

Low-Cost Air Estimation

Raymond Turin
SimuQuest Inc.

Oguz Dagci and Man-Feng Chang
General Motors Corp.

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ABSTRACT

The focus of this paper is an air charge estimator for engine control system applications which do not feature a mass air flow (MAF) sensor. The proposed approach, beyond its independency of a MAF sensor, is designed to be compatible with the confines of a typical production control system configuration. The air charge estimation algorithm is based on mean-value models for the manifold pressure dynamics and the gas flows through the throttle and valve orifices. It involves nominal static models for the volumetric efficiency of the engine and for the throttle discharge coefficient. The static models for those parameters are complemented with correction factors that are adjusted on-line. The update of the volumetric efficiency correction is implemented in the form of a Kalman-filter which uses the difference between the measured and the modeled manifold pressure as an error metric. The discharge coefficient correction, on the other hand, is implemented in the form of an adjustable correction look-up table. The adjustment of the correction table evolves as a function of the operating condition and is based on an error metric which is derived from the value of the closed-loop fuel control correction factor. The performance of the algorithm is tested on the basis of various realistic driving maneuvers, i.e., maneuvers that include the first 18 cycles of the FTP test schedule and the US06 driving schedule. The experimental tests confirm that the algorithm meets typical performance requirements and that the overall system performance is indeed even comparable to the performance of MAF sensor based

systems. The proposed approach is certainly a prime candidate for deployment in markets with moderate emission standards. However, in light of its cost effectiveness and especially its adaptive capabilities, which make up for many of the long-term detriments commonly seen among MAF-sensor-less low-cost solutions, the approach may even have potential in markets with more stringent emission standards.

INTRODUCTION

The basic objective for fuel metering in gasoline engine applications is to track the amount of air in the cylinder in terms of a predefined air-fuel (A/F) ratio (typically stoichiometric A/F-ratio). Therefore, precise air charge assessment is a critical precondition for any viable open-loop fuel control policy. Since the air charge cannot be measured directly, its assessment, one way or another, depends on information acquired via an intake manifold pressure sensor, or on information acquired via a MAF sensor upstream of the throttle plate, or on both. The choice of a particular sensor configuration reflects a compromise between ultimate system cost and minimum performance requirements. Currently, high cost solutions involving both a pressure and a MAF sensor are commonly found in markets with stringent emission standards while low cost solutions, mostly involving just a pressure sensor, are deployed in emerging markets with more moderate emission standards. In general, in order to meet a given set of emission standards, systems without a MAF sensor require an increased amount of system calibration effort and/or the capability

to assess and correct long term system performance degradation (for example coking of the throttle plate, etc., see, e.g., [7]).

The approach presented here reflects an effort to address these shortcomings. The proposed approach is entirely model-based. Model uncertainty is captured in terms of two sensible physical model parameters, i.e., the volumetric efficiency and the throttle discharge coefficient, respectively. These two parameters are individually assessed and tracked on-line with relative high estimation accuracy. For this purpose the approach relies on sensing information from merely a pressure sensor and a switch type oxygen sensor.

The paper is organized as follows: First, a conveniently parameterized, physics-based mean value model for the relevant flow and manifold filling dynamics is derived. Next, algorithms to instantaneously estimate both the volumetric efficiency of the engine and the throttle discharge coefficient are introduced. Then, the experimental test setup including hardware, software, and test procedures is described. The paper concludes with an illustration and a discussion of experimental test results.

INTAKE MANIFOLD MODEL

There are numerous publications dealing with controls oriented models for the intake manifold filling dynamics and/or cylinder air charge estimation (see e.g., [1], [2], [3]). While the details of a particular modeling approach may differ from case to case the underlying physical assumptions of most solutions remain the same (see Figure 1).

