Modelling Combustion Variability in LPG Injected Engines for Improved Engine Performance at Idle

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ABSTRACT
The variability of in-cylinder combustion of gasoline at idle has been investigated previously, culminating in the development of a model relating the past and future indicated torque deviations from the mean at given engine operating conditions of intake manifold pressure, engine speed and spark advance. The developed model has the potential to be used in an idle speed control algorithm to improve vehicle noise vibration and harshness (NVH) at low engine speeds and loads. While environmental considerations have spawned the development of liquefied petroleum gas (LPG) as a viable alternative fuel, adaptation of the variability model to multipoint LPG injected automotive engines is complicated by the fact that the fuel mixture concentrations of propane and butane are subject to wide variations depending on a variety of factors including geographic location and local market pricing. Furthermore, evidence on one engine family suggests that the variability of torque production using LPG injection under cold start conditions is significantly higher than that observed with gasoline injection, a condition that is improved through enhanced modeling of the cyclic torque production process, and subsequent compensation. This paper investigates the development of a model relating past and present indicated torque deviations from the mean at a given engine operating condition in LPG, and explores the variability inherent in the model as a function of temperature. The implication is the model may be incorporated into an idle speed control algorithm, which will provide improved idle speed regulation in multipoint injected LPG vehicles.

INTRODUCTION
The proposed use of advanced algorithms for control of automotive idle speed has led to significant improvements in the disturbance rejection capability of modern spark ignition engines at idle, references [1-4] contain several different examples of proposed controllers for exactly this purpose. Faster disturbance rejection allows a reduction of the engine idle speed set point to a level at which engine noise, vibration and harshness (NVH) is not compromised, even during the application of a step load such as air-conditioner engagement, which temporarily reduces engine speed.

Reductions in the engine idle speed set point can have major impacts on fuel economy, particularly in urban driving conditions where a large proportion of driving time is spent at idle. Results presented in [5] suggest that reductions of 200 rpm in engine idle speed will result in fuel economy improvements of 1-4% over an entire urban drive cycle.

With the advent of these sophisticated idle speed controllers, the lower limit of idle speed set point is no longer characterized by the disturbance rejection capability of the controller to (often known or anticipated) step loads, but the variability of the combustion process itself. The variability of combustion dictates that, even given identical engine operating conditions, the same indicated torque is not observed between consecutive combustion events. At high engine speeds, the variation has been shown to approximate a Gaussian white noise process [6], where the absence of correlation between consecutive events provides no possibility of performance improvement through modeling. However, at idle conditions the effects of internal exhaust gas recirculation and residual gas temperatures are more significant and correlation between consecutive samples has previously been demonstrated in [7, 8].

In previous work [8], the authors looked at producing simple autoregressive moving average (ARMA) models to describe the combustion variability that were based on deviations from the expected, or mean indicated torque at the given operating conditions. Using this model, it was possible to predict the next deviation from the mean combustion torque and adjust the idle speed control variables to take the predicted deviation into account. The benefits of this approach include reduced vehicle NVH at idle engine speeds, and/or further reductions in the idle speed set point for a given NVH quality.

As an alternative automotive fuel, liquefied petroleum gas (LPG) has a simpler chemical composition than gasoline which leads to several well-established desirable properties including better emissions performance and reduced spark plug and combustion chamber deposits. LPG fuelled vehicles have traditionally been run with a gas carburetor system, however tightening emission legislation and performance requirements means the greater degree of control afforded by fuel injection will be a necessity in future
generations of LPG fuelled vehicles. The first generation of LPG injected engines are likely to involve gaseous phase injection at a higher compression ratio, so as a result in this paper the combustion variability characteristics of multipoint-injected, gaseous phase injected engines at idle will be investigated. Earlier work reported in a companion paper [8] allows comparisons of the variability characteristics between gasoline injection and LPG injection on the same engine family, detailed in the Appendix.

