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Cylinder air-charge estimation for advanced intake valve operation in variable cam timing engines

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Abstract

High efficiency of three-way automotive catalysts is achieved by regulating cylinder air–fuel ratio in a narrow band around the stoichiometry. Due to a delay present in the exhaust gas oxygen sensor signal, performance of the air–fuel ratio regulation depends on accuracy of the cylinder air-charge estimate. Variable cam timing (VCT) introduces a challenge to the air-charge estimator due to its effect on engine pumping and, in some cases, due to unmeasured back-flow of the exhaust gas into the intake manifold. The objective of this paper is to illuminate some of these issues and suggest methods to improve accuracy of air-charge estimation in VCT engines.

1. Introduction

Variable valve timing (VVT) systems are used in spark ignition automotive engines to improve fuel economy, reduce emissions, and increase peak torque and power [1–6]. Here we shall consider only the variable cam phasing (timing) systems, as opposed to other VVT systems such as cam profile switching [7], variable intake/exhaust duration [8], variable valve lift [9,10], and camless (electro-magnetic valve drive) engine systems [11]. In conventional (non-VCT) engines, the relative phase between the camshafts and the crankshaft is fixed at a value that represents a compromise between optimal phases at different operating conditions. In a VCT system, a mechanism varies the relative phase as a function of engine operating conditions [12]. Depending on the camshafts being actuated (exhaust, intake, or both), there are four variable cam timing system types: intake-only, exhaust-only, dual-equal, and dual-independent [2]. In each of these cases the VCT system alters engine pumping and affects cylinder air charge.

A significant reduction in tailpipe emissions of regulated exhaust gases (NO_x, HC, and CO), has been achieved with exhaust after-treatment devices such as three way catalysts. High efficiency of a three-way catalyst in removing regulated gases requires that the internal combustion engine be operated at the stoichiometric air–fuel (AF) ratio (approximately equal to 14.6). Consequently, a high priority task of the engine control system is to maintain the air–fuel ratio at stoichiometry. A key component of this control system is the feedback of the measured oxygen content (and, hence, air–fuel ratio) in the exhaust gas by a HEGO or a UEGO sensor. Because of the inherent delay in the sensor measurement (induction to exhaust delay plus the transport delay from the exhaust port to the sensor that may be as large as 0.5 s), regulation of air–fuel ratio in transients mostly relies on a feedforward component provided by the air-charge estimator.

The mass of air inducted into the cylinders during the intake stroke (air-charge) is estimated from the signal of a sensor located upstream from the cylinder ports as illustrated in Fig. 1. This is either a mass air flow (MAF) sensor (hot wire anemometer) located upstream from the throttle body or an air pressure sensor located in the intake manifold (MAP). The air-charge estimate divided by the stoichiometric air–fuel ratio largely determines the amount of fuel to be injected. Feedback from the HEGO sensor is needed to maintain accurate AF regulation in steady state. A consistent AF ratio regulation error as small as one percent may result in manifold increase in tail-pipe emissions. To compensate for the time it takes to inject fuel, a predicted value of air charge is used for fueling [13]. In addition, an AF

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Nomenclature

N	engine speed
P_{m}	intake manifold pressure (MAP)
P_{a}	ambient pressure (BP)
P_{air}	partial pressure of air
P_{egr}	partial pressure of EGR
T_{m}	intake manifold temperature
T_{a}	ambient temperature
T_{e}	engine coolant temperature
R	specific gas constant for air
V_{m}	intake manifold volume
ζ_{cam}	cam timing variable (intake, exhaust, or dual-equal)
θ	throttle angle (TP)
W_{th}	throttle mass air flow (MAF)
W_{cyl}	mass air flow into cylinders
$W_{\mathrm{egr}}^{\mathrm{in}}$	mass EGR flow into the intake manifold
$W_{\mathrm{egr}}^{\mathrm{out}}$	mass EGR flow out of the intake manifold
m_{cyl}	cylinder air charge
Δt	duration of one engine event (equal to $\frac{120}{Nn_{\mathrm{cyl}}}$)
n_{cyl}	number of engine cylinders.

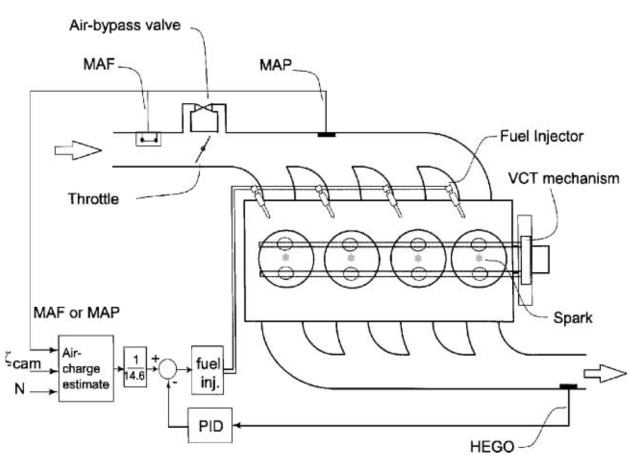

Fig. 1. Configuration of an air–fuel ratio control system in a spark ignition variable cam timing engine.

regulation system may include compensation for fuel-puddling dynamics in the intake port, and parameter adaptation to reduce the effect of slowly varying ambient conditions, aging, and sensor drift on AF ratio regulation accuracy [14].