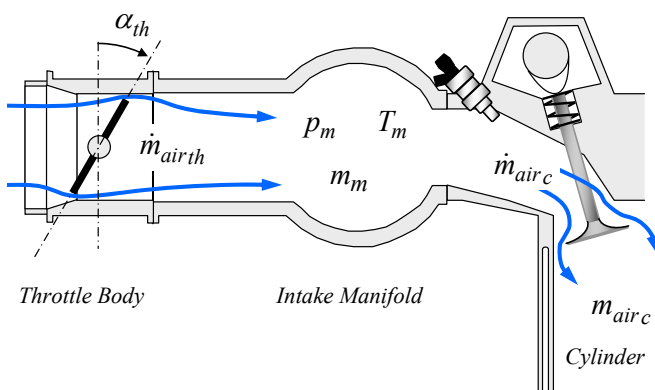


Figure 1 Illustration of manifold and cylinder filling dynamics

Here, a mean-value approach similar to the one outlined in [2] is pursued. The cylinder air charge m_{airc} ingested during one intake stroke is expressed as a function of the manifold pressure p_m , the manifold temperature

T_m , the gas constant for air R , the total engine displacement volume V_d , the number of cylinders N_c , and an empirical coefficient η_{eff} (volumetric efficiency) as follows:

$$m_{airc} = \eta_{eff} \frac{p_m V_d}{RT_m N_c} \quad (1)$$

The volumetric efficiency coefficient η_{eff} thus relates the actual fresh air charge mass to the fresh air mass that could occupy the cylinder if the displaced cylinder volume were entirely replaced with fresh air under manifold conditions. The value of this factor depends on the thermodynamic conditions during the ingestion process and on the valve timing and the valve lift profile. In traditional engine control applications it has been common practice to model the volumetric efficiency factor in the form of a look-up table (as a function of the load and speed conditions). This approach falls short in the case of control applications for modern engines involving variable cam phasing and/or variable valve lift actuation. In these cases the derivation of an accurate volumetric efficiency model is still feasible but is quite more involved (see e.g., [6]).

A mean-value model for the manifold pressure can be obtained from applying the principle of mass conservation to the intake manifold volume. For this purpose, expressions for the mass flows entering and leaving the manifold volume are needed. If the engine speed n is known the mean air mass flow out of the intake manifold into the cylinders can be derived from (1) as follows:

$$\dot{m}_{airc} = \eta_{eff} \frac{p_m V_d}{RT_m} \frac{n}{2} \quad (2)$$

Expression (2) embodies the principles which underlie older, speed-density-based fuel metering systems. The gas mass flow through the throttle orifice into the intake manifold, on the other hand, can be expressed in terms of the compressible flow equation for isentropic flow conditions, i.e.:

$$\dot{m}_{airth} = A_{th} C_d \frac{p_a}{\sqrt{RT_a}} \psi \left\{ \frac{p_m}{p_a} \right\} \quad (3)$$

Here, the variable A_{th} is the throttle orifice area, C_d the throttle area discharge coefficient, p_a and T_a the ambient pressure and temperature, respectively, and ψ is the dimensionless flow coefficient, which is a function

of the isentropic exponent κ and the pressure ratio across the throttle plate, i.e.,

$$\psi = \sqrt{\frac{2\kappa}{\kappa-1} \left[\max\left(\frac{p_m}{p_a}, \beta\right)^{\frac{2}{\kappa}} - \max\left(\frac{p_m}{p_a}, \beta\right)^{\frac{\kappa+1}{\kappa}} \right]} \quad (4)$$

$$\beta = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$$

The orifice area A_{th} can either be modeled geometrically as a function of the throttle dimensions and the throttle opening angle or experimentally in terms of a look-up table. In the former case (geometrical model) the symbol A_{th} reflects the geometric orifice area and the discharge coefficient C_d an empirical factor that accounts for flow contractions in the orifice and thus for the difference between the geometric orifice area and the effective orifice area. In this case, the discharge coefficient is traditionally implemented in the form of a lookup table (e.g., as a function of throttle opening angle and pressure ratio across the throttle plate). In the latter case (experimental model) the symbol A_{th} reflects the stored look-up table values which in turn reflect the effective orifice area. The discharge coefficient C_d then has a nominal value of one.

