An idle speed controller for reduced cyclic variability and fuel consumption

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Abstract

In this paper a proposed idle speed controller consisting of decoupled manifold pressure and spark retard control loops is presented. The controller incorporates a model predictive element in the selection of the bypass valve duty cycle, which is used to set the intake manifold pressure. A similar approach is easily adaptable to throttle by wire systems. A simulation model incorporating the cyclic variability of combustion generated torque is presented, and simulation results are provided that illustrate the effectiveness of the proposed control algorithm at not only handling cyclic variability but also potentially reducing the engine idle speed. This latter point ensures fuel economy of a given engine can be improved without adversely impinging on vehicle noise, vibration and harshness quality.

Key-Words: Idle speed control, model predictive control, cyclic variability, fuel economy.

1 Introduction

The automotive engine at idle is a perfect example of the tradeoffs that must be made in order to satisfy a number of different objectives. From solely a fuel economy and noise, vibration and harshness (NVH) perspective the optimal idle speed could be considered as zero, i.e. when the engine is not running, since there is no combustion and therefore no fuel use or vibration concerns. Practically this is not a realistic scenario, because fast takeoff requires the engine to be turning over at idle and so a non-zero engine speed constraint is placed on the problem. At low speeds, while fuel economy is close to optimal the engine is more susceptible to cyclic variations in combustion increasing the vehicle NVH and therefore reducing passenger and driver comfort. The goal of idle speed control can therefore be stated as ensuring that the engine speed remains low enough to provide good fuel economy whilst rejecting not only those disturbances from ancillary load torques such as the air-conditioner and power steering, but also reducing the effect of cyclic variations in combustion to provide good vehicle noise, vibration and harshness (NVH) and avoid engine stall.

In the modern automotive engine there are three control variables that may be used to increase combustion generated (also called indicated) torque and hence reject torque disturbances of the type mentioned in the previous paragraph. The air fuel ratio (AFR) in the cylinder at the time of combustion, the intake manifold pressure and the spark retard from Maximum Brake Torque (MBT) advance all influence how much torque will be produced. Emission regulations require that the air fuel ratio must be kept at or close to stoichiometry which may be achieved using control schemes such as the one documented in [1]. Thus AFR is generally not considered in idle speed control schemes except as a last resort. The manifold pressure is controlled by the air flow around the closed throttle, which is maintained through a bypass air valve (BPAV). Thus the duty cycle set to the BPAV influences the combustion torque produced by dictating how much air enters the cylinder. The advent of drive by wire and in particular throttle by wire systems is likely to eliminate the need for a BPAV in favor of an electronically controlled throttle. In this case the principle of controlling the intake manifold pressure remains the same, although it is accomplished by directly adjusting the throttle opening. No modifications to the control structure are required for this eventuality – only the transfer function of the air flow past the throttle plate would need to be updated.

Manifold pressure has a very large influence on the amount of torque produced, but has the disadvantage of being slow when compared with the disturbance, as the air fuel mixture is induced two piston strokes prior to combustion, and the intake manifold must be filled (or emptied). A much faster control variable is the spark advance. Although there is a spark advance for best torque production at different engine operating points, the spark is generally offset from this advance at idle to give another possible source of torque. The advantage of using spark advance is there is no delay in the torque production yet there is a more limited amount of torque available as dictated by the degree of spark offset from MBT. Furthermore, as the spark is retarded further from MBT the resulting drop in combustion torque must be made up by increasing the air and fuel charge into the engine, hence fuel efficiency decreases with an increase in spark offset from MBT.

In current production automobiles, the multivariable nature of the problem is handled by using PI control on the BPAV and proportional control on the spark advance [2]. While these schemes work, they tend to be far from optimal and require higher idle speeds to ensure smooth operation, and this has spawned the proposal of more advanced algorithms for the problem. Unfortunately the lack of suitable MIMO control tools has meant that most of the advanced idle speed control schemes proposed in the literature use only the bypass air valve as the control variable, [3, 4] or require additional hardware to accomplish the disturbance rejection properties required [5]. One exception is Ford
where a fixed H-infinity control applied to a linearised version of the engine scheme used a load preview to increase the torque reserve temporarily to better cope with load applications.

In the algorithm presented here, the spark advance is used to compensate for immediate torque disturbances, while a model predictive algorithm smoothly controls the bypass air valve to set the manifold pressure to a desired steady state level to completely compensate for the torque disturbance. This decoupling of the control scheme enables the transient variations solely by the spark advance, whilst the use of feedforward information is also encompassed to ensure good “known disturbance” rejection.

2 Idle speed simulation model

The engine at idle is well described by event based, mean value engine models throughout the literature. Event based models are typically used since the control variables (e.g. spark ignition) can only be set once per combustion so it is an intuitively pleasing construction. On the other hand, the use of an event based model means that the torque to speed process must be discretised from the continuous equation, which is a continuous time process during which the piston is driven down by pressure released during the combustion gases in the cylinder thus turning the crank.

There are three states used in the mean value model, engine speed, manifold pressure and indicated torque. The manifold pressure state equation is derived from conservation of mass principles in the intake manifold and is written as

$$P_m(k+1) - P_m(k) = k_1 (m_{in}(k) - m_{ao}(k))$$  \hspace{1cm} (1)

The mass flow into the manifold is controlled by the duty cycle of the BPAV according to some relation determined by the solenoid driver circuit

$$m_{in}(k) = f(u_{BP})$$ \hspace{1cm} (2)

However as discussed earlier, in a throttle-by-wire system this would become a function of the throttle angle itself, or the magnitude of the control signal sent to the throttle.

The mass flow out of the manifold is determined by the engine speed and manifold pressure, however discretising the process into the event domain removes the engine speed dependence

$$m_{ao}(k) = k_mB(k) + k_3$$ \hspace{1cm} (3)

The engine speed state equation is controlled by the net torque (i.e. the combustion generated torque minus frictional effects and any ancillary load torque). In discrete time this becomes

$$N(k+1) = N(k) + \frac{900}{\pi JN(k)} \left( T_c(k) - T_d(k) \right)$$  \hspace{1cm} (4)

The expected or average torque generated by a combustion is a fraction of the maximum torque at the current operating point. The fraction is determined both by the spark offset from the optimal (in terms of torque production) and the air fuel ratio, ie

$$T_c(k) = MBT \left( P_a(k-1), N(k) \right) \times SI\left( \theta(k) - \theta_{MBT}(k) \right) \times AFI \left( \frac{m_{ao}(k)}{m_{ao}(k)} \right)$$  \hspace{1cm} (5)

A mapping is used for the MBT torque, while the spark influence function, $SI\left( \theta(k) - \theta_{MBT}(k) \right)$, has been found to fall off faster with spark retard from MBT at low manifold pressures [7], a region consistent with idle engine operation.

While this equation describes the expected torque produced for a given operating point, internal exhaust gas recirculation and other factors ensure that the actual torque produced fluctuates about this value on a cyclic basis. The next section discusses this phenomena and completes the torque production model.

3 Cyclic variability of torque at idle

It is well known that the torque generated from consecutive combustion events is not identical, even at the same operating conditions. The following diagram illustrates the cyclic variation in torque in one cylinder at an operating point of 600 r/min, manifold pressure of 32 kPa with a spark advance of 15°BTDC.

![Figure 1: Fluctuations in indicated torque at a typical idle operating point (engine speed 600 r/min, intake manifold pressure 32 kPa, spark advance 15° BTDC).](image)

Clearly, this variation affects the lower limit of speed at which the engine can be confidently run, as consistent severe variation will adversely impact on vehicle NVH.
Even partial compensation of these variations which reduces the severity of the oscillations will have a beneficial impact on both lowering the engine idle speed and reducing the amount of damping needed in the engine mounts and subsequently the cost. The relationship between consecutive torque events has created some interest in the literature. Rizzoni [8] investigated the relationship at wide open throttle for different speed set points and found the process was described by a Gaussian white noise sequence no correlation between consecutive events, however as the following figure demonstrates at low engine speeds with low manifold pressures (i.e. at higher relative internal exhaust gas recirculation levels) there is a clear correlation between consecutive events.

![Figure 2: Correlation in indicated torque fluctuations (engine speed 600 r/min, intake manifold pressure 32 kPa, spark advance 15° BTDC).](image)

Ford and Collings [9] proposed that the fluctuations were related due to the impact of the temperature changes in the exhaust residual gases and used an IIR model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input. Instead the process we will adopt here will use the previous figure to demonstrate that an ARMA model with the expected torque at the engine operating point as an input.

\[
\begin{bmatrix}
T_e(p_m^{k+C}, \theta^k) - T_e(p_m^{k+C}, \theta^k) \\
T_e(p_m^{k-C}, \theta^k) - T_e(p_m^{k-C}, \theta^k)
\end{bmatrix}
\]

\[
= \begin{bmatrix}
a & b \\
1 & 0
\end{bmatrix}
\begin{bmatrix}
T_e(p_m^{k-C}, \theta^k) - T_e(p_m^{k-C}, \theta^k) \\
T_e(p_m^{k-2C}, \theta^k-C) - T_e(p_m^{k-2C}, \theta^k-C)
\end{bmatrix}
+ \begin{bmatrix}
\sigma_w \\
0
\end{bmatrix}w_k
\]

The parameters of the model - \(a\), \(b\) and \(\sigma_w\) can be identified at the engine operating point.