This paper describes effects of variable cam timing on cylinder air-charge and air–fuel ratio regulation for SI engines with an intake-only, exhaust-only, or dual-equal VCT system. Though mechanically similar, these three systems differ in the way they recirculate the exhaust gas. In conventional engines, exhaust gas recirculation (EGR) is provided by a pulse-width modulated EGR valve between the high pressure exhaust manifold and the low pressure intake manifold. The inert exhaust gas dilutes the air–fuel mixture in the cylinder and reduces combustion temperature, thus reducing formation of oxides of nitrogen (NO_x). VCT engines usually do not require an EGR valve. Engines with dual-equal or exhaust-only VCT recirculate the exhaust gas directly from the exhaust port by delaying the exhaust valve closing into the intake stroke. In contrast, intake-only VCT recirculates the exhaust gas through the intake manifold. Advancing intake valve opening into the exhaust stroke creates a back-flow of EGR into the intake manifold which is then re-inducted during the subsequent intake stroke. Because the mechanisms of internal exhaust gas recirculation are the same for dual-equal and exhaust only VCT, we shall only analyze the air-charge estimation problem for the former; the algorithm should apply to the latter.

The VCT system changes the engine volumetric efficiency and the amount of air charge. At the very least, this requires that the volumetric efficiency coefficients depend on the cam phase, in addition to the conventional dependence on engine speed. If the exhaust gas recirculation is provided by a back-flow, deriving an air-charge estimator for a MAF based system becomes a more challenging problem. The model of intake manifold filling now includes an unmeasured amount of the exhaust gas. The problem can be avoided if the amount of back-flow EGR is measured, but this may be difficult. Another option, pursued in this paper, is to use the model of the manifold filling for the partial pressure of air to derive the estimator. This model requires an unconventional method to determine volumetric efficiency coefficients. Alternatively, if a MAP sensor with a simple speed density calculation is used to determine the air charge, the back-flow effect will be accounted for. Both the MAF and MAP based systems must deal with the temperature change introduced by residual back-flow, and this report addresses this issue. The method of accounting for back-flow residual must take into account the engine mapping process as well.

The paper is organized as follows. Section 2 reviews a generic model of a VCT engine pumping. Section 3 provides an analysis of the engine air intake for a dual-equal VCT engine and derives appropriate air-charge estimation algorithms for both MAF and MAP based systems, including temperature effects. Section 4 addresses air-charge estimation issues related to EGR back-flow in early intake opening VCT engines for a MAF based system. Section 5 presents a simple tuning procedure for calibration of the engine pumping coefficients needed for air-charge estimation.

2. Model of engine air intake

For the purpose of air-charge estimation, we have adopted a simple mean-value model of the intake manifold filling [15]. The dynamics governing the intake manifold pressure rate of change are obtained by differentiating the ideal gas law $PV = mRT$:

$$\dot{P}_{\mathrm{m}} = \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}}\dot{m} + \frac{mR}{V_{\mathrm{m}}}\dot{T}_{\mathrm{m}}$$

$$= \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}}(W_{\mathrm{th}} + W_{\mathrm{egr}}^{\mathrm{in}} - W_{\mathrm{cyl}} - W_{\mathrm{egr}}^{\mathrm{out}}) + \frac{\dot{T}_{\mathrm{m}}}{T_{\mathrm{m}}}P_{\mathrm{m}}. \quad (2.1)$$

VCT engines considered in this paper are not equipped with an EGR valve. Under certain conditions, exhaust gas is introduced into the intake manifold by back-flow through the intake valve. Regardless of the source, the presence of EGR affects the pressure in the intake manifold. The $\dot{T}_{\mathrm{m}}$ term contributes little to the dynamics of the intake manifold pressure. Therefore, as it is standard in the literature on engine modeling and control (cf. [16,17]), we have decided to neglect it.

The mass flow rate into the cylinder is an affine function of the intake manifold pressure, with the slope and offset coefficients that depend on engine speed N, cam timing ζ_{cam}, and the temperatures of the gas in the intake manifold and cylinder walls (or the engine coolant temperature we use as the substitute). In addition, the offset coefficient depends on the exhaust manifold pressure which can be accurately represented as a function of ambient pressure and the cylinder mass flow rate. Typically, the slope and offset coefficients are obtained by regressing engine mapping data which means that the dependence of exhaust pressure on the cylinder mass flow is already taken into account. For the sake of simplicity, we shall neglect the dependence on the ambient pressure. Consequently, the general form of the expression relating the mass flow rate into the cylinders and the intake manifold pressure is given by

$$W_{\mathrm{cyl}} = \mu(T_{\mathrm{m}}, T_{\mathrm{e}})\,[\alpha_1(N, \zeta_{\mathrm{cam}})P_{\mathrm{m}} + \alpha_2(N, \zeta_{\mathrm{cam}})]. \quad (2.2)$$