Mass conservation applied to the manifold volume yields:

$$\frac{d}{dt} m_m = \dot{m}_{air_{th}} - \dot{m}_{air_c} \quad (5)$$

A relationship between the gas mass m_m , the manifold pressure p_m , and the manifold volume V_m can be found via the generic gas equation, i.e.,

$$m_m = \frac{p_m V_m}{RT_m} \quad (6)$$

Hence

$$\frac{d}{dt} \left(\frac{p_m V_m}{RT_m} \right) = \dot{p}_m \frac{V_m}{RT_m} - \dot{T}_m \frac{p_m V_m}{RT_m^2} = \dot{m}_{air_{th}} - \dot{m}_{air_c} \quad (7)$$

Conversely, applying the principle of energy conservation (under negligence of heat transfer effects) yields:

$$\begin{aligned} \frac{d}{dt} (m_m c_v T_m) &= \dot{m}_m c_v T_m + m_m c_v \dot{T}_m \\ &= \underbrace{\dot{m}_{air_{th}} c_p T_{th}}_{\text{enthalpy flow throttle}} - \underbrace{\dot{m}_{air_c} c_p T_m}_{\text{enthalpy flow engine}} \end{aligned} \quad (8)$$

The symbols c_p and c_v stand for the isobaric and isochoric heat capacities for air, respectively, and T_{th} is the gas temperature in the throttle orifice. Combining (7) and (8) leads to

$$\dot{T}_m = \frac{1}{m_m} (\dot{m}_{air_{th}} (\kappa T_{th} - T_m) - \dot{m}_{air_c} (\kappa - 1) T_m) \quad (9)$$

Finally, combining (7) and (9) yields

$$\dot{p}_m = \frac{R\kappa}{V_m} (\dot{m}_{air_{th}} T_{th} - \dot{m}_{air_c} T_m) \quad (10)$$

Equation (10) along with (2) and (3) and (9) form the basis for the charge estimator design. The coefficients η_{eff} and C_d in expressions (2) and (3) both are empirical correction factors for the respective model equations. Note that these coefficients embody the entire uncertainty associated with the estimator model. As mentioned earlier, both coefficients depend on various physical or geometrical quantities (e.g., load, speed, and valve timing in the case of the volumetric efficiency, and pressure, temperature and throttle geometry in the case of the discharge coefficient) and invariably vary with the engine operating conditions. The coefficients can be calibrated on the basis of a given set of static engine performance data. However, there is no guarantee that the calibrated values will be universally applicable during the engine lifetime (due to aging effects) and across an entire fleet of engines (due to manufacturing tolerances). To establish high model accuracy for arbitrary engines at any time during the engine lifespan the values of the correction factors must be adjusted on-line. For this purpose it is convenient to characterize the model uncertainty embodied in the correction factors as follows:

$$\begin{aligned} \eta_{eff} &= \eta_{eff}^0 + \Delta \eta_{eff} \\ C_d &= C_d^0 + \Delta C_d \end{aligned} \quad (11)$$

In other words, the empirical correction factors are both split into a sum consisting of a known nominal part and an unknown (uncertain) corrective part. The nominal part of either coefficient can be modeled in terms of a constant value, a set of look-up tables, or an analytical function that characterizes the physical nature of the respective coefficient. The corrective part, on the other hand, captures effects which are difficult to describe.

For example, it is very difficult to accurately characterize the volumetric efficiency of an engine with variable valve actuation across the entire engine operating envelope. Similarly, it is next to impossible to accurately model the influence of the throttle body temperature or the influence of throttle plate coking on the throttle flow at small throttle angles. Of course, the more a priori knowledge is used to characterize the nominal part of the empirical factors, the smaller the average parameter errors will be and the less corrective action must be taken to achieve an accurate estimate for the transient air charge in the cylinder.