Building on the earlier work, the aims of this paper are principally as follows:

1. To investigate the effect injection timing and engine operating point have on combustion variability in a multipoint LPG-injected engine operating at a higher compression ratio than an otherwise equivalent gasoline powered engine from the same engine family,
2. To develop a simple model describing the combustion variability phenomenon in the LPG injected engine and to compare with the one developed previously for gasoline
3. To investigate the feasibility of compensation of combustion variability in an idle speed controller in the LPG fuelled vehicle.

Unless otherwise stated, all results in this paper are for stoichiometric \((\lambda = 1)\) operation due to the post-catalyst emissions restrictions imposed on production vehicles.

DEVELOPMENT OF A MODEL FOR COMBUSTION VARIABILITY BASED ON TORQUE DEVIATIONS

As mentioned earlier, a model relating indicated torque variations between adjacent combustion events was presented in [8]. In order to ascertain the feasibility of a similar model with LPG, the indicated torque deviations were analyzed according to the following procedure.

In-cylinder pressure was obtained over 250 consecutive engine cycles from the same cylinder at a nominal engine operating point. Note that the term ‘nominal’ is used as small fluctuations in manifold pressure and engine speed may occur as the indicated torque changes on a cyclic basis. The indicated mean effective pressure (IMEP) was calculated for each cycle, and then post processed by high pass filtering to remove DC bias and slow drift, thus isolating the variations in IMEP of the process (which are analogous to the deviations in indicated torque).

A sample autocorrelation was calculated allowing for correlations between samples separated by up to 20 combustion events. Given that only 250 data points were taken at each operating point, a threshold of 0.2 was used as the limit to indicate any meaningful correlation between samples. Two sample autocorrelations are shown in Figure 1, illustrating the relationship between samples at two separate operating points.

![Sample autocorrelations](image)

**Figure 1.** Sample autocorrelations between combustion events at (top) 600 rev/min, and (bottom) 800 rev/min, with MAP 30kPa and MBT spark advance.

The results illustrated in Figure 1 are typical of those calculated throughout the idle operating range considered (speeds in the range 600 – 1000 rev/min, at low or no-load operation). Since a correlation is consistently apparent with adjacent combustion events (i.e. a lag of one), a first order model is proposed for the deviations in indicated torque as shown in equations (1) - (3).

\[
\Delta T_e (k) = a \Delta T_e (k - 1) + w(k) \quad (1)
\]

where,

\[
\Delta T_e (k) = T_e \left( P_m, \alpha, N_{idle} \right) - \bar{T}_e \left( P_m, \alpha, N_{idle} \right) \quad (2)
\]

\[
w(k) \sim N(0, \sigma_w^2) \quad (3)
\]

A Gaussian white noise process was verified to model the variability in the absence of correlation at high engine speed and load in petrol fuelled engines in [6], and was used somewhat arbitrarily in [7] so a Gaussian...
white noise process, \( w(k) \), has been used in equation (1) to provide continuity of the model across different engine operating conditions.

In the earlier work using gasoline [8], a two-parameter model was used to describe the variability process. The difference in this case is most likely due to the increased compression ratio used in this case (the compression ratio was increased from 9.65:1 for gasoline to 12.9:1 for LPG). With a higher compression ratio the residual mass fraction will be decreased and hence correlations between events will be diminished. This results in the removal of any apparent relationship between the current combustion event and one occurring in the same cylinder at two previous combustion events.

**EFFECT OF INJECTION TIMING ON VARIABILITY MODEL**

Injection timing relative to intake valve open duration gives an indication of the mixture characteristics of the fuel and air charge. If good mixing is achieved, a relatively low variation in indicated torque would be produced, as the burn will be more even and residuals will be less.

In order to gain some insight into appropriate injection timing at idle, the intake manifold pressure was kept close to 30 kPa, representing a no-load situation, and the engine speed setpoint was changed over three levels – 650, 800 and 1000 rev/min. In each case the spark advance was kept constant close to MBT timing and the fuel injection timing was varied throughout the cycle.

For the engine being used in the experiments, the intake and exhaust valve lift and open durations are provided in Figure 2 below.