The behavior of the slope and offset coefficients with respect to cam timing, and the effect on accuracy of air-charge estimation are discussed in the subsequent sections. The temperature correction factor $\mu(T_{\mathrm{m}}, T_{\mathrm{e}}) = 1$ under nominal (engine dynamometer mapping) conditions when $T_{\mathrm{m}} = T_{\mathrm{m_0}}$ and $T_{\mathrm{e}} = T_{\mathrm{e_0}}$. Typically, it is of the form

$$\mu(T_{\mathrm{m}}, T_{\mathrm{e}}) = \frac{T_{\mathrm{e_0}}}{T_{\mathrm{e}}}\sqrt{\frac{T_{\mathrm{m_0}}}{T_{\mathrm{m}}}}.$$

3. Air-charge estimation for dual-equal and exhaust-only VCT engines

Let us first briefly consider the problem of air-charge estimation for dual-equal VCT. In a dual-equal VCT system, internal EGR is provided by retarding intake and exhaust valve timing equally (there is only one VCT actuator per cylinder bank) resulting in the late closing of the exhaust valve, allowing burned exhaust gas to be redrawn into the cylinder as depicted in Fig. 2.

The presence of exhaust gas reduces the air-charge (at a given intake manifold pressure) and, hence, reduces volumetric efficiency. In addition, closing of the intake valve later in the compression stroke reduces the amount of fresh air available for combustion, further reducing the volumetric efficiency of the engine. These two mechanisms provide most of the fuel economy benefits of dual-equal VCT by allowing part load operation at higher intake manifold pressure (thus decreasing pumping losses). Fig. 3 shows experimentally measured values of cylinder mass air flow versus intake manifold pressure at $N = 1500$ RPM and six values of cam phase retard.

Once steady state engine mapping data are available, the slope and offset coefficients can be obtained as functions of engine speed and cam timing and represented in the form of look-up tables or polynomials to be used for air-charge estimation. Depending on the sensor selection, manifold air pressure or (throttle) mass air flow, we shall consider two cylinder air-charge estimation problems.

The main advantages of the MAP sensor, compared to MAF, are its relative proximity to the engine air intake and lower cost. Given the measurements of engine speed N, cam timing ζ_{cam}, and manifold air pressure P_{m}, one can use the affine relationship (2.2) to

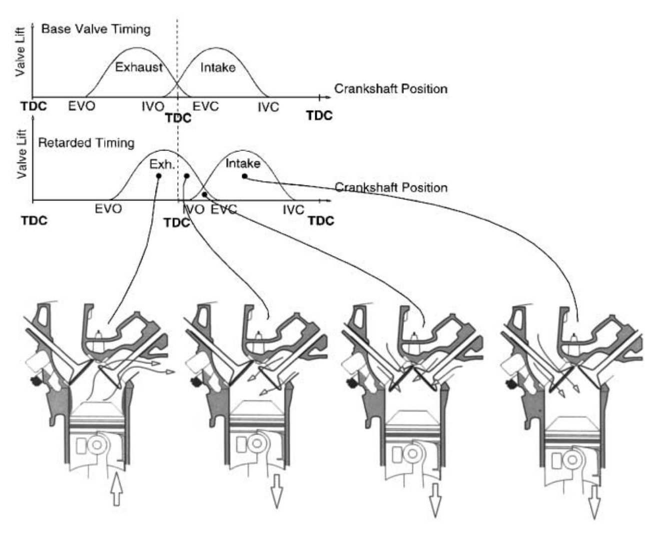

Fig. 2. Flows of air and exhaust gas in and out of a cylinder in a dual-equal VCT engine.

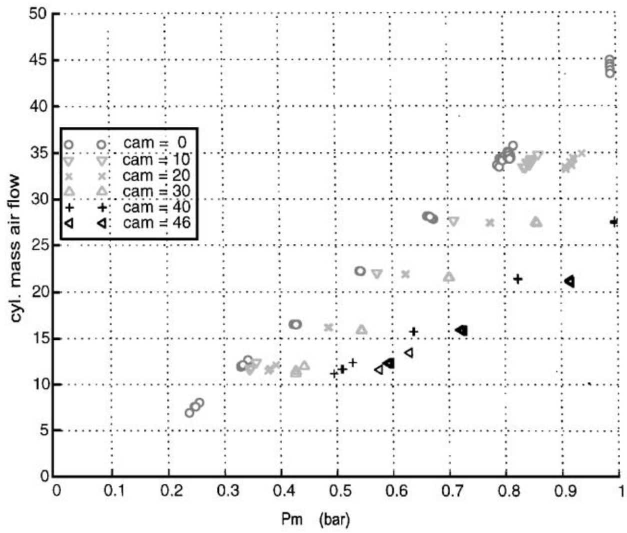

Fig. 3. Cylinder mass air flow remains an affine function of the manifold pressure at different values of cam timing (shown at 1500 RPM).

estimate the cylinder mass air flow:

$$\hat{W}_{\text{cyl}} = \mu(T_{\text{m}}, T_{\text{e}})\,[\alpha_1(N, \zeta_{\text{cam}})P_{\text{m}} + \alpha_2(N, \zeta_{\text{cam}})]. \tag{3.1}$$

Averaging the estimated mass air flow over one engine event gives us an estimate of the cylinder air charge:

$$\hat{m}_{\text{cyl}} = \int_t^{t+\Delta t} \hat{W}_{\text{cyl}}(\sigma)\,\mathrm{d}\sigma \approx \Delta t \hat{W}_{\text{cyl}} = \frac{120}{Nn_{\text{cyl}}}\hat{W}_{\text{cyl}}. \tag{3.2}$$

Equations (3.1) and (3.2) define the so-called "speed-density" air-charge estimator for VCT engines based on the MAP sensor measurements.