ESTIMATOR DESIGN

From a theoretical viewpoint, (11) along with (2), (3), (9), and (10) could be used to formulate a system identification problem involving a dynamic model with known input A_{th} , known output p_m , and the two unknown parameters $\Delta\eta_{eff}$ and ΔC_d . However, it is well known that in estimation problems involving multiple unknown parameters, simultaneous convergence of all parameters can only be achieved if the system inputs satisfy the criteria for persistency of excitation. In the present case, persistency of excitation is satisfied if the spectral energy associated with the throttle motion has sufficiently large magnitude over a sufficiently large bandwidth. Of course, this criterion will not necessarily be satisfied by natural, driver induced throttle transients as the driver operates the engine without any regard for persistency of excitation. Hence, to guarantee fast parameter convergence under any operating condition, the driver-induced throttle motion would have to be complemented with an appropriate external perturbation signal. Clearly, such a solution is not very practical as it not only compromises transient fuel control but may also lead to unacceptable effects on the perceived driveability. Hence, for all practical purposes, the unknown parameters $\Delta\eta_{eff}$ and ΔC_d in (11) must be estimated in a way that avoids the need for explicit external perturbation signals. In other words, a solution must be found that permits to estimate the two unknown parameters $\Delta\eta_{eff}$ and ΔC_d in independent steps. Such a solution is presented hereafter.

VOLUMETRIC EFFICIENCY ESTIMATION

Assuming for the moment that the throttle flow is known, (2), (3), (10), and (11) can be combined to yield

$$\dot{p}_m = -\eta_{eff}^0 \kappa \frac{V_d}{V_m} \frac{n}{2} p_m - \Delta\eta_{eff} \kappa \frac{V_d}{V_m} \frac{n}{2} p_m + \frac{R\kappa}{V_m} \dot{m}_{air_{th}} T_{th} \quad (12)$$

Subtracting and adding an arbitrary constant k from the first and second term in (12), respectively, leads to

$$\dot{p}_m = -(\eta_{eff}^0 + k) \kappa \frac{V_d}{V_m} \frac{n}{2} p_m + (k - \Delta\eta_{eff}) \kappa \frac{V_d}{V_m} \frac{n}{2} p_m + \frac{R\kappa}{V_m} \dot{m}_{air_{th}} T_{th} \quad (13)$$

In most generic mathematical terms, expression (13) reflects a first order linear, parameter varying differential equation (DE) for the manifold pressure. Now, viewing the first term of the right hand side of (13) as the homogeneous part and the last two terms as two independent inhomogeneous parts of the DE (13), it is easy to show that the first order system (13) is input-output equivalent to the following second order system ([2], [4]):

$$\begin{aligned} \dot{\omega}_1 &= -(\eta_{eff}^0 + k) \kappa \frac{V_d}{V_m} \frac{n}{2} \omega_1 + \kappa \frac{V_d}{V_m} \frac{n}{2} p_m \\ \dot{\omega}_2 &= -(\eta_{eff}^0 + k) \kappa \frac{V_d}{V_m} \frac{n}{2} \omega_2 + \frac{R\kappa}{V_m} \dot{m}_{air_{th}} T_{th} \\ p_m &= (k - \Delta\eta_{eff}) \omega_1 + \omega_2 \end{aligned} \quad (14)$$

The system (14) is said to be a non-minimum order state-space representation of the system (13). It can be viewed as a filter for the estimation of the manifold pressure signal with the parameter k as a filter design parameter.

The output equation in (14) can be simplified using the following substitution

$$\theta = k - \Delta\eta_{eff} \quad (15)$$

The significance of (14) along with (15) lies in the fact that the a priori unknown parameter θ appears exclusively in the output equation of (14). Moreover, the unknown parameter θ appears linear in the output equation of (14). An output equation in this form is called an output equation in linear regression form and is an intrinsic prerequisite for the application of standard linear parameter estimation techniques [4], [5]. Since the state variables ω_1 and ω_2 in (14) are independent of the unknown parameter θ their calculation would be straightforward if the throttle mass flow was known. In the absence of a mass flow sensor this is obviously not the case. Thus, to calculate the second state-variable of (14) the throttle mass flow must be replaced with a suitable approximation. Such an approximation is derived from (3) along with (11) as follows:

$$\hat{m}_{air_{th}} = A_{th} (C_d^0 + \Delta\hat{C}_d) \frac{P_a}{\sqrt{RT_a}} \psi \left\{ \frac{\hat{p}_m}{P_a} \right\} \quad (16)$$

Likewise, the following approximation for the system (14) is introduced: