuel charge we are about to give it and, due to spillover effects, the fuel we have just given other cylinders (i-j). Thus, the acceleration in cylinder (i) at time step (k+1) may be predicted by:

$$\text{Acceleration}_{(i)}^{(k+1)} = f(\text{Fuel}_{(i)}^{(k)}, \text{Fuel}_{(i-j)}^{(k)})$$
accel\_i\,(k+1)\,=\,a\,accel\_i\,(k)\,+\,b_i\,dlmb\_i\,(k+1)\,+
\quad b_{i-1}\,dlmb\_i\,-\,1\,(k)\,+
\quad b_{i-2}\,dlmb\_i\,-\,2\,(k)\,+
\quad b_{i-3}\,dlmb\_i\,-\,3\,(k)\,+
\quad b_{i-4}\,dlmb\_i\,-\,4\,(k)\,+
\quad b_{i-5}\,dlmb\_i\,-\,5\,(k)\,+
\quad b_{i-6}\,dlmb\_i\,-\,6\,(k)\,+
\quad b_{i-7}\,dlmb\_i\,-\,7\,(k)\,\quad (7)

where a is the (negative for anti-) correlation coefficient for subsequent accelerations on the same cylinder, \( b_{i-j} \) is the correlation between the \( i \)-th acceleration and the \( (i-j) \)-th change in equivalence ratio, \( dlmb\_i\,-\,j\,(k) \), and \( (k) \) refers to the cycle that has just passed, \( (k+1) \) to the cycle just ahead. For our experiments at an average equivalence ratio of 0.77 and low speed (400 to 700 rpm), we used an \( a \) value of -0.9 and \( b_{i-j} \) obtained from the average response of Figure 4. For any cylinder at the point in time when we wish to determine \( dlmb\_i\,(k+1) \), we have the previous acceleration, \( accel\_i\,(k) \), all the \( dlmb\_i\,-\,j\,(k) \)'s we have produced leading up to the moment, and the target acceleration, \( accel\_i\,(k+1) \), we wish to get. The equation may then be solved for \( dlmb\_i\,(k+1) \), the next fuel change to give cylinder \( i \).

To compensate for the potential deleterious effects of individual cylinder positive feedback, it is necessary to apply negative feedback. The extent of coordination between individual cylinder accelerations may be monitored by looking at the phase of acceleration (the difference between the current acceleration and a running average of accelerations) averaged over the past eight events. Applying a modification to as-injected fuel-to-air ratio opposite to this phase is effective in suppressing cycle to cycle oscillations. If this modification is neglected, Figure 5 shows the global engine oscillations that may result, particularly at higher individual gains.

Figure 5: mean \( \phi \,=\,0.77 \), individual \( Gp \,=\,0.70 \)
Figure 6: Combined strategy, $\phi = 0.77, \ G_p = 0.4$
RESULTS

Combining individual cylinder positive feedback with compensation for cylinder spillover effects and negative feedback on phase results in reducing overall variations in accelerations as well as anti-correlation. The results of the combined strategy imposed on a '94 Grand Marquis vehicle are shown in Figure 6. Comparing to Figure 2, we see a noticeable reduction in anti-correlation and overall variations in acceleration resulting from applying a certain level (Gp = 0.4) of this control. Figure 7 shows how the overall variations in acceleration change with Gp. (Although the minimum variation occurs for Gp = 1.0, operation at that level of gain is not particularly desirable. Further examination of the data shows that high gain leads to a semi-stable control situation of a few cylinders with higher than average fuel and the rest starved. The reduction in variation occurs because the starved cylinders misfire completely and their accelerations are nil.)

![Figure 7: Variation vs. Gain](image)

Figures 8 shows the distribution of accelerations (for cylinder synchronization index 1) with control, and Figure 9 shows the distribution without any control.

![Figure 8: Normal Probability Plot](image)

![Figure 9: Normal Probability Plot](image)

The effect of control is to change an almost bimodal distribution to one that is near Gaussian. We believe that these results are the first successful demonstration of improving cycle to cycle combustion variation by imposing a modified control strategy on a production vehicle.

CONCLUSIONS

We have demonstrated a control approach for reducing cycle-to-cycle variations in an internal combustion engine incorporating a number of unique features.

- We have used our model of the coupling between cycles to exploit a deterministic nonlinear instability. Unlike previous investigators, we recognize that the dominant combustion instability arises from nonlinear bifurcations near the lean limit. We exploit that knowledge to recognize when the instability begins and how it can be countered with feedback perturbations.

- We also recognize the importance of stochastic perturbations. The observed combustion instabilities result from the combined effects of stochastic perturbations and nonlinear determinism. While the basic deterministic structure is not destroyed by the high-dimensional engine parameter perturbations (e.g., fluctuations in the fuel injector), these perturbations do modify the instabilities and the region of air/fuel ratios where they occur. Engine design or operating changes that impact the parameter fluctuations can thus affect cyclic combustion variations.

- We reduce the combustion variation severity by means of cycle resolved feedback perturbations. Because we are able to recognize the deterministic component in combustion variations at lean conditions, we are able to reduce the instability with explicit and simple real-time control algorithms. Computational complexity and overhead are thus minimized. The nonlinear sensitivity of the combustion to small changes in parameters such as fuel
injection pulse width and spark timing allows effective control with very small control inputs. This makes it possible to improve engine operation with little or no net change to time average parameter values.

- We have demonstrated the implementation of this control approach in a commercial automobile using currently available sensors and control systems. By using torsional acceleration as the measured dynamic variable, we have been able to detect the dominant combustion patterns despite the complex transformations imposed by the crankshaft and its dynamics. Further, our control algorithms have been adapted for implementation with an existing EEC computer.

- We have implemented the control simultaneously on multiple cylinders. We have adapted our feedback control algorithm to successfully deal with fuel spillover effects from cylinder to cylinder and phasing interactions between cylinders. Without this refinement, control of commercial multi-cylinder engines would not be practical.

REFERENCES


CONTACT

L. I. Davis, Jr., ldavis3@ford.com, Ford Research Laboratory, P. O. Box 2053, MD2036/SRL, Dearborn, Michigan 48121.