The main advantage of the MAF sensor is that, in steady state, it actually measures the cylinder mass air flow. The challenge is, therefore, to accurately account for the intake manifold filling and emptying during transients. We define the new slope and offset coefficients

$$\beta_1(N, \zeta_{\text{cam}}) = \frac{Nn_{\text{cyl}}}{120}\frac{1}{\alpha_1(N, \zeta_{\text{cam}})},$$

$$\beta_2(N, \zeta_{\text{cam}}) = -\frac{\alpha_2(N, \zeta_{\text{cam}})}{\alpha_1(N, \zeta_{\text{cam}})}$$

that characterize the relationship between the intake manifold pressure and cylinder air charge:

$$P_{\text{m}} = \frac{\beta_1(N, \zeta_{\text{cam}})}{\mu(T_{\text{m}}, T_{\text{e}})} m_{\text{cyl}} + \beta_2(N, \zeta_{\text{cam}}). \tag{3.3}$$

Under the assumption that the temperatures change relatively slowly, by differentiating both sides of (3.3) with respect to time and rearranging the terms we obtain

$$\begin{aligned}\dot{m}_{\text{cyl}} &= \frac{\mu}{\beta_1}\left(\dot{P}_{\text{m}} - \frac{\dot{\beta}_1}{\mu}m_{\text{cyl}} - \dot{\beta}_2\right)\\ &= \frac{\mu}{\beta_1}\left[\frac{RT_{\text{m}}}{V_{\text{m}}}\frac{Nn_{\text{cyl}}}{120}(m_{\text{th}} - m_{\text{cyl}}) - \frac{\dot{\beta}_1}{\mu}m_{\text{cyl}} - \dot{\beta}_2\right],\end{aligned} \tag{3.4}$$

where $m_{\text{th}} = \Delta t W_{\text{th}}$. In conventional (non-VCT) engines, the slope and offset coefficients depend only on engine speed; therefore, $\dot{\beta}_i = (\partial\beta_i/\partial N)\dot{N},\ i = 1, 2$. Considering relatively moderate effect of engine speed on volumetric efficiency and, thus, on air charge, one could argue that the time derivatives of the slope and offset coefficients can be neglected. On the other hand, for a VCT engine where the slope and offset coefficients also depend on cam timing, that is, $\dot{\beta}_i = (\partial\beta_i/\partial N)\dot{N} + (\partial\beta_i/\partial\zeta_{\text{cam}})\dot{\zeta}_{\text{cam}},\ i = 1, 2$, neglecting the time derivatives of β_i would result in a more significant error. With a typical range of 40–60°, cam timing has a more significant effect on volumetric efficiency than the engine speed. Moreover, cam phase may go through its full range of travel in as little as 0.3 s. Therefore, for VCT engines the time derivatives of the coefficients must be retained. Taking all this into account, the discrete time version of the air-charge estimator takes the form

$$\hat{m}_{\text{cyl}}(k) = \frac{\frac{\beta_1(k)}{\mu}\hat{m}_{\text{cyl}}(k-1) + \frac{RT_{\text{m}}}{V_{\text{m}}}m_{\text{th}}(k) - \beta_2(k) + \beta_2(k-1)}{\frac{1}{\mu}(2\beta_1(k) - \beta_1(k-1)) + \frac{RT_{\text{m}}}{V_{\text{m}}}}, \tag{3.5}$$

where the sampling time is equal to the duration of one engine event Δt.

Because the exhaust-only and dual-equal VCT systems have the same mechanism for recirculating the exhaust gas, the air-charge estimation is fundamentally the same. Thus, the topic of air-charge estimation for exhaust-only VCT engine will not be discussed further in this paper.

4. Air-charge estimation for intake-only VCT engines

With respect to air-charge estimation, a simplifying advantage of the dual-equal/exhaust-only system over the intake-only case is that the back-flow of the exhaust gas into the intake manifold is small. Retarding the cam timing results in the intake valve opening later, which makes the amount of back-flow even smaller. Hence, the measured pressure in the intake manifold is solely due to fresh air, and the manifold filling model (3.4) accurately represents the engine air intake.