![Figure 2. Measured valve lift as a function of degrees after TDC of compression stroke.](image)

In order to investigate the effect variability has on the correlations between consecutive samples, the standard definition of coefficient of variation of IMEP, given in the following equation is used.

\[
COV_{\text{IMEP}} = \frac{\text{Standard deviation of IMEP}}{\text{Mean IMEP}} \times 100\%
\] (4)

Note that the standard deviation of the white noise, \( \sigma_w \), in equation (1) is directly related to the \( COV_{\text{IMEP}} \) by the mean torque at the given operating point. However, because it is a dimensionless quantity the same \( COV_{\text{IMEP}} \) at two different engine loads does not mean that there is the same magnitude of torque variations.

In order to investigate the dependence of both combustion variability and correlation between combustion events, the fuel injection timing was swept through a complete cycle at different engine speeds. The results are plotted in Figure 3.

Not surprisingly, injections after the intake valve have closed will have a much longer interval in which turbulent effects can aid mixing uniformity. Conversely, the variability exhibits a peak in each case when the injection is during the intake valve open duration. Interestingly however, not all injections through the open intake valve result in high \( COV_{\text{IMEP}} \) readings, as there is a sharp peak at approximately 480\(^\circ\) ATDC of the compression stroke, and the variability is roughly constant either side of the peak. This indicates that there is a lack of time for the turbulence to provide a uniform mixture of the fuel and air charge, resulting in higher variability of mixture concentrations and more dependence on fluid dynamic effects such as swirl, which are not modeled in an ARMA approach such as that given in equation (1). This observed peak could be reduced through optimization of the cylinder head and port design, which would also potentially reduce the variability throughout the entire operating range.

Meanwhile, the correlation between consecutive combustion events peaks when the combustion variability is worst. This again is explained by the uneven mixture distribution and the small amount of residuals, even at the relatively high compression ratio used. If the valve overlap was increased, or alternatively the compression ratio was decreased, it is anticipated that residuals would have an even greater effect on the next combustion event, and the correlation between events would be higher.

Any increase in correlation leading to better modeling and potentially improved compensation through control is likely to far outweighed by the increase in the noise, so fuel injection should take place away from the 480\(^\circ\) ATDC peak. As a result, for the remainder of this work, a fuel injection timing of 100\(^\circ\) BTDC of compression will be used.
Figure 3. Coefficient of variation of IMEP and sample first order lag correlation as functions of end of injection timing in degrees ATDC of the compression stroke. (Top) 650 rev/min, (mid) 800 rev/min, (bot) 100 rev/min.

EFFECTS OF ENGINE TEMPERATURE, MANIFOLD PRESSURE, SPARK ADVANCE AND ENGINE SPEED ON COMBUSTION VARIABILITY

The next stage is to investigate the combustion variability as a function of engine operating point, where engine temperature is the first variable to be considered. Knowledge of the variability of combustion as a function of engine temperature allows a temperature threshold to be determined, above which the engine speed can be safely reduced to a warm-engine idle setpoint. This warm idle setpoint is typically several hundred revolutions per minute less than the cold start idle setpoint.

The engine speed was held constant at approximately 1000 rev/min, a no-load manifold pressure of 30 kPa was used along with MBT spark advance and consecutive indicated pressure traces were monitored as the engine coolant temperature was increased from 35°C to a warm temperature at 85°C. The COV$\text{IMEP}$ as a function of temperature at this operating point is plotted in Figure 4.

From Figure 4 it is clear that the variability decreases as the engine coolant temperature increases, which agrees with anecdotal evidence. The first point in the series (at an engine coolant temperature of 35 degrees) shows an artificially low COV$\text{IMEP}$ due to difficulty in maintaining the 30 kPa manifold pressure setting at this temperature. (It will be demonstrated in a subsequent section that higher manifold pressures lead to lower coefficients of variation in IMEP).

A consequence of this result is a lower idle speed set point should not be considered until the variability has reached a reasonable level, i.e. in a production application, cold start would require a higher idle speed to ensure variability and consequently vehicle NVH remain at acceptable levels. This set point could then be reduced as the engine warms up.

Figure 4. Variability as a function of engine temperature at 1000 rev/min and 30 kPa manifold pressure.