4 Model predictive control scheme

A multivariable idle speed controller using spark advance and BPAV typically has to find a balance between disturbance rejection and fuel economy. Typically, for very good disturbance rejection, the spark should be retarded a reasonable amount from the MBT spark advance, in order to give a fast source of additional torque (often referred to as the torque reserve). Naturally the torque reserve obtained by offsetting the spark must be made up using other control variables – in this case the air charge into the manifold, which in turn decreases fuel economy. However, using a very small torque reserve typically necessitates the use of a higher idle speed to maintain vehicle NVH quality in the presence of a step torque disturbance.

One of the methods to maintain good disturbance rejection with a smaller torque reserve is to incorporate feedforward information into the control scheme when applicable. For instance, in the case of the air conditioner the driver may switch the system on, but the actual system may be delayed by some small delay (say 0.3 sec) while the idle speed controller takes preemptive action to prevent a large speed deviation.

The model predictive controller used in this paper uses a decoupled two stage process to minimize speed fluctuations. The spark advance is set to minimize speed deviations due to transient torque loads (these include cyclic variability as well as the effects of an ancillary load application) while the BPAV duty cycle is obtained using a model predictive control scheme to bring the manifold pressure to a level calculated to overcome the torque disturbance and return the spark advance to a setpoint providing the desired torque reserve.

Information about anticipated known disturbances is incorporated into the BPAV control, while the cyclic variability model is inverted and included in the spark advance control. The entire control algorithm is illustrated in the following two figures.

![Figure 3: Determination of desired retard from MBT spark advance](image)
5 Results

5.1 Compensation of cyclic variability

The controller used in the simulations has a feedforward load preview of two engine cycles along with a desired torque reserve of 6 Nm. The desired engine speed was set to be 600 r/min, which represents a level considerably lower than the idle setpoint of the current generation engine controller for the test engine.

In the first simulations, compensating for cyclic variability in torque production is considered. The compensation was achieved by inverting the torque variability model given in equation (6), to predict the next cyclic variation. This is then added to the disturbance torque estimate to produce the total torque that should be compensated for by the spark advance.

A 10 Nm ancillary torque step load was applied to the engine, which is widely regarded in the literature as consistent with the application of the air conditioner. The disturbance rejection capabilities of the proposed control algorithm to such a load application have previously been established in the absence of cyclic variability with deviations of less than 10 r/min, and so the main purpose of this simulation is to demonstrate the results that can be achieved using compensation of cyclic variability.

In Fig 5(a), it is observed that there is quite a large spread in the instantaneous engine speed caused by the variability in combustion-generated torque. Implementing the compensation for this variability provides substantial improvement in the amount of variation observed, as illustrated in Fig 5(b).

Quantitatively, the improvement is seen in a reduction in the standard deviation of the instantaneous engine speed from 27 r/min to 21 r/min, or a reduction of nearly 23%. This improvement has the potential to reduce the cost of engine mounts that diminish the force fluctuations transmitted to the car body.

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It is also worth noting that the engine speed is normally measured over an entire engine revolution, so in a six-cylinder engine (such as the test engine) the measured speed variations are averaged over three consecutive events and therefore more likely to be of the order shown in Fig 6.
5.2 Fuel economy improvements

In this section the fuel economy improvements available using the proposed control scheme are demonstrated. As a point of comparison, the fuel usage at two different speed set points is plotted over a period of 60 seconds in Fig 7. To mimic an existing production controller, at the idle speed setting at 800 r/min a small torque reserve of 2Nm with no load preview is used while the proposed controller is simulated using the lower 600 r/min set point a 6 Nm torque reserve with 3 event load preview. A 10 Nm load is activated after 30 seconds and maintained for the remainder of the simulation.

The total fuel used after one minute at the higher idle speed is 17.3 g while only 14.5 g is used after one minute at the lower idle speed, representing an improvement of almost 20% in fuel economy at idle using no additional hardware. To quantify the effect of the fuel economy improvement at idle has on the fuel economy of the entire drive cycle, the percentage of time and typical fuel use at idle for different international urban drive cycles are shown in Table 3.

Table 1: Percentage of time spent and approximate fuel use at idle engine operation under different international urban drive cycles [10].