The situation is more complex in the case of intake-only VCT. The mechanism for supplying the (internal) EGR, making deletion of the external EGR valve possible, is to advance the intake valve timing so that the intake valve opens earlier during the exhaust stroke, allowing exhaust gas to enter the intake manifold as depicted in Fig. 4. The lower the pressure in the intake

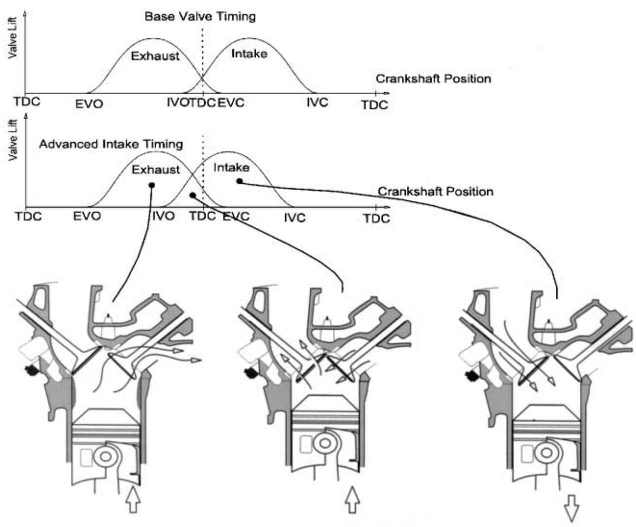

Fig. 4. In intake-only VCT engines EGR is provided by advancing the intake valve timing, resulting in a back-flow of the exhaust gas into the intake manifold.

manifold, the larger the amount of exhaust gas recirculated (at given engine speed and cam timing). The EGR is then drawn from the intake manifold back into the cylinder during the intake stroke.

Another effect of the advanced intake cam timing is that, as the intake valve closes earlier,[1] more air is retained in the cylinder for combustion, increasing the air charge. Both effects are visible in the mass air flow versus manifold pressure plot shown in Fig. 5. At a low manifold pressure, the increase in EGR back-flow dominates and the cylinder mass air flow decreases with cam advance. At high pressure, early intake valve closing has a dominant effect and the air charge increases with cam advance.

Returning to the air-charge estimation problem, we note that, as before, the relationship between the total (measured) intake manifold pressure P_{m} and the cylinder air charge is affine:

$$W_{\mathrm{cyl}} = \mu(T_{\mathrm{a}}, T_{\mathrm{e}})\,[\alpha_1(N, \zeta_{\mathrm{cam}})P_{\mathrm{m}} + \alpha_2(N, \zeta_{\mathrm{cam}})]. \tag{4.1}$$

The slope and offset coefficients α_1 and α_2 are found by measuring steady state values of the manifold pressure and throttle mass air flow, which is equal to the cylinder mass air flow, at a given engine speed/cam timing operating point. If the engine is operated under nominal conditions, the temperature correction factor $\mu = 1$ even if the intake manifold temperature changes due to exhaust gas backflow. The temperature change is closely correlated with intake timing advance and its effects are already accounted for in the mapped values of the slope and offset coefficients. For this reason one should select μ to depend on T_{a} rather than T_{m}.

This relationship between the cylinder mass air flow and the total (air + exhaust gas) manifold pressure allows us to estimate the cylinder mass air flow and cylinder air-charge from the MAP measurements in full analogy with the dual-equal VCT case:

$$\hat{m}_{\mathrm{cyl}} = \frac{120}{N n_{\mathrm{cyl}}}\mu(T_{\mathrm{a}}, T_{\mathrm{e}})(\alpha_1(N, \zeta_{\mathrm{cam}})P_{\mathrm{m}} + \alpha_2(N, \zeta_{\mathrm{cam}})). \tag{4.2}$$

Unfortunately, it has turned out that the MAF based air-charge estimator (3.5) is not directly applicable. The problem can be traced back to the fact that P_{m} dynamics, used in the derivation of the estimator (3.5), now satisfy

$$\dot{P}_{\mathrm{m}} = \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}}(W_{\mathrm{th}} - W_{\mathrm{cyl}} + W_{\mathrm{egr}}^{\mathrm{in}} - W_{\mathrm{egr}}^{\mathrm{out}}). \tag{4.3}$$

Going back to the derivation of the MAF based air-charge estimator, one can see that only the time derivative of the manifold pressure, not the manifold pressure itself, enters the expression. In steady state $W_{\mathrm{egr}}^{\mathrm{in}} = W_{\mathrm{egr}}^{\mathrm{out}}$, so the EGR back-flow does not have an effect on the air-charge estimate. In transients, however, in particular when ζ_{cam} changes rapidly, $W_{\mathrm{egr}}^{\mathrm{in}} \neq W_{\mathrm{egr}}^{\mathrm{out}}$, affecting the accuracy of the air-charge estimate. If the amount of EGR backflow is known, one could be able to compensate for its effect on air charge. But measuring EGR content with a CO_2 sensor placed in the intake manifold may not give the desired result because most of the back-flow gas is redrawn into the cylinder before it becomes mixed with air in the intake manifold.