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The dependence of $\text{COV}_{\text{IMEP}}$ on manifold pressure and engine speed was considered next. The $\text{COV}_{\text{IMEP}}$ was measured at for a warm engine and MBT spark advance throughout the idle operating range considered. The results are plotted in Figure 5.

Figure 5. $\text{COV}_{\text{IMEP}}$ as a function of manifold pressure and speed.

From Figure 5, it appears that there is a slight increase in $\text{COV}_{\text{IMEP}}$ with decreasing engine speed, and an apparently exponential relationship between the degree of variability and manifold pressure observed in Figure 5 was also noted for gasoline injection in [8]. In the same body of work with gasoline, it was found that the standard deviation of the noise term, $\sigma_w$, remained independent of engine speed as the increase in $\text{COV}_{\text{IMEP}}$ was offset by a decrease in the mean indicated torque. The reasoning behind these relationships lies in the connection between residual mass fraction and these parameters, namely:

1. The residual mass fraction decreases linearly with increasing manifold pressure. Assuming sonic flow through the exhaust valve during valve overlap, there will be an inverse relationship between engine speed and residual mass fraction.

2. The heat capacitance effect gives a linear relationship between flame temperature and mass of the residuals in the cylinder.

3. The square of the laminar flame speed is exponentially related to the flame temperature, i.e. $u_L^2 \propto \exp\left(-\frac{E}{RT}\right)$, where $E$ is the activation energy, $R$ is the universal gas constant and $T$ is the gas temperature.

4. The turbulent flame speed is proportional to the laminar flame speed for a given turbulence intensity.

5. The crank angle for total burn is inversely proportional to the turbulent flame speed (by flame travel over time) although in the region of operation the low degree of curvature means this is almost a linear relationship.

6. The torque produced by the combustion is linearly related to crank angle for total burn, i.e. $T_c \propto -\theta_{\text{burn}}$.

Thus we can conclude that since the mass fraction of the residuals affects the engine torque exponentially, so in regions where there are high residual mass fractions (i.e. at low manifold absolute pressures or low engine speeds) we expect to see much larger combustion variability.

The final parameter that we can expect to affect the combustion variability is the spark advance, or more correctly the spark retard from MBT spark advance. In Figure 6, the $\text{COV}_{\text{IMEP}}$ is shown for both gasoline and LPG injection on the same engine family.

Figure 6. $\text{COV}_{\text{IMEP}}$ as a function of spark retard from MBT at nominal 30kPa load condition using (top) gasoline (bottom) LPG as the injected fuel.

The first point to note is that the magnitude of $\text{COV}_{\text{IMEP}}$ is significantly reduced in the LPG-injected engine at the
same operating conditions. This is primarily a consequence of the increased compression ratio of the engine used for LPG injection.

Secondly, the variability in combustion appears to be roughly constant regardless of spark retard using LPG. Knowledge of the decrease in indicated torque with spark retard from MBT is vital for idle speed control algorithms, as spark advance is one of two parameters used to control the engine speed. At high load conditions it has been shown previously [9] that the spark influence function (defined to be the indicated torque divided by the MBT torque) can be reasonably well represented by the relationship

\[ SI_{\text{high-MAP}} = \left( \cos(\alpha_{\text{MBT}} - \alpha) \right)^{2.875} \]  

However, high combustion variability can act to increase the rate at which the spark influence function decreases. In [8], it was observed that the standard deviation of the noise increased with spark retard from MBT when gasoline was used as the fuel. This had the effect of rapidly increasing the rate at which the spark influence function decreased with increasing retard from MBT. As a result, at 30kPa the following phenomenological model was found to apply

\[ SI_{\text{idle-gasoline}} = \left( \cos(\alpha_{\text{MBT}} - \alpha) \right)^9 \]  

Because the same degree of variability is not observed when using LPG as a fuel with the higher compression ratio, the same power term used in (6) is not applicable. Also, because the magnitude of the variability is approximately constant regardless of engine speed, the different data sets can be combined to provide a modified spark influence function for LPG at 30kPa. The following modified spark influence function was found to reasonably represent the decrease in spark influence function as the retard is increased.

\[ SI_{\text{idle-LPG}} = \left( \cos(\alpha_{\text{MBT}} - \alpha) \right)^5 \]  

Figure 7 shows both the indicated torque to MBT torque ratio, along with the two estimates used of spark influence function described in equations (6) and (7). It is clear that the spark influence function obtained for gasoline does not apply for LPG injection.