<table>
<thead>
<tr>
<th>Drive cycle</th>
<th>Time at idle (% of total time)</th>
<th>Fuel use at idle (% of total fuel)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ADR 37 (US FTP) city cycle</td>
<td>17.9</td>
<td>8.0</td>
</tr>
<tr>
<td>Euro 1</td>
<td>33.5</td>
<td>19.9</td>
</tr>
<tr>
<td>Euro 2/3</td>
<td>25.6</td>
<td>12.8</td>
</tr>
<tr>
<td>Australian Urban</td>
<td>17.8</td>
<td>6.6</td>
</tr>
<tr>
<td>Japanese mode 10</td>
<td>30.9</td>
<td>16.9</td>
</tr>
<tr>
<td>Japanese mode 11</td>
<td>23.0</td>
<td>11.6</td>
</tr>
</tbody>
</table>

Figure 7: Comparison of fuel consumption with different idle speed set points. A load event of 10 Nm is activated at t = 30 seconds. (a) Instantaneous fuel flow rate (b) Cumulative fuel flow rates.

From the data presented in Table 1, improvements in idle fuel consumption of the order demonstrated in Fig 7 will result in a fuel economy improvement over the entire drive cycle of up to 3.5% depending on the drive cycle used. It should be emphasized that this improvement has been achieved by reducing the idle speed using the more sophisticated control algorithms described earlier without compromising the vehicle NVH or requiring any additional hardware.

6 Conclusions

A model for cyclic variability has been presented and results from a test engine obtained that demonstrate a correlation between consecutive torque disturbances. Using this model, future torque fluctuations can be predicted and potentially compensated for in advance.

The proposed control algorithm presented in the paper has demonstrated that the predicted torque fluctuations can be easily incorporated into the controller. This integration of the torque fluctuations model into the control scheme was shown to have the benefit of a substantial reduction in the instantaneous engine speed fluctuations. Reductions in engine speed fluctuations potentially reduce the cost of engine mounts required to dampen the effect of torque fluctuations in the engine on the car body.

Furthermore, reducing the magnitude of the engine speed deviations and improving the disturbance rejection properties of the controller can reduce the desired idle speed set point reduced without sacrificing vehicle NVH or risking engine stall. In the simulated engine it was demonstrated that using the proposed controller the idle speed could be reduced by 200 r/min, which has a flow on effect of improving the fuel consumption by nearly 20% at idle.

Given the amount of time spent at idle in typical urban drive cycles, this improvement equates to nearly 3.5% improvement over the entire drive cycle. Again it is emphasized that no extra hardware was required for these improvements, only modifications to the control scheme at idle. Finally it is reaffirmed that the proposed control structure is applicable to the next generation of
passenger vehicles encompassing drive by wire technology.

7 Nomenclature

\begin{align*}
a, b & \quad \text{Parameters of combustion model, (6)} \\
A F I & \quad \text{Air fuel ratio influence function} \\
C & \quad \text{Number of cylinders} \\
J & \quad \text{Engine inertia (kg m}^2) \\
k & \quad \text{Current event} \\
k_i & \quad \text{Constant, } i = 1, 2, 3 \\
m_{ai} & \quad \text{Mass of air flow into manifold (g/-)} \\
m_{ao} & \quad \text{Mass of air flow out of manifold (g/-)} \\
m_{fo} & \quad \text{Mass of fuel flow out of manifold (g/-)} \\
N & \quad \text{Engine speed (r/min)} \\
\theta & \quad \text{Spark advance (deg. BTDC)} \\
\theta_{MBT} & \quad \text{Spark advance for MBT (deg. BTDC)} \\
P_i & \quad \text{Intake manifold pressure (kPa)} \\
\sigma_w & \quad \text{Standard deviation of combustion (Nm)} \\
SI & \quad \text{Spark influence function} \\
T_c & \quad \text{Actual combustion torque (Nm)} \\
\overline{T_c} & \quad \text{Expected combustion torques (Nm)} \\
T_d & \quad \text{Disturbance torque (Nm)} \\
T_{MBT} & \quad \text{Combustion torque at MBT (Nm)} \\
u_{BP} & \quad \text{Bypass valve duty cycle (-)} \\
\omega_w & \quad \text{White noise}
\end{align*}

8 Engine specifications

The test engine on which the simulation is based has the following specifications:

<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. Op.</td>
<td>12º BTDC, CI: 72º ABDC</td>
</tr>
<tr>
<td>Exh. Op.</td>
<td>58º BBDC, CI: 24º ATDC</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>370 Nm @ 4200 r/min</td>
</tr>
<tr>
<td>Maximum Power</td>
<td>172 kW @ 4800 r/min</td>
</tr>
</tbody>
</table>

9 References