Consequently, we have to find a different way to design an air-charge estimator from a MAF sensor measurement. Recall that, by Dalton's Law of partial pressures, the total pressure in the intake manifold is equal to the sum of partial pressures of air and EGR, $P_{\mathrm{m}} = P_{\mathrm{air}} + P_{\mathrm{egr}}$. Because each constituent gas satisfies the ideal gas law $PV = mRT$, we can write

$$\dot{P}_{\mathrm{air}} = \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}}(W_{\mathrm{th}} - W_{\mathrm{cyl}}). \tag{4.4}$$

This equation has the same form as the total manifold pressure equation for dual-equal VCT. Moreover, one can reasonably assume that there is an affine relationship between P_{air} and m_{cyl}:

$$P_{\mathrm{air}} = b_1(N, \zeta_{\mathrm{cam}})m_{\mathrm{cyl}} + b_2(N, \zeta_{\mathrm{cam}}), \tag{4.5}$$

where b_1 and b_2 are the new slope and offset coefficients for air. For the sake of simplicity we have set the temperature correction factor $\mu = 1$. Following the derivation for the dual-equal case, one obtains an air-charge estimator for an intake-only VCT engine of

[1] In conventional engines, intake valve closes well into the compression stroke, typically around 50° after the bottom dead center [18].

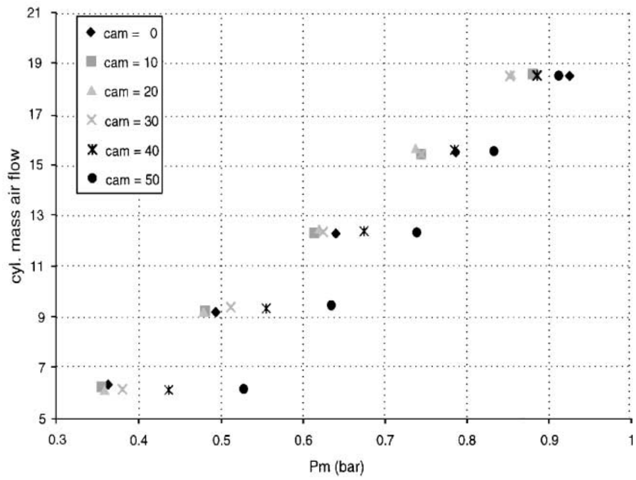

Fig. 5. Cylinder mass air flow as a function of intake manifold pressure and intake cam advance at 1000 RPM.

the same form as (3.5):

$$\hat{m}_{\mathrm{cyl}}(k) = \frac{1}{2b_1(k) - b_1(k-1) + RT_{\mathrm{m}}/V_{\mathrm{m}}} \times \left[b_1(k)\hat{m}_{\mathrm{cyl}}(k-1) + \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}} m_{\mathrm{th}}(k) - b_2(k) + b_2(k-1) \right]. \tag{4.6}$$

The significant difference compared to the dual-equal case is that the steady state engine mapping data cannot be used to determine the slope and offset coefficients b_1 and b_2 because P_{air} is not known (cannot be measured). In this report, we propose two parameter identification methods to determine the coefficients, assuming that transient engine data containing both MAF and MAP, as well as engine speed and cam timing, signal traces are available. First we choose a linear parameterization of the slope and offset coefficients:

$$b_1(N, \zeta_{\mathrm{cam}}) = \sum_1^n b_{1i}\varphi_i(N, \zeta_{\mathrm{cam}}),$$

$$b_2(N, \zeta_{\mathrm{cam}}) = \sum_1^n b_{2i}\varphi_i(N, \zeta_{\mathrm{cam}}), \tag{4.7}$$

where $\varphi_i(N, \zeta_{\mathrm{cam}})$ are appropriately selected basis functions (polynomials, RBFs, etc). Second, from the intake manifold pressure, engine speed, and cam timing signals we can obtain an air-charge estimate $\hat{m}_{\mathrm{cyl}}$ using (4.2) and define a known quantity

$$y = \frac{RT_{\mathrm{m}}}{V_{\mathrm{m}}}(m_{\mathrm{th}} - \hat{m}_{\mathrm{cyl}}).$$

Rearranging (4.6) we obtain

$$y(k) = b_1(k)(2\hat{m}_{\mathrm{cyl}}(k) - \hat{m}_{\mathrm{cyl}}(k-1)) - b_1(k-1)\hat{m}_{\mathrm{cyl}}(k) + b_2(k) - b_2(k-1). \tag{4.8}$$

Using linear parameterization (4.7), we can rewrite (4.8) as

$$y(k) = \phi^{\mathrm{T}}(k)B, \tag{4.9}$$

where B is a $2n$-dimensional vector of unknown parameters and $\phi(k)$ is a known regressor vector:

$$B = \begin{bmatrix} b_{11} \\ \vdots \\ b_{1n} \\ b_{21} \\ \vdots \\ b_{2n} \end{bmatrix},$$

$$\phi(k) = \begin{bmatrix} (2m_{\mathrm{cyl}}(k) - m_{\mathrm{cyl}}(k-1))\varphi_1(k) - m_{\mathrm{cyl}}(k)\varphi_1(k-1) \\ \vdots \\ (2m_{\mathrm{cyl}}(k) - m_{\mathrm{cyl}}(k-1))\varphi_n(k) - m_{\mathrm{cyl}}(k)\varphi_n(k-1) \\ \varphi_1(k) - \varphi_1(k-1) \\ \vdots \\ \varphi_n(k) - \varphi_n(k-1) \end{bmatrix}.$$