![Figure 7. Real and estimated spark influence functions at no-load operation for two speed set points with LPG and gasoline injection.](image)

**POTENTIAL IMPROVEMENTS IN VEHICLE NVH WITH FEEDFORWARD APPLICATION OF THE VARIABILITY MODEL IN IDLE SPEED CONTROL ALGORITHMS**

To ascertain whether the model described in equation (1) can be of any use in compensating variability, the variability of the torque deviations are compared both with and without the underlying model ARMA model.

The parameter \( a \) in equation (1) was estimated via a least squares error approach under the assumption that the noise is zero mean. Using different data sets the torque perturbation vectors and least squares estimates can be generated by defining:

\[ Y = \begin{bmatrix} \Delta T_c(k) & \Delta T_c(k-1) & \ldots & \Delta T_c(k-N) \end{bmatrix}^T \]

\[ X = \begin{bmatrix} \Delta T_c(k-1) & \Delta T_c(k-2) & \ldots & \Delta T_c(k-N) \end{bmatrix}^T \]

Hence it can be easily shown that the least squares estimate of the parameter \( a \) is given by:

\[ a_{LS} = \left( X^T X \right)^{-1} X^T Y \]  

Given an estimate of the underlying process, the white noise can be isolated from the torque production process using the relationship

\[ w(k) = \Delta T(k) - a\Delta T(k-1) \]  

The potential improvements in variability through appropriate predictive compensation can then be ascertained by comparing the variance of the torque perturbations, to the variance of the noise term alone. Two operating points were chosen, the first in a high \( \text{COV}_{\text{IMEP}} \) situation and the second in a low \( \text{COV}_{\text{IMEP}} \) region. These are referred to as Case 1 and 2 respectively, and their operating points are given in Table 1.
Table 1: Engine operating points used

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed (rev/min)</td>
<td>650</td>
<td>800</td>
</tr>
<tr>
<td>Manifold Pressure (kPa)</td>
<td>30</td>
<td>50</td>
</tr>
<tr>
<td>Fuel injection timing</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>Spark retard from MBT</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Mean IMEP</td>
<td>55</td>
<td>272</td>
</tr>
<tr>
<td>COVIMEP</td>
<td>45</td>
<td>3.8</td>
</tr>
</tbody>
</table>

A least squares estimate was generated from data sets at several different operating points, and the same value of $a_{ls} = -0.3$ was used to isolate the torque perturbation noise according to equation (9). This ensures that the parameter is not simply optimally tuned for a given data set, and thus gives a more realistic assessment of the achievable gains. The standard deviations of the torque and the white noise process are shown in the Table 2.

Table 2: Comparison of standard deviation of torque both with and without underlying inter-sample correlations.

<table>
<thead>
<tr>
<th></th>
<th>Case 1 (High COVIMEP)</th>
<th>Case 2 (Low COVIMEP)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Std dev of torque perturbations</td>
<td>7.7</td>
<td>3.3</td>
</tr>
<tr>
<td>Std dev of white noise process</td>
<td>6.9</td>
<td>2.4</td>
</tr>
</tbody>
</table>

Reducions in the standard deviation of the combustion process will have a beneficial effect on the vehicle NVH, potentially lowering the allowable idle speed setpoint and reducing the cost in engine mounts. From the Table above it is clear that an improvement in standard deviation of around 10% in the high COVIMEP region is possible by incorporating equation (1) without the noise term in a feedforward component of an idle speed control algorithm for this engine. This potential improvement is despite the least squares estimate not being based solely on that operating point, so is not completely optimised.