The expression (4.9) is in the standard form for the application of the least squares estimation techniques (see, for example, Chapter 3 of [19] and the references therein). Thus, given engine transient data collected at time instants $t = k\Delta t$, $k = 0, 1, \ldots, K$, one can obtain the least squares estimate of B as

$$\hat{B} = (\Phi^{\mathrm{T}}\Phi)^{-1}\Phi^{\mathrm{T}}Y, \tag{4.10}$$

where

$$\Phi = \begin{bmatrix} \phi^{\mathrm{T}}(1) \\ \vdots \\ \phi^{\mathrm{T}}(K) \end{bmatrix}, \qquad Y = \begin{bmatrix} y(1) \\ \vdots \\ y(K) \end{bmatrix}.$$

On line recursive estimation of the B parameters is also possible.

Under ideal conditions (i.e. no measurement noise, "sufficiently rich" regressor vector [19], etc.) the parameters $\hat{B}$ obtained by (4.10) when used in the MAF based air-charge estimator (4.6) should produce an estimate that coincides with the MAP based estimate (4.2).

5. Calibration of slope and offset coefficients

If one wants to avoid collecting and processing transient engine data as required for the identification algorithm described above, an alternative, described in this section, is a simple calibration method that can be performed in a dynamometer test cell. Let us assume that the slope b_1 and offset b_2 coefficients are affine functions of cam timing:

$$b_1(N,\zeta_{\rm cam}) = b_{11}(N)\zeta_{\rm cam} + b_{10}(N),$$
$$b_2(N,\zeta_{\rm cam}) = b_{21}(N)\zeta_{\rm cam} + b_{20}(N). \tag{5.1}$$

This form is convenient for determining the coefficients in a dynamometer setting where the engine speed can be held approximately constant so that b_{ij} parameters are constant. A nonlinear dependence on cam timing can also be accommodated as described below. If the data are collected in a vehicle where the engine speed cannot be held constant, a different representation of the slope and offset coefficients may be more appropriate.

When the cam advance is 0 (base timing), exhaust gas backflow is negligibly small and one can determine the parameters $b_{i0},\ i = 1, 2$, directly from the conventional static engine mapping data because $P_{\rm m} = P_{\rm air}$. Hence, we shall consider these two parameters known across the speed range. If the engine speed is kept constant, the parameters $b_{ij}(N)$ are constant. In this case, Eq. (4.8) can be rewritten as

$$\begin{aligned} &y(k) - b_{10}(N)(\hat{m}_{\rm cyl}(k) - \hat{m}_{\rm cyl}(k-1)) \\ &\quad = b_{11}(N)[\hat{m}_{\rm cyl}(k)(2\zeta_{\rm cam}(k) - \zeta_{\rm cam}(k-1)) \\ &\qquad\qquad - \hat{m}_{\rm cyl}(k-1)\zeta_{\rm cam}(k)] \\ &\qquad + b_{21}(N)(\zeta_{\rm cam}(k) - \zeta_{\rm cam}(k-1)). \end{aligned} \tag{5.2}$$

Note that, in steady state, when the variables at times k and $k-1$ are the same, the right-hand side is 0 and (5.2) collapses to $y(k) = 0$. Thus, the only possibility of identifying the parameters b_{11} and b_{21} from (5.2) is during transient operation of the engine.

The procedure for tuning b_{11} and b_{21} parameters can be simplified by observing that, when $\zeta_{\rm cam}(k) = \zeta_{\rm cam}(k-1)$, the parameter b_{21} does not affect the air charge. By setting $\zeta_{\rm cam}(k) = \zeta_0 = \text{constant} \neq 0$, we obtain

$$\begin{aligned} &y(k) - b_{10}(\hat{m}_{\rm cyl}(k) - \hat{m}_{\rm cyl}(k-1)) \\ &\quad = b_{11}\zeta_0(\hat{m}_{\rm cyl}(k) - \hat{m}_{\rm cyl}(k-1)), \end{aligned} \tag{5.3}$$

which shows that by tuning a single parameter b_{11} one can achieve equality between the quantities on the left- and right-hand side and, thus, accurate air-charge estimation. We emphasize that the variables on both sides of (5.3) are known except for the parameter b_{11}. Reliable identification of b_{11} requires that the air charge is varied sufficiently vigorously by changing the throttle angle, that engine speed is held approximately constant (possible in the dynamometer setting), and that ζ_0 is not chosen too small. One can check several different values of ζ_0 (for example, 10, 30, and 50 degrees advance) to test the assumption that b_1 is indeed affine function of cam timing (5.1).

Now with the parameter b_{11} known, we turn to the estimation of b_{21}. To this end, we rewrite (5.2) by moving all known quantities on the left-hand side:

$$\begin{aligned} &y(k) - b_{10}(\hat{m}_{\rm cyl}(k) - \hat{m}_{\rm cyl}(k-1)) + b_{11}[\hat{m}_{\rm cyl}(k)(2\zeta_{\rm cam}(k) \\ &\qquad - \zeta_{\rm cam}(k-1)) - \hat{m}_{\rm cyl}(k-1)\zeta_{\rm cam}(k)] \\ &\quad = b_{21}(\zeta_{\rm cam}(k) - \zeta_{\rm cam}(k-1)). \end{aligned} \tag{5.4}$$

Now, to tune the parameter b_{21}, the cam advance is varied while the engine speed is held constant (since there is nothing gained by varying the throttle angle, we suggest that it also be held constant). Good tuning of b_{21} is achieved when the left- and right-hand side of (5.4) remain approximately equal as the cam timing changes.

In summary, at each fixed engine speed N equal, say, $600, 1000, 1500, \ldots$ RPM, the parameters $b_{ij}(N)$ are tuned manually by applying the following method:

1. Obtain the nominal, that is, 0-cam timing calibration (including $b_{10}(N)$ and $b_{20}(N)$) from the conventional engine mapping data.
2. Set up a data acquisition system that provides real time access to MAF, MAP, engine speed, and cam timing signals. Using the latter three signals compute the speed density air-charge estimate (Eq. (4.2)).
3. Run the engine with the dynamometer in the speed control mode set at $N = 600$ RPM.
4. At a fixed cam timing, say $\zeta_0 = 10^\circ$, induce air-charge variation by changing the throttle angle.
5. By repeating step 4 as many times as needed, tune the parameter b_{11} to make the left and right hand sides of (5.3) approximately equal during transients. Alternatively, tuning of b_{11} can be accomplished by making the speed density air-charge estimate (4.2) equal to the MAF based one (4.6) with b_1 and b_2 set according to (5.1).
6. Repeat steps 4 and 5 with different values of cam advance ζ_0 (say 30° and 50°). If the calibrated values of b_{11}, at different values of ζ_0, are sufficiently close, set final b_{11} value equal to the average of the three and proceed to the next step. If the difference is significant enough, use the nonlinear cam-dependence algorithm described below.
7. At a fixed throttle opening, vary the cam advance sufficiently to induce noticeable air-charge variation.
8. Adjust the parameter b_{21} to minimize the difference between the left- and right-hand side in (5.4) or to minimize the difference between the air-charge estimates given by (4.2) and (4.6).
9. Repeat steps 4–8 for $N = 1000, 1500$, etc. to obtain $b_{i1}(N)$.

In the case when the dependence of b_1 and b_2 on cam is not well approximated by straight lines, as assumed in (5.1), one can proceed as follows. Subdivide the range of possible values for cam into several intervals. The case when two segments are used is shown in Fig. 6.

To estimate the parameter b_{11}, ζ_0 should be chosen less than ζ_1 (say $\zeta_0 = \zeta_1/2$) for the steps 4 and 5 of the

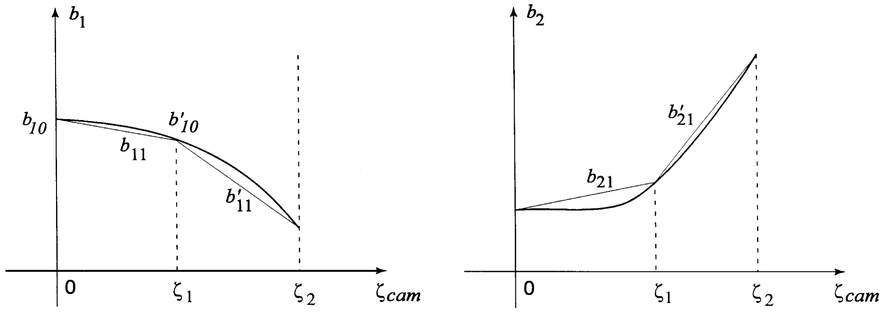

Fig. 6. By appropriately selecting the parameters, one may obtain a good approximation of b_1 and b_2.

tuning algorithm. With b_{11} known, to obtain b'_{11} we use $\zeta_0 = (\zeta_2 + \zeta_1)/2$ and $b'_{10} = b_{10} + b_{11}\zeta_1$ instead of b_{10} in the steps 4 and 5. After b'_{11} is obtained, the calibration of b_1 is complete at a given engine speed. We identify parameters b_{21} and b'_{21} by varying ζ_{cam} between 0 and ζ_1 (for b_{21}) and between ζ_1 and ζ_2 (for b'_{21}) in steps 7 and 8 of the calibration algorithm. Finally, the process is repeated at different engine speeds to obtain the complete calibration for the slope and offset coefficients for air.

6. Conclusion

Variable cam timing systems, introduced in modern automotive engines to improve fuel economy, emissions, torque, and power present a more challenging problem with respect to air-charge estimator design. In this paper we have considered estimator design problems for two VCT systems, dual-equal and intake-only, that rely on different mechanisms for providing exhaust gas recirculation. With respect to air-charge estimation, exhaust-only VCT resembles the dual-equal case. For intake advanced VCT engines we have argued that the conventional steady state engine mapping data may not provide sufficient information to select estimator parameters. By appropriately parameterizing the engine pumping coefficients, we have shown that the standard least squares technique can be applied to transient engine data to determine the estimator parameters. In a dynamometer test cell where engine speed can be held constant, the parameters can be efficiently tuned one at a time. Experimental verification of these results remains a topic for future research.

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