Interestingly though, the low COVIMEP region shows an even greater percentage improvement in standard deviation through elimination of the underlying correlated process. The reason for this is a fundamental limitation of the approach – namely that the model is good at predicting ‘recoveries’ from a bad combustion, but a priori does not predict the occurrence of a bad combustion event. These bad combustion events are significantly more common at low engine speed and low manifold pressure conditions and hence the standard deviation will remain higher in this case.

Whilst this is by no means a conclusive result, it does indicate there may be some substantial benefit to vehicle NVH quality at idle through incorporation of such a model into idle speed control strategies. This would have the effect of potential reductions in engine idle speed setpoint, leading to potential improvements in fuel economy, or reduced costs in engine mounts.

**CONCLUSIONS**

This paper has investigated the combustion variability of an LPG injected engine, with particular focus on the behaviour at idle speed operation.

It was found that the timing of fuel injection relative to intake valve opening, and hence the turbulence and mixing experienced by the fuel charge was important to minimise the variability in combustion. In particular, extremely high variability was observed when the injection finished while the intake valve was fully open.

It was also shown that the engine operating point had a large impact on the combustion variability observed. It was noted that:

- Variability decreased as the engine coolant temperature increased. This indicates that the idle speed set point should be a function of temperature in order to maintain constant cyclic variability.
- The combustion variability increased with decreasing manifold pressure and engine speed. This was attributed to the internal gas recirculation of the engine, since it was noted that the residual mass fraction is increased at lower manifold pressures and engine speeds.
- The combustion variability did not exhibit the same dependence on spark retard from MBT as was previously observed for gasoline operation. This implies that a separate spark influence function must be used in idle speed control applications for LPG injection.

Furthermore a model was presented based on correlations between consecutive combustion events. Application of this model in a feedforward compensation approach to idle speed control may offer benefits such as improved vehicle NVH of the order 10% at idle.

**REFERENCES**


CONTACT

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DEFINITIONS

\( \alpha \) Spark advance, degrees BTDC.

\( \alpha_{MBT} \) Spark advance for MBT, degrees BTDC.

\( \sigma_w \) Standard deviation of indicated torque, Nm.

\( \text{COV}_{\text{IMEP}} \) Coefficient of variation of IMEP.

\( N(\mu, \sigma^2) \) Normally distributed process with mean, \( \mu \), and variance, \( \sigma^2 \)

\( N_{\text{cyl}} \) Number of cylinders.

\( P_m \) Manifold absolute pressure, kPa.

\( SI(-) \) Spark influence function.

\( T_c \) Indicated torque, Nm.

\( \bar{T}_c \) Expected indicated torque, Nm.

\( \Delta T_c \) Deviation of indicated torque from expected indicated torque.

\( w_k \) I.I.D. Gaussian variable at time \( k \).

ACRONYMS

ABDC After bottom dead center

ATDC After top dead center

BBDC Before bottom dead centre

BTDC Before top dead centre

IEGR Internal exhaust gas re-circulation

IMEP Indicated mean effective pressure

MAP Manifold absolute pressure

MBT Minimum advance for best torque

NVH Noise, vibration and harshness

ACKNOWLEDGEMENTS

The authors would like to thank Paul Baker and Don Halpin for their efforts in conversion of the engine to LPG injection, without which this paper would not have appeared.

APPENDIX - SPECIFICATIONS FOR THE TEST ENGINE (FORD AU FALCON)

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement</td>
<td>3.9835 litres</td>
</tr>
<tr>
<td>Bore and stroke</td>
<td>92.26 x 99.31 mm</td>
</tr>
<tr>
<td>No. cylinders</td>
<td>6, inline</td>
</tr>
<tr>
<td>Valve timing: Int.</td>
<td>Opens at 12 degrees BTDC</td>
</tr>
<tr>
<td></td>
<td>Closes at 72 degrees ABDC</td>
</tr>
<tr>
<td>Exh.</td>
<td>Opens at 58 degrees BBDC</td>
</tr>
<tr>
<td></td>
<td>Closes at 24 degrees ATDC</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12.9:1</td>
</tr>
</tbody>
</table>